Coupler Concept

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by

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ABSTRACT

An idea for a coupler that slipped when a torque was applied was conceptualized after seeing a hydraulic gerotor motor tear down. The interface within the hydraulic motor influenced the initial concept leading to the form of the coupler interface. The concept was chosen as a Senior Design project because it was a challenge, an original design, and a good test of engineering abilities. Concepts were thoroughly analyzed and a design route was chosen. The concept was finalized and knowledge of engineering design, manufacturing, and testing was challenged throughout the process. The outcome of the assembled coupler was better than expected from a design and manufacturing standpoint. Once tested, the functionality of the coupler met the intended goal even when the testing methods were gauche. In spite of the concept not having a direct application, the overall results of the project have been gratifying.

BACKGROUND

Couplers have a wide variety of uses whether it is in a manufacturing environment or within a mechanical assembly. The interface between the driver and driven shaft can be manipulated in many ways to diminish or enhance certain effects that may be caused by the driven end. For example, flexible couplers are used to reduce vibrations and/or alleviate misaligned shafts; rigid couplings are used when the driven shaft needs to replicate the driving shaft. Shaft couplers can also account for axial misalignment and torque-limiting properties. The use and form of shaft couplers are vast and can be found wherever a rotating shaft is used.

PROBLEM STATEMENT

One issue between a driving and driven shaft interface may be when the driving shaft accelerates faster than intended or too high a torque value is applied on either side of the shaft transmission. Without the proper coupler or component, either of these cases could cause the one end to react with the same speed or torque, possibly damaging components or assemblies that lie at either end.

CUSTOMER PROFILE

DEMOGRAPHICS:

- Age: 21 – 60+
  - [Aiming for those who are in an engineering or design industry]
- Gender: Male or Female
- Income: Medium+
  - [I assume it will rarely be used for personal use, it would be applicable for any mechanical applications within an industry/company.]
- Skill/Education Level: Technician or Engineer
  - [Intended for those who have completed upper-level education with knowledge and/or an understanding of mechanical applications.]

GEOGRAPHY:

- Region: Anywhere industry is prevalent.
- Relevant Climate Conditions: Manufacturing environment or within a mechanical assembly.

BUDGET RESTRICTIONS:

- Cost – Under $1,000
  - [Aiming for a balance between affordability and high quality.]

INDUSTRY REQUIREMENTS:

- Safety Requirements/Standards – Compliant
  - Complies with any OSHA regulations regarding couplings if applied in manufacturing environment.
  - Design will prevent any safety hazards from occurring.

CUSTOMER FEATURES

Most needs given below were assumed. The features were chosen from a viewpoint of the concept being applicable in the industry and the function being utilized within a design of a potential product. Each feature is in order from purchase of product to full usage of product.

- Affordable
- Durable & high quality
- Easy installation/implementation
- Little to no maintenance
- Replaceable parts
- Design not overly complex
- Accounts for small misalignments in shafts
- Components adequately covered for protection against debris

SURVEY RESULTS

Three survey responses were gathered, one of which was beneficial towards the design of the product. It had a small amount of influence on the design choices made. *See Appendix A for full survey responses.*

RESEARCH, TECHNOLOGY AND CURRENT STATE OF THE ART

COMMONLY USED SHAFT COUPLERS

As stated in the background, couplers cover a large area of application. Different kinds of couplers perform different duties. Below are the most common types of couplers applied to shafts and their functions. *All information below gathered from* (Carr).

- Bellows coupling:
    - "Flexibility to handle all three types of shaft misalignment with low vibration at high speeds."
    - "good for precision stepper and encoder motion-control applications."
    - Zero backlash.
- Flexible shaft couplings:
    - "Cuts in the coupling body allow flexibility to handle shaft misalignment. Good for applications with frequent starts and stops."
    - Zero backlash.
- Lovejoy Coupling:
    - "Cushion between two hubs to reduce shock and handle minor shaft misalignment."
- Oldham coupling:
    - "Slotted disc that allows hubs to slide independently, which is especially good for applications with parallel shaft misalignment."
    - Zero backlash.
- Over-torque protection coupling:
    - "Protect[s] power-transmission components from damage, these couplings will shear or tear in over-torque conditions (such as when a machine jams) to sever connections between shafts."
    - Compensates for some mis-alignment.
- Roller chain coupling:
    - "Allows excellent torque and angular misalignment capacities."
- Schmidt Coupling:
    - "Handle higher angular misalignment than other replaceable-center couplings."
    - Zero backlash.
- Vibration damping coupling:
    - "A flexible tire on these couplings safeguards components on shafts by damping vibration and shock. Couplings also handle moderate parallel, angular, and axial shaft misalignment as well as fair amounts of torque."

The couplers discussed above are the current state of common shaft couplings. Many are used for axial misalignments, parallel misalignments, and vibration damping.

The Over-Torque Protection coupler relates the closest to the coupler concept out of the common couplers mentioned above. If the driving shaft's torque reaches a certain threshold or it goes over that given amount, the coupler will shear and it will no longer be able to transmit torque. This same application could be used in the Coupler Concept. If the driven end is locked, the driving end will spin within the coupling.

TORQUE-LIMITING COUPLERS

The Coupler Concept's intended function is most similar to torque-limiting couplers. There are numerous forms of torque-limiting couplers such as friction plates, shear pins, and ball-detent designs.

Friction Plates

In a friction plate assembly, when the torque on one end reaches an amount higher than what the plates are rated, the plates will slip because the torque value will overtake the force of friction between the two plates. (Birnstiel, Bowden and Foerster) (Mott)

Shear Pin

Shear Pin limiters rely on the shear strength of a shear pin to determine when a torque value is too high. When the torque exceeds the desired amount, the torque load is transferred to the pin, shearing the pin rather than the load being transferred the driven end. (Ondrus Jr.)

Ball-Detent

A current variety of torque-limiting and over-load limiting couplers are currently on the market utilizing a ball detent design, one of which is manufactured by Mayr. This line of couplers is closest in design to my Coupler Concept. As with my concept, it utilizes notches and ball-bearings (ball-detent) to determine when there is an overload or undesirable torque value being input into the shaft. However, in the Mayr design, a proximity sensor is tripped when the bearings move out of their assigned notches. As a torque or load is applied, the bearings will move out of their notches, pushing against a spring washer, which in turn moves an outer ring on the coupler. The movement of the outer ring on the coupler trips a proximity sensor connected to the PLC. This trip will stop the motor or power source providing movement to the shaft. Although this design is similar to the Coupler Concept, the product was not discovered until after the final concept selection. (mayr)

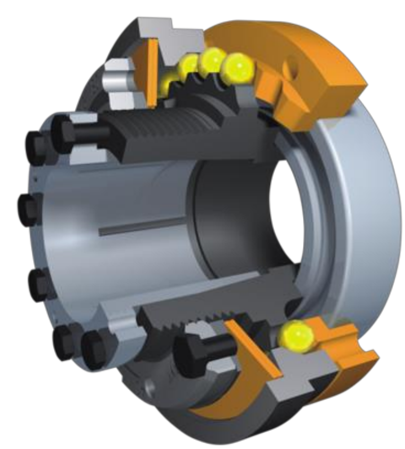

Figure 1: Mayr Torque Limiter

A more common used ball-detent torque limiting design is within a cordless power drill. Many drills have a feature that allows the operator to adjust the desired torque transferred to the object being drilled. Once a torque value is met or exceeded, ball-detent or interfacing detent components within the drill housing, slip out of the detents. This causes the coupling to slip and does not allow the applied torque from the drill's motor to transfer to the driven output of the drill. (Aeberhard) (Roehm)

Both the Mayr Torque-Limiter and the detent designs within a drill or other rotational power tool are very similar to that of the Coupler Concept. The primary difference is that in both designs, the spring and ball-detent movement is parallel to that of the rotational axis of the shaft. In the Coupler Concept, the spring and ball-detent movement is perpendicular to said axis. The other large difference is the adjustability in design within the Coupler Concept. This topic is covered in the *Design* section.

Torque Converter

A vaguely similar product to the Coupler Concept is a torque converter, most commonly found in automotive transmissions. The intent of a torque converter is to smoothly transmit torque between the engine and the transmission. The end attached to the engine providing power quickly rotates while the other end attached to the transmission gradually builds up its rotational speed. This makes the transition of torque much smoother than if it was directly attached to the engine output. (Heisler)

The use of a torque converter relates to the design of the coupler through limiting the rotational speed between the driving end to the driven end. The torque converter does so hydraulically and the acceleration limiter would do so mechanically.

PRODUCT OBJECTIVE

I plan to approach the issue of too high a torque value or sudden acceleration on either end of a shaft coupling by designing a coupler that utilizes a slipping function to reduce or eliminate possible damage from occurring at either end of the transmission.

This coupler will be intended for use as a functioning prototype and proof of concept more so than a product to be put on the market.

DESIGN

CONCEPTUALIZATION

Concept 1

This is the original concept that led to the design of this coupler concept. The interface design came from seeing the internals of a hydraulic gerotor. The internal circular part with 8 smaller circles included in it (1A) would be attached to the driving shaft. The outer parts with the larger half circles (1C) falling in between the spaces in the internal piece, would be connected with an elastic or pliable band of material (1B). If there was an undesirable rotation on the driving end, the thought was that the pliable material holding the outer pieces together (1B) would expand due to

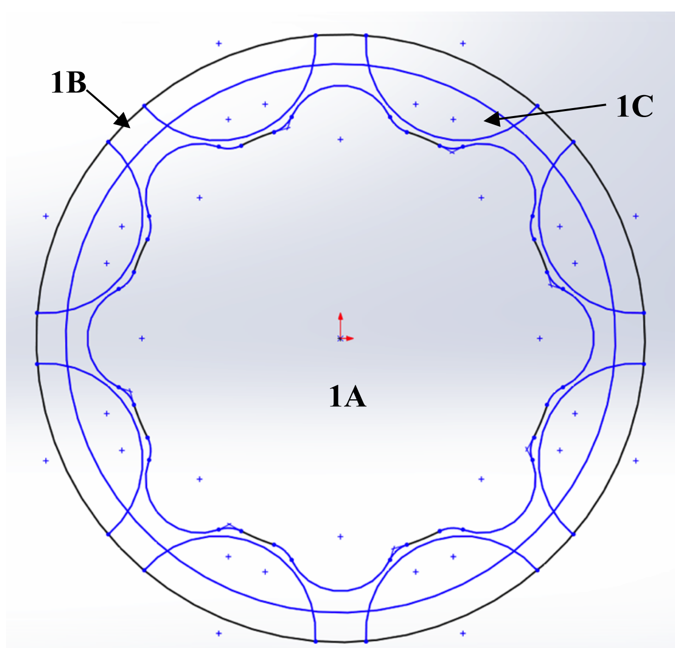

Figure 2: Concept 1 Sketch

the force applied by the internal rotating part, causing it to slip. The main issue with this design is attaching the moving and flexible outer parts (1B & 1C) to the driven shaft. This issue led to further concepts.

Concept 2

Concept two was the second ideation built off of the original idea keeping the slipping properties in mind. In this design, the driving shaft would turn the internal part (2A). An outer housing (2D) was implemented so a solid component could be attached to the driven shaft. Between the housing (2D) and the turning part (2A) there is an assembly that is able to give and replicate the slipping properties intended. A ball bearing would be attached to a small cylinder (2C). The cylinder would go through the outer housing, to provide the turning force from the driving component. A spring is placed between the ball and the ID of the outer housing (2B), supported internally by the cylinder to provide a force onto the rotating part (2A). Once the undesirable rotation took place, the spring would compress as the ball would move out of the indent, allowing the internal driven component to slip within the assembly.

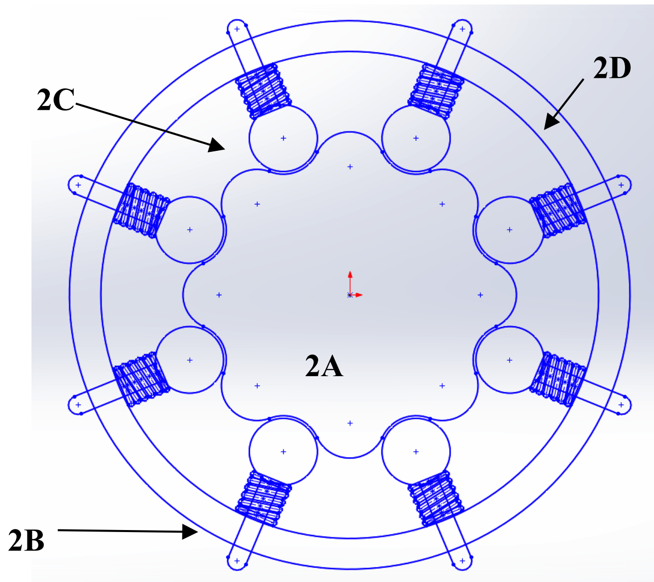

Figure 3: Concept 2 Sketch

Concept 3

In this concept, concept two's same spring (3C) and bearing assembly (3D) was taken and applied to the internal component (3A) attached to the driving shaft. The same affects caused by the spring compressions interacting with the indents takes place but the indents are within the outer housing (3B). This concept allows the driven shaft to be turned by the bearings and indents rather than the cylinders attached to the ball bearings.

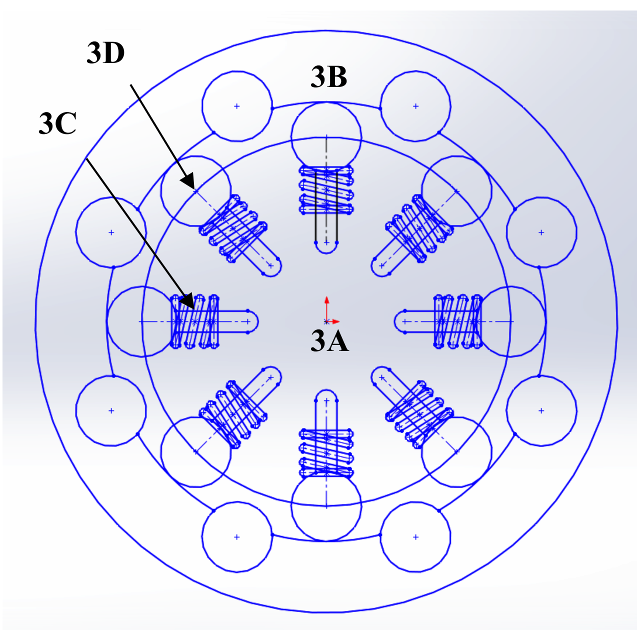

Figure 4: Concept 3 Sketch

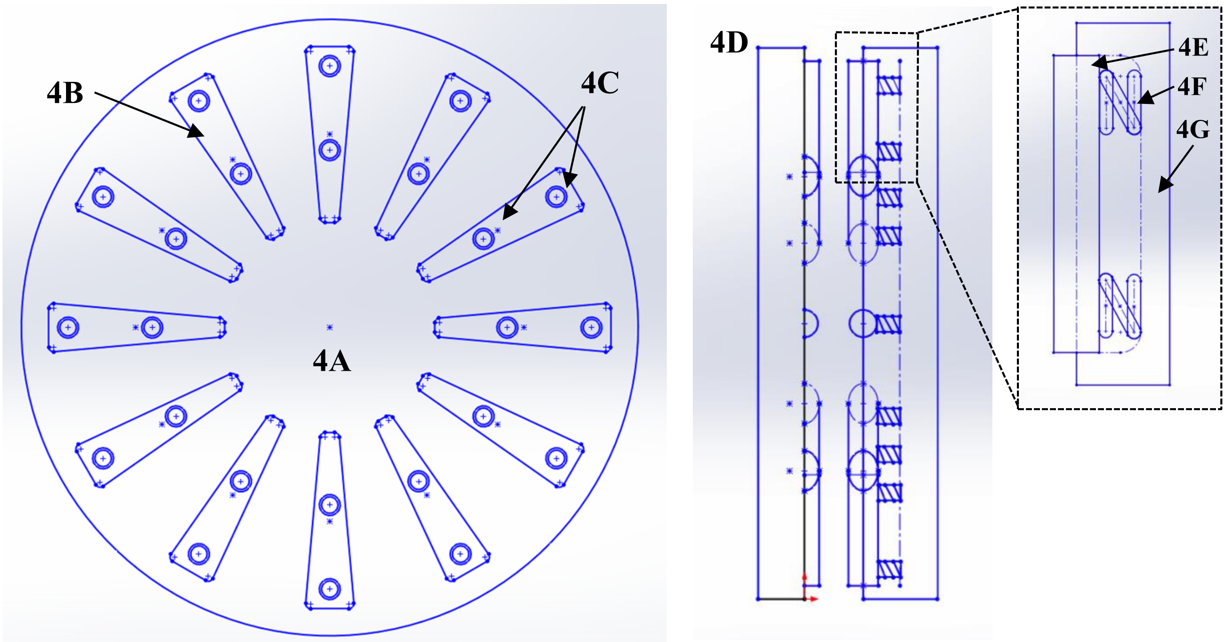

Figure 5: Concept 4 Sketch

Concept 4

This was the last concept that was formed before selection of the design to approach. The bearing and spring interface is carried over but in this concept, the bearings used are roller bearings in please of ball bearings. Both the driving and driven plates would include roller bearings but the driving plate would have springs applying a force behind the roller bearings. This would create the same slipping effect when the driving end would perform incorrectly. The drawing with 4A-4C is the front view, looking at the interacting interface of the driving component. 4A is the plate holding the bearings, 4B points to the roller bearings, and 4C shows the two springs behind the bearings, applying the load. 4D is a side view of the two interfaces interacting, with the right drawing being the driving end. 4E-4G is a view of a single bearing and spring interaction within the housing.

CHOSEN CONCEPT

After consideration of the strength and stability, ease of fabrication, estimated cost, and complexity of each design, **Concept 3** was chosen as the final design route for the following reasons:

- Best Spring Stability
- Best Bearing Stability
- Most Achievable to Fabricate without excessive cost
- Not overly complex
- Driven side of coupler was the best for the driven shaft connection – no moving or changing parts, an issue that was seen in other concepts.

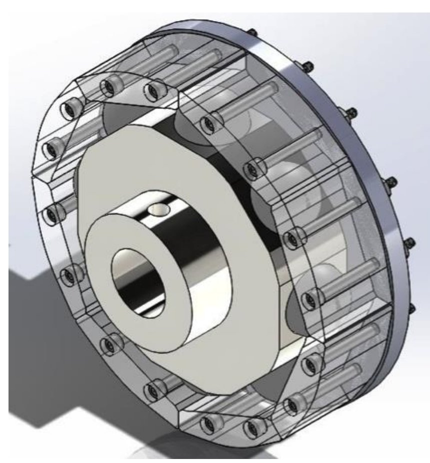

Figure 6: Coupler Isometric View

Coupler Concept Terminology

Throughout the remainder of the report, terminology of components of the coupler and positions of the bearings will be used. The names of the components that will be used can be found in *Figure 7*and *Table 1*. The terms of the bearing position can be found in *Figure 8* and *Table 1*.

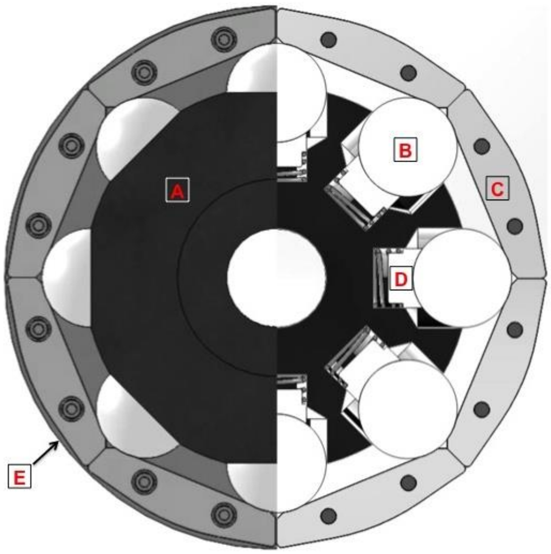

Figure 7: Coupler Term Reference

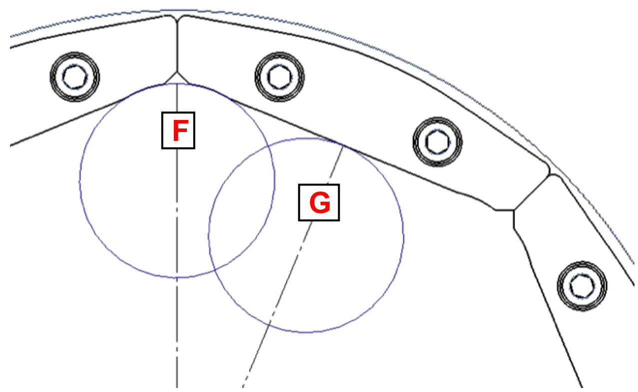

Figure 8: Bearing Position Term Reference

Letter	Name
A	Housing
B	Bearing
C	Interface *(Piece)*
D	Insert
E	*(Interface)* Plate
F	Normal Position
G	Center of Interface

Table 1: Coupler Terminology Reference

Design Process

The following section is an overview of the design process of going from the Concept 3 selection to a finalized design just prior to fabrication. The actual process likely differed in certain areas such as iterations of dimensioning. The detailed design steps were not recorded with detail due to time and many repeated steps or iterations of small changes that would have been unnecessarily repetitious. The following steps also overlapped each other multiple times throughout the process but for organizational purposes, they are discussed individually. The overlapping steps that required one another will be mentioned within the steps if necessary.

Progressing Concept 3's Design

- Once the concept was selected, more practical and detailed concepts were stemmed from the design.
    - *These refined concept sketches can be seen in Appendix F.*
- A component to build off of and base the dimensions on was selected so there was a starting point and so that there was not a never-ending cycle of iterations.
- The component chosen was the 1.0" ball bearing. It was an appropriate diameter for the general coupler size desired and it was a component that was not going to vary in shape throughout the design.
- Next, the bearing was placed in a circle eight times to form the size of the housing the bearings would be in.
    - Eight bearings waere chosen because when only four were used, the interface between the bearings was too large and too far apart. An odd number of bearings would be more difficult to manufacture.
- After the size of the housing formed, it was decided that the center point of the ball bearing would be on the same plane as the edge of the housing in the normal position. This feature was chosen so that the largest OD of the bearing would always be in contact with the housing. This also set the outer limits of the interface size.
- Next, the dimensional limits of the depth of the hole for the spring and bearing in the housing were set.
- *Without knowing the final form of the interface, it was difficult to have a torque value to design for. The bearing and spring interaction was known but many interface designs were considered. Only until an interface was finalized could a torque value be calculated. Because of this, the design choices were reliant on the results of spring calculations. No specific torque value was designed for but a range was targeted. The torque wanted needed to slip the coupler when a small amount of manual force was applied but not so high of a torque that it would be difficult to slip manually and so that components would not be subject to large stresses.*

Spring Selection

- Knowing the limits of the hole depth in the housing, a spring was to be chosen. A spring with a smaller OD could go deeper into the housing but was likely to be unstable during performance as springs with a length larger than the OD are prone to buckling. A spring with a larger OD would be shallower but have a better distribution of force onto the bearing.
- A limit of the spring OD of 0.75" was set due to force distribution onto the bearing and so there would be no interference of the spring holes within the housing. A limit of 0.75" was also set for the length to also limit the potential interference of the spring holes.
- Having these limits, multiple springs (about 25 in total) were chosen for consideration.
- An excel sheet was set up where the necessary values would be plugged in (spring length, spring rate, wire diameter, number of coils, etc.) and the characteristics of the spring would be calculated.

- The following values were influential during spring selection:
    - Spring Rate (given value from manufacturer)
    - Spring Force (at a compressed length)
        - $F = -kx$
    - Spring Index (used for spring verification)
        - $C = D_{mean}/D_{wire} - (5 \leq C \leq 12)$
    - Wahl Factor (used for spring verification)
        - $K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$
    - These equations were selected during the equation selection process before the concept was refined.
- After spring characteristics were calculated and analyzed, the best springs were chosen.
- Using the spring rate and dimensions, the interface designs were applied and the different spring loading conditions were calculated.
    - This step is interchangeable with the interface design step. The two had a close interaction with each other.
- Once the interface design was settled, the best spring rate was calculated and chosen for the design. The spring rates chosen for consideration were chosen based more on a feeling of correctness rather than in-depth theoretical calculation.
- *All spring equations and references used:* (Mott)

Design for Spring & Bearing Interaction

- With the bearing size and position set and the spring hole limits set, a 2D sketch tool (the final sketch tool can be seen in *Figure 9*) of the bearing and spring hole in the housing was drawn in Solidworks. The proper constraints were set and this was used as a tool to determine appropriate and suitable dimensions for different springs (and later on inserts) within the housing.
- In the early concept, the spring would be pressing directly against the bearing but after some thought, this would cause uneven distribution of force onto the bearing (all spring considerations had squared and ground ends) and could potentially damage the bearing surface.
- An insert design was drawn up where one end would be put into the ID of the spring and the other would push on the bearing. The OD of the insert would not exceed the OD of the spring.
    - This design has not changed since the first form except for lengths within the insert, a design choice that is discussed in the *Adjustability within Design* section.

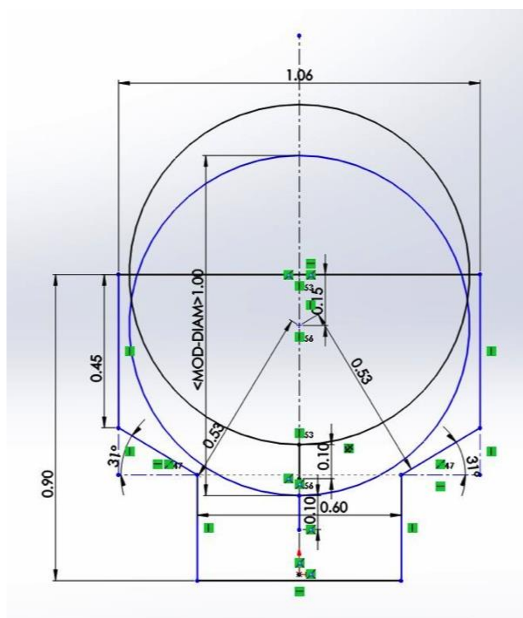

Figure 9: Dimensioning Sketch Tool

Housing Formation

- The design of the housing started off with eight parts, almost like a hexagonal cake cut into eights. This boxy design would originate from a cylindrical piece
- of stock but the diameter would be cut down to flat faces. This design can be seen in *Figure 10*.
- After further definition, the faces featuring the holes were made the same width as the bearing hole. The area between the holes was also flattened out and simply connected to the next hole face. This was easy to draw in CAD but would be difficult to manufacture. This design is seen in *Figure 11*.
- A final form of the flattened faces with the feature holes and keeping the round OD of the stock was chosen.
- No shaft hole was considered or dimensioned until the hole dimensions were finalized. The shaft size would be over designed to ensure it would not fail before the coupler during testing.

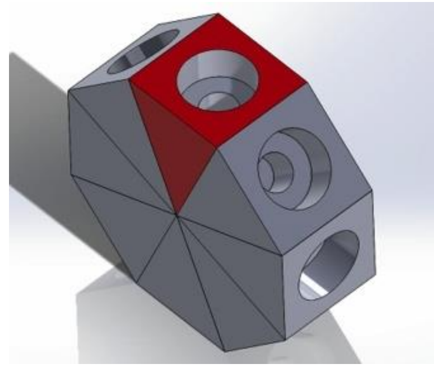

Figure 10: Housing Model 1

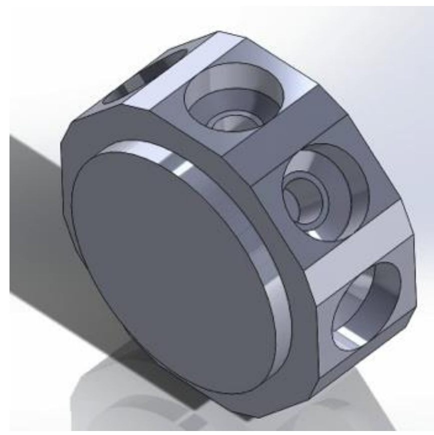

Figure 11: Housing Model 2

Interface Design

- Initially, the interface was designed as being one solid casing around the housing assembly. The full casing was not chosen as it was too complex and costly to machine.
- Originally, the interface component was intended to be fabricated from stock.
- A few different forms of the interface were conceptualized but the triangular detent was chosen as a design route, mainly for ease of fabrication and minimal complexity.
- With the triangular detent design being used, the design started with a 45-degree and a 60-degree angle off of the centerline of the bearing.
- After calculation, both of these designs were too aggressive in the gradient of the spring compression for the coupler to slip. Both the 45 and 60 angle also made the line of bearing travel between detent to detent too difficult to flow easily.

Figure 12: Interface Casing

- Both angles were dropped and then a tangent line was simply drawn from one bearing diameter to the adjacent bearing.
- These tangent lines were then slightly tightened so an indent was formed in the interface. The purpose of the indent was to hold the bearing in place, as the initial torque would be applied. It was assumed that without the indent, the bearing would move forward a small amount every time a torque was applied which is not ideal. This indent can be easily seen in *Figure 8*.
- The interface design was then finalized.
    - This step is interchangeable with the spring design step. The two had a close interaction with each other.

Design Iterations

- As the steps above were performed and often overlapped each other, multiple iterations were performed to analyze and refine on the hole size, spring rate choice, insert dimensions, and the role of the interface design. There were many iterations of calculation and ensuring the dimensions were correct and attainable in the design. All things were considered (cost, manufacturing, complexity, etc.) during this process to have the best design result. Unfortunately, these details of the design iterations were not recorded and cannot be recalled with a high level of confidence.

Adjustability within Design

- Throughout the design, adjustability was taken into account more progressively as the concept came closer to finalization.
- The expected performance being unknown led to planning for some adjustability, if necessary, after testing was performed. This also led to a unique design aspect: availability for change and adjustability for the future customer.
- The following are the features of the coupler where adjustability was implemented into the design:
    - The holes for the springs within the housing were designed to be shallow in case a larger or longer spring is needed to provide a greater force. Extra depth can be machined out of the housing in case of a change in dimensions or a larger spring is desired.
    - On the Insert, thickness was added to the component between the top of the spring and the surface of the bearing, to leave room for a potential dimension or spring change.
    - Filler inserts (different from the component Insert) could be easily created in CAD if springs smaller than the finalized design wanted to be used. They would likely be printed and inserted into the existing hole.
    - The Interfaces can be re-designed or re-shaped and easily implemented onto the driven plate.
- These aspects can be easily changed and recalculated with-in CAD. In case of any changes, they can be implemented into the final assembly accordingly and with ease. This is advantageous both in testing and as an applicable design feature of the component.

See Appendix C for Detailed Drawings

STRESS ANALYSIS

CALCULATIONS

All calculations of forces applied were taken from the compression of the spring at different stages. Below is an example calculation along with descriptions of the different loads applied in simulation.

$$\underline{Hooke's\ Law}$$

$$Force_{Normal\ Position} = -(Spring\ Rate)(\Delta Length) = \left(23.2\frac{lb}{in}\right)(-0.2\ in) = 4.64\ lb$$

Spring Rate	23.2 lb/in

Position	Force from Spring	Comment
Normal Position (-0.200in)	4.62 lbs	Rounded up to 5 lb in most simulations
Edge of Indent (-0.275 in)	6.38 lbs	-
Center of Interface (-0.350 in)	8.12 lbs	-
Safety Factor = 1.97	*16.0 lbs*	Used to test the strength beyond applied load

Table 2: Loads Applied to Components in Simulation

SIMULATIONS

Stability of Simulation

When first attempted, the simulation on the interface pieces and housing would fail from constraint errors therefore unable to give results. The first attempt at resolving this issue was going through each of the materials assigned values and making sure there were no zeroes and the correct values were in place. After multiple failed attempts after this, the simulations were rebuilt and rerun. The bearing design used previously had a flat spot to apply the force through the bearing. In the rebuilt simulations, a small shaft was added into the bearing to provide additional stability for the bearing movement. Once this shaft was added to the bearing for analysis, the simulations ran with ease and gave expected results. A picture of the modified bearing for analysis can be found in *Figure 13*.

Figure 13: Simulation Bearing

Material Properties Used in Simulation

Three materials needed to be manually input into Solidworks to replicate the material's mechanical properties that were planned on being used. The following table shows the values of those materials. The material data for the Glass Filled Nylon was taken from a different supplier than the printing service used.

Material:	**40% GF Nylon**		**Spring Wire (A228)**		**S2 Tool Steel (Bearing)**	
Property:	Value	Unit	Value	Unit	Value	Unit
Elastic Modulus	2240796160	N/m^2	2.1029e11	N/m^2	210000000000	N/m^2
Poisson's Ratio	0.4079	-	0.3130	-	0.285	-
Shear Modulus	-	-	7.998e10	N/m^2	81000000000	N/m^2
Mass Density	1245.6	kg/m^3	7861.1	kg/m^3	7.8	g/cm^3
Tensile Strength	35852736	N/m^2	2132582912	N/m^2	1990000000	N/m^2
Compressive Strength	35852736	N/m^2	-	-	-	-
Yield Strength	32000000	N/m^2	2364081152	N/m^2	1650000000	N/m^2
Therm. Ex. Coeff.	0		-	-	10.9e-6	
Therm. Conduc.	0.2256	W/mK	0.2256	W/mK	24.4	W/mK
Sp. Heat	1386	J/kgK	-	-	0.460	J/gC
Source	(Stratasys) (Amado-Becker, Ramos-Grez and Yanez)		(MatWeb, ASTM A228)		(AZoM)	

Table 3: Simulation Material Properties

Housing

- The housing material was intentionally overdesigned to ensure the function of the spring and bearing assembly was tested properly.
- Because there was a difference in steel quality between the housing and the bearing, (low carbon steel and S7 tool steel, respectively) simulation was run to see if deformation would occur.
- Simulation showed that there were small areas of stress but no deformation would occur on the housing.
- All housing simulations ran with the fixed geometry being the ID of the shaft hole within the housing.

Component	Condition	Loading Description	Load Applied	Figure
Housing	Stress	force of the bearing on wall	16.0 lbs	*Figure 14*
	Stress	force of the bearing on corner edge of housing		*Figure 15*
	Deformation	bearing force on wall of housing		*Figure 16*

Table 4: Housing Simulation

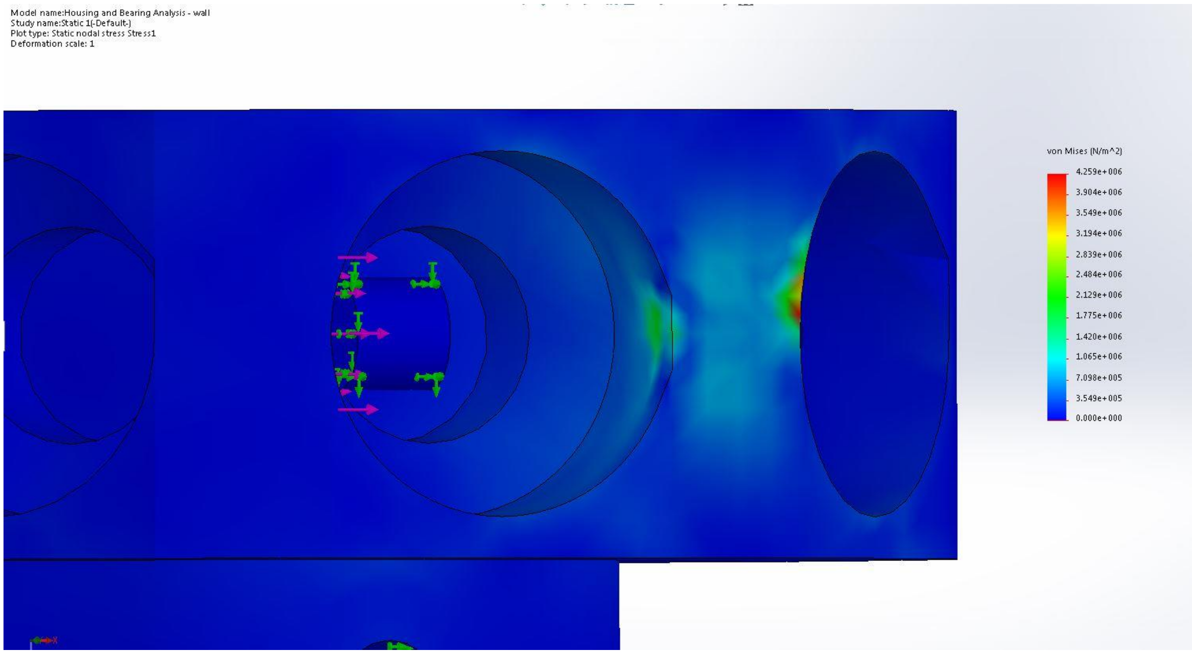

Figure 14: Stress on Housing Wall from Bearing – Center of Interface

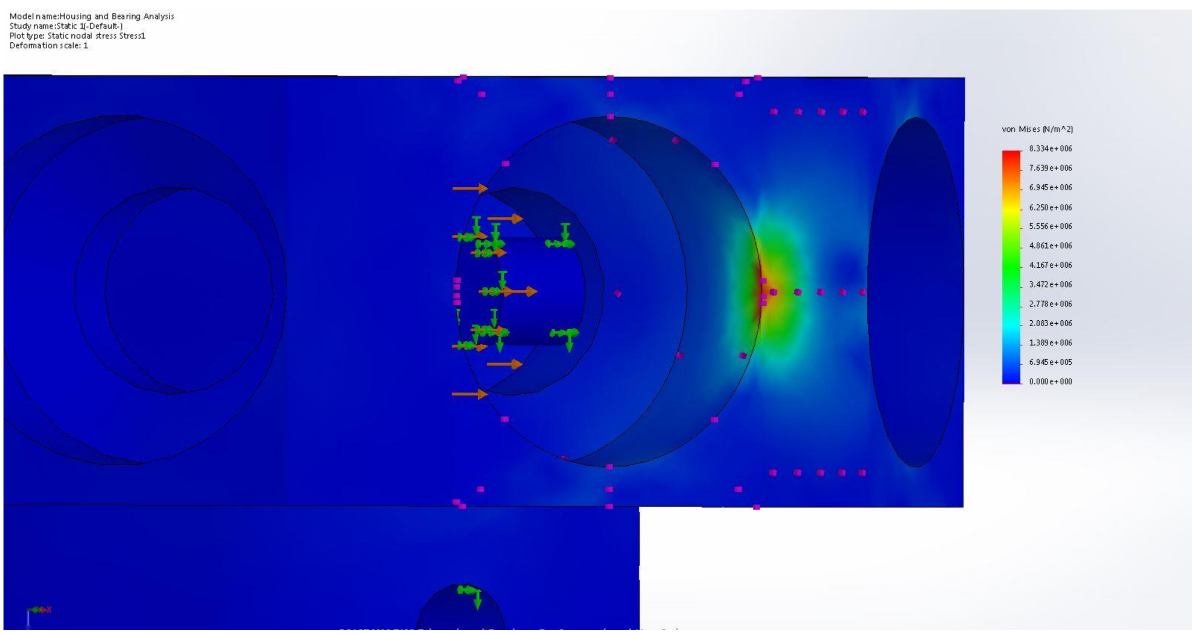

Figure 15: Stress on Housing Edge from Bearing - Normal Position

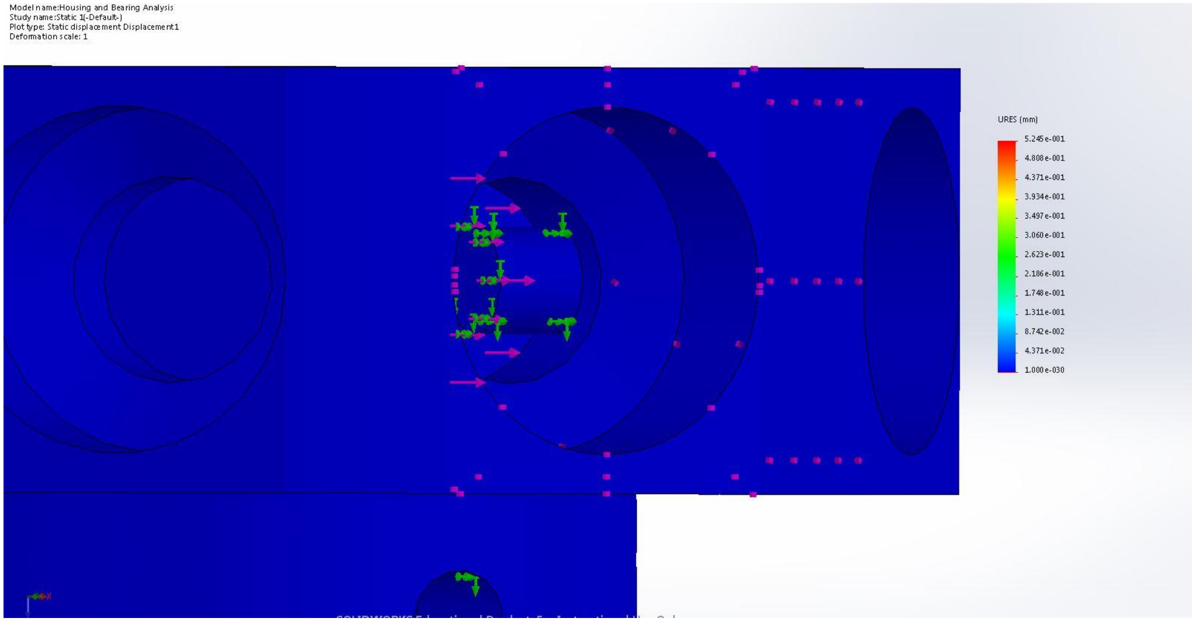

Figure 16: Deformation of Housing from Bearing

Insert

- In the first design, a 90-degree angle was between the large OD and the small OD. (Dashed line in *Figure 17)*. Once the simulation was run, a stress concentration was seen at this corner. Internal fillet was implemented to reduce the stress concentration without interfering with keeping the spring flush to the insert. This design change can be seen in *Figure* .
- Slight stress is seen on the small OD due to the designed press fit of the spring onto the insert.
- In all simulations, the spring was the fixed geometry. A spring compression simulation was attempted but not successful.

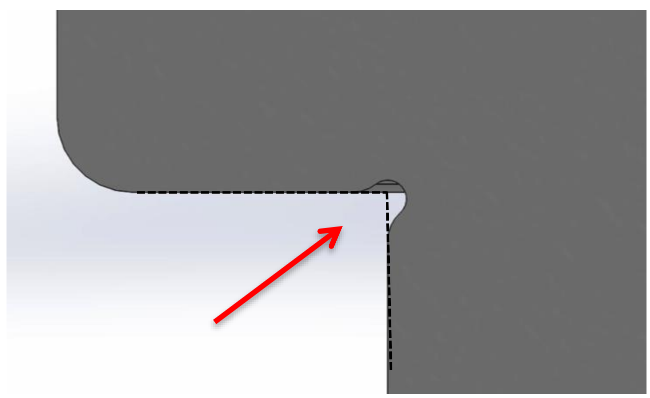

Figure 17: Insert Corner Design Edit

Component	Condition	Loading Description	Load Applied	Figure
Insert	Strain	fixed spring with force applied at bearing surface	8.12 lbs	*Figure 18*
	Stress			*Figure 19*
	Stress			*Figure 20*
	Stress			*Figure 21*

Table 5: Insert Simulation

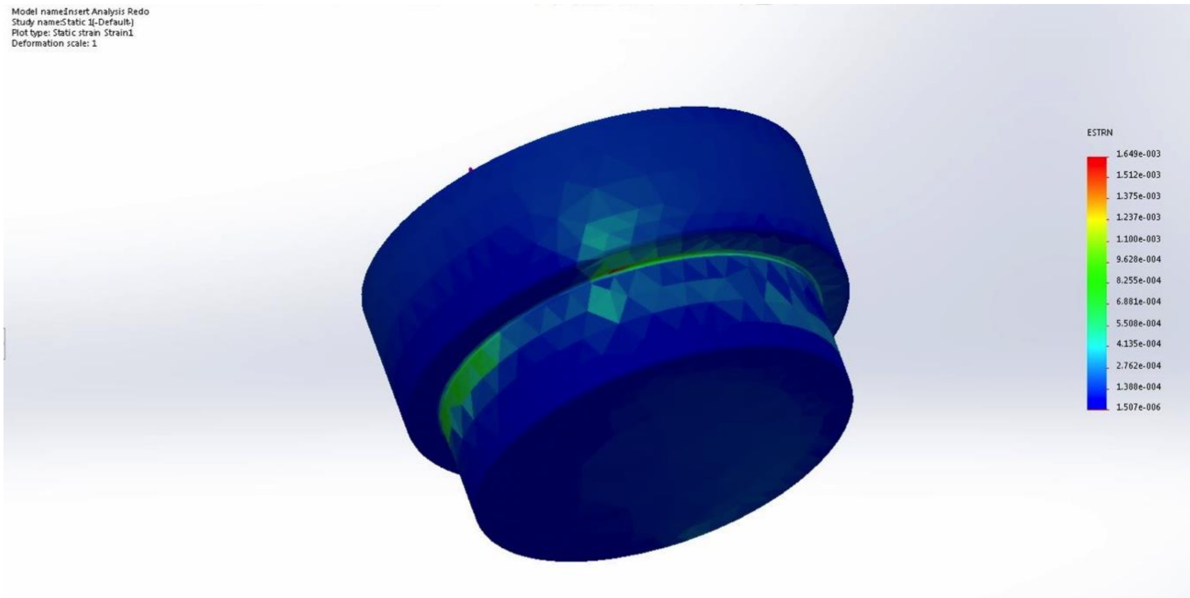

Figure 18: Insert Strain

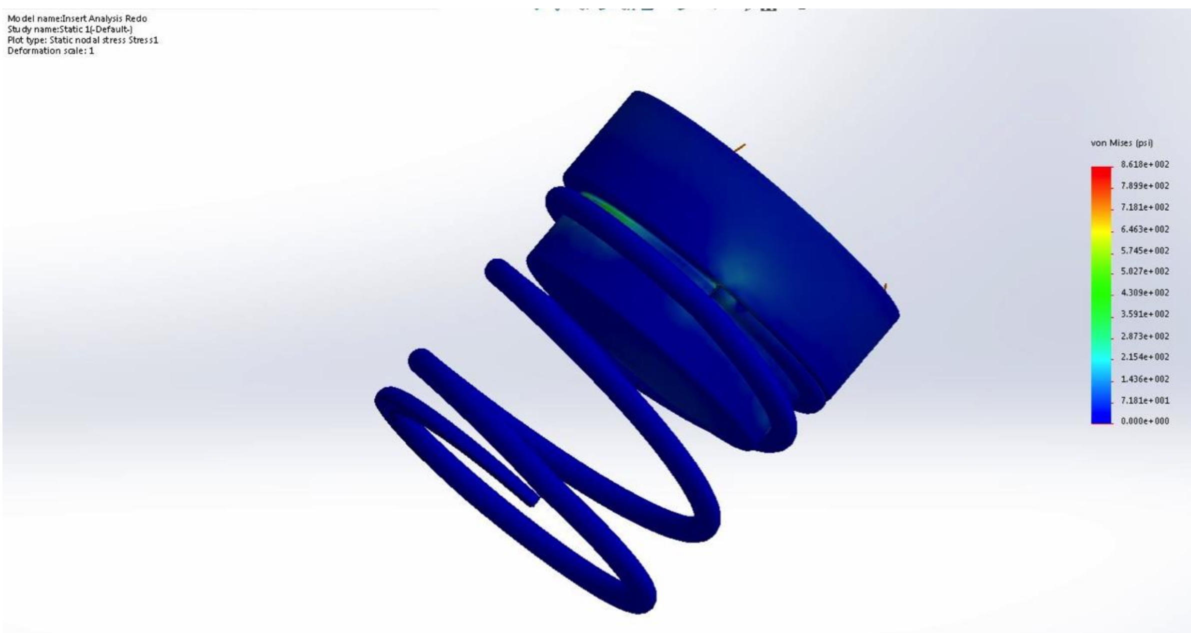

Figure 19: Insert Stress - With Spring Shown

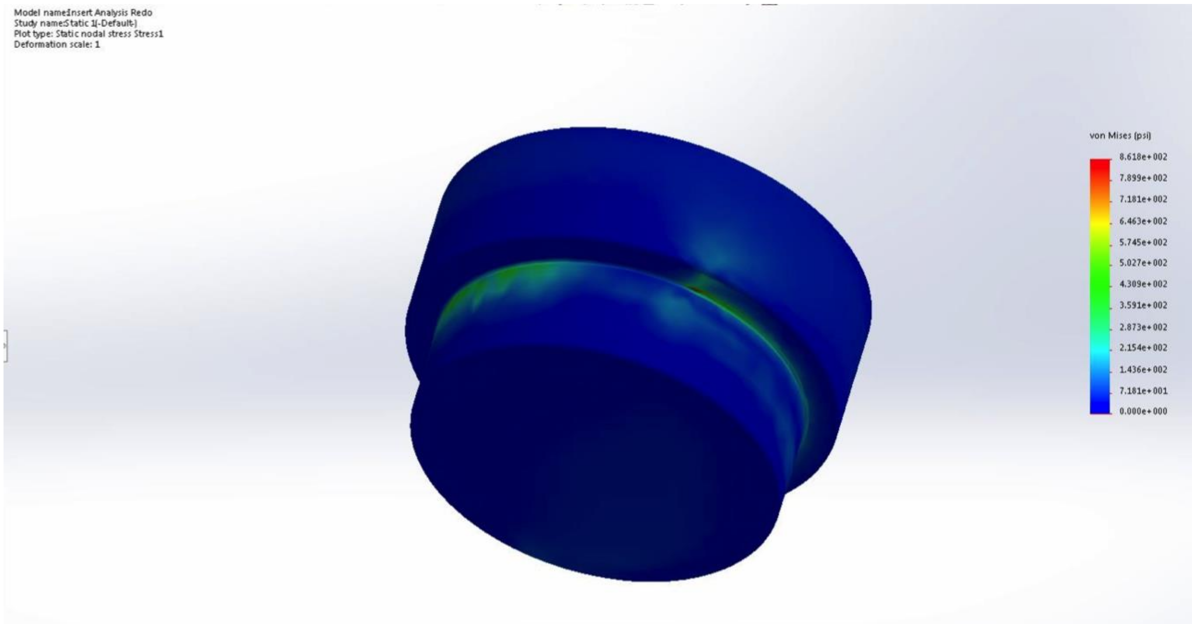

Figure 20: Insert Stress - Angled View

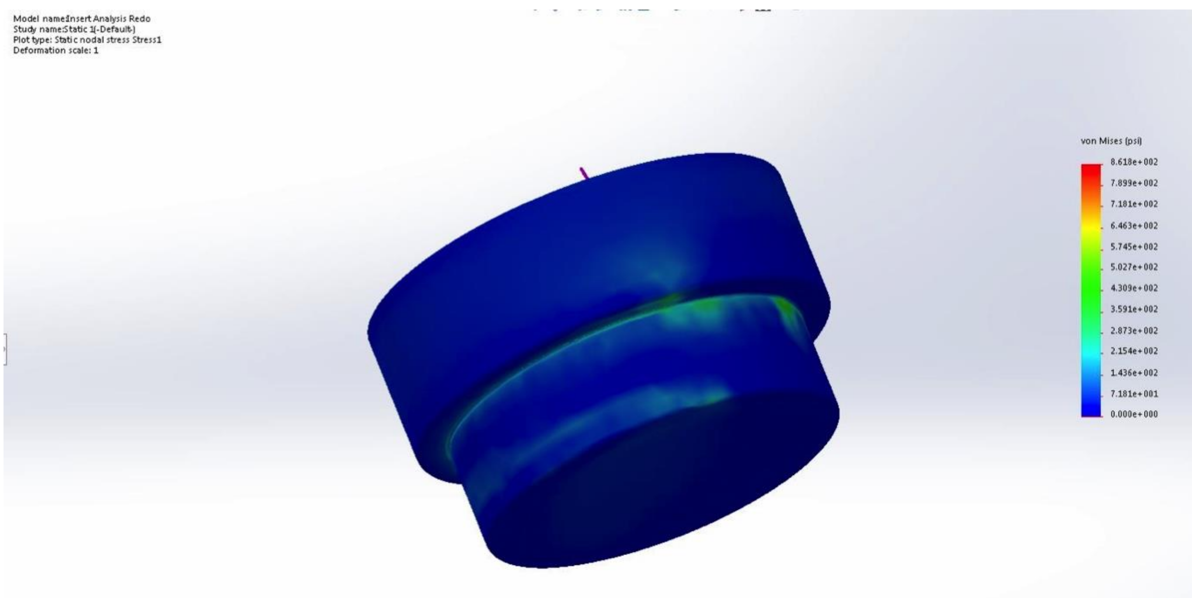

Figure 21: Insert Stress - Angled View 2

Interface

- This was the most critical simulation that was run because it is the known weak point in the design.
- Although the edges of each interface piece show stress while the bearing is in the normal position, no deformation occurred.
- The design was modified to reduce the volume of material used due to the quote for 3D printing being higher than planned, but each modified design failed in simulation. This also shows that the geometry of the interface piece is just the correct size because when material was removed, simulations under the same conditions were considerably worse.
- The slightly offset stresses shown during the simulation in the normal position is likely due to the imperfect meshing of the bearing.
- All interface pieces were fixed by means of the face that would be against the plate and the bolt holes. Simulations were run without fixing the bolt holes but little change was seen.

<table>
<tr><th>Component</th><th>Condition</th><th>Loading Description</th><th>Load Applied</th><th>Figure</th></tr>
<tr><td rowspan="7">Interface</td><td>Deformation</td><td rowspan="2">bearing force in normal position</td><td rowspan="2">5.00 lbs</td><td>Figure 22</td></tr>
<tr><td>Deformation</td><td>Figure 23</td></tr>
<tr><td>Deformation</td><td>bearing force on center of the interface</td><td>8.12 lbs</td><td>Figure 24</td></tr>
<tr><td>Stress</td><td>bearing force in normal position</td><td>5.00 lbs</td><td>Figure 25</td></tr>
<tr><td>Stress</td><td rowspan="2">bearing force in normal position with high force</td><td rowspan="2">10.0 lbs</td><td>Figure 26</td></tr>
<tr><td>Stress</td><td>Figure 27</td></tr>
<tr><td>Stress</td><td>bearing force onto center of interface piece</td><td>8.12 lbs</td><td>Figure 28</td></tr>
</table>

Table 6: Interface Simulation

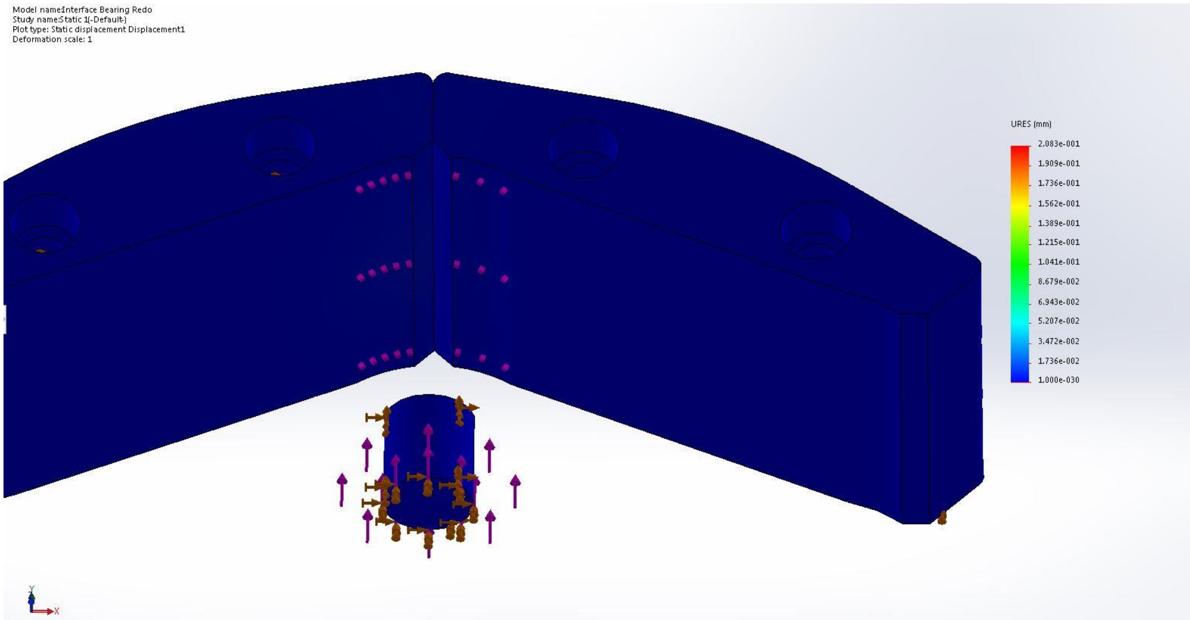

Figure 22: Interface Deformation – Normal Position - Bottom View

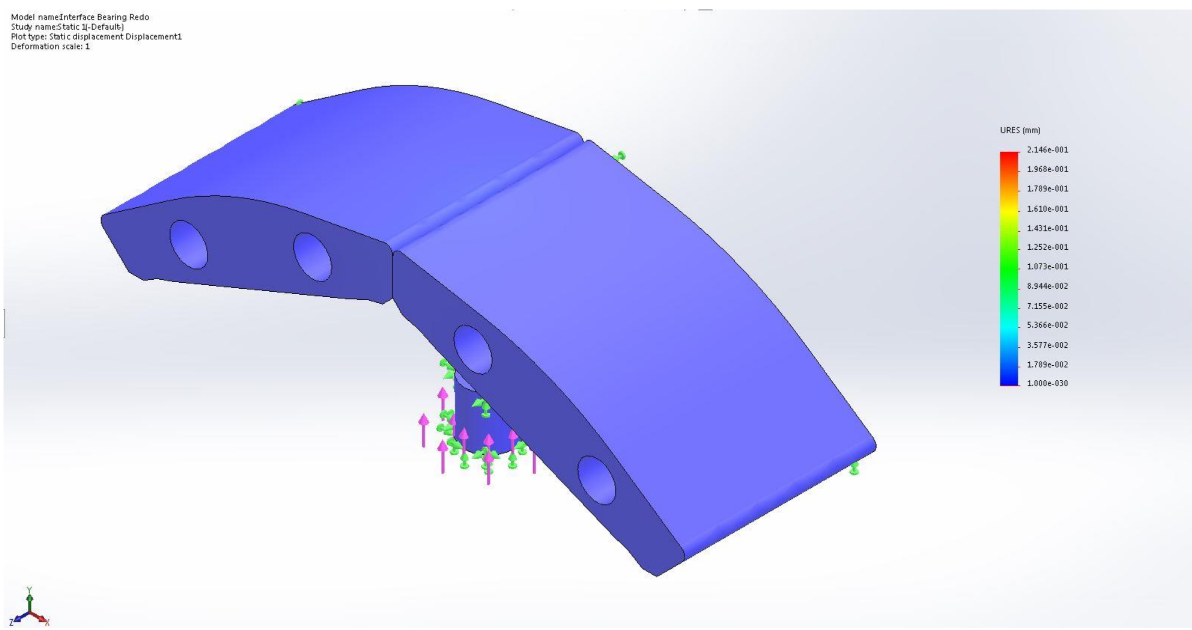

Figure 23: Interface Deformation - Normal Position - Top View

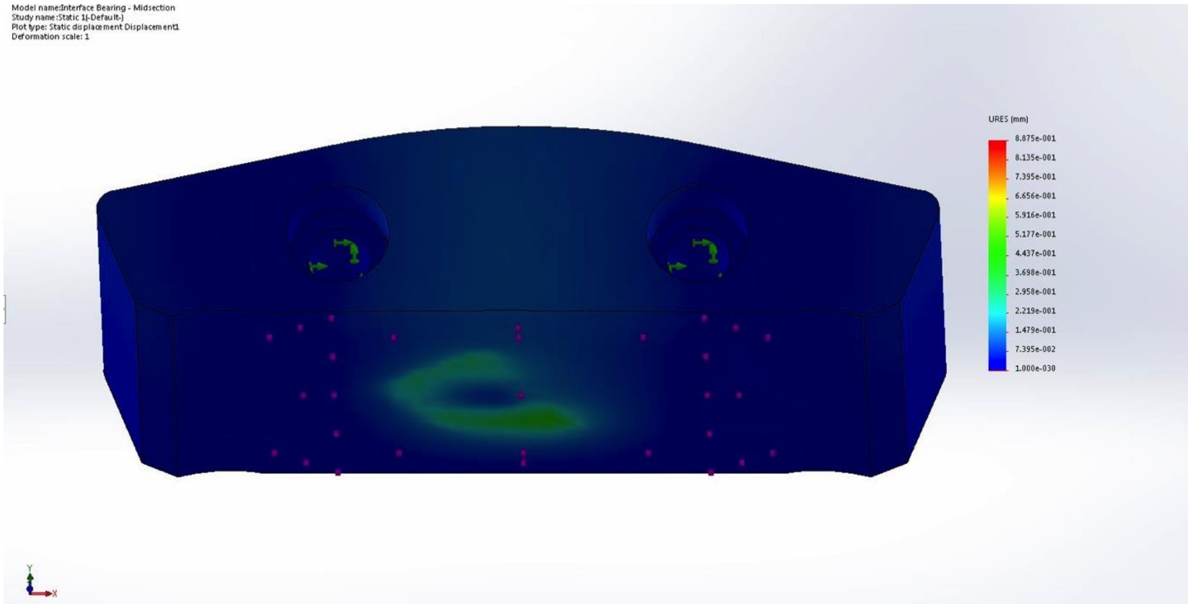

Figure 24: Interface Deformation - Center of Interface - Bottom View

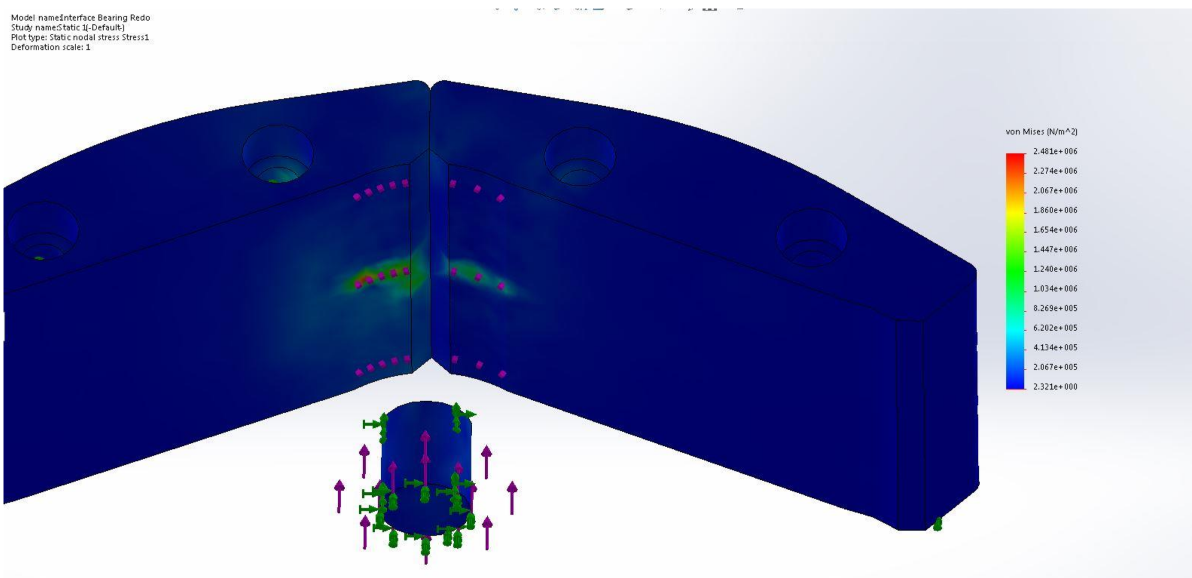

Figure 25: Interface Stress - Normal Position - Bottom View

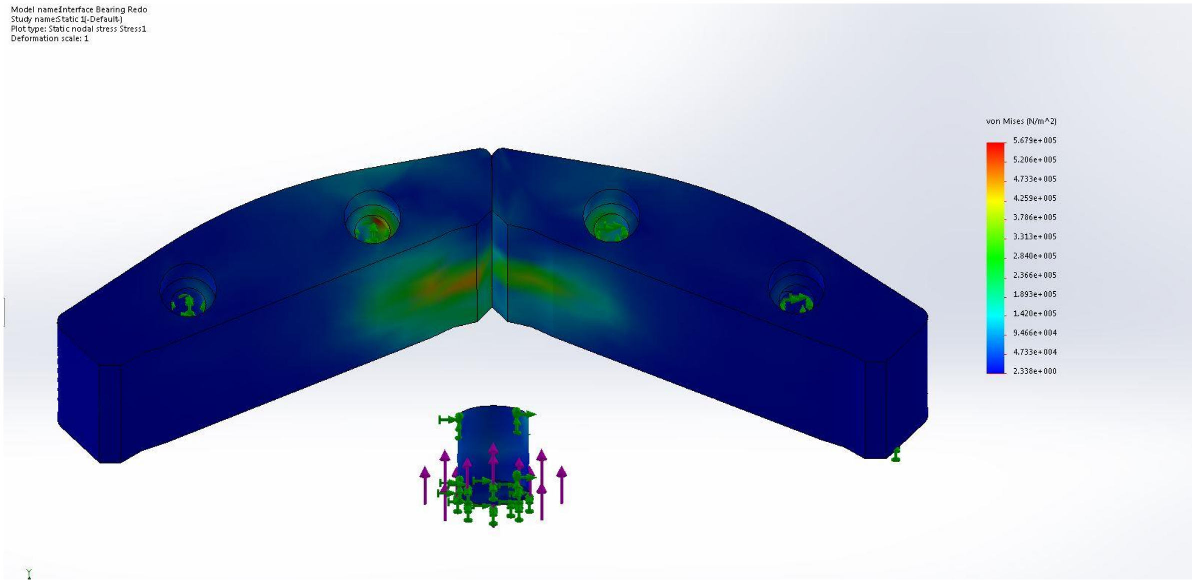

Figure 26: Interface Stress - Normal Position - Bottom View

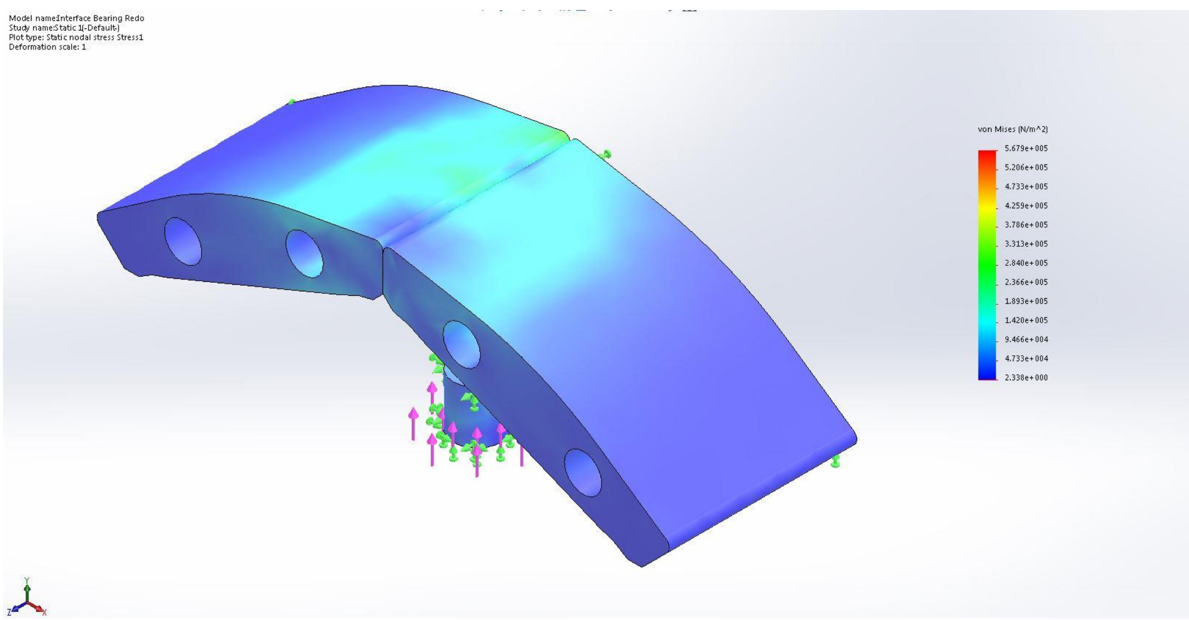

Figure 27: Interface Stress - Normal Position - Top View

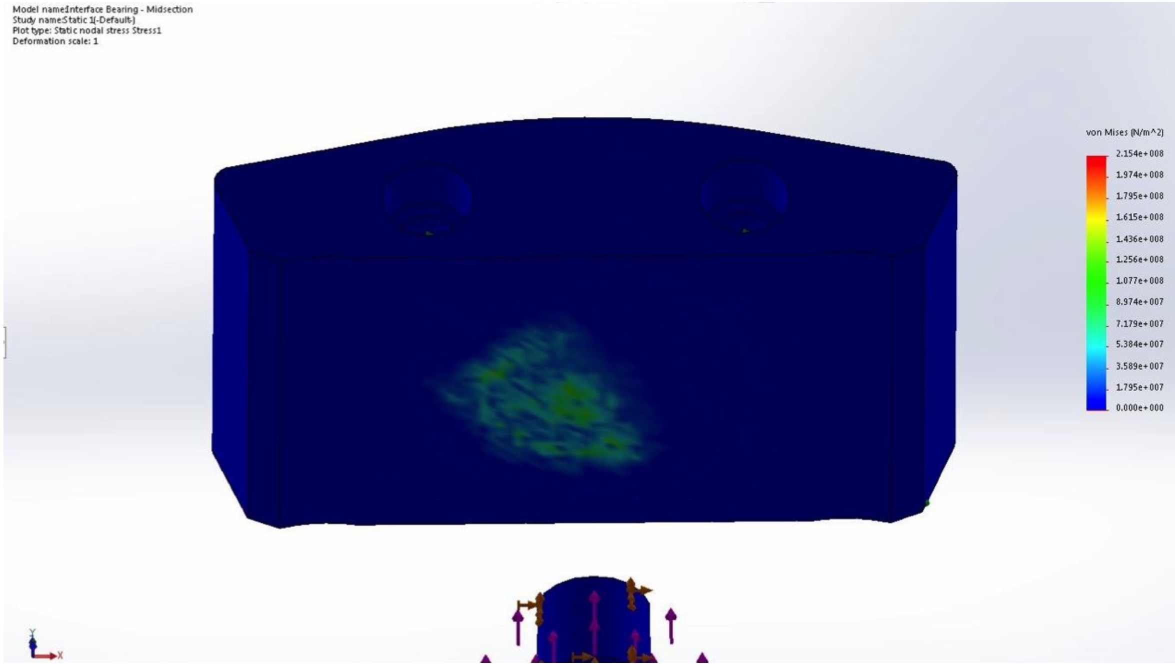

Figure 28: Interface Stress - Center of Interface - Bottom View

FABRICATION

BILL OF MATERIALS

Material	Description	Quantity
Shaft	D-Profile	2
Mounted Bearings	-	4
Set Screws	-	2
Raw Billet – Cylindrical	for Housing	1
Raw Billet – Plate	for plate	1
Raw Billet – Tube	for driven shaft housing	1
Raw Billet - Rectangle	to Mount Shaft bearings	1
Springs	-	8
Inserts	-	8
Bearing	-	8
Interface	-	8
Bolts	-	16
Nuts	-	16
Bushing	driven shaft in housing	1

Table 7: Bill of Materials for Fabrication

HOUSING

Planned

i. Raw - Round Billet
ii. Chuck into lathe and turn both circular surfaces.
iii. OD will be turned (Raw material will account for step out and chucking space in lathe.)
iv. Opposite side will be held, and remaining metal will be turned down to step out facing the Shaft.
v. ID/Shaft Hole will be drilled and then bored out on the lathe.
vi. Part will be put onto a mill and one of the eight faces will be machined off.
vii. Part will be dyed with purple machining ink and dimensions for the remaining flat cuts will be etched and checked multiple times to ensure conformance to dimensions.
viii. Once all eight flat spaces are machined, parts will be marked for hole location.
ix. The smaller hole (housing the spring) will be drilled out first in 2-3 drilling steps.
x. Next, the larger hole (housing the bearings) will be drilled out. The angle on the end of the drill is accounted for in the design. If measured angle is different, a redesign will take place.
xi. Repeat on all 8 faces.
xii. With all eight holes drilled, the part will be cleaned up and finished off.

Actual

1. Round billet of 4.0" diameter and 3.0" thick.
2. Cut down to ~2.2" using horizontal band saw
3. Both ends turned down on a lathe to 2.0" length.
4. Center hole for shaft drilled (*Figure 29*)
    a. Initial spot drill
    b. 5/16 hole drilled
    c. 15/16ths drilled
5. Smaller OD of housing turned down to proper size. (*Figure 30*)
    a. Manual turning attempt at leaving a chamfer between small OD and face on housing to reduce the stress concentration
6. Center shaft hole reamed to 1.0" (*Figure 31*)
7. Piece flipped around so the smaller OD is held in the chuck and larger OD of housing turned to correct diameter.
8. Part placed into mill.
9. Using 2-flute end mill, zeroed out on larger OD surface. (*Figure 33*)
10. Milled off -0.068" in z axis direction to create a flat surface. (*Figure 34)*
11. Removed part and flipped 180 degrees to utilize the newly machined flat surface.
12. Repeated milling step –
    a. The part was not zeroed out properly; realized after a few thousands were taken off. Calculations were performed to determine the rest of the material that needed to be removed.
13. Part flipped around so that the newly machined flat ends were between the vice jaws.
14. Milled a third flat surface.
15. Part flipped 180 degrees utilizing the third flat surface machined.

16. Using a third hand and a 45-45-90 triangle gage block, the part was set-up in vice jaws so that the next 4 faces could be started. (Faces that are 45 degrees off from the previous 4 faces.) (*Figure 35*)
17. Once complete, the new face was used to mill the other side.
18. For the remaining two, the 45-45-90 was used again and then the seventh flat face was used to mill the final face. (*Figure 36*)
19. Part taken to Prototrak milling machine.
20. Using an SPI LED indicator and a 1.0" hardened shaft, center axis of center shaft hole was set to zero. (Reset with every turn of the part. Part turned for new hole to be drilled.) (*Figure 20*)
21. Using the indicator, the rear vice jaws, the ones that would touch the large OD flat end was zeroed out. (only performed once)
22. Vertical zero set each time new tooling was changed.
23. With zeros set and the drills axis in the correct position, 0.5" hole drilled anywhere from 0.865" to 0.880" deep. (Final hole will be 0.900" deep, a buffer was left and will be finished off with the end mill.) (*Figure 38*)
24. Using a 1" drill bit (the slight oversizing drill bits tend to have was utilized, hole needed to be just over 1.0" in diameter.) angled surface of the drill was pressed into the part until the vertical corner of the bit reached the surface. Z axis zeroed out and then drilled down 0.45". (*Figure 39*)
    a. After the second time through this, it was measured and calculated that from the tip of the bit on the flat surface to the desired depth, the drill needed to go down 0.78" so that the machine only needed to be zeroed at the start and the zeroing of the z axis at the corner of the bit each time was not needed.
25. A sharpened 5/8ths end mill (sharpened = smaller diameter) was found that met the diameter I needed for my hole. This end mill was used to finish off the smaller and deeper hole for the spring.
    a. Chips were blown out every 0.20" due to the fast buold up of chips from the end mill.
26. Steps 19 & 21-24 were repeated for the next 7 holes. (*Figure 40*)
27. Spot drill on smaller OD for set screw. 5/16th hole drilled. 3/8-16 thread tapped. (*Figure 41)*

Figure 29: Housing - Step 4

Figure 30: Housing - Step 5

Figure 31: Housing - Step 6

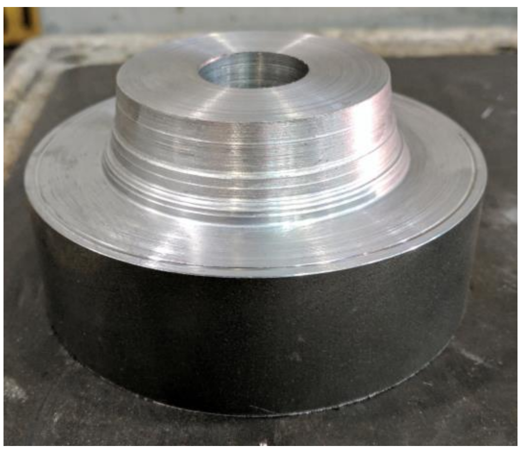

Figure 32

Figure 33: Housing - Step 9

Figure 34: Housing - Step 10

Figure 35: Housing - Step 16

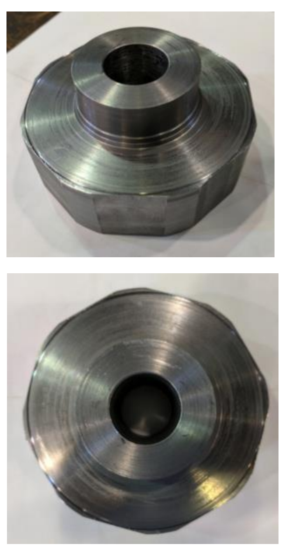

Figure 36: Housing - Step 18

Figure 37: Housing - Step 20

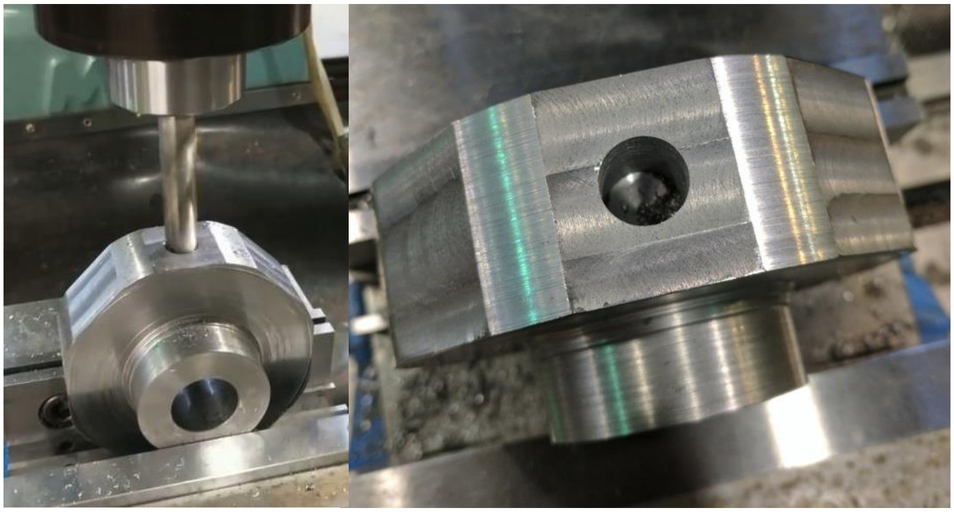

Figure 38: Housing - Step 23

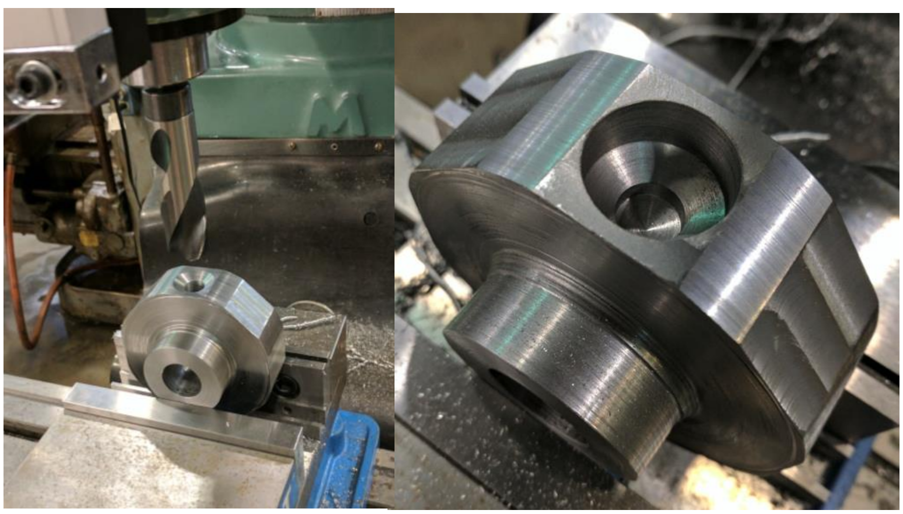

Figure 39: Housing - Step 24

Figure 40: Housing - Step 27

Figure 41: Housing – Step 27

Mid-Fabrication Design Change (*Figure 42*)

28. On larger OD flat face of the housing, the center hole for the shaft was widened with 1 3/32" drill 0.5" deep. (*Figure 43*)
29. The 0.5" section was bored out for press fit of bushing. Aimed for a few thousandths under 1 1/8" but due to a large amount of play in the tool advancement mechanism on the lathe, the boring was not carried out with precision. However, the end result was successful. (*Figure 44* & *Figure 45*)

30. After some testing, 0.0550" was taken out of holes to account for extra 0.1110" on the OD of the housing. This will be covered in the *Testing* section.

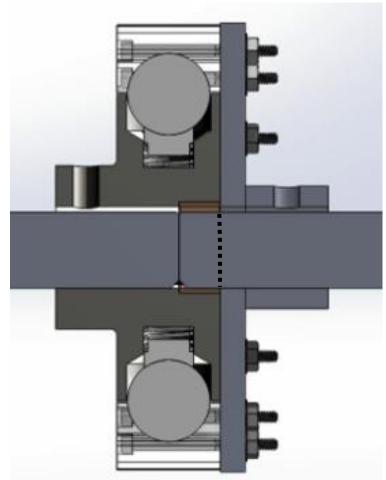

Figure 42: Housing - Shaft & Coupler Cross-Section

INSERT

Planned

- This component will be outsourced to a rapid prototyping service.

Actual

- 3D Printed by ProtoLabs

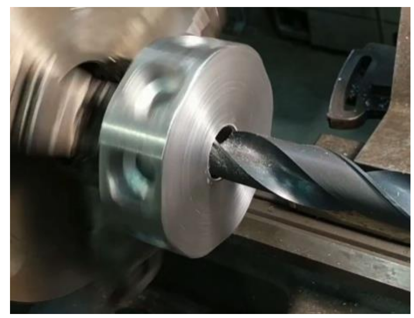

Figure 43: Housing - Step 28

INTERFACE

Planned

Two options are available for this manufacturing process. The final choice will be decided after stress analysis, performance analysis, budgeting, and level of difficulty.

- Machined
  - Raw - Long stock metal
  - Outer dimensions will be machined down using a mill
  - Pieces will then be cut into appropriate sizes
  - Holes for the bolts will be drilled into each of the pieces.
- 3D Printed
  - 3D Model will be sent to rapid prototyping service
  - Cost will be added to budget

Actual

- 3D Printed by ProtoLabs

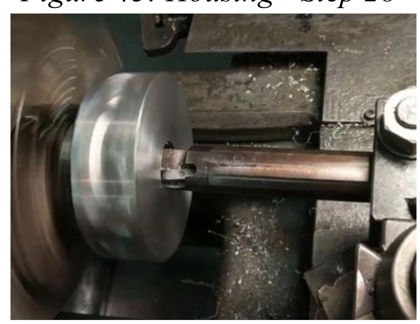

Figure 44: Housing - Step 29

Figure 45: Housing - Step 29

PLATE WITH SHAFT HOUSING

<u>Planned</u>

i. Raw – Plate will be purchased as close to dimensions of actual part as possible
ii. Part will be cleaned up and machined to meet dimensions
iii. Center shaft hole will be drilled and bored out on lathe.
iv. Interface pieces will be placed onto plate and numbered. Hole locations will be transferred from the pieces to the plate.
v. Holes will be drilled into plate
vi. Once complete, Shaft Housing will be welded onto plate.
    a. MIG or outsource to friction welder

<u>Actual</u>

1. Billet with 6" diameter and 0.5" length
2. On lathe, center hole drilled, same process as housing.
3. Using expanding mandrel, turned part to an external diameter of desired 5.5"
4. Began facing plate on the lathe. Facing off was taking too long and the part was moved to a mill. The part was fixed by means of using a clamp on the center hole. Once milling began, fixture broke loose and took a small chunk out of the part.

5. Using a steel billet found in the shop (approx. 1.5' long and dia. Of 5.5") cut rusty face off and using the horizontal band saw,
    - Cut two plates (just in case another incident occurred) from billet measuring just over 0.3"
6. Center drilled 1" hole on the lathe, the same process as previously stated. (*Figure 46*)
7. Using Prototrack mill, part set up and computer programmed to move to the hole positions desired on the x & y axis. Program run with spot drill, then moved to the desired drill size of 9/64ths and drilled holes. (*Figure 47*& *Figure 48*)

8. Metal Tube was cut to length.
9. Tube was reamed (as the shafts were not fitting into the ID) and ends were faced and finished on a lathe.
10. Taken to a mill, centered, and spot drilled, 5/16ths hole and 3/8-16 tapped.
11. Shaft Housing was MIG welded to plate with 1" shaft as place holder and a bolt in the set screw hole to prevent spatter from entering into the threads. (*Figure 50*)
12. Debris and spatter removed with air powered rotary tool. (*Figure 51*)
13. D-profile shaft became difficult to put onto the plate because of a very slight mis-orientation between the tube and the plate after welding.
14. Plate and tube taken to the lathe and re-reamed and shaved down for an easy fit onto the D-profile shaft.

Figure 46: Plate - Step 6

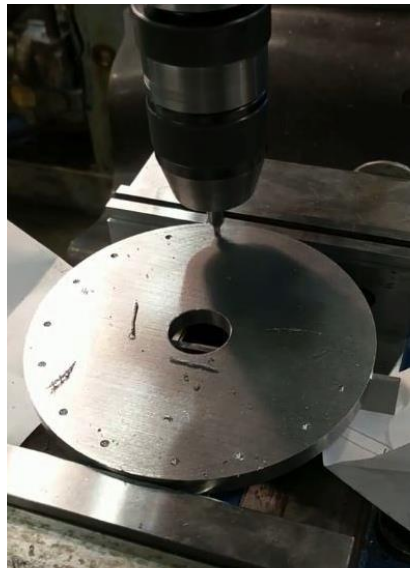

Figure 47: Plate - Step 7

Figure 48: Plate - Step 7

Figure 49

Figure 50: Plate - Step 11

Figure 51: Plate - Step 12

Assembly

Planned

The following is a prediction of the assembly process. Actual assembly may be different and the actual process will be documented.

i. The Springs and the Spring inserts will be pressed together (press fit)
ii. With the Plate laying flat on a table, one interface piece will be applied to the
iii. The Housing will then be placed on top of it.
iv. The shaft will be inserted through the A-Housing.
v. One of the spring assemblies will be inserted into the Housing and will be held in by the interface piece already attached to the plate.
vi. The adjacent interface piece will be attached to the plate and the next spring/bearing assembly will be inserted.
vii. This will continue until 2 interface pieces remain, where the last assembly will be pushed in, and then using a clamp or partner, the last interfaces will be applied.
The Shaft will be inserted and casing applied.

Figure 52: Assembly - Step 1

Figure 53: Assembly - Step 5

Actual

1. All interface pieces were bolted to the plate, hand turning the nuts onto the bolts.
2. Once all nuts were on the bolts, the nuts were hand tightened with a wrench.
3. Housing without spring-bearing assemblies placed onto plate.
4. One by one, the spring with the press fit insert was placed into the hole and then a bearing was placed into it.
5. It was partially pressed in and continuing onto the next hole, the housing was tilted so a bearing could be fit into the intended hole but the surrounding bearings were still partially held in. It may take some finesse but it is a painless process.
6. Setscrews were partially threaded into their holes.
7. After the coupler is together, the mounted D-profile shafts were inserted into the coupler, making sure the correct shaft was traveling to the correct depth.
8. Setscrews were tightened down.

Figure 54: Assembly - Step 8

TESTING PLAN

i. A variable speed motor will be attached to the driving shaft end to represent the torque that would be applied during use.
ii. Once the motor has been properly fashioned to the shaft, different speeds and accelerations will be tested. Simple observation will determine whether the assembly is functioning as planned.
    a. If the design performs as predicted, the following experiments will be recorded.
        i. Torque limits of coupler.
        ii. Acceleration properties.
        iii. Performance of assembly when driven shaft is fixed.
    b. If the experiment *fails* to perform as predicted, the following steps will be taken.
        i. The failure mode(s) will be determined.
        ii. With the failure modes and performance recorded, re-calculation of the failed component, material, etc. will be completed.
        iii. The assembly will be adjusted or fixed, depending on the failure mode and the recalculation.

TESTING & RESULTS

INITIAL TESTING

1. Blocks were built to support to coupler and shafts. Mounted bearings were placed and fixed onto aluminum blocks.
2. Coupler springs and bearings were attempted to be compressed by turning the shafts by hand but the force of the springs was too strong.
3. To increase the torque applied, large adjustable wrenches were put onto to the shafts, utilizing the D shape and twisted against each other.

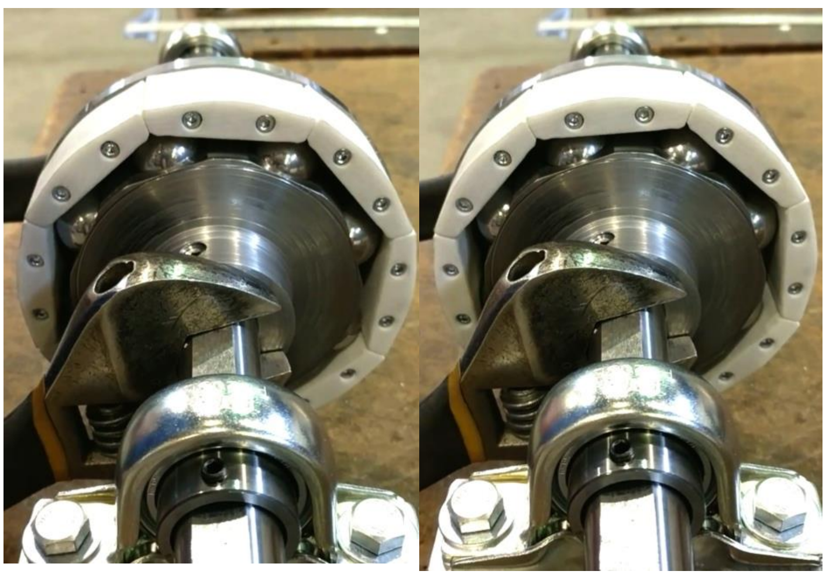

Figure 55: Initial Testing – Step 4
left is normal position and right shows position with bearings bottoming out

4. Ball bearings were compressed but would stop before reaching lowest/half point of the external interface. The bearings only traveled about one third of the way to the center of the interface before stopping. Multiple attempts tried but something was wrong because the torque being applied was much higher than intended and the interface inserts were beginning to bend as more force was applied.
5. Next, the coupler was removed from the shafts. The interfaces were loosened so that they were still fixed to the plate by bolt and nut. Because they were loose, the spring-bearing assembly would push them out to their furthest point. The thought was that the interface was too tight around the housing, causing the bearings to compress more than calculated, forcing them to bottom out.
6. Interfaces tightened down at this state.
7. The wrench test was attempted again. Minimal improvement, still no complete slippage, the ball bearings still seemed to be bottoming out.

MODIFICATION CALCULATION

8. Measurements were taken and bearings were placed into housing without the spring.
9. Housing OD (pre-flat face mill – Housing Step #6) was 0.1110” too large. This caused the holes (which were dimensionally correct) to be 0.050” too shallow. With a ball travel of only 0.15” this made a considerable impact on the performance of the bearing movement.

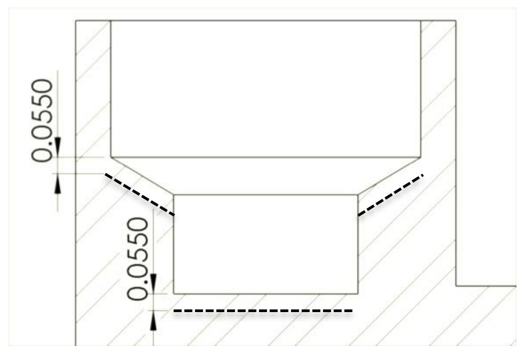

Figure 56: Testing - Step 11

10. This fault was likely caused by measuring the large OD with calipers due to the proper measurement devices not being readily available.
11. Re-measurement and calculations were performed to determine how much material to to remove from the holes. All taken into account, 0.0550” +/- 0.0010” was removed from the holes. This included the 0.60625” hole for the spring and the 1.000” hole for the bearing. Both holes were simply made deeper; no changes in the geometry of the ball travel were affected.

POST-MODIFICATION TESTING

12. After the 0.0550” was removed from two opposing holes in the housing, two spring-bearing assemblies were placed into the newly machined holes and the coupler was hand twisted to assure the calculations were correct. The reduced torque (only two balls) made it possible to turn by hand.
13. This hand test passed and it slipped within the assembly, showing the calculations were correct.
14. All 8 holes then had 0.0550” removed on the Prototrak. This removal process was a simplified version of the initial drilling of the holes, but only the 1.0” drill bit and the end mill were needed.
15. Coupler was re-assembled onto shafts and mounted bearings.
16. Wrenches were taken to the shafts (same as the initial testing) and the assembly slipped as intended.

17. The driving shaft was taken to a lathe and turned down to an OD just under 0.5” and a length of ~1.0” so it would be able to fit into the chuck of the power drill.
18. The power drill was then applied to housing shaft to verify it would spin under the load of a small motor.
    a. Drill Used – DeWalt Cordless Drill (DC970K-2)
        i. 18 V, 380 W, 1500 RPM Max Speed, ½” Chuck.
        ii. Source: (DeWalt)
19. Small vibrations while spinning at a high speed caused the driving and driven assemblies to gently start to slip apart from each other.
20. A clamp was applied to both ends of the mounted shaft bearings to provide additional support so the coupler did not slip out and fall apart during high speed testing.
21. Using an optical tachometer, the top-spinning speed that was achieved by the coupler was 1170 rpm. All rotary motion was transferred to the driven end flawlessly.
22. Next, the motor was set at a slow speed, around 100 rpm, and the speed was rapidly increased to it’s max speed of 1170 rpm. The coupler followed through and transferred all motion. No slippage occurred. Different speeds and accelerations were applied but no slippage could be achieved.

23. To test the amount of torque needed to make the coupler slip, the driving shaft was taken to a mill.
24. 4 sides of the shaft were milled off to create a square feature that was just ast the small OD used for the power drill application.
25. 7/8ths deep socket on a torque wrench was applied to the end of the driving shaft as the driven shaft was held still.
26. The torque measurement required to fully slip the coupler resulted on just under 20 inlb.

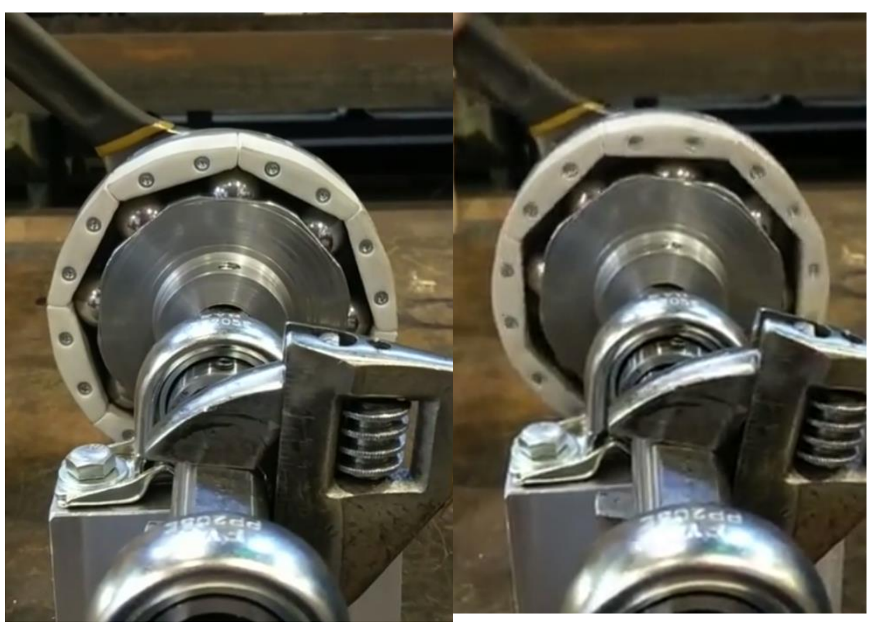

Figure 57: Testing - Step 16
left is normal position and right shows position with bearings slipping, as intended, after modification.

FINAL TESTING

27. After some thought, it was thought that since the driven end had no load and was not driving anything, a load or resistance needed to be applied onto the driven shaft to determine if slippage would occur.
28. A small cylindrical piece (about 2” OD, 1” ID, 1” thick) was attached to the driven shaft to act as a makeshift brake rotor.
29. This makeshift rotor would be used to apply a load onto the shaft by means of pliers or Vice Grips along with some form of friction paper.
30. Adjustable pliers and 1” strips of P40 sandpaper were applied around the rotor but no force was applied.
31. Using the power drill, the coupler assembly was spun at high speed and then the pliers with sand paper was applied to the rotor.
    - Because of time constraints and limited access to materials, the makeshift brake assembly was not the best or most reliable device to use to test the functionality for the coupler. About two out of every three times, the sandpaper would be carried out of the grip of the pliers due to the spinning of the rotor.
32. When the sandpaper did stay between the rotor and the pliers, a gripping force was applied and the coupler slipped at a high speed as intended.
33. Different forms of holding the sandpaper in place were attempted but holding the strips between the jaws of the pliers proved to be the best.
34. This test was run many times with different lengths of time and forces applied to the rotor. Based on these tests, the coupler performed better than expected but no load values or torque values could be recorded.

Figure 58: Final Testing Set-Up

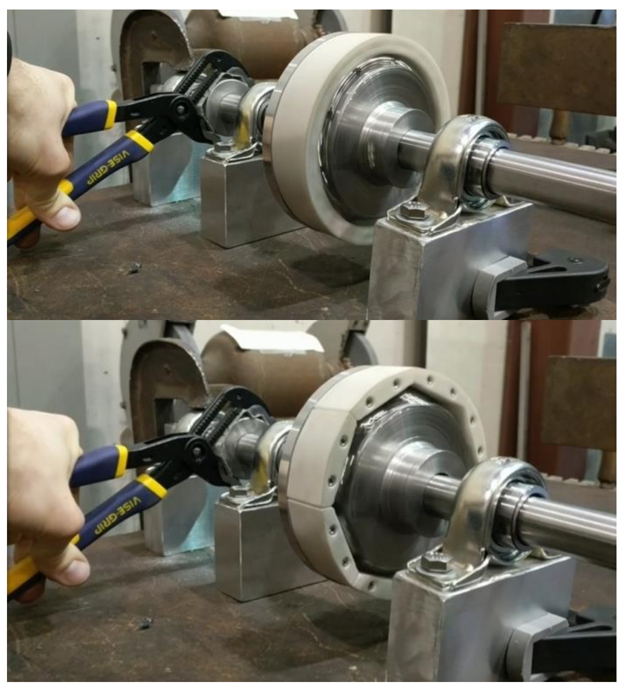

Figure 59: Testing - Step 32
top is coupler rotating with no load on driven end and bottom shows driven end stopped while coupler slips

RESULTS DISCUSSION

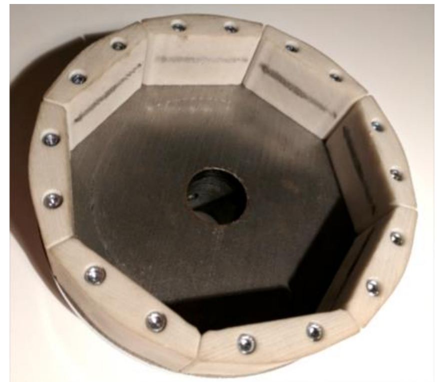

Figure 60: Interface after Testing

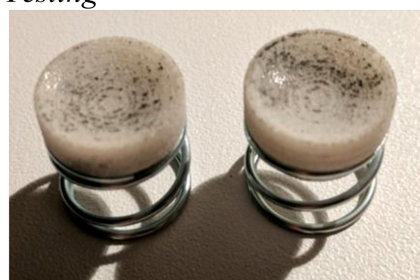

Figure 61: Inserts after Testing

During fabrication, no trouble was predicted based on the progress of the build, but after the initial testing failed, the proper steps were taken to solve the issue and the testing plan for a failed component was followed. After the modification and implementation of the makeshift brake rotor, many successful tests were run, proving the concept. One design issue that was seen during testing was when the bearings would hold position in the middle of the interface. This happened numerous times, especially when the braking load was applied suddenly. The bearings would slip in the coupler during applied load on the driven end and when that load was released, the bearings were stuck in the fully compressed state at the middle of the interface. After a few seconds the bearings would release and go back into the normal position. This is a minor design issue brought to light by testing. It did not have a great affect on the transmission of the coupler, but it is not desired.

After multiple successes of the coupler functioning with a variable load on the driven end (constant speed from the power drill and intermittent loads on the brake rotor) testing the coupler to see if it would perform properly under constant load on the driven end and variable speed was attempted but no results were gathered. The issue during this testing was the lack of control and inability of the operator to apply a constant loading on the driven end. No results were gathered. Due to time constraints, refinement of this testing was not attempted.

After all testing was performed, the coupler was disassembled to observe the wear of the components. In the interface pieces, a line was made that appears to be wear but no deformation was measured. The line (*Figure 60*) is residue from the bearings and work environment where the coupler was fabricated and tested. The same results are on the inserts. No deformation was found, only residue. (*Figure 61*)

If a second testing plan is formed, it will take into account measurement devices to obtain numerical results on the torques, speed, and loads applied onto the coupler. Proper testing equipment and rigging will also be accounted for.

CONCLUSION

AREAS OF IMPROVEMENT

- The biggest issue with the project was not having a specific application for the coupler. Different applications were researched but nothing significant was found where the coupler could be utilized. This is due in part to the fact that the specific dynamic properties of the coupler were unknown until testing was performed.
- Time management had much room for improvement, which would have led to better performance in design, fabrication, and testing.
- A detailed documentation of the design process was intended but not performed because of poor time management and being too caught up in getting the design finalized perfectly.
- Looking back, some areas that should have gotten more attention are calculating theoretical values and better, more organized testing methods. Obtaining theoretical values would have provided numerical values to compare the actual results against. This would provide a measure of success of the coupler. Another consequence of poor time management was the testing being left to the last minute. Because of this, little to no time was left for proper means of testing, especially testing that involved applying a load on the driven end.
- A design that would be implemented in the next step is a means of fixing the two coupler ends together. Occasionally, if not handled correctly, the coupler housing and the interface would be shaken or pulled apart, which left the operator with bearings and springs everywhere. A casing to enclose the moving parts and hold the two assemblies together was considered but not implemented due to cost and unnecessary complexity. After handling and testing the product, a simple means of holding the coupler together could be implemented.

FINAL THOUGHTS

Considering the entire process of the project, from ideation to final testing, the result is better than expected. The spring selection and amount of torque required to slip the coupler is exactly where it was intended to be. Even if there are some design issues that need to be worked out, the concept functioned and proved its ability to slip while under the proper conditions. Despite the project being stressful, it was enjoyable and the work paid off in the end.

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Zinc-Plated Steel Music Wire Compression Spring. (n.d.). Retrieved from McMaster Carr: https://www.mcmaster.com/#9657k279/=17a81sw

APPENDIX A (SURVEY RESULTS)

Q1: What is your current job title?

Product Development Engineer

Q2: What kind of couplers are currently used in your industry or workplace?

Lovejoy Coupling

Roller-Chain Coupling

Over-Torque Protection Coupling

Torque Limiting Coupling (Mechanical or Electrical)

Flexible Shaft Coupling

Q3: If you have any torque-controlling couplers in use, could you specify which kind?

Please specify.	I primarily use and specify couplings for lab use. I perform design work for an industrial pump manufacturer. Any torque limiting coupling I use is a custom fabricated design that utalizes sacrificial pins.

Q4: If you have couplers currently in use, how satisfied are you with their performance?

Satisfied

Q5: Why are you or why aren't you satisfied with your current coupler? (If you do not have any in use, put N/A)

I am usually satisfied with lovejoy type couplings for Lab use. However, we do get a lot of warranty returns for broken shafts. These are always a result of shaft misalignment. I can see real benefit to any design that incorporates aligning features or is potentially self-aligning.

Q6: Please select the following characteristics of the proposed coupler from not important to very important

Afforability	Very Important
Durability	(no label)
Ease of Installation	Neutral
Amount of Maintenance Required	Neutral
Complexity of Design	Not Important
Accounts for Shaft Misalignment	Very Important
Protection from Outside Contaminants	Not Important
Product Lifetime	Very Important

Q7: On a scale of 1-5, how likely would this concept be implemented into your industry or workplace? (1 - Not likely, 5 - Very likely)

56

Q8: What is the highest price you would be willing to pay for a coupler assembly?

50

Q1: **What is your current job title?**

Noise & Vibration Performance Engineer

Q2: **What kind of couplers are currently used in your industry or workplace?**

Over-Torque Protection Coupling

Torque Limiting Coupling (Mechanical or Electrical)

Vibration Damping

Flexible Shaft Coupling

Q3: **If you have any torque-controlling couplers in use, could you specify which kind?**

Please specify.	Lots of vib rubber dampers

Q4: **If you have couplers currently in use, how satisfied are you with their performance?**

Satisfied

Q5: **Why are you or why aren't you satisfied with your current coupler? (If you do not have any in use, put N/A)**

N/A

Q6: **Please select the following characteristics of the proposed coupler from not important to very important**

Afforability	Very Important
Durability	Very Important
Ease of Installation	Neutral
Amount of Maintenance Required	Neutral
Complexity of Design	(no label)
Accounts for Shaft Misalignment	(no label)
Protection from Outside Contaminants	Very Important
Product Lifetime	Very Important

Q7: **On a scale of 1-5, how likely would this concept be implemented into your industry or workplace? (1 - Not likely, 5 - Very likely)**

5

Q8: **What is the highest price you would be willing to pay for a coupler assembly?**

2

Q1: **What is your current job title?**

Process Engineer - (Peter K at Bilstein)

Q2: **What kind of couplers are currently used in your industry or workplace?**

Lovejoy Coupling

Q3: **If you have any torque-controlling couplers in use, could you specify which kind?**

I do not have any in use.

Q4: **If you have couplers currently in use, how satisfied are you with their performance?**

I do not have any in use

Q5: **Why are you or why aren't you satisfied with your current coupler? (If you do not have any in use, put N/A)**

Satisfied, no complaints

Q6: **Please select the following characteristics of the proposed coupler from not important to very important**

Afforability	(no label)
Durability	Very Important
Ease of Installation	(no label)
Amount of Maintenance Required	Neutral
Complexity of Design	Not Important
Accounts for Shaft Misalignment	(no label)
Protection from Outside Contaminants	Very Important
Product Lifetime	(no label)

Q7: **On a scale of 1-5, how likely would this concept be implemented into your industry or workplace? (1 - Not likely, 5 - Very likely)**

1

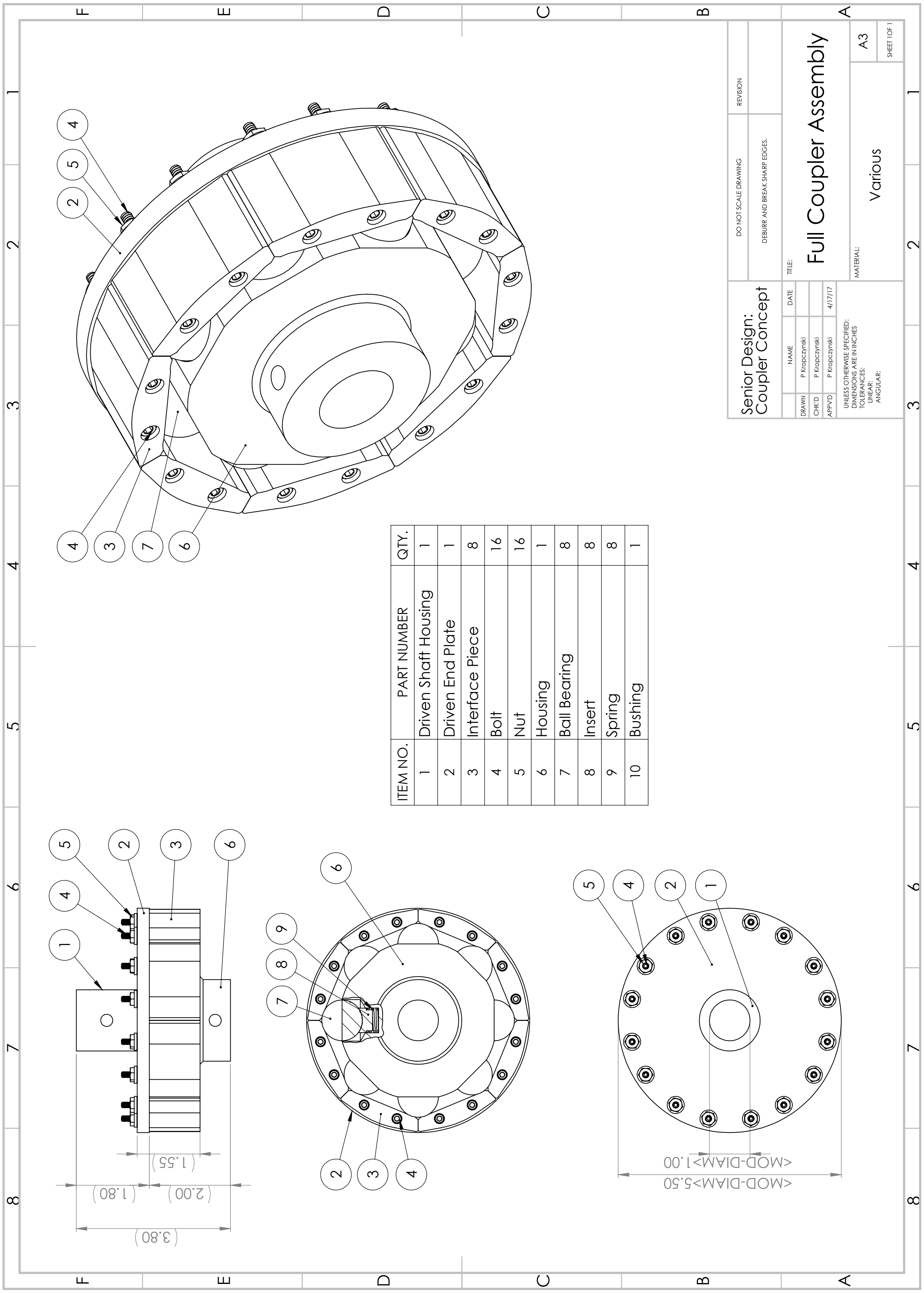
Q8: **What is the highest price you would be willing to pay for a coupler assembly?**

45

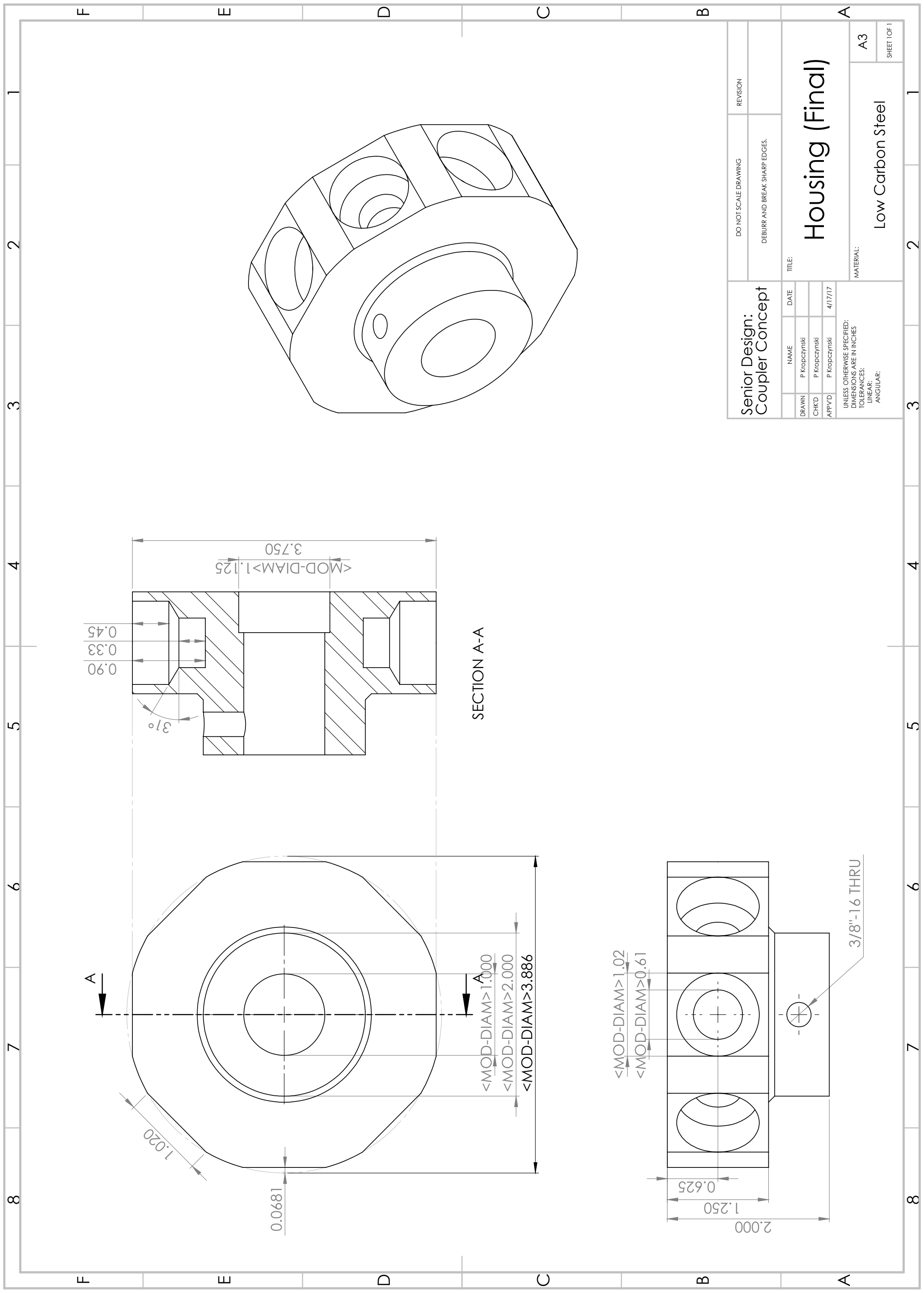
APPENDIX B (QFD)

	Customer Requirements	Importance wt.	Torque (Nm)	Angular Acceleration (rad/sec^2)	Strength of Material (N/m^2)	Spring Constant (N/m)	
			Engineering Requirements				
			1	2	3	4	5
1	Easy to install	0.15			1		
2	Allows small misalignments	0.10	1			1	
3	Durable	0.25	3	3	9	9	
4	Clean during operation	0.10			1		
5	Little to no maintenance	0.15			9		
6	Slips at specified value	0.25	9	9	3	9	
7							
8							
9							
10							
	Total Importance	1.00					
	Engineering requirement importance		3.1	3	4.6	4.6	

APPENDIX C (DRAWINGS)



ITEM NO.	PART NUMBER	QTY.
1	Driven Shaft Housing	1
2	Driven End Plate	1
3	Interface Piece	8
4	Bolt	16
5	Nut	16
6	Housing	1
7	Ball Bearing	8
8	Insert	8
9	Spring	8
10	Bushing	1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DO NOT SCALE DRAWING
DEBURR AND BREAK SHARP EDGES.
REVISION
TITLE:
Housing (Final)
MATERIAL:
Low Carbon Steel
A3
SHEET 1 OF 1
NAME
DATE
DRAWN
P Kropczynski
CHK'D
P Kropczynski
APPV'D
P Kropczynski
4/17/17
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
LINEAR:
ANGULAR:
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3.750
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0.33
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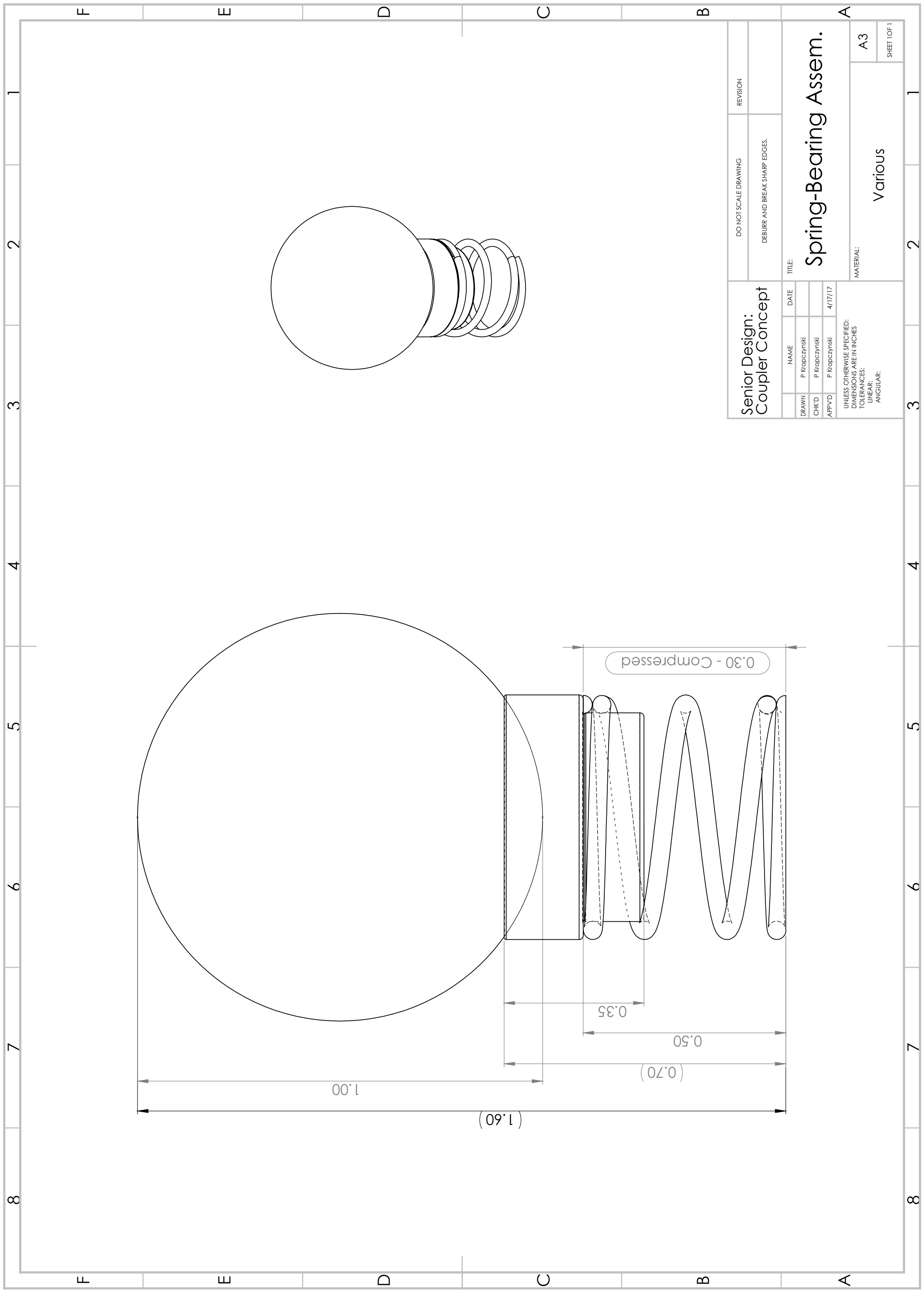
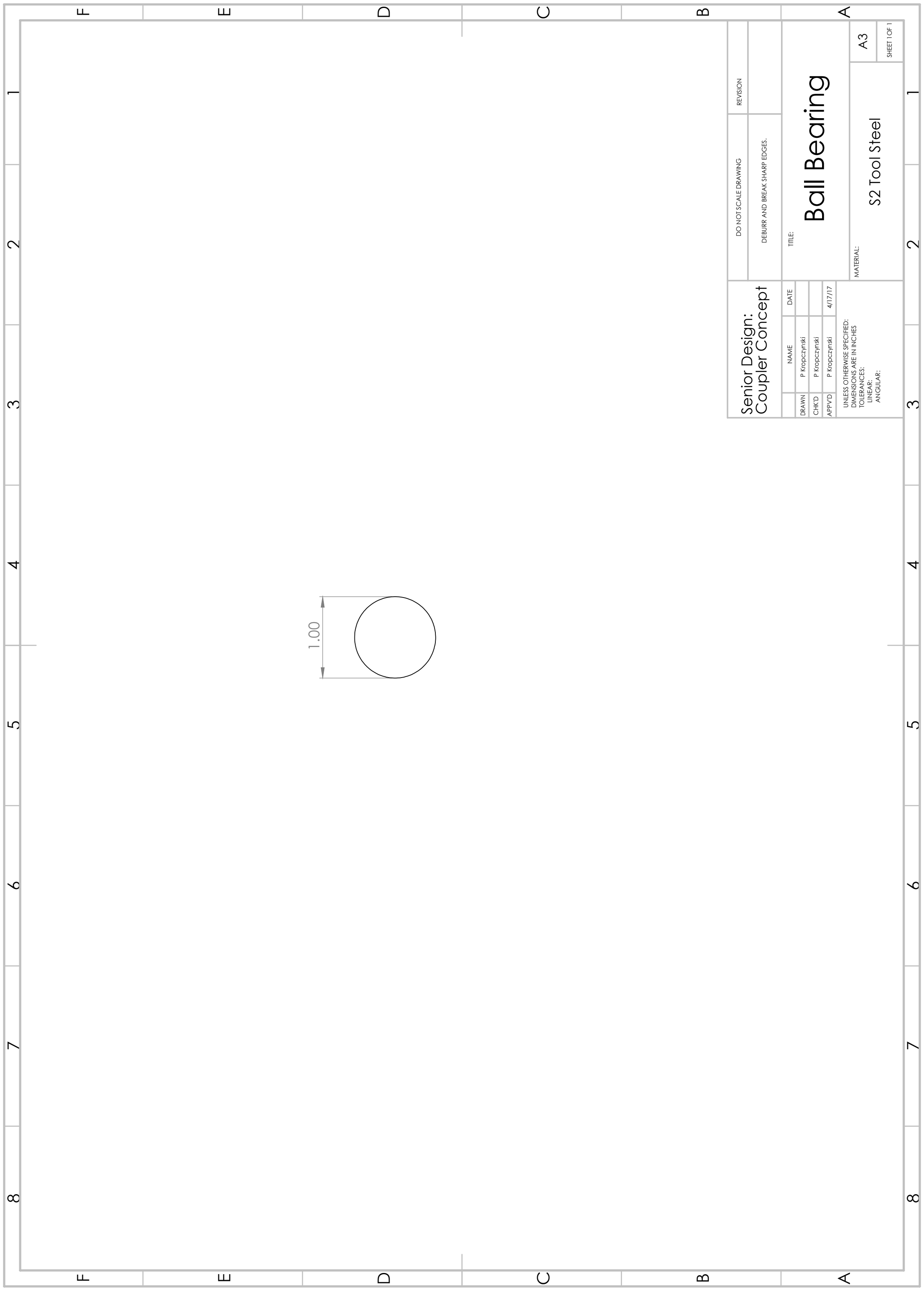
2.000



Senior Design:
Coupler Concept
DO NOT SCALE DRAWING
DEBURR AND BREAK SHARP EDGES.
REVISION
NAME
DATE
DRAWN
P Kropczynski
CHK'D
P Kropczynski
APPV'D
P Kropczynski
4/17/17
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
LINEAR:
ANGULAR:
TITLE:
Spring-Bearing Assem.
MATERIAL:
Various
A3
SHEET 1 OF 1
0.30 - Compressed
0.35
0.50
(0.70)
1.00
(1.60)


1.00
Senior Design:
Coupler Concept
NAME
DATE
DRAWN P Kropczynski
CHK'D P Kropczynski
APPV'D P Kropczynski 4/17/17
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
LINEAR:
ANGULAR:
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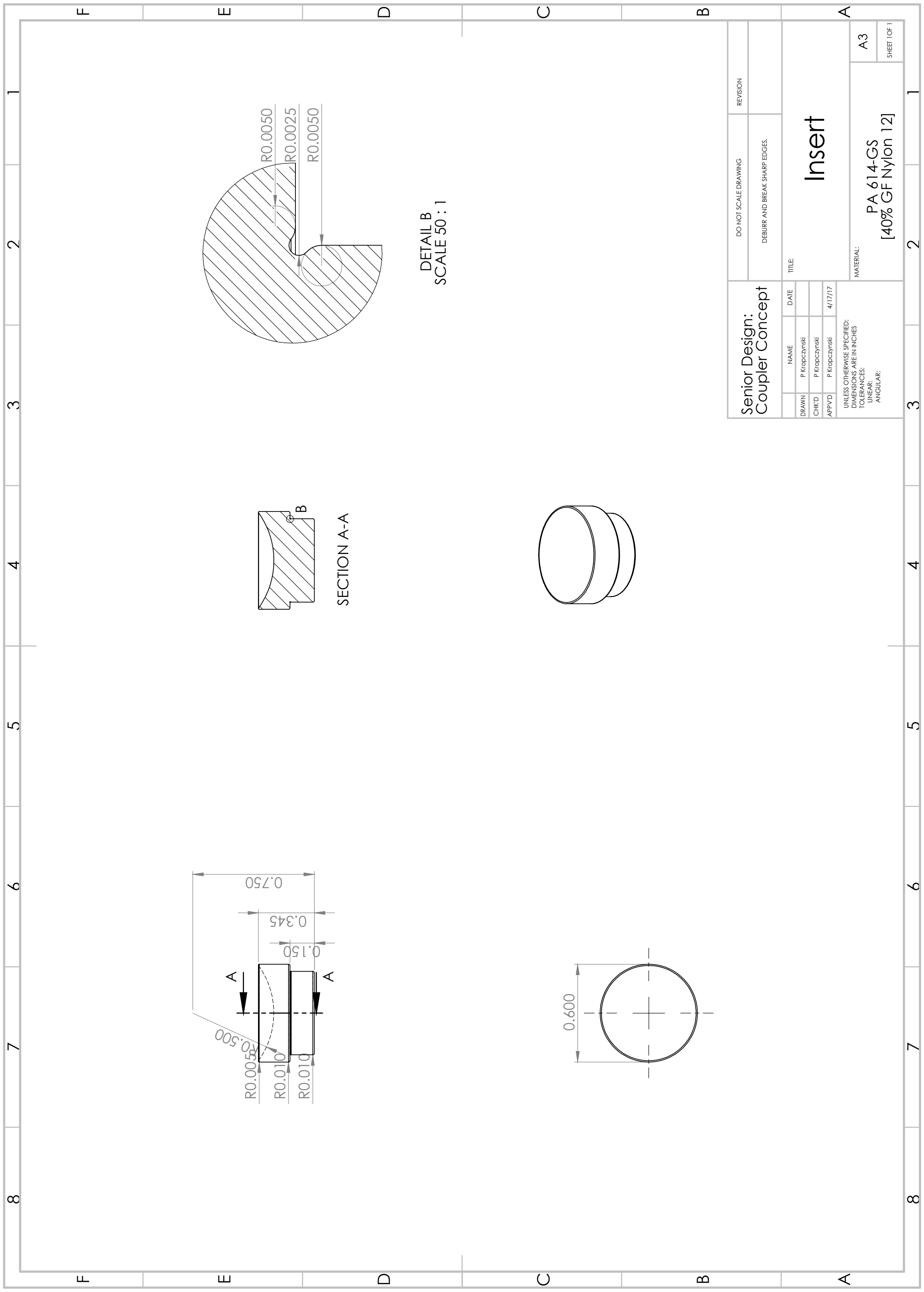
REVISION
DEBURR AND BREAK SHARP EDGES.
TITLE:
Ball Bearing
MATERIAL:
S2 Tool Steel
A3
SHEET 1 OF 1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Insert
MATERIAL:
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DEBURR AND BREAK SHARP EDGES.
REVISION
TITLE:
NAME
DATE
DRAWN
P Kropczynski
CHK'D
P Kropczynski
APPV'D
P Kropczynski
4/17/17
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
LINEAR:
ANGULAR:
A3
SHEET 1 OF 1
DETAIL B
SCALE 50 : 1
R0.0050
R0.0025
R0.0050
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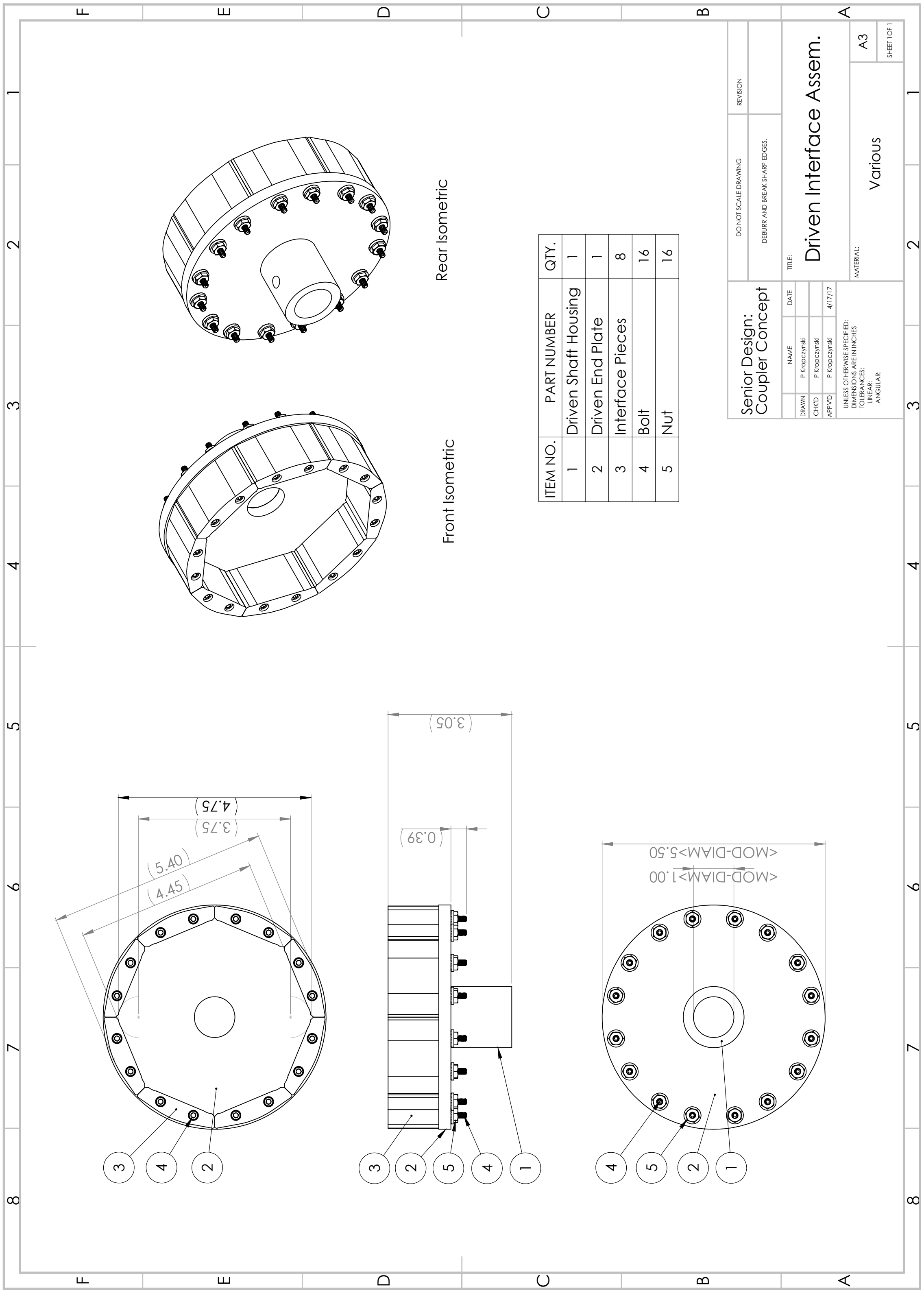
R0.500
R0.005
R0.010
R0.010
0.600



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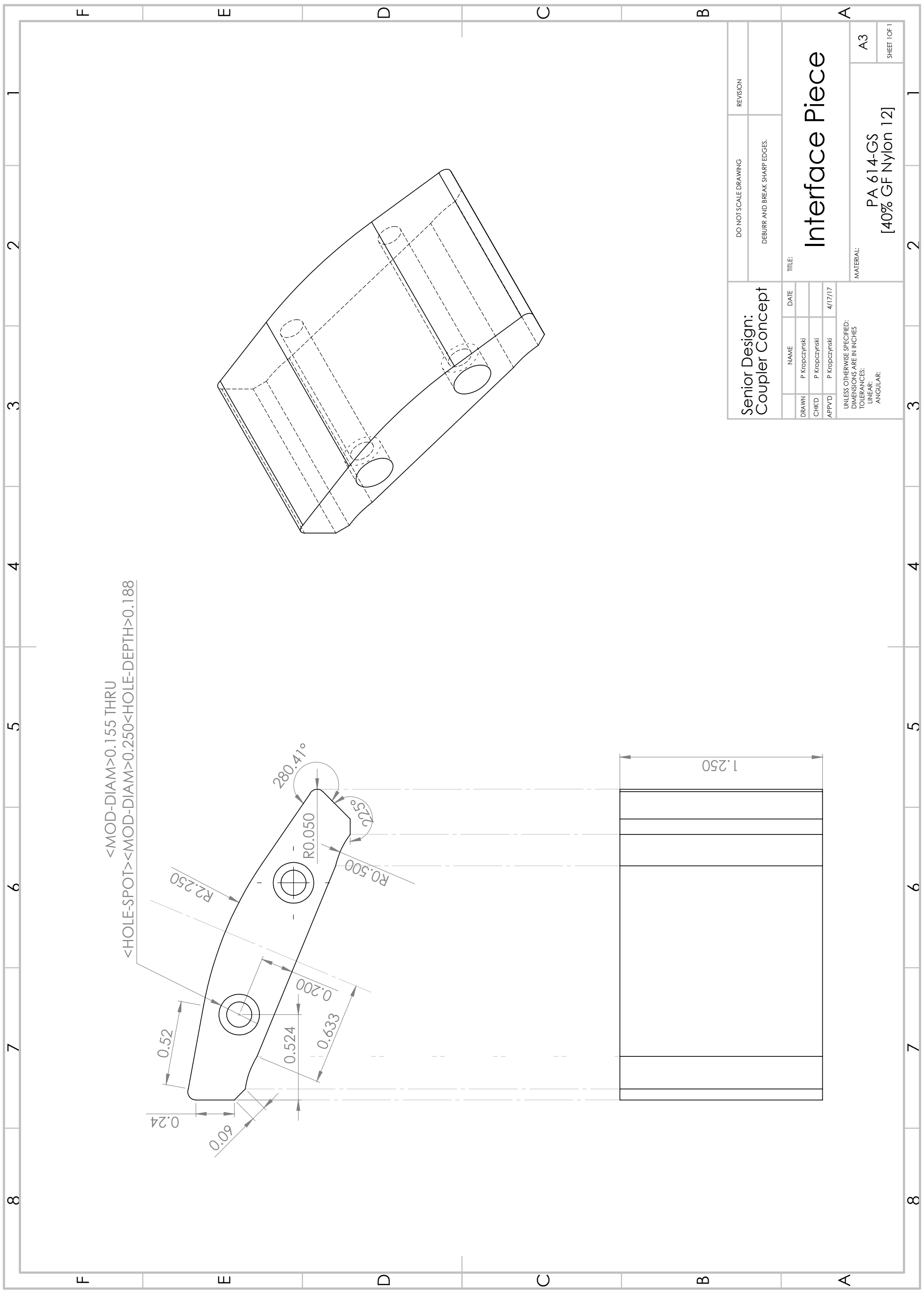
2	Driven End Plate	1
3	Interface Pieces	8
4	Bolt	16
5	Nut	16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REVISION
DEBURR AND BREAK SHARP EDGES.
TITLE:
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NAME
DATE
DRAWN
P Kropczynski
CHK'D
P Kropczynski
APPV'D
P Kropczynski
4/17/17
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
LINEAR:
ANGULAR:
MATERIAL:
PA 614-GS
[40% GF Nylon 12]
A3
SHEET 1 OF 1
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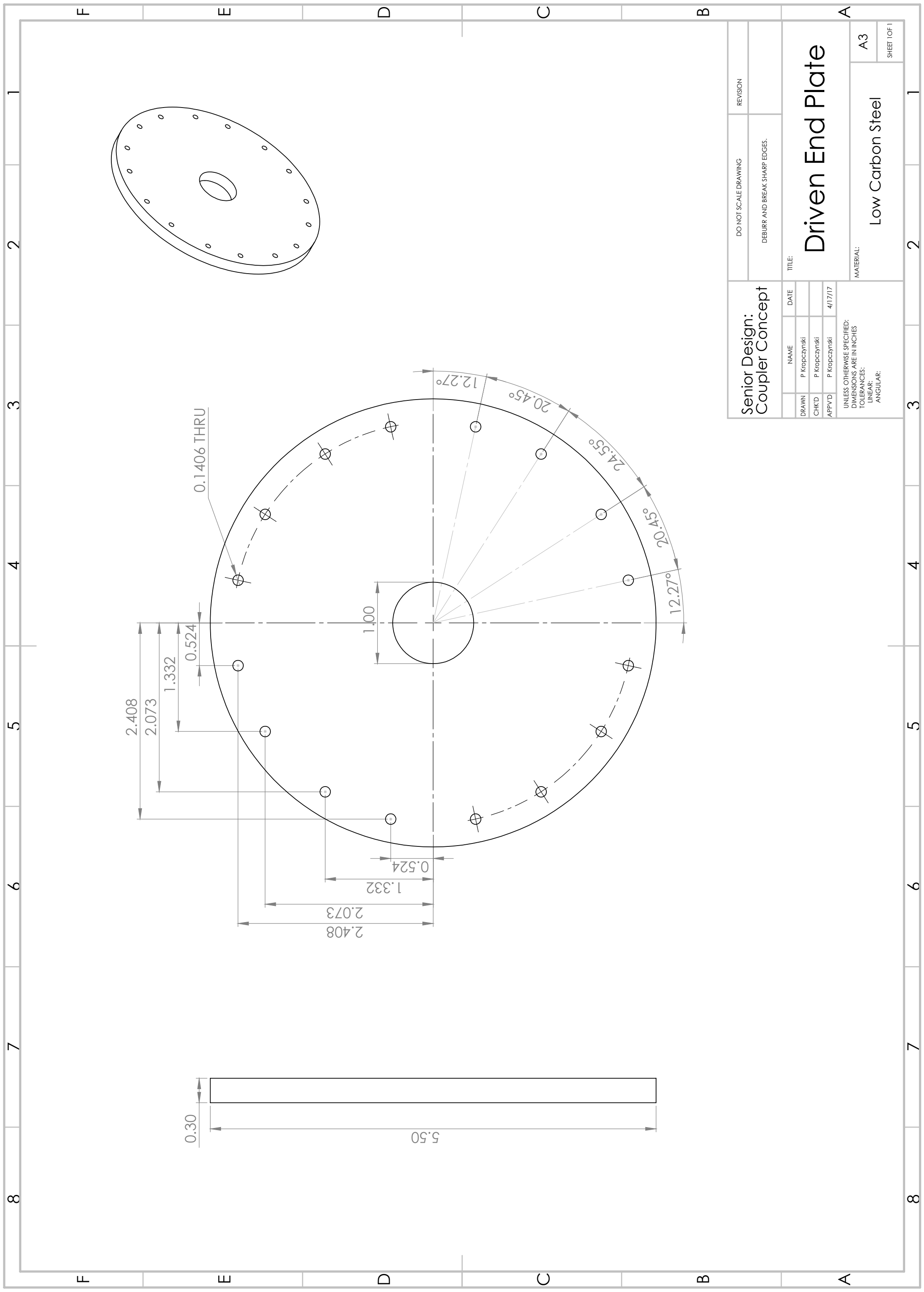
25°
R0.050
R0.500
R2.250
0.200
0.633
0.524
0.52
0.24
0.09
60°
1.250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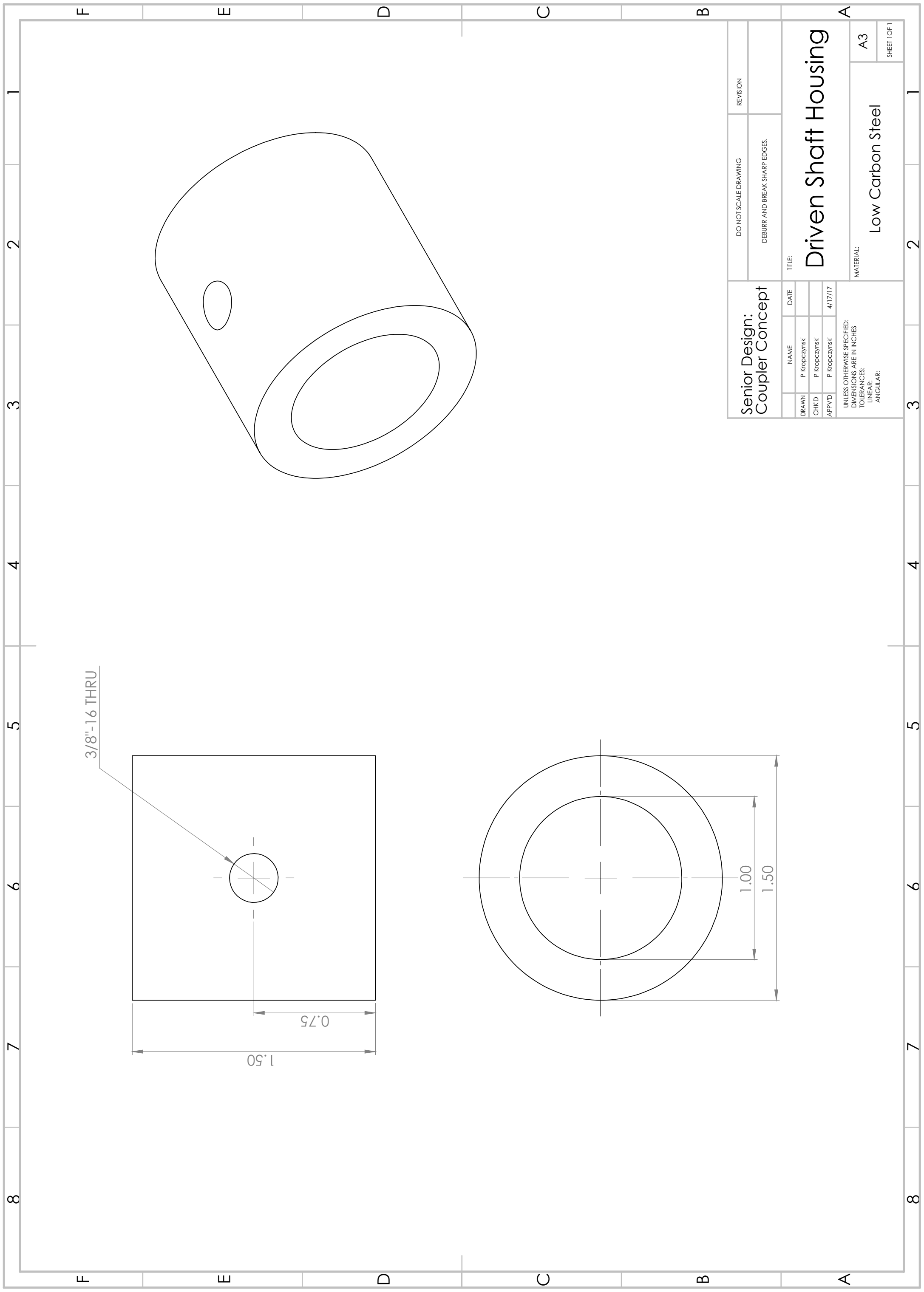
Coupler Concept
DO NOT SCALE DRAWING
DEBURR AND BREAK SHARP EDGES.
REVISION
TITLE:
Driven End Plate
MATERIAL:
Low Carbon Steel
A3
DRAWN P Kropczynski
CHK'D P Kropczynski
APPV'D P Kropczynski 4/17/17
0.1406 THRU
12.27°
20.45°
24.55°
20.45°
12.27°
1.00
0.524
1.332
2.073
2.408
0.524
1.332
2.073
2.408
0.30
5.50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REVISION
DEBURR AND BREAK SHARP EDGES.
TITLE:
MATERIAL:
A3
SHEET 1 OF 1
NAME
DATE
DRAWN P Kropczynski
CHK'D P Kropczynski
APPV'D P Kropczynski 4/17/17
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
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ANGULAR:
3/8"-16 THRU
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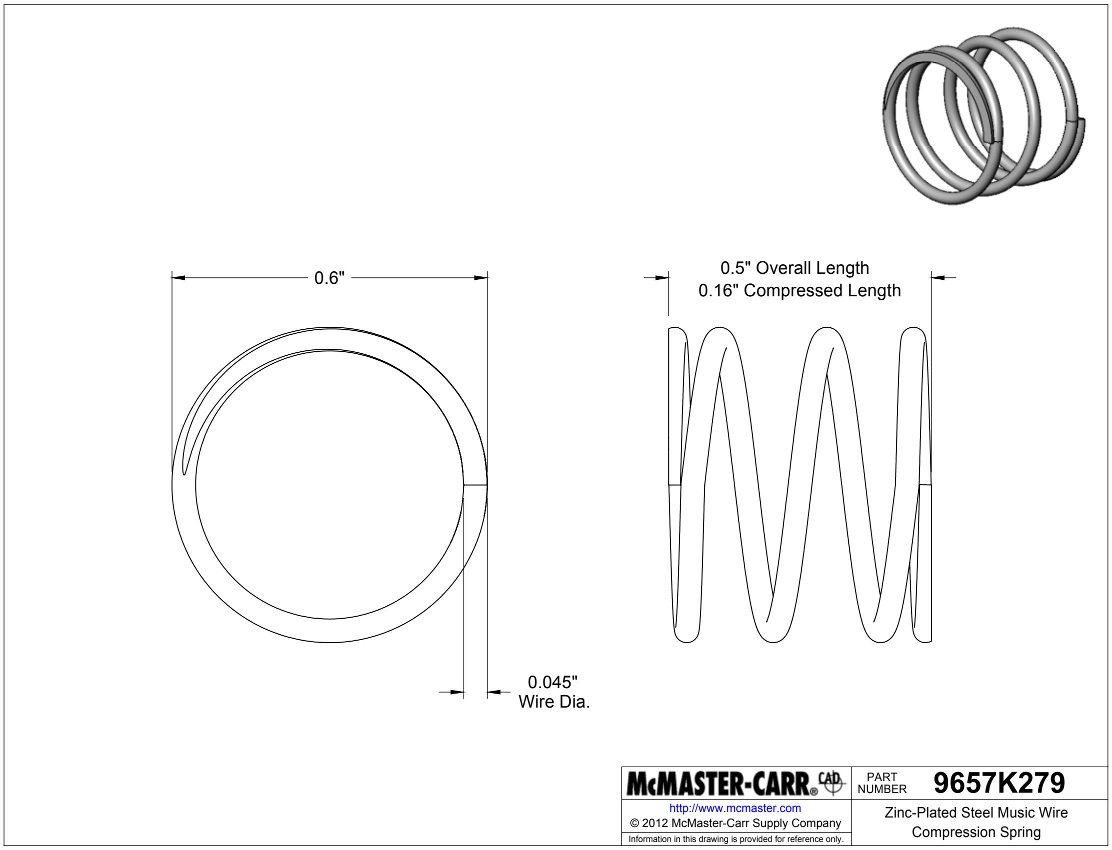
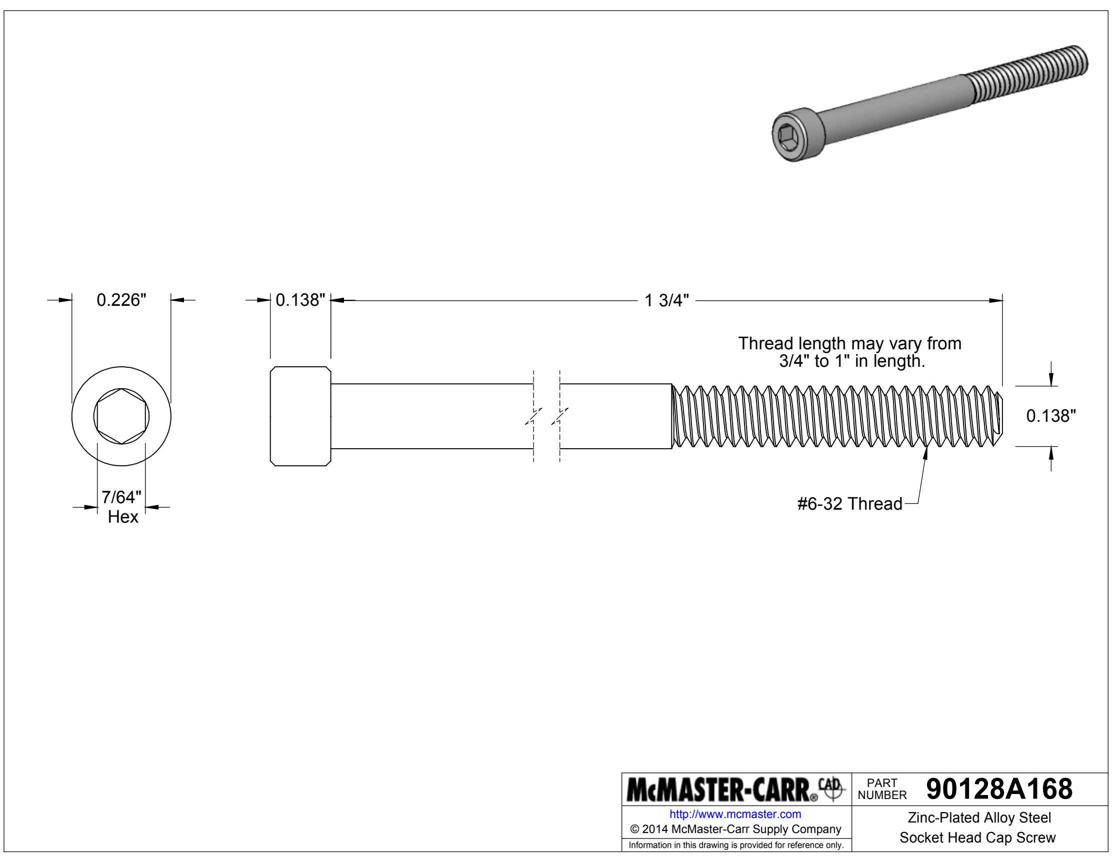
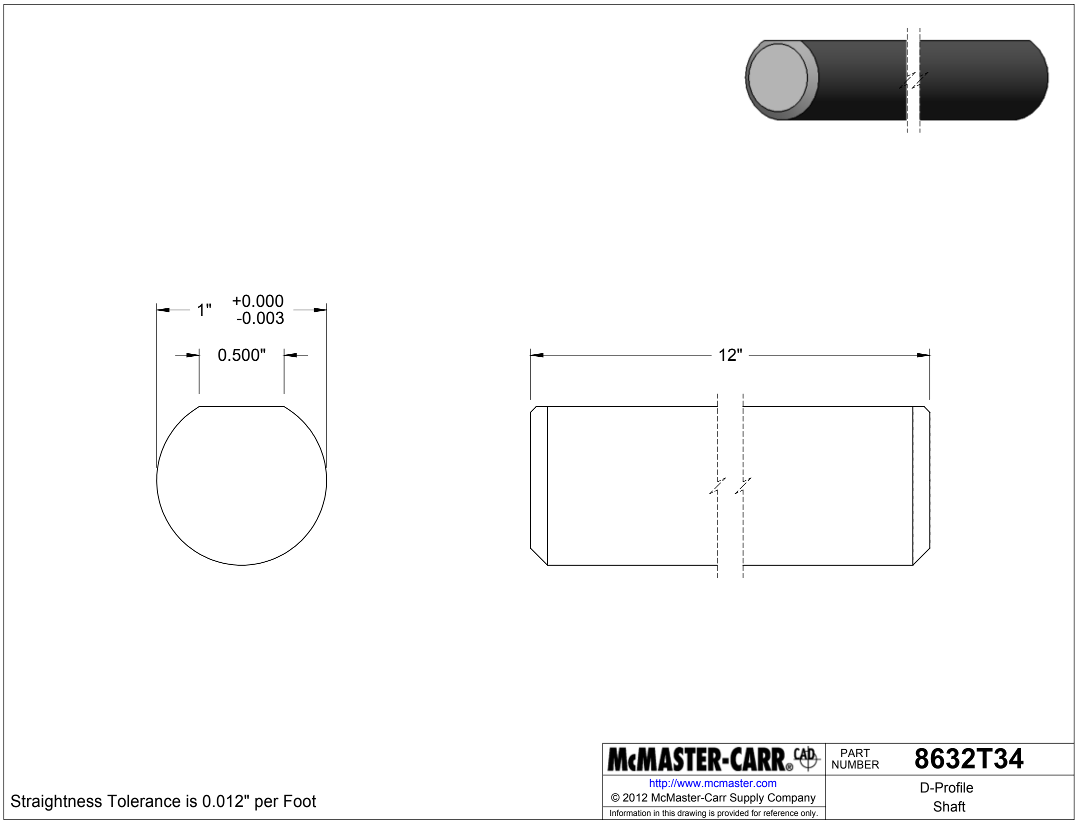
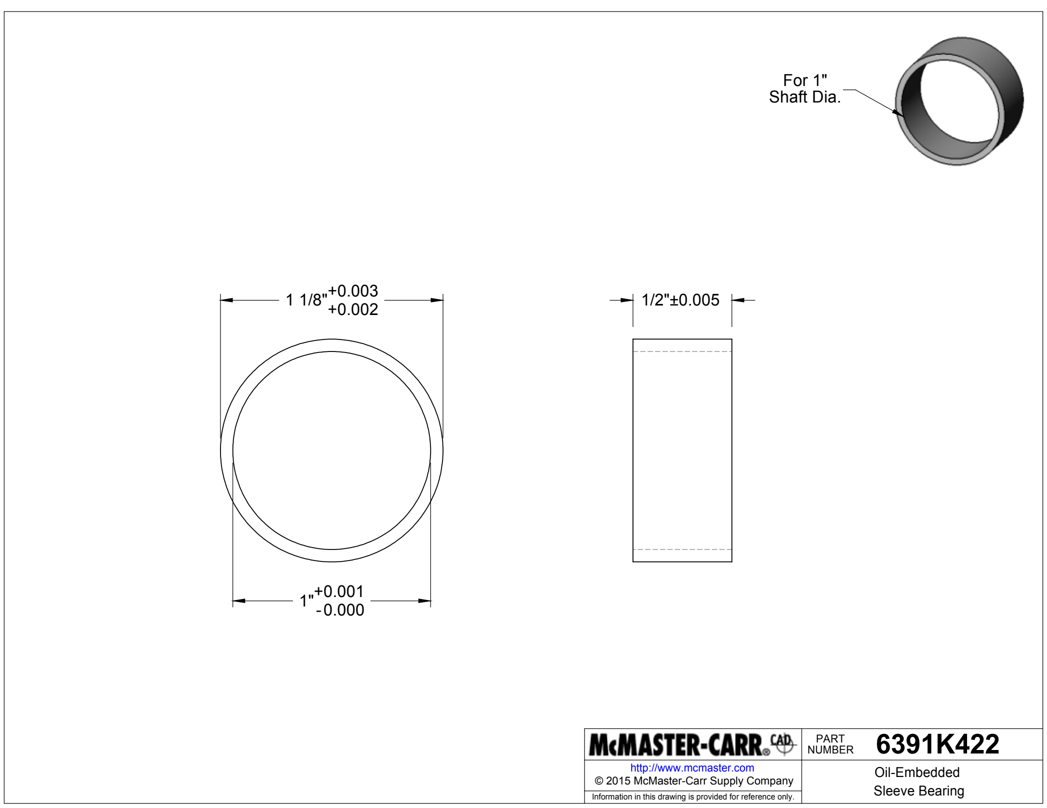
1.50
1.00
1.50

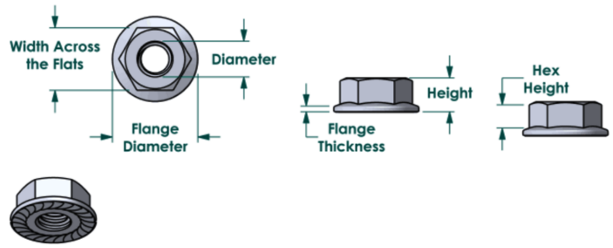

0.6"
0.5" Overall Length
0.16" Compressed Length
0.045"
Wire Dia.
McMASTER-CARR CAD
PART NUMBER 9657K279
http://www.mcmaster.com
© 2012 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
Zinc-Plated Steel Music Wire
Compression Spring


0.226"
0.138"
1 3/4"
Thread length may vary from
3/4" to 1" in length.
0.138"
7/64"
Hex
#6-32 Thread
McMASTER-CARR CAD
PART NUMBER 90128A168
http://www.mcmaster.com
© 2014 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
Zinc-Plated Alloy Steel
Socket Head Cap Screw


1" +0.000 -0.003
0.500"
12"
Straightness Tolerance is 0.012" per Foot
McMASTER-CARR CAD
http://www.mcmaster.com
© 2012 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
PART NUMBER
8632T34
D-Profile Shaft


For 1" Shaft Dia.
1 1/8" +0.003 +0.002
1/2"±0.005
1" +0.001 -0.000
McMASTER-CARR CAD
http://www.mcmaster.com
© 2015 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
PART NUMBER
6391K422
Oil-Embedded Sleeve Bearing

Product Images



Product details

Bolt Depot Product #:	9072
Units:	US
Category:	Nuts
Subcategory:	Flange nuts serrated
Dimensional standard:	IFI-145 (2002) / ASME B18.2.2 (2011)
Material:	Stainless steel
Grade:	18-8
Thread direction:	Right hand
Thread density:	Coarse
Diameter:	#6
Thread count:	32
Width across the flats:	5/16"
Width across the flats Max:	0.312"
Width across the flats Min:	0.302"
Flange diameter Max:	0.422"
Flange diameter Min:	0.408"
Flange thickness Min:	0.02"
Height Max:	0.171"
Height Min:	0.156"
Hex Height Min:	0.10 "

Product Images

NOTE: Image not to scale

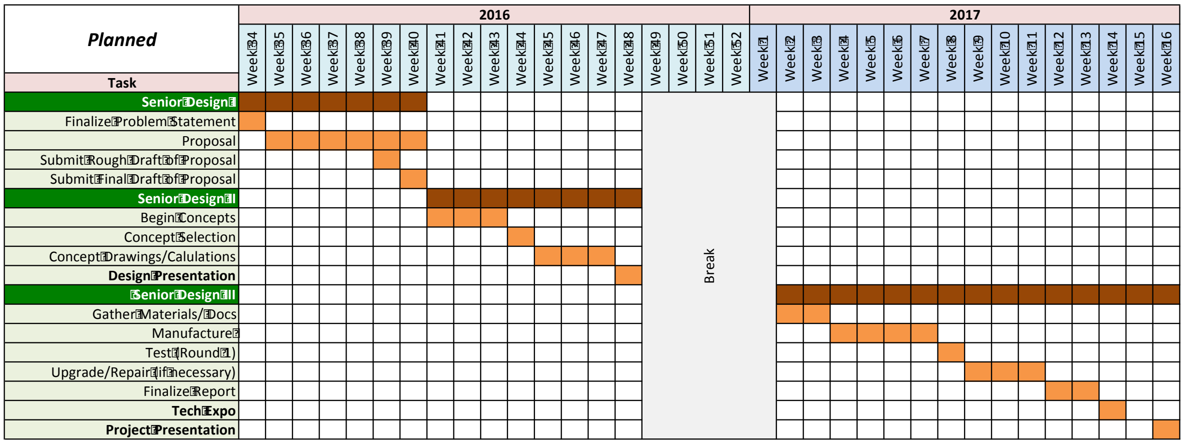
Product details

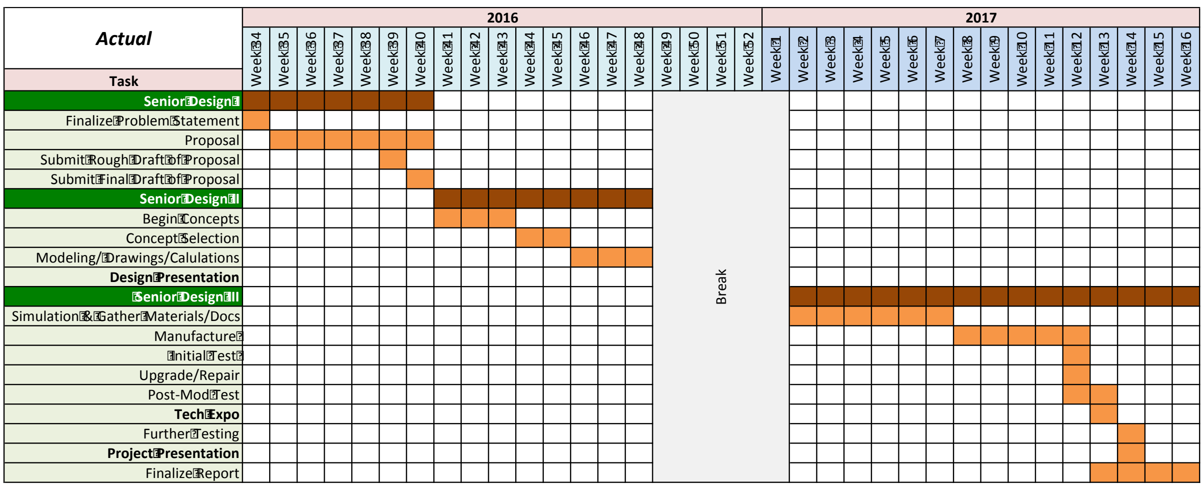
Bolt Depot Product #:	7050
Units:	US
Category:	Set screws
Drive depth Min:	0.188"
Dimensional standard:	ASME B18.3
Drive type:	Allen
Point type:	Cup
Material:	Stainless steel
Grade:	18-8
Thread direction:	Right hand
Thread density:	Coarse
Diameter:	3/8"
Thread count:	16
Length:	1/2"
Drive size:	3/16"

DRAWING CITATIONS:

Spring
(Zinc-Plated Steel Music Wire Compression Spring)
Bolts
(Zinc-Plated Alloy Steel Socket Head Cap Screw)
D-Profile Shaft
(D-Profile Shaft)
Bushing
(Oil-Embedded Sleeve Bearing)
Nuts
(Flange nuts serrated, Stainless steel 18-8, #6-32)
Set Screws
(Set Screws, Allen, Cup point, Stainless steel 18-8, 3/8"-16 x 1/2")

APPENDIX D (SCHEDULE)





APPENDIX E (BUDGET)

Initial Spending Budget (Personally Funded)	$1,000.00

Rough Budget Estimate (Based on Concept 3)			
Material	Pieces Needed	Cost	Total
Drive & Driven Shaft	2	$30.00	$60.00
Bearing Mounts	4	$50.00	$200.00
Round Steel Stock	2	$75.00	$150.00
Springs	6	$2.00	$12.00
Bearings	6	$1.00	$6.00
Fasteners	30	$0.30	$9.00
Mounting Plate	1	$35.00	$35.00
Motor	1	$80.00	$80.00
Control	1	$30.00	$30.00
			$582.00

Refined Budget Est.					
Component	ID	Quantity	Cost	Summed Cost	Details
Shaft	A-Shaft,B-Shaft	2	$13.14	$ 26.28	*D-Profile, Dia.=1"*
Set Screws	SScrews	1	$8.00	$ 8.00	*Pack of 10*
Raw Billet	A-Housing	1	$40.00	$ 40.00	*Main Component Housing*
Raw Billet	B-Plate	1	$40.00	$ 40.00	*Plate*
Raw Billet	B-Shousing	1	$25.00	$ 25.00	
Springs	A-Springs	1	$10.35	$ 10.35	*Pack of 12*
Bearings	A-Bearings	1	$11.59	$ 11.59	*Pack of 50*
Inserts	A-Inserts	10	$10.00	$ 100.00	*Need quote from RP Service*
Interfaces	B-Interface	10	$10.00	$ 100.00	*Qutoes from Machinist or RP Service*
Interface Bolts	B-Bolts	1	$10.75	$ 10.75	*Pack of 25*
Interface Nuts	B-Nuts	16	$0.10	$ 1.60	
Case	Casing			$ -	*RP Service?*
Seal	-	1		$ -	*Needed?*
Grease	-	1	$15.00	$ 15.00	*Needed?*
			Total	$ 388.57	
			Tax Est.	$ 46.63	
			Shipping Est.	$ 19.43	
			Net Total	$ 454.63	

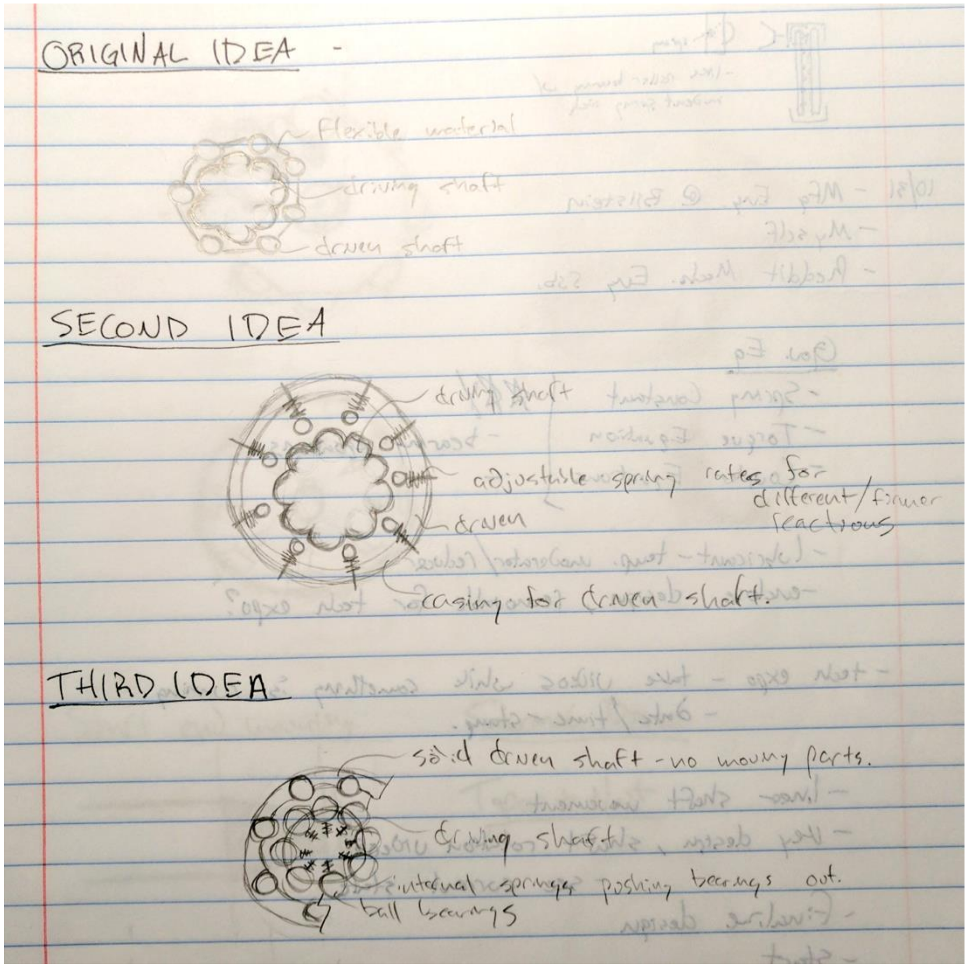
Estimated Cost	
Total Cost	

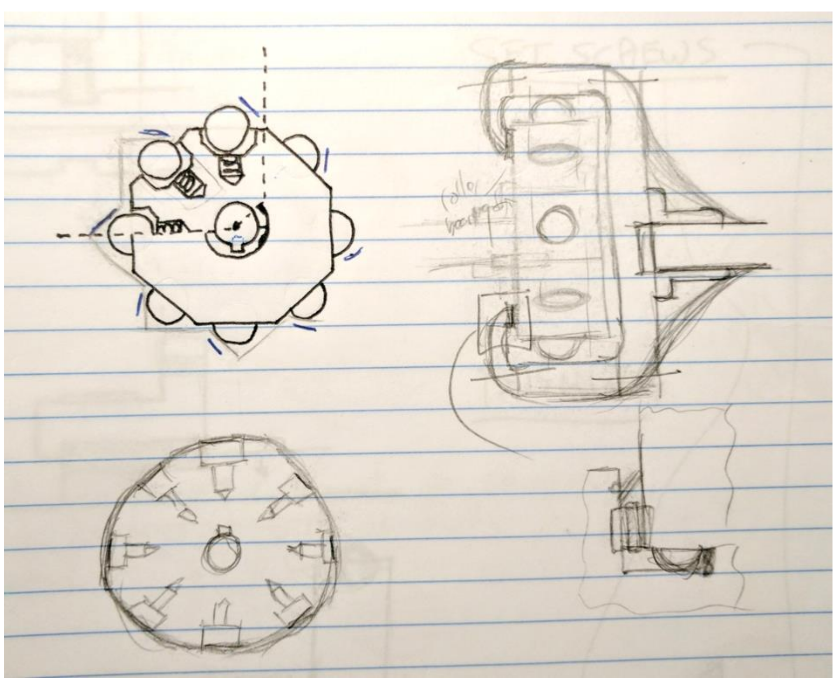
Component	ID	Quantity	Cost	Summed Cost	Details
Shaft Mounts	-	4	$20.00	$ 80.00	
Motor w/ Control	-	2		$ -	
Motor-Shaft Coupler	-	1		$ -	*if needed*
Plate	-	1		$ -	*fix mounts and motor*
			Total	$ 80.00	
			Tax Est.	$ 9.60	
			Shipping Est.	$ 4.00	
			Net Total	$ 93.60	

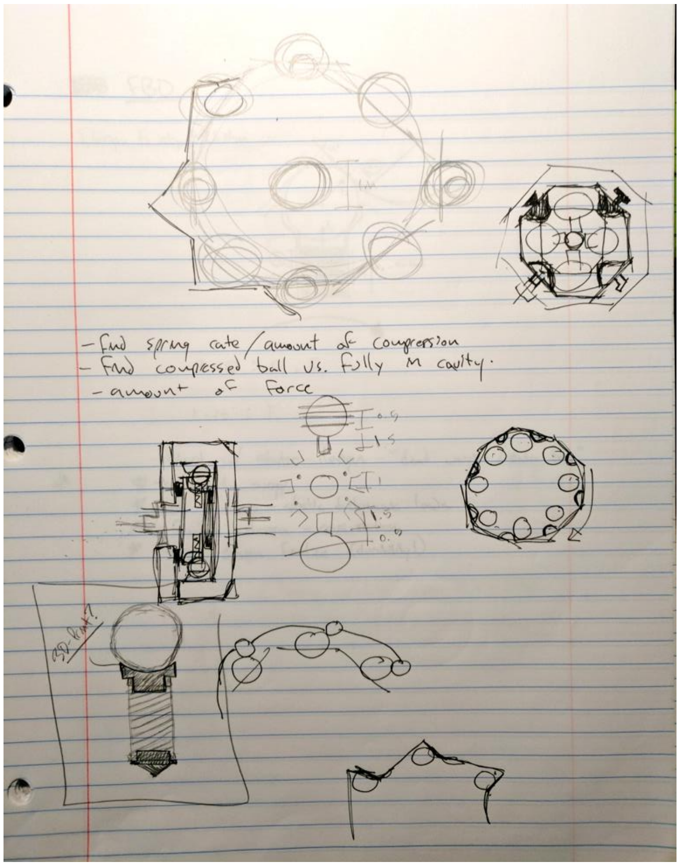
Est. Net Total	**$ 548.23**

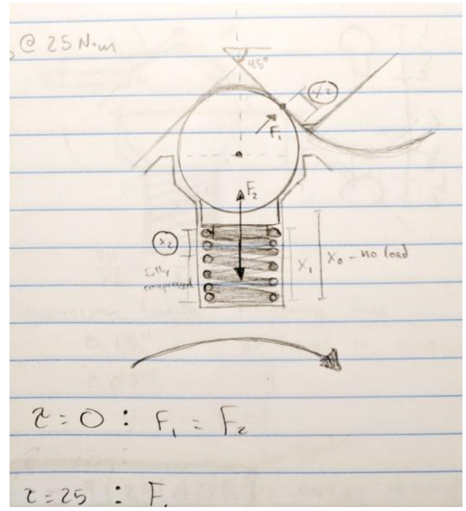
Actual Budget					
Component	Quantity	Cost	Summed Cost	Details	Source
Springs	1	$10.35	$ 10.35	*pack of 12*	Mcmaster Carr
Ball Bearings	2	$7.88	$ 15.76	*packs of 5*	Mcmaster Carr
Bolts	1	$10.75	$ 10.75	*pack of 25*	Mcmaster Carr
Shaft Housing Tube	1	$23.68	$ 23.68	*OD=1.5" ID=1" WallThick=0.25"*	Mcmaster Carr
D-Profile Shaft	2	$13.14	$ 26.28	*1" Diameter, 12" Length*	Mcmaster Carr
Plate Stock	1	$17.14	$ 17.14	*Diameter=6" Length=1/2"*	Mcmaster Carr
Housing Stock	1	$37.04	$ 37.04	*Diameter=4" Length=3"*	Mcmaster Carr
Oil Embedded Sleeve	1	$1.74	$ 1.74		Mcmaster Carr
Mounted Bearings	4	$12.69	$ 50.76		Mcmaster Carr
3D Printed Intfce/Insrt	1	$159.40	$ 159.40	*10 of each component, 50% discount*	ProtoLab
Nuts	1	$7.95	$ 7.95	*30 Nuts*	Bolt Depot
Set Screws	1	$6.70	$ 6.70	*5 Screws*	Bolt Depot
Aluminum Block	1	$0.00	$ -	*Made into bases for Mounted Bearings*	Donated Metal UC
Stainless Steel Plates	2	$0.00	$ -	*Re-fab. of driven end plate*	Donated Metal UC
Tech Expo Poster	1	$45.00	$ 45.00		FedEx Printing Service
		Total	$ 412.55		

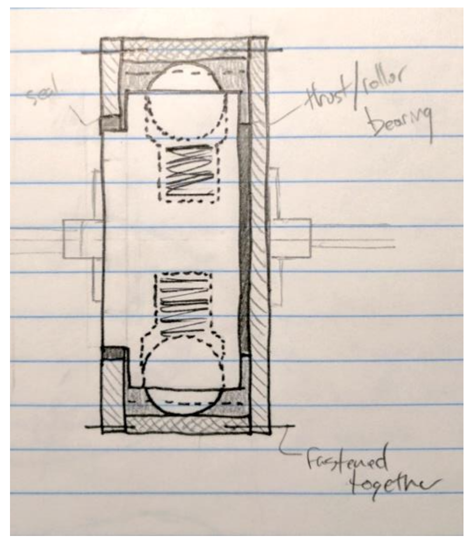
APPENDIX F (SKETCHES AND NOTES)

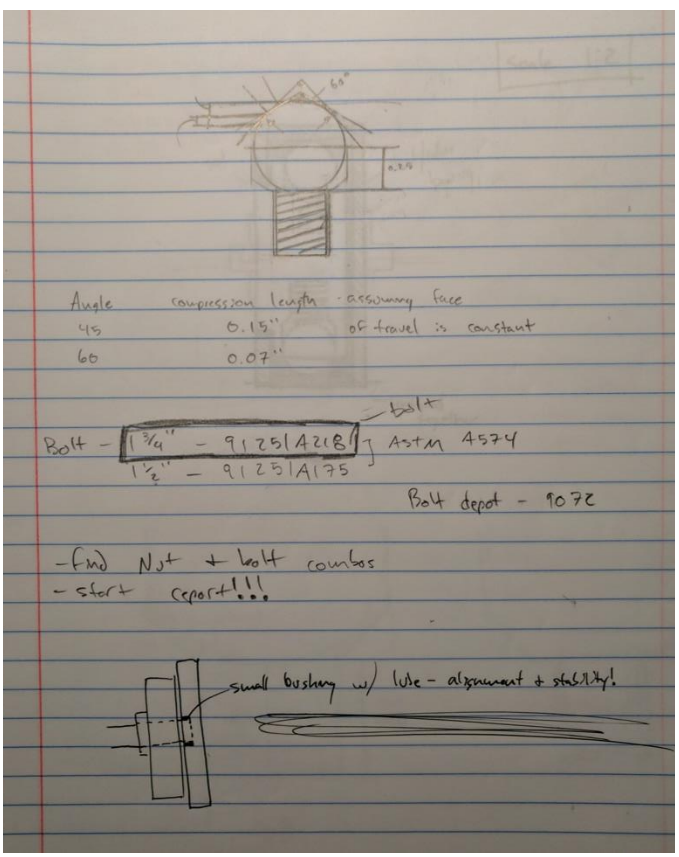


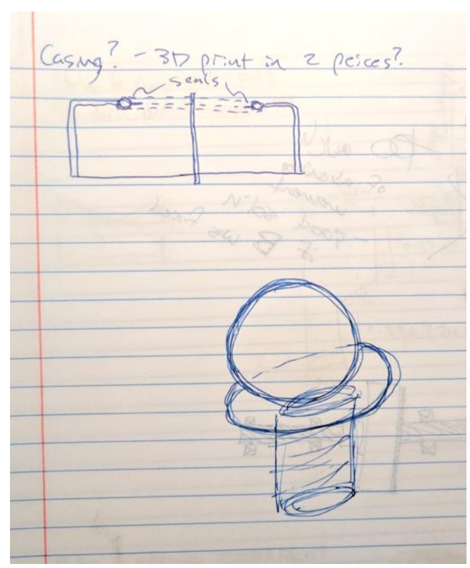


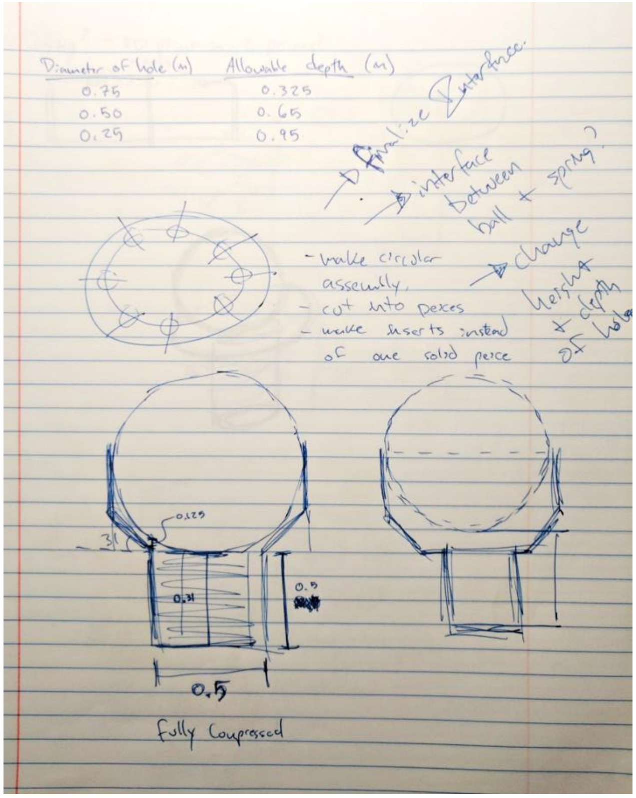

- find spring rate / amount of compression
- find compressed ball vs. fully in cavity.
- amount of force

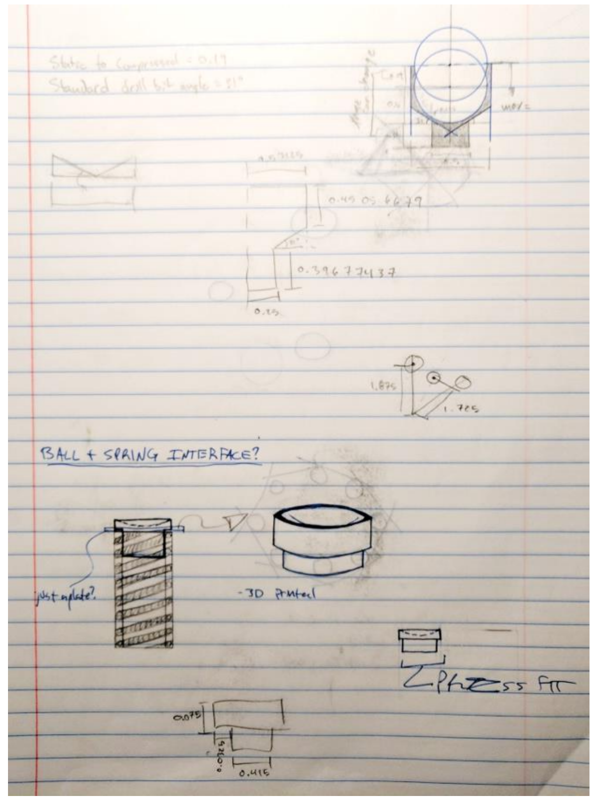

@ 25 N·m
45°
F1
F2
fully compressed
X1
X0 - no load
τ = 0 : F1 = F2
τ = 25 : F1

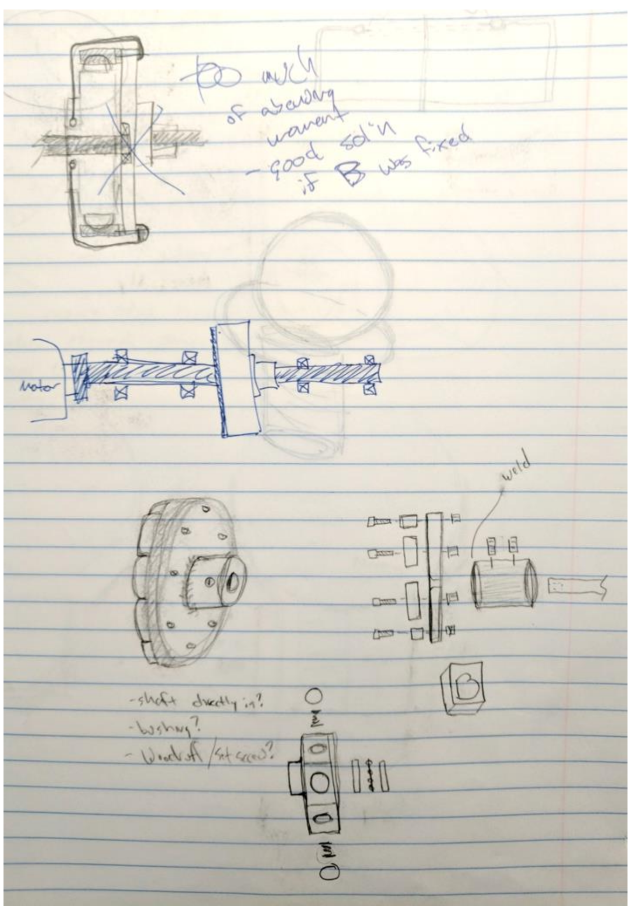

seal
thrust/roller bearing
fastened together

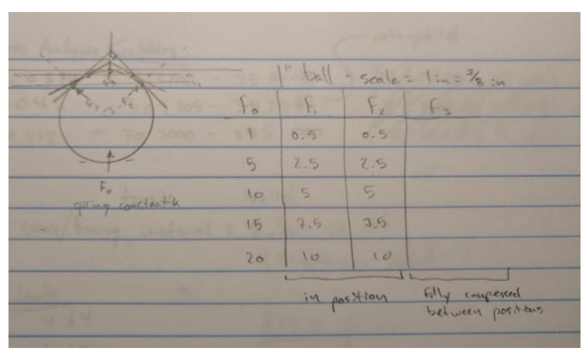

60°
Angle compression length - assuming face
45 0.15" of travel is constant
60 0.07"
bolt
Bolt - 1 3/4" - 91251A218] Astm 4574
1 1/2" - 91251A175
Bolt depot - 9072
- Find Nut + bolt combos
- start report!!!
small bushing w/ lube - alignment + stability!

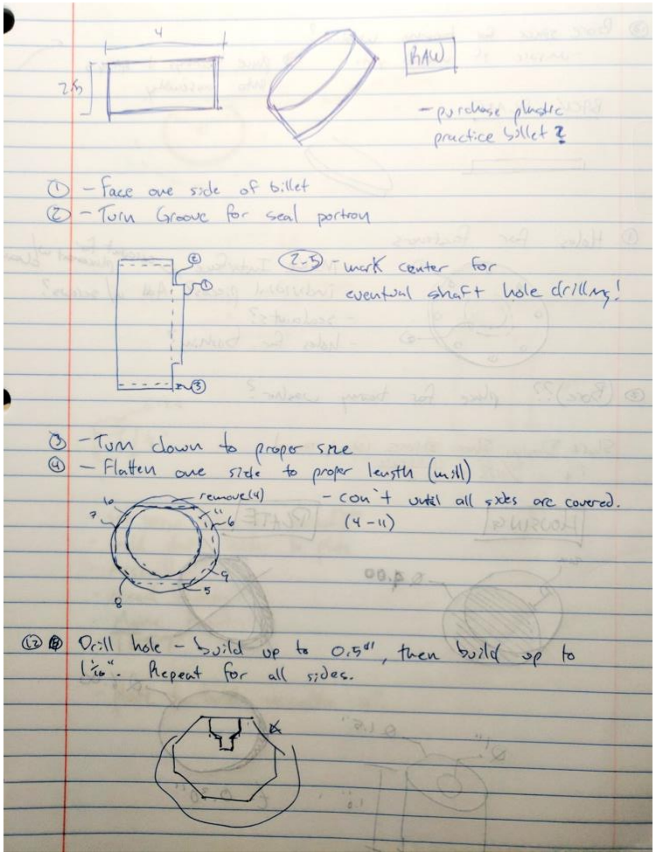

Casing? - 3D print in 2 peices?
seals

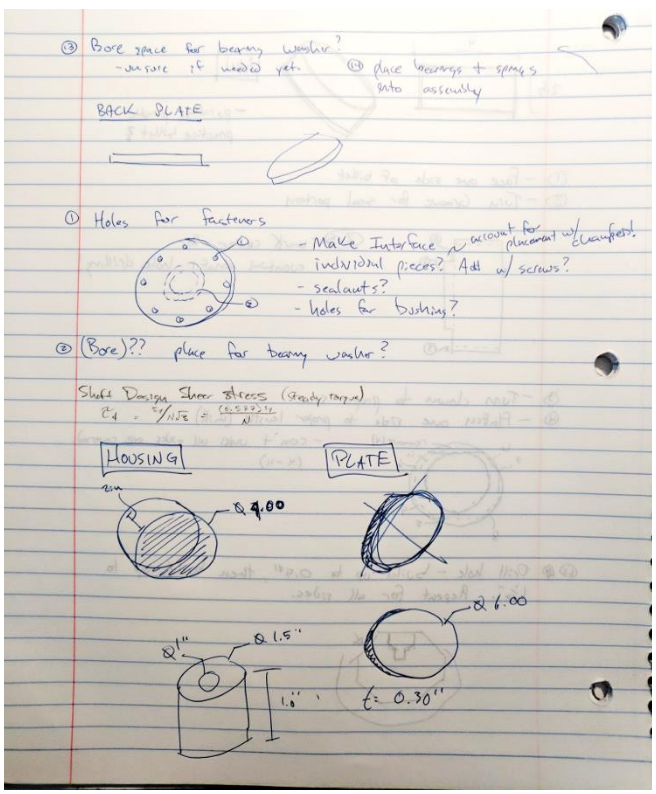

Diameter of hole (in) Allowable depth (in)
0.75 0.325
0.50 0.65
0.25 0.95
→ finalize interface
→ interface between ball + spring?
- make circular assembly
- cut into pieces
- make inserts instead of one solid piece
→ change height + depth of holes
0.325
0.5
0.5
fully Compressed

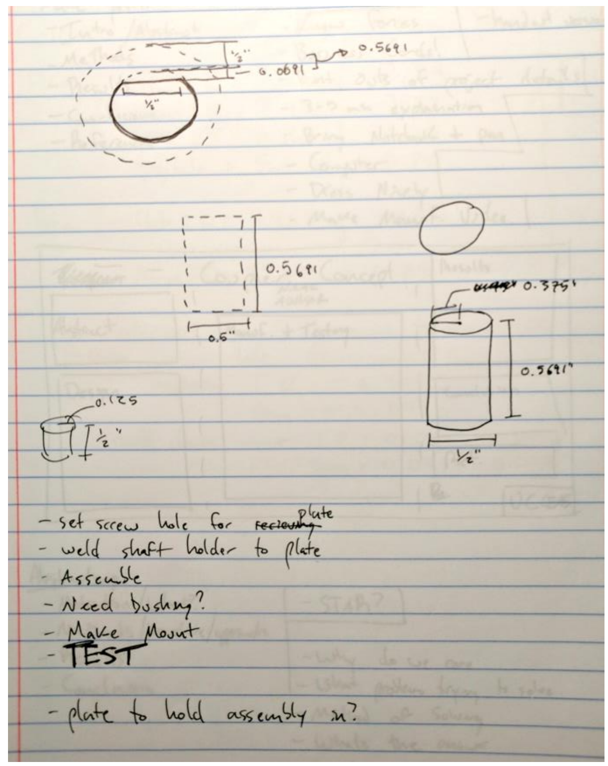

BALL + SPRING INTERFACE?
just a plate?
-3D printed
PRESS FIT
0.396772437
1.725

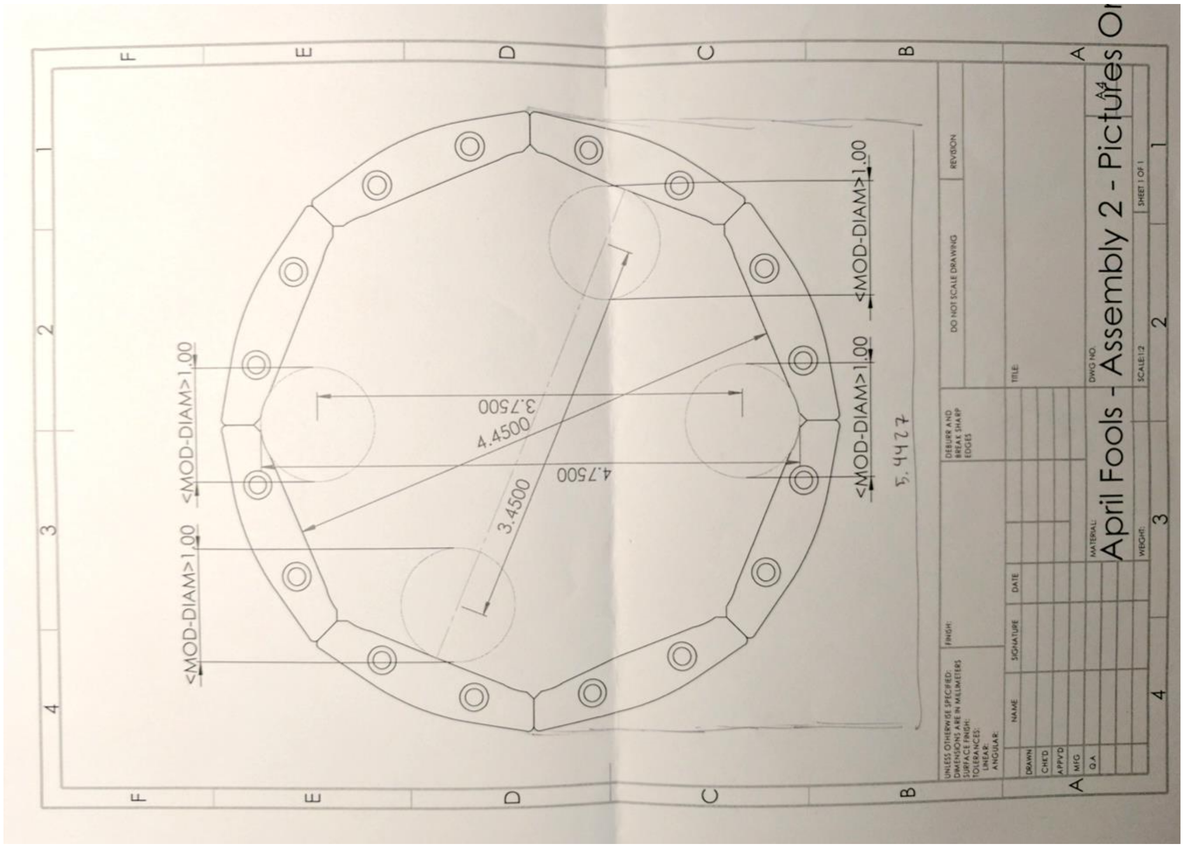

of avoiding moment
- good sol'n if B was fixed
Motor
weld
-shaft directly in?
-bushing?
- Woodruff/set screw?


1" ball - scale = 1 in = 3/8 in
F0 F1 F2 F3
1 0.5 0.5
5 2.5 2.5
10 5 5
15 7.5 7.5
20 10 10
in position
fully compressed between positions
F0
spring contact

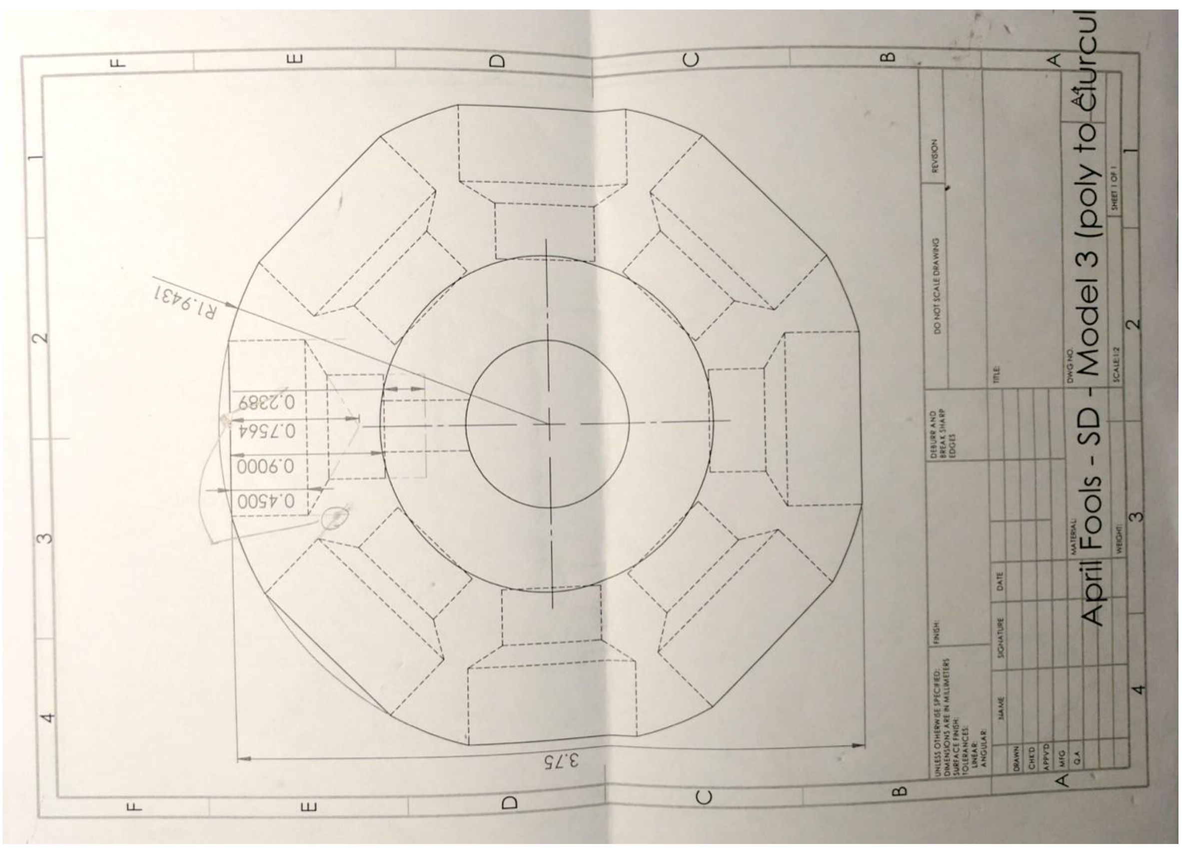

RAW
- purchase plastic practice billet?
① - Face one side of billet
② - Turn Groove for seal portion
2.5 - mark center for eventual shaft hole drilling!
③ - Turn down to proper size
④ - Flatten one side to proper length (mill)
- con't until all sides are covered. (4 - 11)
Drill hole - build up to 0.5", then build up to 1 1/16". Repeat for all sides.


③ Bore space for bearing washer?
- unsure if needed yet.
⑭ place bearings + springs into assembly
BACK PLATE
① Holes for fasteners
- Make Interface account for placement w/ clampers!
individual pieces? Add w/ screws?
- sealants?
- holes for bushing?
② (Bore)?? place for bearing washer?
Shaft Design Shear Stress (Steady torque)
HOUSING
PLATE
Ø 4.00
Ø 6.00
Ø 1.5"
Ø 1"
1.0"
t = 0.30"

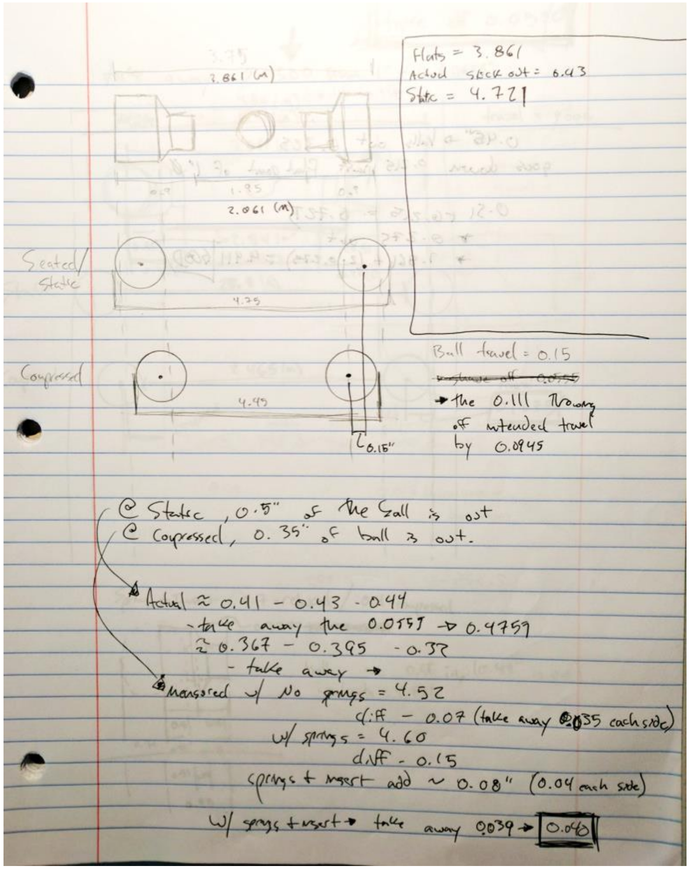

0.5691
0.0691
1/2"
0.5691
0.5"
0.375"
0.5691"
1/2"
0.125
1/2"
- set screw hole for plate
- weld shaft holder to plate
- Assemble
- Need bushing?
- Make Mount
- TEST
- plate to hold assembly in?



April Fools - Assembly 2 - Pictures On
<MOD-DIAM>1.00
<MOD-DIAM>1.00
<MOD-DIAM>1.00
<MOD-DIAM>1.00
3.7500
4.4500
4.7500
3.4500
5.4427



April Fools - SD - Model 3 (poly to clurcul
R1.9431
0.2389
0.7564
0.9000
0.4500
3.75



Flats = 3.861
Actual stick out = 6.43
Static = 4.721

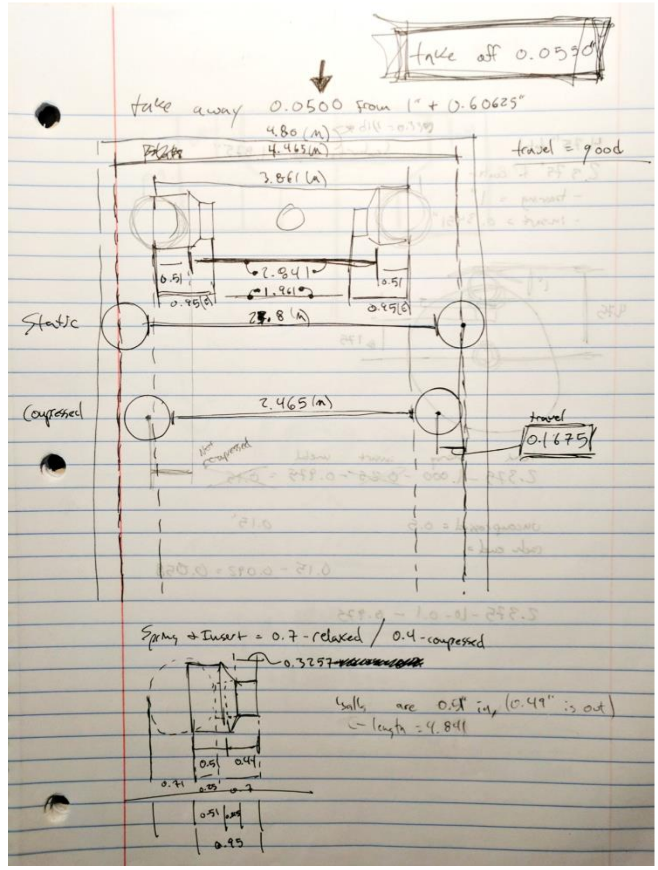
3.861 (m)
1.95
2.061 (m)
Seated/Static
4.25
Compressed
4.45
0.15"

Ball travel = 0.15
~~take off 0.0555~~
→ the 0.111 throwing of intended travel by 0.0945

@ Static, 0.5" of the ball is out
@ Compressed, 0.35" of ball is out.

Actual ≈ 0.41 - 0.43 - 0.44
- take away the 0.0559 → 0.4759
≈ 0.367 - 0.395 - 0.37
- take away →
measured w/ No springs = 4.52
diff - 0.07 (take away 0.035 each side)
w/ springs = 4.60
diff - 0.15
springs + insert add ~ 0.08" (0.04 each side)

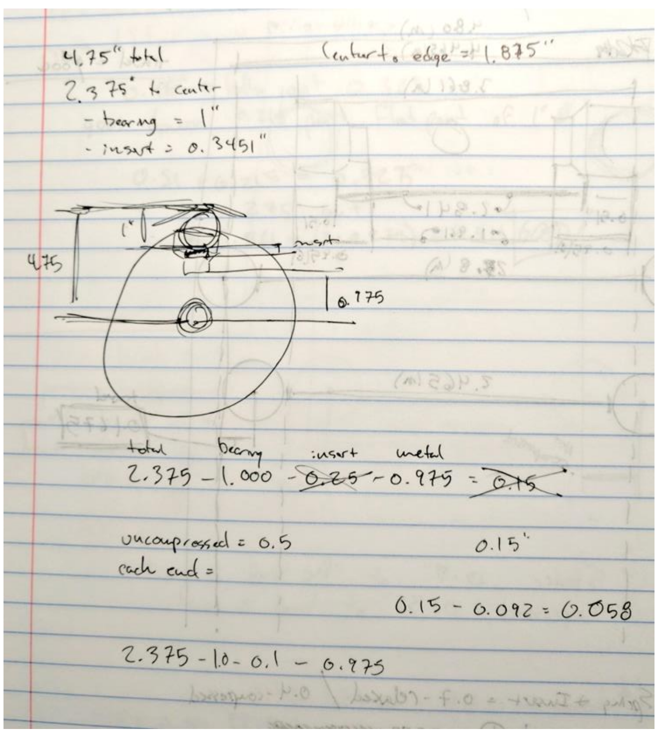
w/ springs + insert → take away 0.039 → 0.040



take off 0.0590"

take away 0.0500 from 1" + 0.60625"
4.80 (m)
4.465 (m)
travel = good
3.861 (m)
2.841
1.961
0.51
0.45(s)
Static
2.8 (m)
Compressed
2.465 (m)
Not compressed
travel
0.1675

Spring + Insert = 0.7 - relaxed / 0.4 - compressed
0.3257
balls are 0.5" in, (0.49" is out)
length = 4.841
0.51
0.44
0.71
0.7
0.95



4.75" total
2.375" to center
- bearing = 1"
- insert = 0.3451"

Center to edge = 1.875"

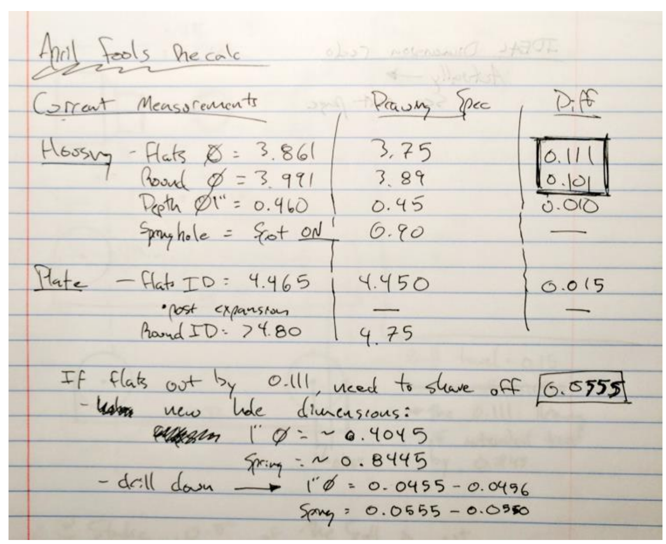
1"
insert
4.75
0.775

total bearing insert metal
2.375 - 1.000 - ~~0.25~~ - 0.975 = ~~0.15~~

uncompressed = 0.5
each end =
0.15"

0.15 - 0.092 = 0.058

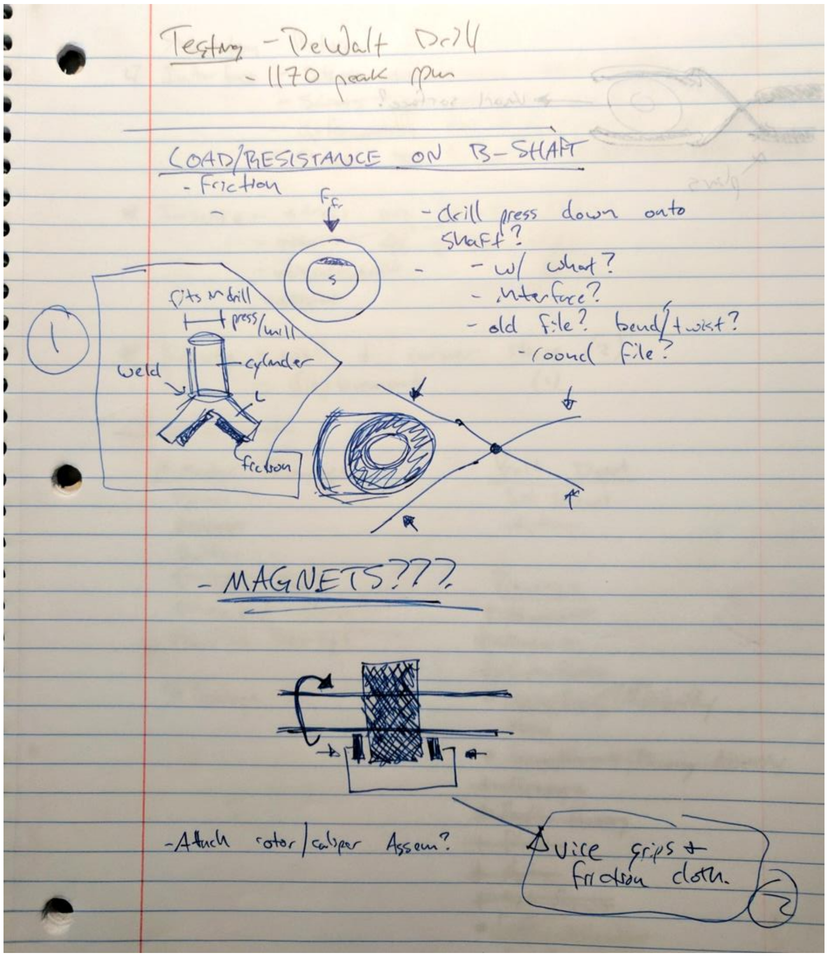
2.375 - 1.0 - 0.1 - 0.975

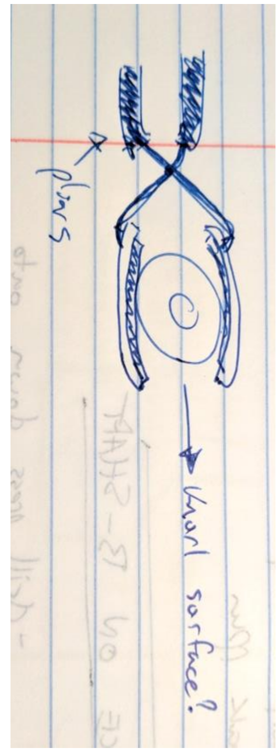


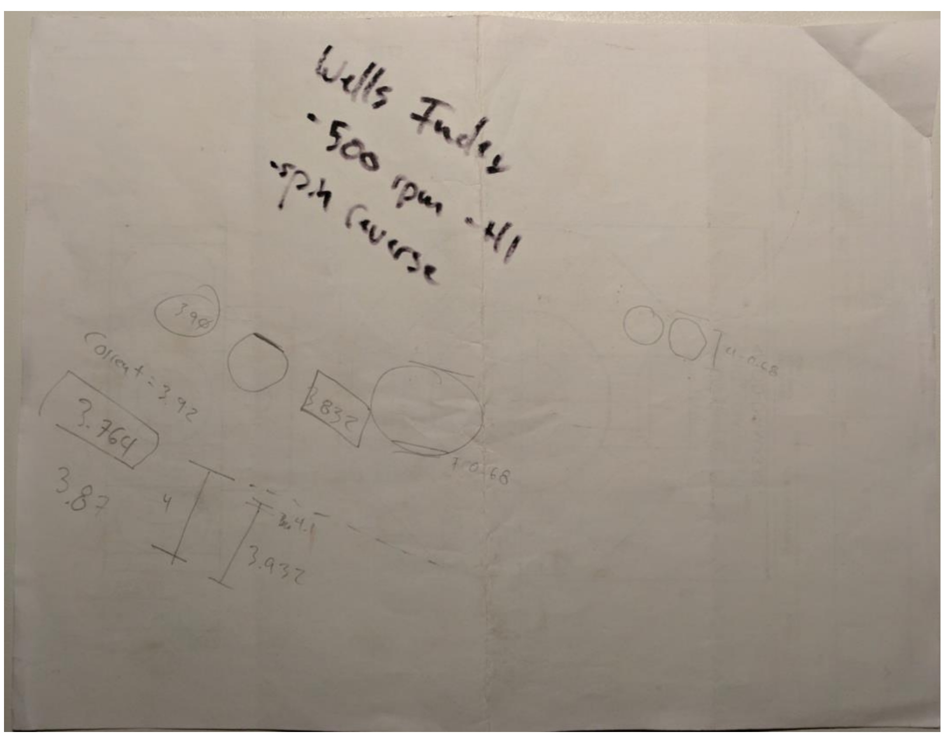
April Fools Recalc

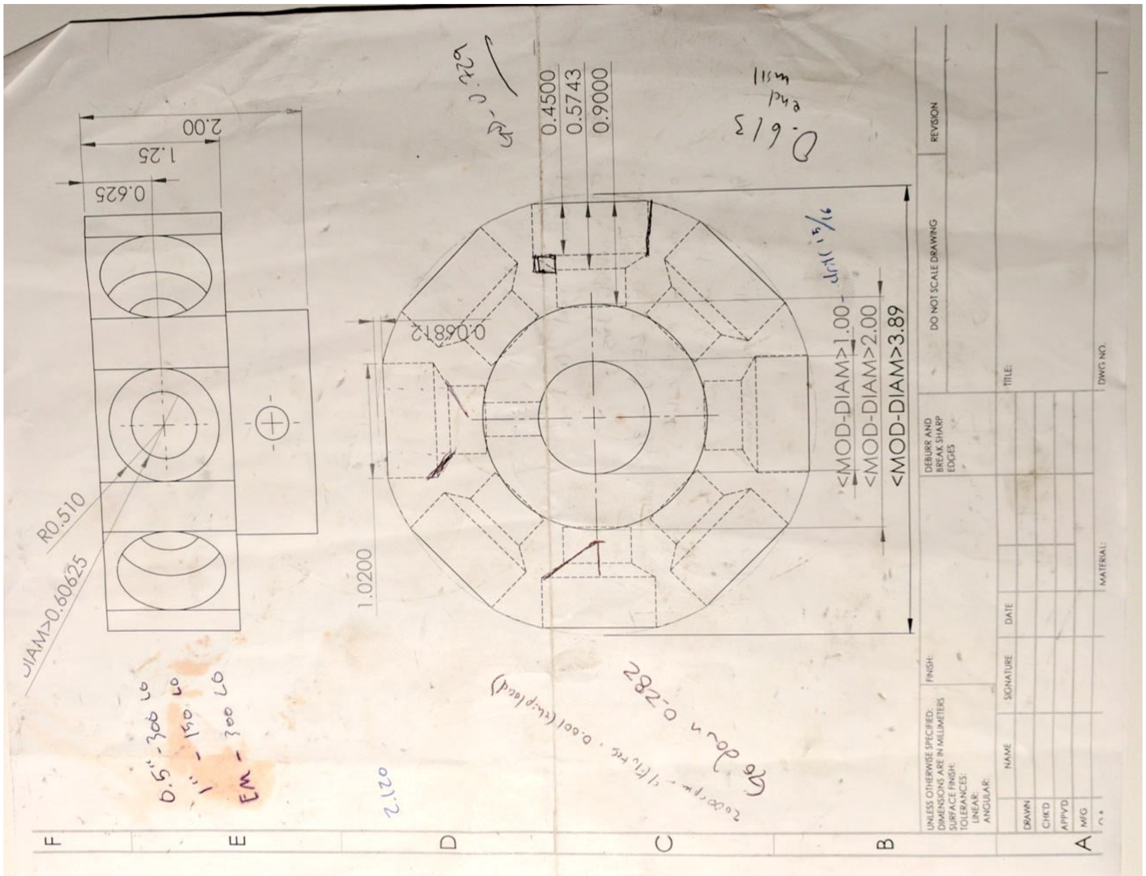
Current Measurements		Drawing Spec	Diff
Housing	- Flats Ø = 3.861	3.75	0.111
	Round Ø = 3.991	3.89	0.101
	Depth Ø1" = 0.460	0.45	0.010
	Spring hole = Spot ON	0.90	—
Plate	- Flats ID = 4.465	4.450	0.015
	*post expansion	—	—
	Round ID = >4.80	4.75	

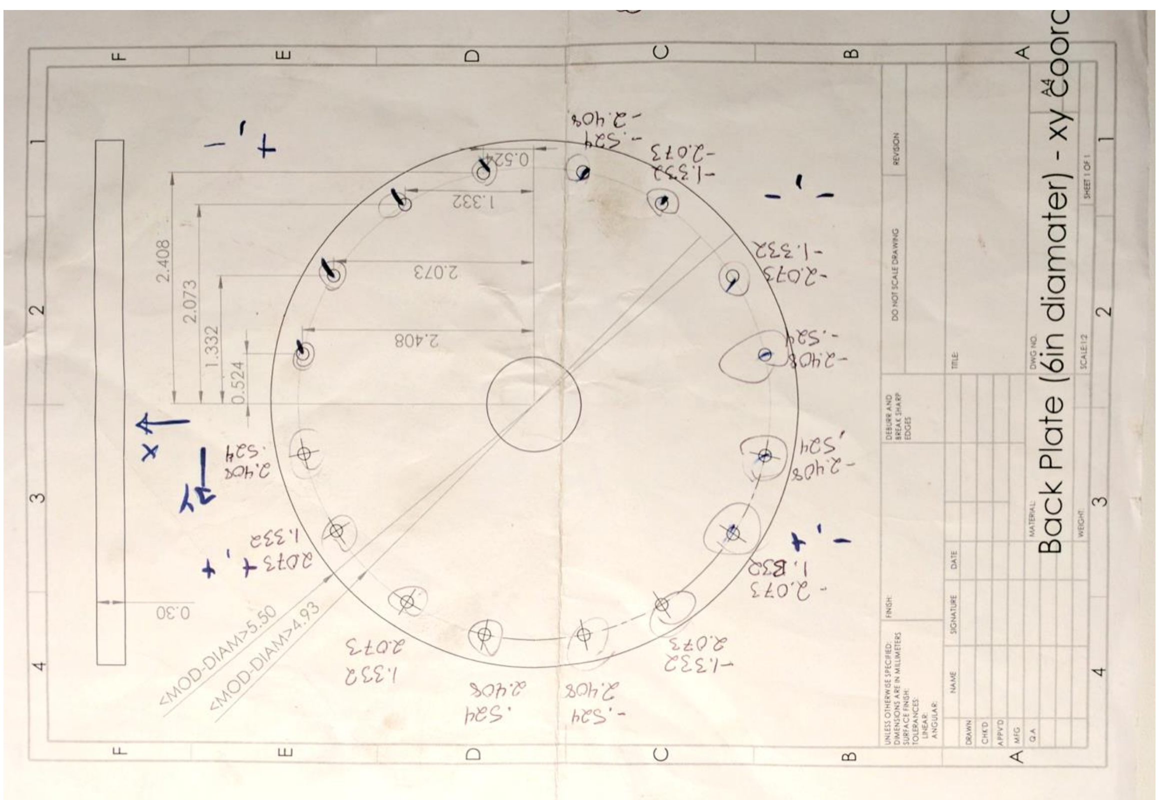
If flats out by 0.111, need to shave off 0.0555
- new hole dimensions:
1" Ø = ~ 0.4045
Spring = ~ 0.8445
- drill down → 1" Ø = 0.0455 - 0.0496
Spring = 0.0555 - 0.0950


Testing - DeWalt Drill
- 1170 peak rpm
LOAD/RESISTANCE ON B-SHAFT
- Friction
- drill press down onto shaft?
- w/ what?
- interface?
- old file? bend/twist?
- round file?
fits in drill press/mill
cylinder
weld
friction
- MAGNETS???
- Attach rotor/caliper Assem?
vice grips + friction cloth.


pliers
knurl surface?


Wells Index
Correct = 3.92
3.764
3.832
3.87
3.932





Back Plate (6in diamater) - xy Coorc