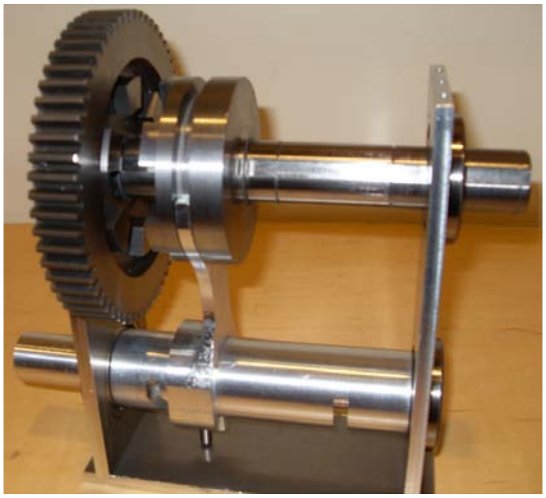
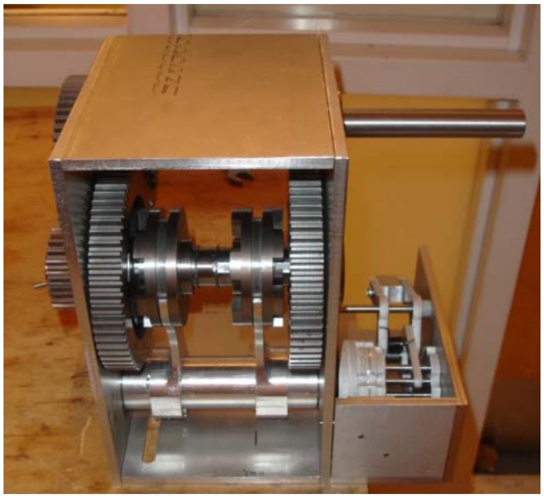
# Baja's Manservants

*Edward Byun, David Gardner, Jay Gorasia, Ryan Harris, Neil Paulson*

*12/14/2009*





# Contents

## TABLE OF FIGURES

## EXECUTIVE SUMMARY

We intend to design and manufacture a prototype high-low-neutral-reverse gearbox for the Olin College MiniBaja Team over a period of 2 months involving a midway (Gate 1) and final (Gate 2) status presentation. The current system used in the Baja car includes only a high gear. The low and reverse capability of the gearbox will enhance hill climbing ability and allow the car to recover from going off track. The primary focus for this design experience is to create a reliable shifting mechanism between gears that are appropriately sized. Sizing of gears is based on an analysis of the power source prescribed by SAE MiniBaja, a Briggs and Stratton 10 HP engine, and the reaction of the 460 lb Baja car to this engine. We hope to minimize weight and volume of this transmission while maximizing reliability.

Our design has three subsections: power, transmission, and structure. Our power source is defined for us by SAE MiniBaja so we used this information to determining an appropriate gear reduction for optimizing speed and torque (high 5:1 and low 8.75:1). As our project is a transmission, most of our component analysis is of the pieces involved in gear selection. The gearbox is designed after a motorcycle transmission consisting of six unique components: shafts, gears, spiders, shift forks, a barrel cam, and an actuation system. Rotation of the barrel cam by the actuation system causes the linear motion of the shift forks. This motion causes the spiders to engage or disengage from gears. For each of these components, we provide a loads analysis, material selection, and manufacturing techniques applied. We chose a barrel cam system because it seemed the most reliable for shifting to 3 sets of gears. Our analysis yielded several design constraints and changes. We had originally considered diametral pitch 16 gears based on the 2008 Olin Baja car design; however, a fatigue analysis indicated these gears would fail. We therefore designed around a system using larger diametral pitch 12 gears. Structurally we chose to make our components out of relatively easy to machine, cheap materials: 6061 aluminum, 1018 steel, and cast iron. While we based each components material selection on loads and purpose analysis, some materials selections were not up to us (we use gears from Boston Gear which only come in cast iron).

Our major fabrication challenge was managing time such that 47 distinct components could be machined over a machining period of 3 weeks. We solved this through daily delegation of specific tasks by a fabrication manager overseeing the timeline of the project. 10 of the components required CNC machining, which had a high setup time but allowed for relatively fast production of similar parts.

We were successful in producing a shifting mechanism that would be a foundation for significant redesign and analysis for inclusion in a Baja car. The actuation system was effective at inducing gear meshing though high friction caused it to slip and fail to shift when entering reverse. Our design was also not suitable for testing under running conditions so we are unsure whether are model is realistically appropriate. Despite not testing our system in operating conditions, we have several suggestions for redesign before this transmission will be ready for inclusion in a Baja car. These include weight reduction, gear material selection to reduce weight and friction significantly, and continued redesign of the actuation system to improve reliability.

## INTRODUCTION

The Olin College MiniBaja Team is consistently looking for innovative solutions to yield improved performance. However, because of time constraints, the team tends to focus on design simplicity and reliability. A high-low-neutral-reverse gearbox is an advanced technical feature, which would make the Olin car more competitive. A low gear would improve results in the tractor pull competition and the endurance competition, which includes steep hill climbs. In the endurance and maneuverability competitions, points are deducted whenever a course worker touches the vehicle. A reverse gear would improve point total as the Olin MiniBaja car would not require course worker intervention if it travels off course.

## POWER DISCUSSION

### GEAR REDUCTION SELECTION

#### OBJECTIVE

The major objective is to find appropriate gear ratios for the gear box. We first calculate the torque needed to drive the Baja car up an incline of 30°, which is a typical gradient in the Baja competition, at constant power from the CVT. We then back out the gear ratios required to produce this torque at the wheels from the maximum torque produced at the engine.

#### GIVEN PROPERTIES

The Baja car is powered by a Briggs and Stratton 205332-0536 10 HP Intek OHV horizontal shaft engine. Figure 1 shows the torque and power curves for the Baja car's engine. The figure is supplied by Briggs and Stratton.

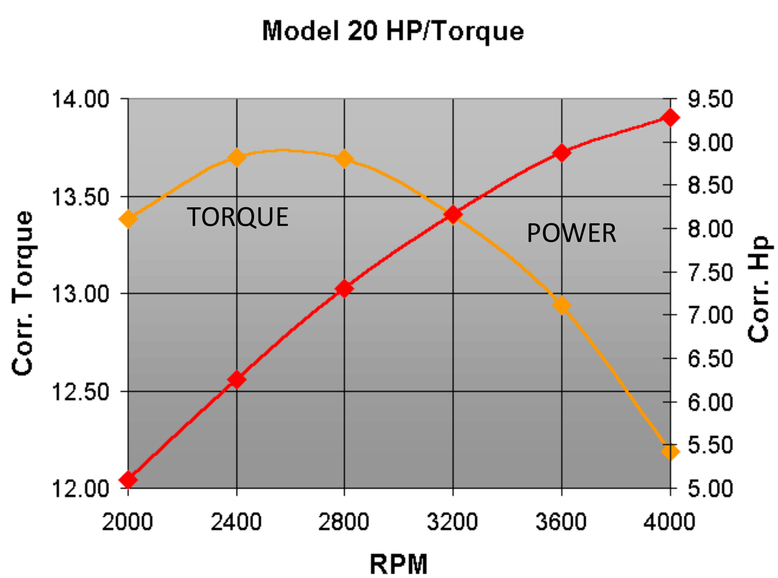



Figure 1: Torque and power curve for the engine used.

The output shaft of the engine is connected to a continuously variable transmission. The maximum and minimum gear ratios for the CVT are 3:1 and 0.98:1.

Subsequently, the output shaft of the CVT is connected to the gearbox. Finally, the gearbox imparts a torque onto the differential which then transfers a load to the wheels.

This relationship is captured in:

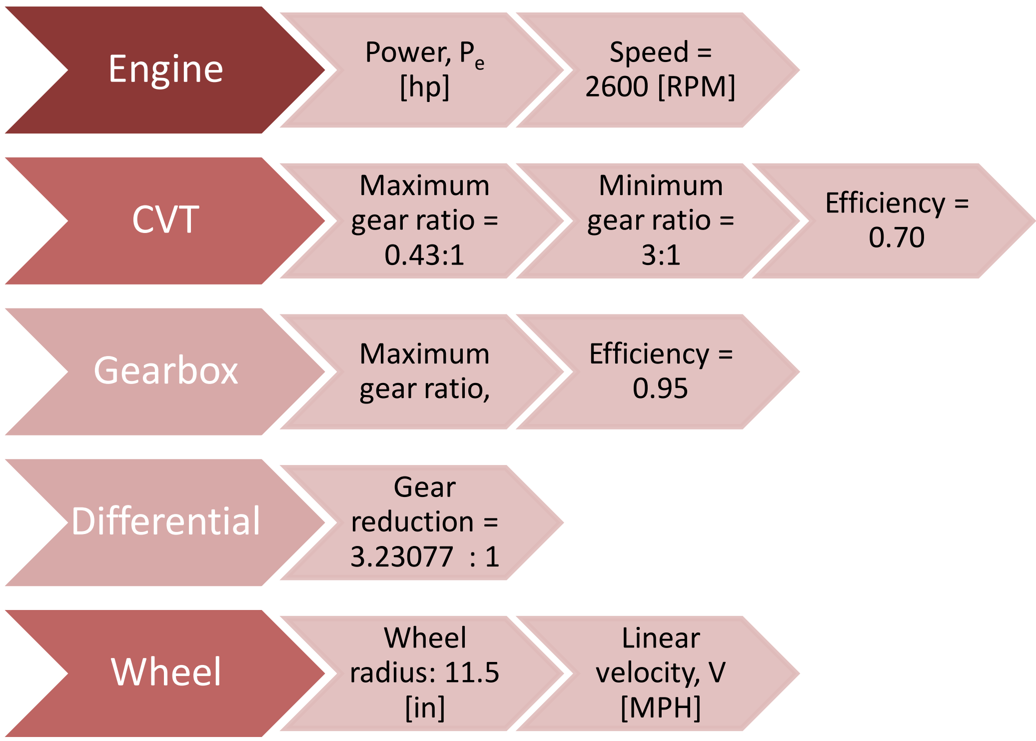



**Figure 2: Given values for the transmission.**

Other important values are:

- Mass of the car and driver: 620 [lbs]
- Frontal area of the car: 1.1 [m^2]

## ASSUMPTIONS

When the Baja car is attempting to climb a hill, we assumed that the CVT would try to maximize torque through the engine. Therefore, the engine would run at 2600 RPM, producing 13.75 ft-lbs of torque at 6.7 hp.

Next, we assumed that the transmission was perfectly efficient. We feel this is valid as the components in the transmission, if well designed would have high efficiencies of about 95%.

## ANALYSIS

We modeled the car as a particle moving up a hill, as we felt the effects of it as a rigid body were negligible.

Since we know the output torque of the engine, we can feed that forward to find out the torque being transmitted to the wheels. From the torque on the wheels, we get a traction, $F_T$, which accelerates the car. From this driving force, we can create the governing differential equation for the car,

$$m\ddot{x} = F_T - F_D - W\sin\theta$$

where x is in the direction of motion for the car.
The drag force, $F_D$ is given by

$$F_D = \frac{1}{2}C_d\rho A v^2$$

We first verified the results by setting Θ to 0° , and ran the simulation, Bajacar.m. As Figure 3 shows, the car would run at 30 miles per hour on a flat incline, which is consistent with observations from pervious Baja vehicles.

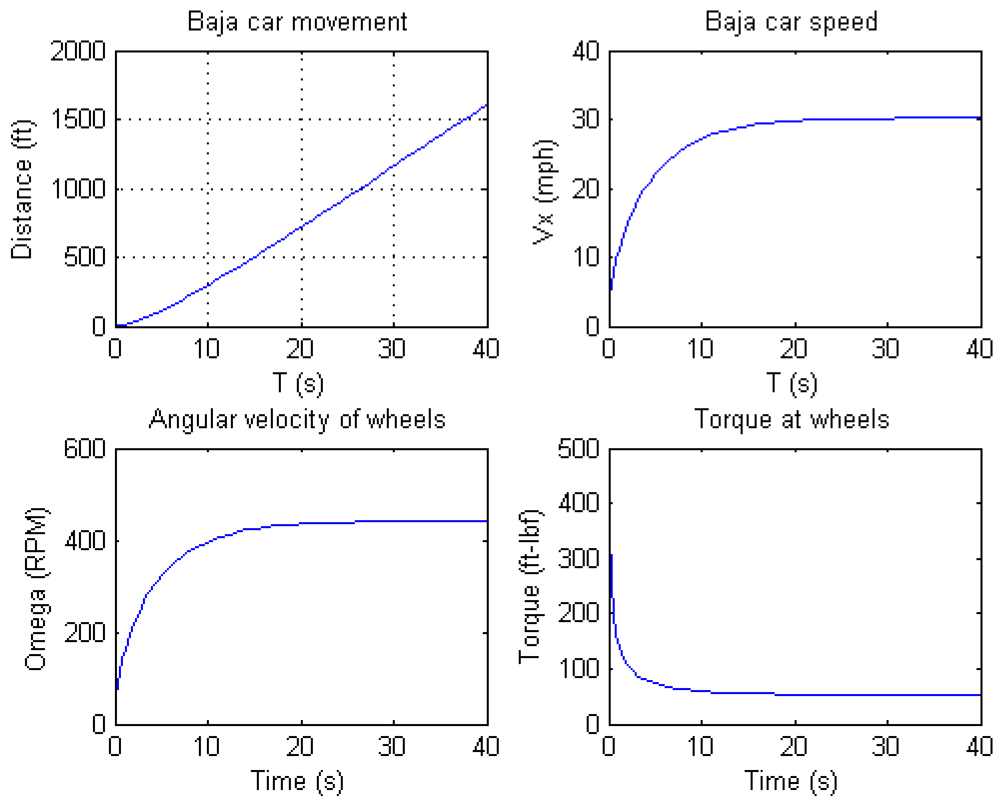



**Figure 3: Results for the car dynamics on a flat plane.**

Next, we changed the incline to 30°. The top speed of the car dropped from 30 MPH to 8 MPH and the output torque at the wheels increased from 53.1 ft-lbf to 298 ft-lbf. These results are shown in Figure 4.

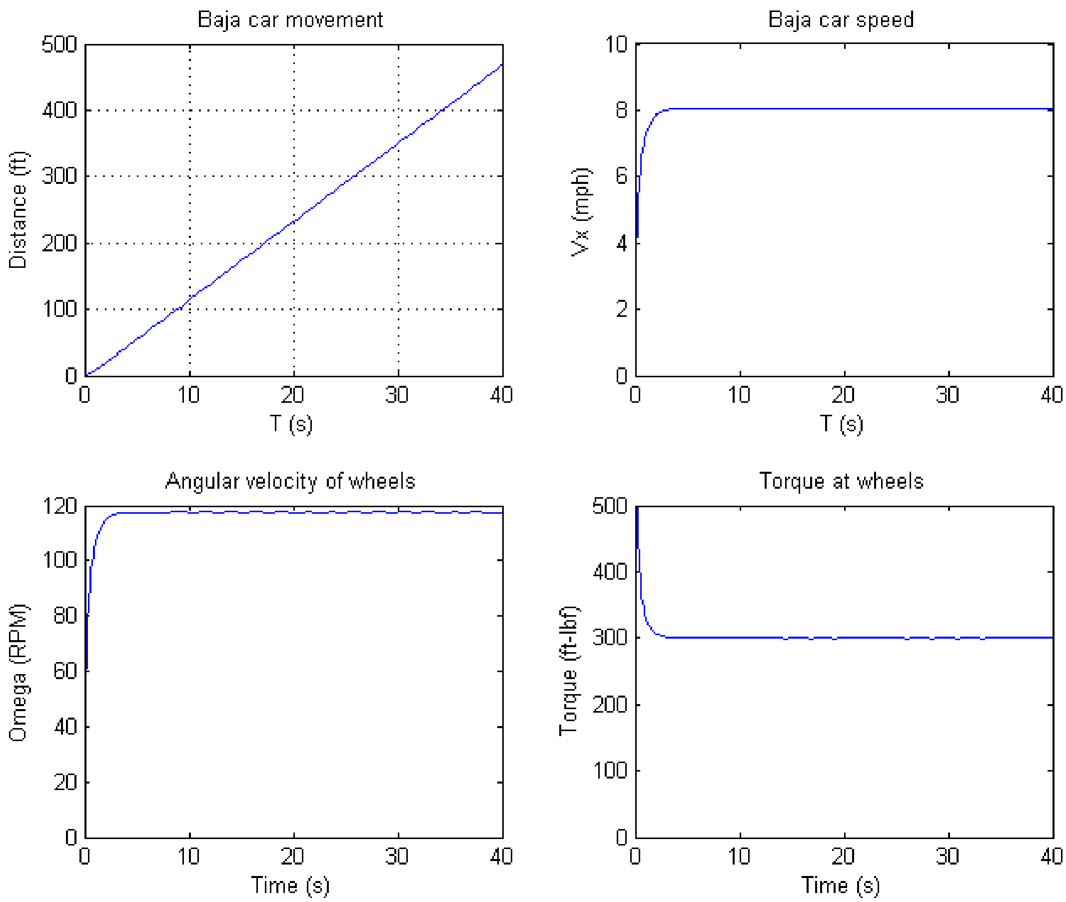



**Figure 4: The drawing Results on a 30° incline.**

Taking these two cases as bounds for the high and low gear setting of the gearbox, we can determine the gear ratios for the gearbox.

In high gear, the total reduction from the engine to the wheels is:

$$\frac{Output\ torque\ of\ wheel}{Output\ torque\ at\ engine} = \frac{53.13\ ft \cdot lbf}{13.75 ft \cdot lbf} = 3.86$$

Since the CVT has a minimum reduction of 0.43:1 and the differential has a reduction of 3:1, the low gear should have a reduction of

$$3.86 \times \frac{0.43}{1} \times \frac{3.23077}{1} = 5.37$$

Similarly, in low gear, the total reduction from the engine to the wheels is:

$$\frac{Output\ torque\ of\ wheel}{Output\ torque\ at\ engine} = \frac{298\ ft \cdot lbf}{13.75 ft \cdot lbf} = 21.6$$

Therefore, the high gear should have a reduction of

$$21.6 \times \frac{0.43}{1} \times \frac{3.23077}{1} = 30.1$$

This is a difficult gear ratio to implement as it would require multiple gear reductions to achieve it. Therefore, we are going to aim for the greatest gear reduction we can fit geometrically.

## CONCLUSION

The gearbox would require a high gear ratio of about 5:1 and a low gear setting of 8.75:1

# TRANSMISSION DESIGN

## OVERALL DESIGN

We based our design off of a sequentially shifting motorcycle gearbox. The gearbox has high, low, and reverse settings which can be selected sequentially by rotating a barrel cam.

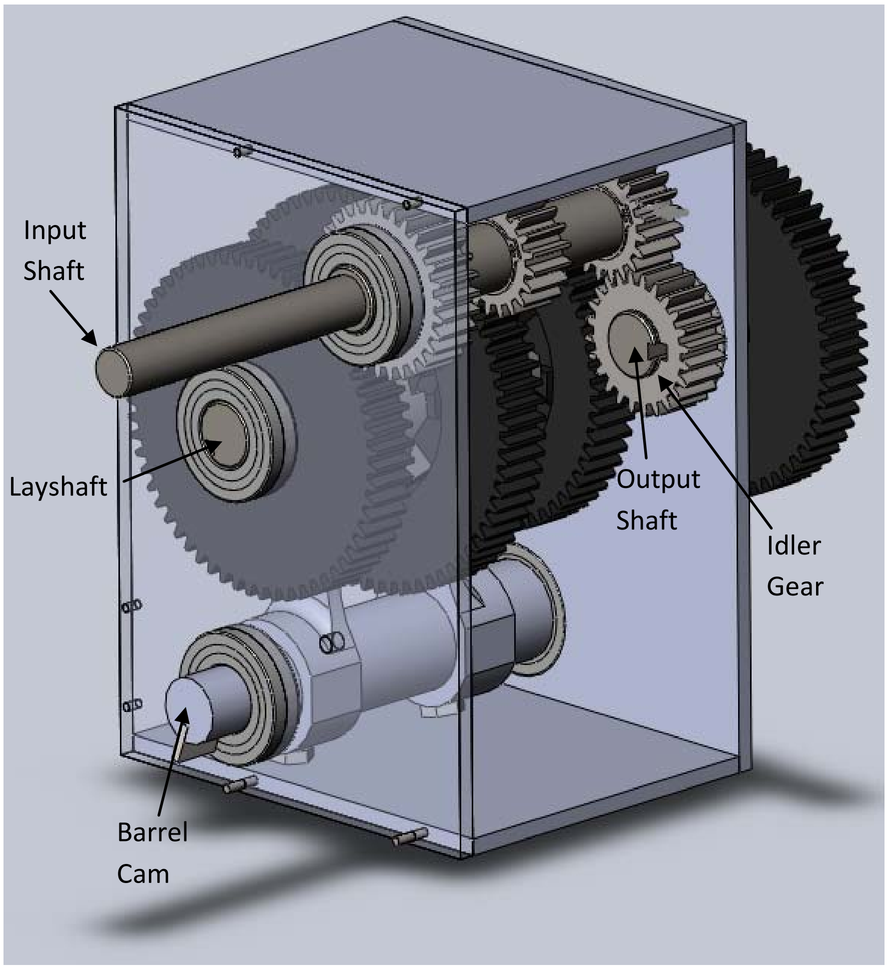



**Figure 5: An isometric view of the gearbox assembly. The gearbox has a barrel cam and three shafts which hold gears. The input shaft holds three pinion gears which are keyed to the shaft. The layshaft holds three spur gears which are free to spin on the shaft, and a pinion gear (not shown in this figure) which is keyed to the shaft. On the output shaft sits an idler gear which is free to spin on the shaft and connects the third pinion gear on the input shaft to the layshaft. Also on the output shaft is a large gear (partially shown) which is keyed to the output shaft.**

The gearbox has three shafts which hold gears: the input shaft, layshaft, and output shaft. When the input shaft is turning, all of the gears inside of the gearbox housing are also turning but since each pair of gears within the housing has a different gear ratio, the angular velocity of the gears on the layshaft are all different. The gears on the layshaft are not keyed to the shaft, and therefore do not transmit any torque. To select a gear ratio, the barrel cam is rotated causing a shift fork to engage one of the spiders which are keyed to the layshaft.

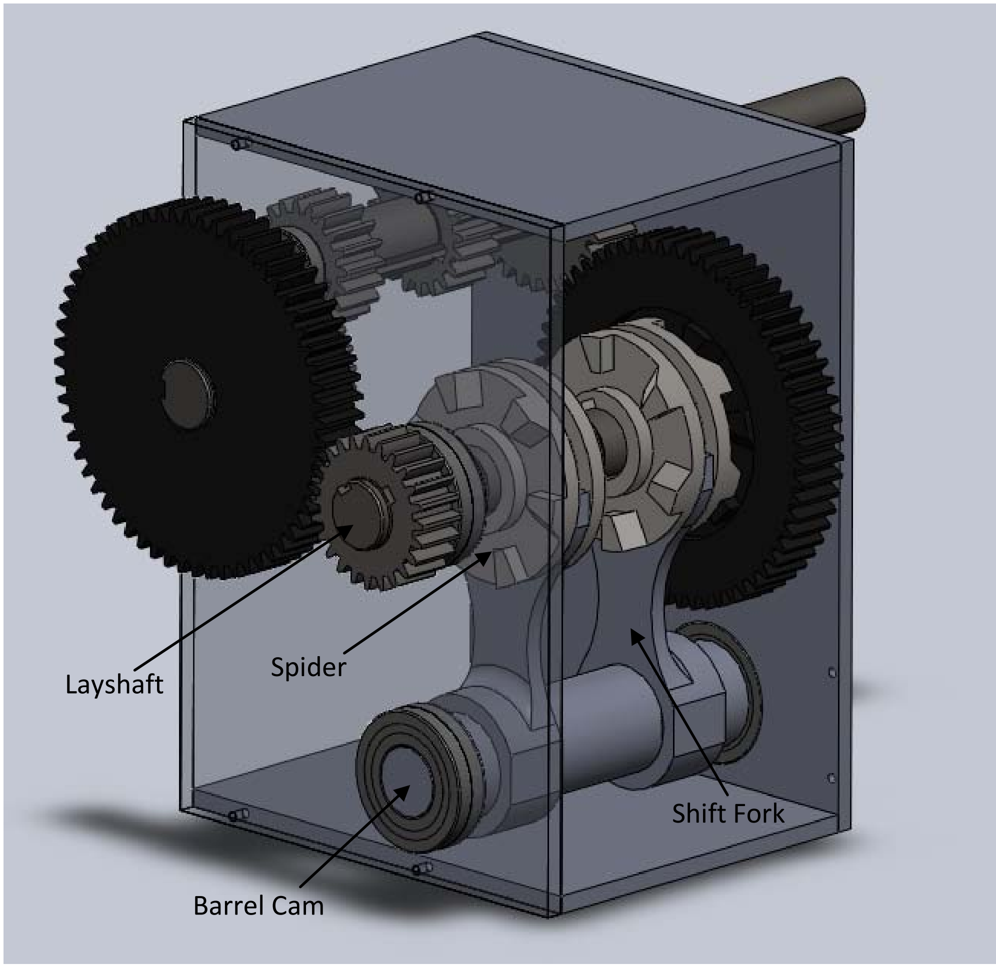



**Figure 6: Another isometric view of the gearbox. In this view two of the gears have been hidden on the layshaft to expose the spiders. The pinion gear on the layshaft can be seen in this view, and the larger gear on the output shaft is also fully visible. The two gears on the outside of the housing act as a final gear reduction for the rest of the gearbox. The spiders are keyed to the layshaft, but are constrained axially by the shift forks. The shift forks can be displaced by rotating the barrel cam. The spiders and gears on the layshaft have matching sets of dogteeth, which allow the spider to transmit torque from the gears to the layshaft.**

The spiders and gears on the layshaft have matching sets of dogteeth. When the barrel cam is rotated, the shift fork forces the spider to engage the dog teeth on one of the layshaft gears. Torque can then be transmitted from the input shaft, through one of the input shaft pinion gears, to one of the layshaft gears, through one of the spiders, to the layshaft, to the pinion gear on the layshaft, through the keyed gear on the output shaft, and finally to the output shaft.

There are four possible positions for the spiders:

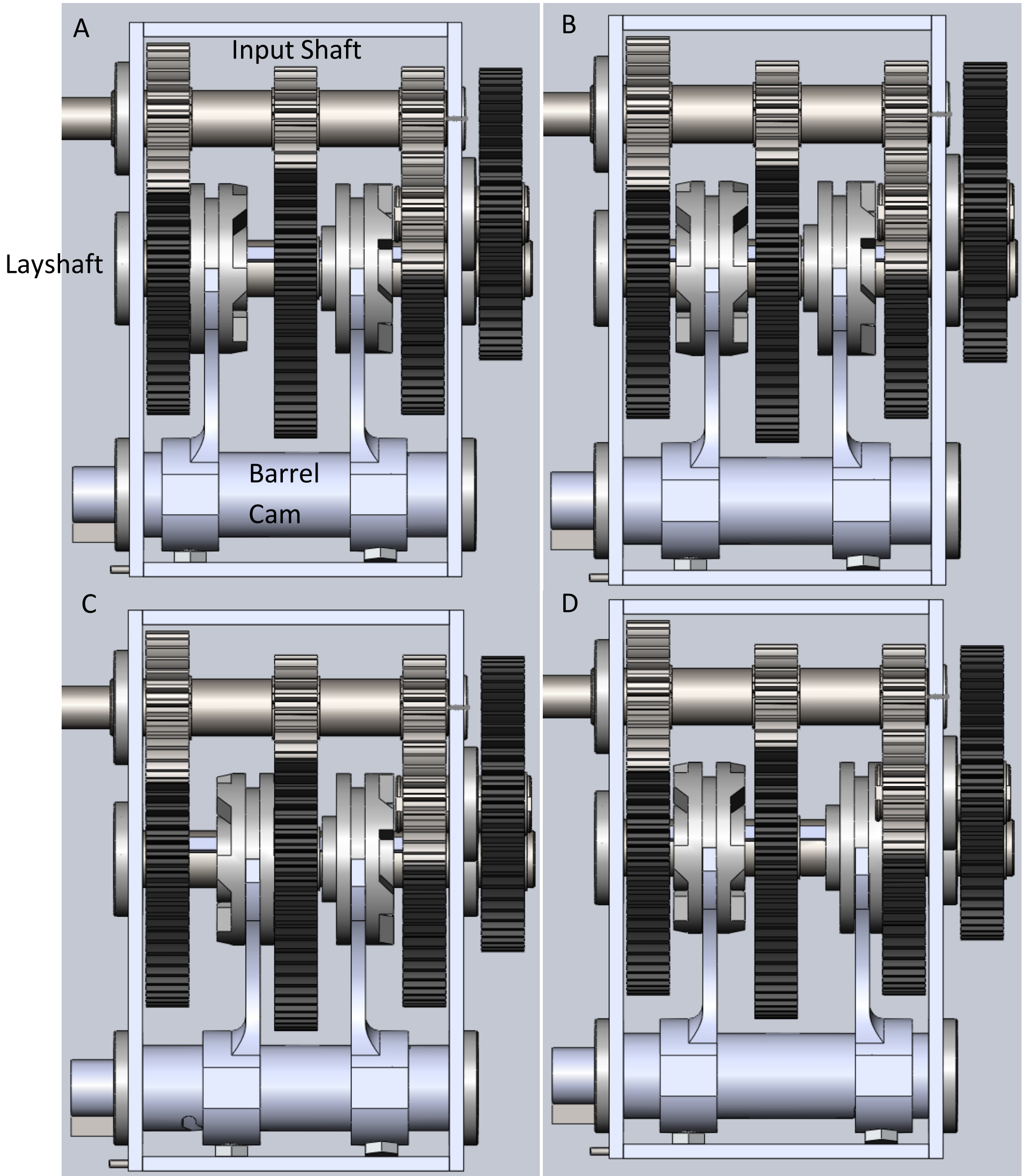



**Figure 7: The four possible positions of the spiders. A) The high gear position B) The neutral position C) The low gear position D) The reverse gear position. As the barrel cam rotates, the gearbox passes through high gear, neutral, low gear, a second neutral, and reverse gear sequentially.**

As the barrel cam rotates, the spiders engage gears in the sequence high-neutral-low-neutral-reverse. The location of the spider changes which layshaft gear is transmitting torque to the output.

## GEAR STRESS ANALYSIS / GEAR SIZING

### OBJECTIVE:

To validate that the gears chosen because of ratio requirements and geometric requirements will be able to withstand the various operational stresses.

### ASSUMPTIONS:

- Low and reverse will be used very briefly, and at low speeds. Analysis will focus on the high gears.

### ANALYSIS

When choosing gear sizes, we needed to ensure that we were choosing sufficiently large enough gears to sustain the main stresses (shear, surface fatigue, and bending) that the gears would experience.

#### *Gear tooth bending stress*

In order to ensure our gear teeth would be able to withstand bending stresses, we compared the predicted bending stress experienced by the teeth with the fatigue strength. The fatigue strength for $10^6$ cycle life is given by:

$$S_n = S'_n C_L C_G C_S k_r k_t k_{ms}$$ [1]

For our application, $C_L$ = 1 (bending loads), $C_G$=1 (for gears with pitch >5), $k_r$=.814 (99% reliability), $k_{ms}$=1.4 for input and output gears.

2% Carbon Steel has a yield strength of 80ksi, which corresponds to an R.R. Moore endurance limit of 40ksi, and (assuming machined steel) $C_s$=.775. With these values, the fatigue strength of the steel gears is 35.33ksi.

Cast iron of ASTM Class 30[2] (as used by Boston Gears) has a tensile strength of 31ksi, corresponding to an R.R Moore endurance limit as 12.4ksi. The cast iron also has $C_s$=1.121 (from $C_s=aS_u^{\,b}$ with a=2.7ksi, b=-.256)[3]. With these values, the fatigue strength of the cast iron gears is 15.84ksi.

We calculated the effective fatigue stress, $\sigma$, using the Lewis stress equation embellished with other factors reflecting operational and geometric considerations:

$$\sigma = \frac{F_t P}{bJ} K_v K_o K_m$$

The tangential force on the input pinion and lay shaft gear, $F_t$, can be found using $F_t$=T/r where T is the torque on the shaft due to the CVT, and r is the radius of the pinion. With a max torque T=13.75 ft-lbs, a reduction of 3 from the CVT, and a pinion radius of 1.25", the force on the teeth is 396lbs. The pitch line velocity is 566.9 ft/min (assuming a speed reduction of 3), which yields a $K_v$ of 1.5 (assuming a precision,

---

[1] Robert Juvinall and Kurt Marshek, *Fundamentals of Machine Component Design, ed.4* p608-619

[2] Boston Gear Manual http://www.bostongear.com/litportal/pdfs/P-1482%20ALL%20PAGESsm.pdf p150

[3] Dan B. Marghitu, *Mechanical Engineer's Handbook,* p174.

shaved and ground gear). We took $K_o$ = 1.5 (assuming light shock on the gears), $K_m$=1.3, b=.75, and P=12. For the steel pinion (30 teeth), J=.38, and for the cast iron gear (60 teeth), J=.43.

With these values, the predicted bending stress on the steel gear is 48.8ksi. For the iron gears, the predicted stress is 43.1ksi.

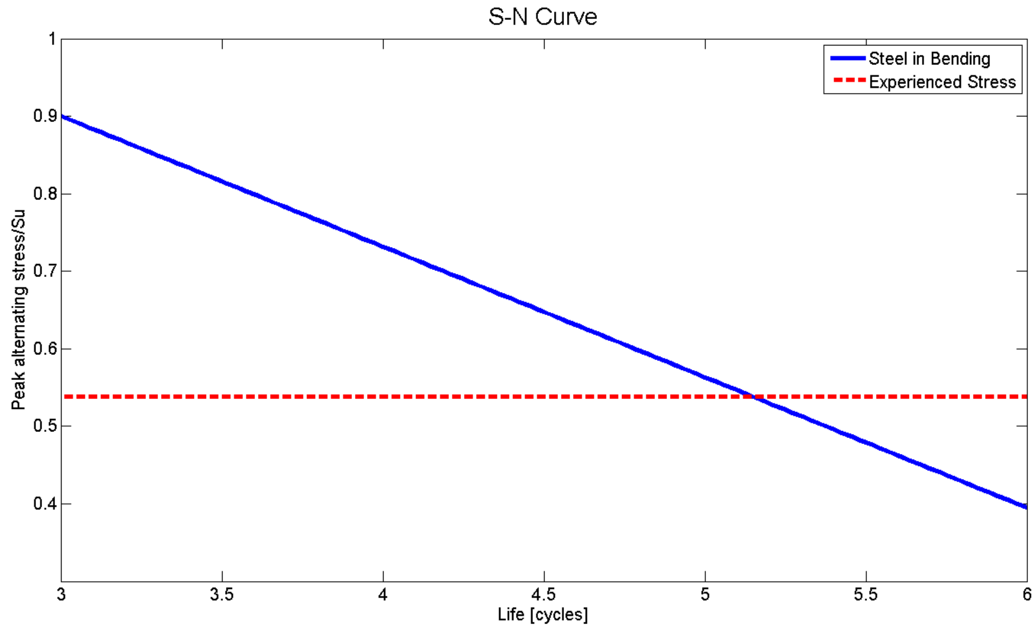



**Figure 8: This figure shows the S-N curve for our steel gears in bending. The red dotted line denotes the experienced stress of the system.**

The normalized steel gear stress ($\frac{\sigma}{S_u}$) is .538. Using the S-N diagram above, we find that the steel gear will last for $10^{5.18}$ cycles. At 2600 RPM, the gear should last 18.8 minutes, or .313 hours. Based on previous experience with the Baja team, we are confident this gear will last longer than the determined life and further analysis should be performed to find errors in our calculation.

### *Gear surface fatigue analysis*

Our gears will also face surface fatigue stresses. The stress due to surface fatigue is found by:

$$\sigma_H = C_P\sqrt{\frac{F_t}{bd_pI}K_V K_o K_m} \quad ^4$$

Where $K_v$ , $K_o$ , $K_m$ , $F_t$ , and b are defined as in the adjusted Lewis stress equation. $C_P$ is 2000, based on the steel pinion and cast iron gear. $d_P$ is the diameter of the pinion, and I is defined as

$$I = \frac{sin\phi\ cos\phi\ R}{2\ (R+1)}$$

Where Φ is the pressure angle (14.5) and R is the ratio of the gear and pinion diameters (2). This gives us a surface fatigue stress of about 178ksi. The surface fatigue strength, $S_H$ based on

$$S_H = S_{fe}C_{Li}C_R$$

[4] Robert Juvinall and Kurt Marshek, *Fundamentals of Machine Component Design, ed.4* p622-625

is 48.6ksi. The experienced fatigue stresses are much greater than the surface fatigue strength. In order to confidently use these gears in the Baja car, we must either surface treat the material to improve the strength or use higher pitch gears. For the analysis we assumed that the car would be running at maximum torque for the entire lifetime, which is unrealistic for the car.

### *Gear tooth shear analysis*

The gear tooth shear stress is approximated by

$$\tau = \frac{F}{A}$$

where F is the force on the teeth, and A is the area of contact of the tooth. For the area, we used circular pitch, p, divided by 2 (to find the gear thickness) times the gear width (.75”). This yields an area of .098 inches$^2$. Based on the 411.84lbs force, the shear stress experienced by the gear is 4.2ksi. The shear strength of 2% Carbon steel is about 31.9ksi, which corresponds to a factor of safety of 7.6.

## CONCLUSION

Previously we had chosen pitch 16 gears, but the analysis showed that the gears would fail under the operational stresses. With this understanding, we increased size to pitch 12 gears.

For the gearbox, we required a high ratio of 5:1, and ratio greater than 5 for the low. With the pitch 12 gears, we wanted to use the smallest possible gear for the low pinion that would fit geometrically with the high gear set. Using these geometry and weight considerations, we chose the1.67” pitch diameter gear and a 5.83” pitch diameter gear for the low gear (corresponding to a ratio of 8.75:1) and a 2.5” pitch diameter pinion and a 5” gear for the high ratio.

Although initial analysis indicated a pitch 12 gearset would be appropriate, a refined stress analysis indicated that the pitch 12 gear set would be problematic for the gearbox as maximum bending and surface fatigue strengths are exceeded. The gearset for the Baja car should either be redesigned so multiple gears are used for the 5:1 ratio, or redesigned with stronger materials such as hardened steel or different style gears such as helical gears though due to time and part availability we continued design assuming the use of pitch 12 gears.

## BEARING SELECTION

### OBJECTIVE:

To select bearings for the output, lay, and input shafts.

### ASSUMPTIONS:

- The gearbox must have a 100 hour life at 8900 RPM (53,400,000 revs). This is a conservative estimate assuming maximum possible speed. (3800 RPM of engine with .43 overdrive from the CVT).
- Loads analysis neglects forces on the bearings caused by the spiders on the layshaft.
- Forces on the gearbox shafts are all aligned.

## ANALYSIS:

In order to select our bearings, we first had to calculate the maximum radial forces on our bearings. We began by creating a MATLAB script that takes the gear sizes and distances between the gears and outputs the radial loads on all of the bearings in our system. The code that we used for this calculation is in the Appendix. Figure 9 shows the FBDs for the input and lay shaft, as well as some of the naming conventions used in the calculations.

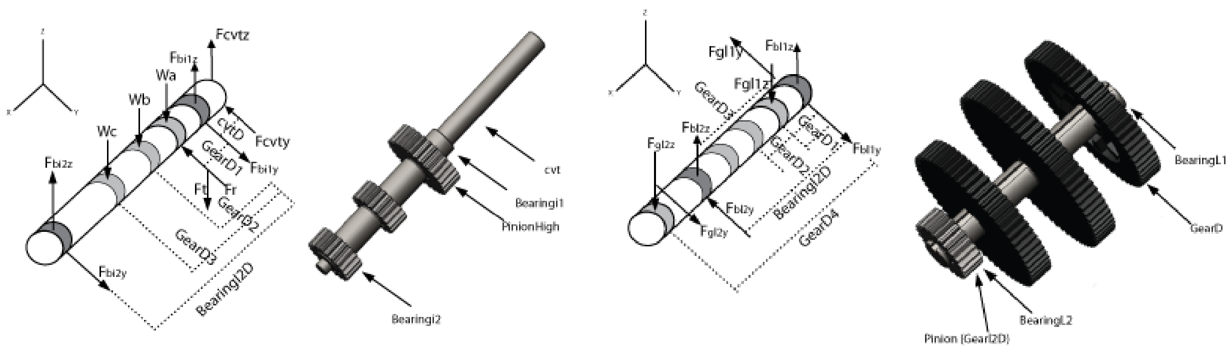



Figure 9: The input shaft and the lay shaft. These two diagrams show the free body diagrams of the input and lay shaft (left and right). These forces will be used to find the reaction forces at the bearings.

Figure 10 shows the naming convention that we used to define our bearings and gears.

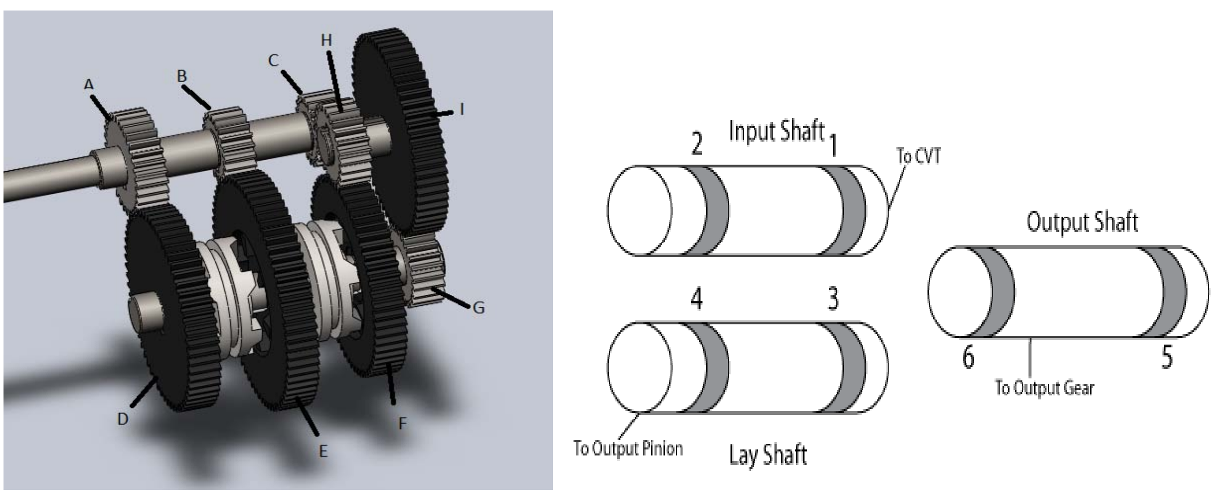



Figure 10: These figures show the naming convention used for the gears (left) and bearings (right).

Below, Table 1 shows the gears' diameters and weights.

| Gear: | A | B | C | D | E | F | G | H | I |
|---|---|---|---|---|---|---|---|---|---|
| Weight (lbs) | .85 | .27 | .27 | 3.21 | 4.6 | 3.21 | .48 | .48 | 3.63 |
| Diameter (inches) | 2.5 | 1.67 | 1.67 | 5 | 5.83 | 5 | 2 | 3 | 5 |

Table 1: This table shows the weights and diameters of the all of the gears used in the gearbox.

We found the forces on the bearings by using the equilibrium equations for the forces and moments in the y and z directions. Using $F_r = F_T \tan \emptyset$ where $F_r$ is the radial force on the shaft (transmitted through the gear), $F_T$ is the tangential force on the shaft, and Ø is the pressure angle of the gears.

| Bearing: | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|
| High | 312 | 97 | 412 | 1318 | 591 | 592 |
| Low | 91 | 278 | 417.0 | 1171 | 511 | 511 |
| Reverse | 214 | 472 | 395 | 1370 | 1683 | 570 |

Table 2: This table shows the resulting force magnitudes (in lbs) in all of the bearings in the system when in high, low, and reverse. Note that bearing 2 is a .5" bore bearing.

To calculate the load life, we found the load life $C_{10:}$

$$C_{10} = F_D \left( \frac{L_D * n_D}{L_R * n_R} \right)^{1/3}$$

Using this formula, the necessary load life for each shaft varies because speed affects the desired life of our bearings. In order to make sure none of the bearings will fail, we considered each shaft, looking at the maximum load each shaft will experience. Specifically, we looked at bearing 2 in reverse, bearing 4 in low, and bearing 5 in reverse. Because bearing 2 is a ½" inner bore bearing, the load life is different. This is reflected in the adjusted speed rating. For each of these cases, the values of interest were:

| Variable: | $F_D$ (lbs) | $L_D$ (revs) | $n_D$ (rpm) | $L_R$ (revs) | $n_R$ (rpm) | C10 (lbs) |
|---|---|---|---|---|---|---|
| B2 Reverse | 472 | 9804000 | 1634 | 1000000 | 30000 | 383 |
| B4 Low | 1370 | 2801100 | 466.86 | 1000000 | 14000 | 621 |
| B5 Reverse | 1683 | 1307200 | 217.87 | 1000000 | 14000 | 459 |

Table 3: This table shows the desired variables and the bearing values that were used in the load life analysis of the bearings.

We were unable to find the information necessary for our McMaster bearings, so we compared our C10 value to similarly constructed (materials and dimension) bearings with the same trade number as our bearings from NTN Bower. The ½" bearing has a C10 value of 1150lbs, and the 1" bearing had a C10 value of 2,260 lbs.

Because we did not have our specific bearing's C10 value, we assumed a maximum variability of 25% between the NTN Bower and McMaster bearings (translating to a maximum C10 value of 863.5lbs and 1695 lbs). This corresponds to a factor of safety of 2.3 for the ½" bearing and 3.2 for the 1"bearing.

## SPIDER DESIGN

### OBJECTIVE:

Determine the dimensions for the "spider" used to couple the gears on the driveshaft to the layshaft.

The crucial dimensions are:

- Diameter (D) of the spider
- Thickness of the teeth (t)
- Height of the teeth (h)

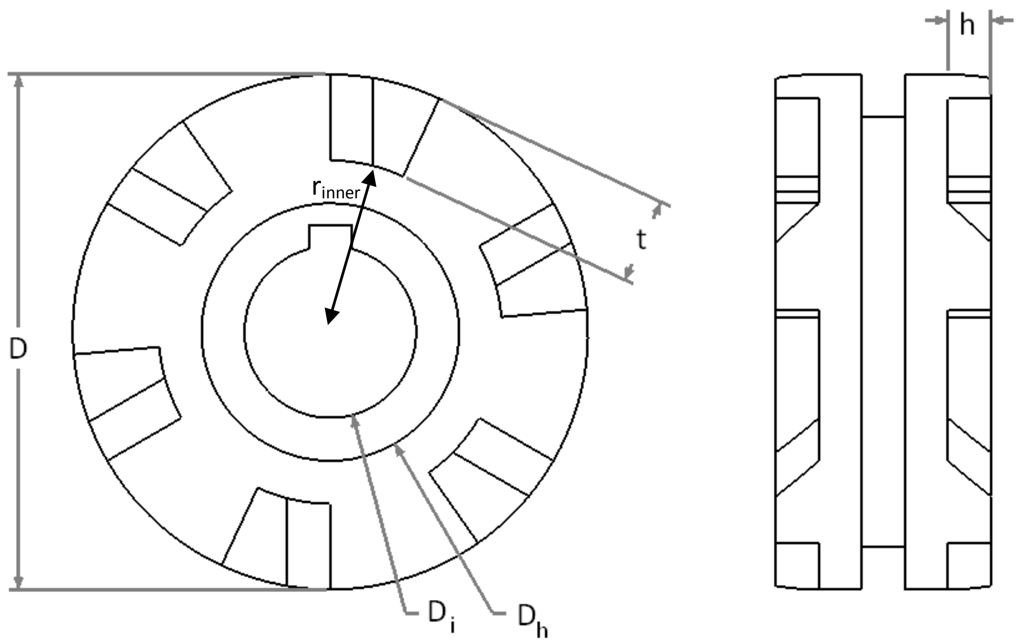



**Figure 11: The crucial dimensions considered in the spider loading analysis are defined.**

### ASSUMPTIONS:

- The spider is made out of AISI 1018 plain carbon steel. This has been chosen as this is the most commonly available steel. Also, we do not expect to exceed the maximum strength or stiffness of the AISI 1018 steel.
- The maximum torque transmitted through the spider is 200 ft-lbs (203 Nm). This comes from our previous calculation of the simplified gearbox.
- The radius of the shaft passing through the spider is 1". This is specified to be reasonable for passing loads through the gears.

### GIVEN PROPERTIES

- Ultimate strength, $S_u$ :49.5 ksi
- Yield strength, $S_y$ : 32 ksi
- Shear strength : 90 ksi

- Modulus of elasticity: 30 Mpsi
- Modulus of Rigidity: 11.5 Mpsi
- Torque transmitted through the spider, *T*: 200 ft-lbs

## ANALYSIS

### *Shear Stress*

Modeling the spider as a tube, we can determine the minimum diameter the spider needs to be.

Since:

$$\tau = \frac{\mathrm{Tr}}{\mathrm{J}}$$

Where

$$J = \frac{\pi}{2}(r^4 - r^4{}_{inner})$$

The outer radius of the spider is given by *r* while its inner radius is given by $r_{inner}$.

Rearranging the above two equations, we can get an expression for the maximum shear stress as a function of radius:

$$\tau = \frac{24Tr}{\pi(r^4 - r_{inner}^4)}$$

Refer to spider_shearstress.m in the Appendix for the calculations. The graph below shows that as the radius of the spider increases, the maximum shear stress experienced decreases. The minimum radius to get a factor of safety of 3 is .51", only .01" greater than the bore in the spider. We chose a radius of 1.5", giving us a factor of safety of 2357, instead. We chose this much larger radius for aesthetics and for machinability.

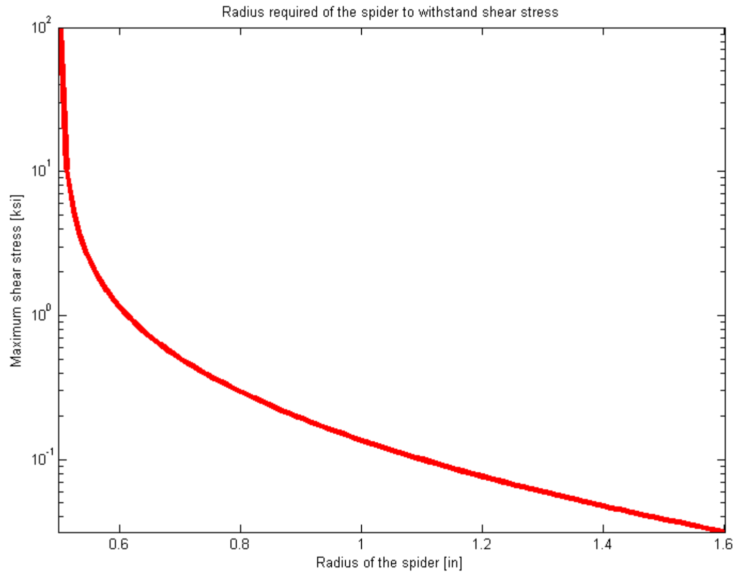



**Figure 12: Radius required of the spider based on the maximum permissible shear stress. The chosen radius, 1.5" gives a factor of safety of 50**

### *Bearing Stress on Teeth and Number of teeth*

To determine the required depth of the teeth, we calculated the bearing stress on an individual tooth. We know that the maximum torque transmitted through the spider is 200 ft-lbs. Therefore, at a radius of *r* out, the force that needs to be resisted is:

$$F_{total} = T/r$$

Let us assume that we will have 6 teeth and that the number of engaged teeth is 1 as a conservative estimate. We can find the maximum force a single tooth will experience by further assuming that the force will be equally distributed between n teeth.

$$F_{per\ tooth} = T/nr$$

The next assumption is that h, the teeth height is ¼ inch. This is arbitrary, as we needed to constrain the geometry.

The bearing stress limit of steel is 56 Mpsi. (Assumed 80-55-06 Grade ductile nodular Iron)

Therefore, the required teeth thickness, t is:

$$t = \frac{T}{nhr\tau_b}$$

Using the script, spider_bearing_stress.m, we determined that t has to be greater than 0.0853”. Again, we chose a much larger value (0.5”) for aesthetics, machinability, and to anticipate of other problems such as slippage and wear.

### *Key Way dimensions*

Since we are keying the spider to the shaft, we need to determine the dimensions of the keyway.

The torque that can be transmitted by key shear is the product of limiting stress, area and radius

$$T = \frac{0.58 S_y L d^2}{8}$$

Since annealed 1018 steel has a yield strength of 32 ksi, and the maximum torque through the key is 200 ft-lbs, we find that the length of the key is L=1.03 in. To increase the safety factor, we are using two keys, and going to a length of 1.25 inches..

### *Thickness of the spider*

Arbitrarily chosen based on ratios we found for linear bearings on McMaster. With more time, we would consider justifying this decision analytically.

## SELECTOR MECHANISM

### OBJECTIVE

Check the dimensions for the selector arm and barrel cam.

The crucial dimensions are:

- Width (w) of selector
- Thickness (t) of the selector
- Diameter (d) of dowel pin

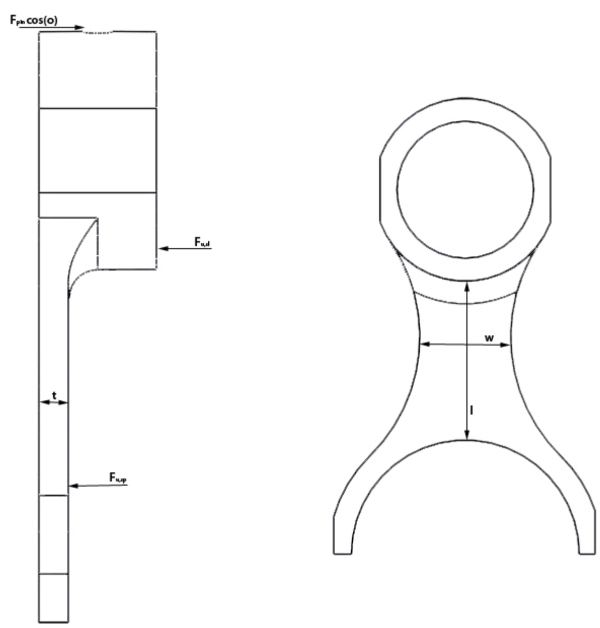



**Figure 13: The crucial dimension of the shift fork are defined. The shift fork is the interface between the barrel cam and the spiders.**

## ASSUMPTIONS:

- The selector is made of 6061 Aluminum and pin is made from a 3/8-16 bolt
- The forces on this system are derived from the friction between components
- The spider steel slides on a steel shaft without lubrication and the aluminum selector slides on a steel shaft without lubrication

## GIVEN PROPERTIES

- Young's modulus of selector, $E_{sl}$: 10,000 ksi
- Friction coefficient of spider on shaft, $\mu_{sp}$: .8
- Mass of spider, $m_{sp}$: 2.20 in
- Length of selector, $L_{sp}$: 1.75 in
- Acceleration of gravity, g: 386 in/$s^2$
- Mass of selector, $m_{sl}$: 0.22 lb
- Friction coefficient of selector on shaft, $\mu_{sl}$: .61
- Angle of contact between pin and barrel cam, θ: 45°
- Yield strength of dowel pin, σ: 21.76 ksi

## ANALYSIS

### *Bending Stress in Selector*

Modeling the selector as a rectangular beam fixed on one end the maximum friction force applied to other end we can determine the deflection of the selector $x_{sl}$.

$$x_{sl} = -\frac{F_{sl} L_{sp}^3}{3E_{sl} I}$$

Where

$$F_{sl} = \mu_{sp} m_{sp} g$$

$$I = \frac{w_{sl}^3 t_{sl}}{12}$$

Rearranging the equation, we can substitute in the values of $w_{sl}$ and $t_{sl}$ as the size they were machined to, 1 in and .25 in, respectively.

$$x_{sl} = -\frac{4F_{sl} L_{sp}^3}{E_{sl} w_{sl}^3 t_{sl}}$$

We find the deflection for our selector is equal to .000239 in which is well within the tolerance we need for proper function. In the future it would be helpful to find the actual force it takes to move the spider since small manufacturing misalignment adds additional forces to the system, in addition to the friction force.

### *Shear Stress of Dowel Pin*

To find the shear stress on the dowel pin we use the equation;

$$\tau_{pin} = \frac{F_{pin}}{A_{pin}}$$

Where

$$F_{pin} = \frac{\mu_{sp} m_{sp} g + \mu_{sl} m_{sl} g}{\sin(\theta)}$$

$$A_{pin} = \frac{\pi\, d_{pin}^2}{4}$$

Simplifying these equations

$$\tau_{pin} = \frac{4(\mu_{sp} m_{sp} g + \mu_{sl} m_{sl} g)}{\pi\, d_{pin}^2 \sin(\theta)}$$

Now if we input the pin diameter .235 in we find a shear force of 0.319 ksi. This yields a safety factor of 1000 which is sufficient for the design.

## CONCLUSION

This analysis shows that all of the components in the selector system are over built; however, from the empirical data we collected while testing the transmission there are larger forces on the system that are not being accounted for in the above analysis, these forces most likely result from imprecision in machining particularly the barrel cams non perpendicular sides. Moving forward, implementing empirical methods of measuring forces would help in developing a more accurate model.

# ACTUATOR MECHANISM

## OVERALL DESIGN

To rotate the barrel cam, a double ratcheting sequential shifter is used. The design was adapted from a US Patent filed by Renato Gavillucci in 2005 (US 6843149). Figure 1 shows the neutral position of the actuator. When the user applies a force and rotates the handle clockwise, pawl 1 will push pin A down forcing the cup to rotate. The force is transferred through pin A and compressing the spring plunger and allowing the ratchet wheel to rotate counter clockwise. The ratchet wheel rotates 60°, until pin D collides with the upper arch of pawl 2 at which point the spring plunger engages with the next detent. Torsion springs are attached to the pawls to ensure that they are always in contact with a pin.

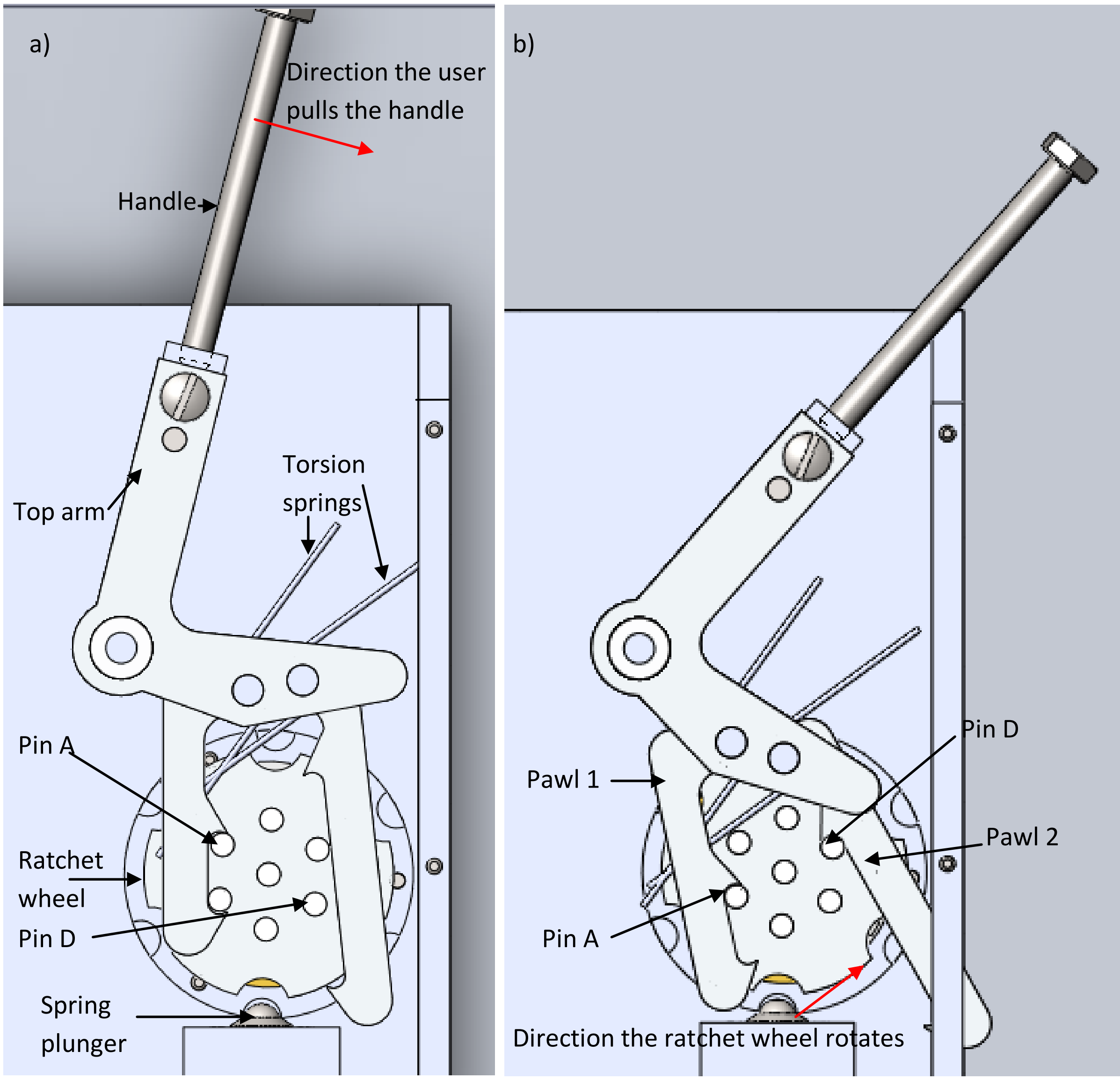



**Figure 14: a) The shift actuator in the neutral position. b) The shift actuator after being shifted counter clockwise.**

Similarly, if the user applies a force in the opposite direction, pawl 1 will pull a pin of the ratchet wheel until another pin collides with the lower arch of pawl 2. This will cause the ratchet wheel to rotate clockwise.

The spring plunger's role is to maintain the position of the ratchet wheel within discrete positions. To rotate the ratchet wheel, the user is primarily applying a force against the spring plunger to compress it out of its detents. When the wheel rotates to a new position, the spring plunger engages into the detents, locking the ratchet wheel in place and leaving the actuator free to move back to the neutral position.

The ratchet wheel is connected to the cup. The cup is a single assembly composed of four pieces of ¼" aluminum plate.

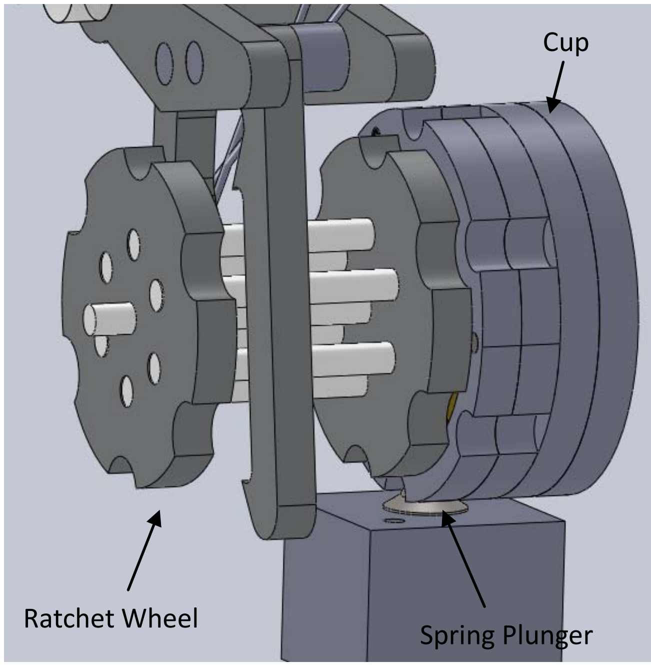



**Figure 15: The Ratchet Wheel is connected to the cup. The cup houses two compressions springs which interface with the barrel cam. These springs allow the ratchet wheel to rotate even when the barrel cam cannot. Also, the spring plunger engages with detents in the cup.**

The cup houses two compression springs which engage with the barrel cam.

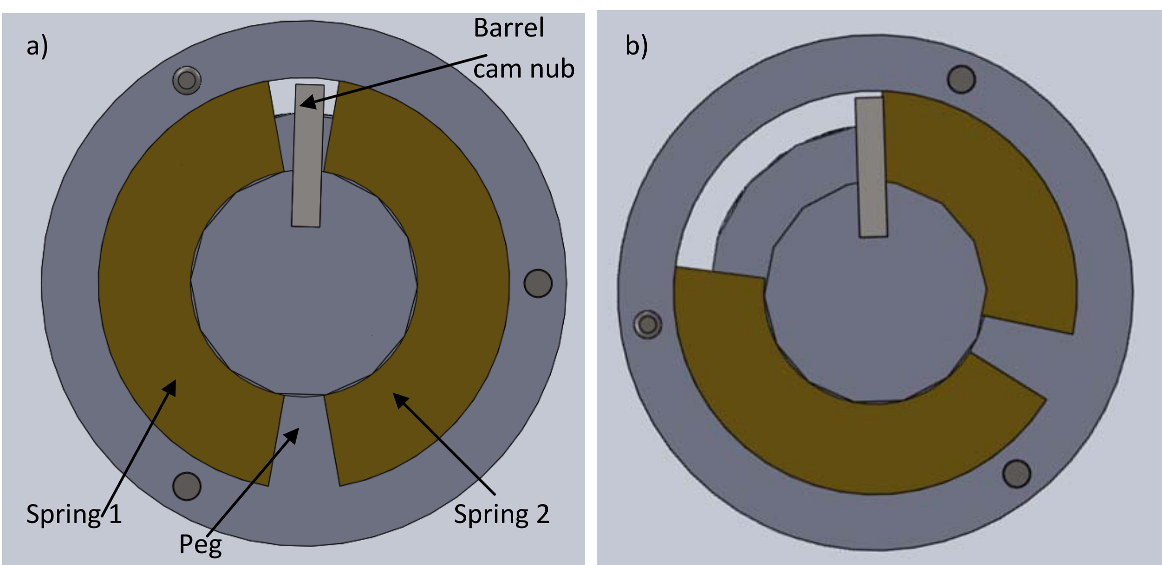



**Figure 16: a) The cup with compression springs which couple the ratchet wheel and the barrel cam. The circular island in the center of figure a is an extension of the barrel cam. A slot is cut into this island, and the barrel cam nub is inserted into it. b) If the barrel cam cannot rotate because a gear and spider cannot mesh, the cup will still be able to rotate and compress one of the springs. The compressed spring exerts a constant torque on the barrel cam which ensures that the spider and gear will mesh when possible.**

When the ratchet wheel rotates counter clockwise, it compresses compression spring 2. If the spider and gear are aligned correctly, there is very little torque preventing the barrel cam from rotating. The compression spring tries to release its stored elastic energy and rotate the barrel cam. However, if the spider and gear are not aligned correctly, the barrel cam will not rotate much, which will cause spring 2 to compress significantly. Once the input shaft starts rotating, the spider and meshing gear will align and the barrel cam will force the shift fork into position.

## CUP

### OBJECTIVE

The cup is designed to house the compression springs that help shift the barrel cam if the spider and the intended gear do not mesh. The springs need to be sized such that they provide enough torque to rotate the barrel cam, while being able to be compressed such that the ratchet wheel can rotate between shift positions.

### ASSUMPTIONS

- The size of the cup is greater than 1" (size of the barrel cam) and smaller than 3" in diameter. We want to conserve space, and make the cup no larger than it needs to be.
- The springs used are compression springs and only compress axially.
- The maximum the springs have to compress is when the cup rotates 60° with respect to the barrel cam because the spring plunger will hold the cup in positions at 60° intervals.
- Friction of the spring rubbing against the inside walls of the cup is negligible.
- Torque required to rotate the barrel cam is 10 in-lbs. This was an estimate made from rotating the barrel cam made in Gate 1. Measuring the torque was not too useful, as the difficulty in shifting lay when the pin of the barrel cam was binding against the slots in the barrel cam. Therefore, this conservative estimate was used to ensure that there would be sufficient torque to shift the barrel cam coming from the actuator.

### GIVEN PROPERTIES

- Torque from barrel cam, $T_{bc}$ : 10 [in-lbs]
- Diameter of barrel cam, $D_{bc}$: 1 [in]
- Angular displacement of the springs, Θ: 1.05 [radians] (60.0°)
- $\Theta_f$ = 1.67 [radians] (95.5°)
- $\Theta_i$ : 2.71 [radians] (155.5°)

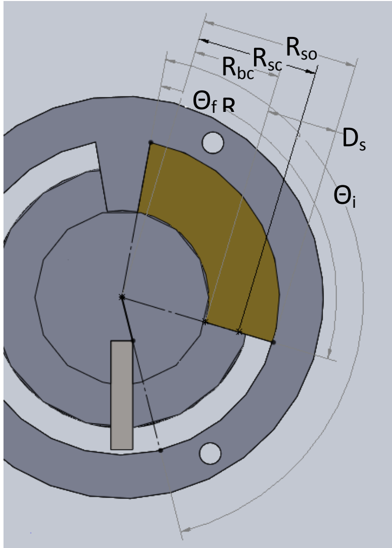



**Figure 17: Cup and barrel cam assembly to determine the compression spring needed to twist the barrel cam. The spring is the brass colored object in the figure. The other compression spring is not shown. $R_{bc}$ is the radius of the section of barrel cam which the cup interfaces with. The compression spring is bent into a partial torus, and has a radius of curvature $R_{sc}$. The outer radius of the spring is $R_{so}$. Note, we can find the diameter of the compression spring $D_s = R_{so} - R_{bc}$ and that the radius of curvature is defined as $R_{sc} = \frac{R_{so} + R_{bc}}{2}$.**

The torque from the barrel cam, $T_{bc}$, compresses the spring, changing the spring's angular displacement from Θi to Θf [radians]. From $D_{bc}$ and $D_s$, we know the spring has to have a minimum initial length of:

$$s_i = \Theta_i * R_{sc} = \Theta_i * (R_{bc} + {}^{D_s}/_2)$$

And a maximum final length of:

$$\Theta_f * R\_sc = \Theta_f * (R_{bc} + {}^{D_s}/_2)$$

The minimum force required of the compression spring at the final length is:

$$F_{s=} \frac{T_{bc}}{R_{sc}} = \frac{T_{bc}}{R_{bc} + D_s/2}$$

With these constraints, it is source a spring. Looking through McMaster-Carr's website, it was clear that we could not use die springs as they would not provide sufficient angular displacement. The spring that we chose was a 2" long steel compression spring with a 13/32" outer diameter, a compressed length of 1.18" and a compressed load of 9.4 [lbf] (Part#: 9657K37). Carrying through the equations above,

- $s_i = 1.91"$
- $s_f = 1.17"$
- $F_s = 15.22\ [lbf]$

The spring we chose almost matches our criteria except that it does not compress quite as far as we would like it to (1.18" compared to 1.17") and has two thirds the required force. We chose to use this spring regardless since we expected that any small amount of slop in our system would compensate for the difference in desired compression length, and since the torque required to shift the barrel cam was a rough estimate.

### COMMENTS

In retrospect, we should have used two torsion springs centered on an axle running through the cup to achieve desired behavior. The compression springs we found were not suited for this application, since they frequently would not have either the force or the compressed length required. Torsion springs are better suited for applications where a high angular displacement and force are needed.

## SPRING PLUNGER

### OBJECTIVE

The spring plunger needs to be able to exert enough force to resist the torque from the compression springs in the cup, and not cause the barrel cam to shift inadvertently. However, the spring plunger cannot be too strong or else shifting the barrel cam would become too difficult. Therefore, we need to find the minimum strength spring plunger that can resist rotation by the barrel cam.

### ASSUMPTIONS:

- Using a ball spring plunger instead of a standard spring plunger as we do not require a lot of travel.
- The spring plunger is mounted rigidly, and does not deform when the spring plunger compresses.
- The minimum diameter of the cup is 1.91” to fit in the barrel cam and the springs, and to leave a ¼” rim to put 1/8” dowel pins through.
- The spring plunger must keep the cup stationary on its own when torque from the barrel cam through the compression springs is applied to the cup.

### GIVEN PROPERTIES:

- Input torque from the barrel cam through the compression springs is : 10 lb-in
- Minimum distance of the spring plunger from the axis of rotation: .953”

## ANALYSIS

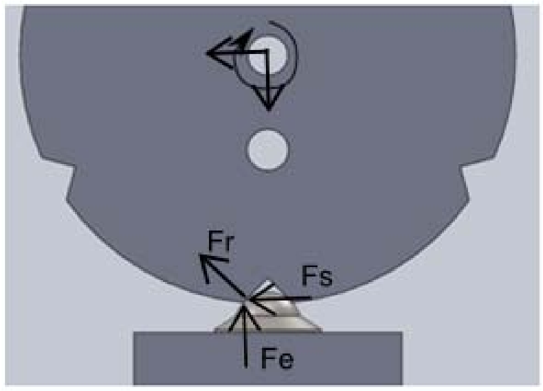



**Figure 18: A free body diagram of the forces acting on the ratchet wheel. The normal force from the spring plunger, $F_r$, can be resolved into two components, $F_s$ which resists the moment due to the compressions springs in the cup, and $F_e$ the end force on the spring plunger. As $F_e$ increases, the ball is depressed into the spring plunger and at a critical displacement the ratchet wheel will be allowed to turn freely.**

Taking moments about the center of the ratchet wheel we find,

$$F_s = \frac{T}{r} = 10.5\ [lbs]$$

We chose a detent angle of 90°, as it was easy to make. Therefore, the end force on the spring plunger was,

$$F_e = F_s \tan {}^{\theta}/_{2} = 10.5\ [lbs]$$

Spring plungers are rated by their initial and final end forces and since we wanted some room to adjust the mechanism we chose to let $F_e$ be the desired average of the initial and final end forces. Therefore, we chose spring plunger with a starting force of 5 [lbs] and an ending force of 14 [lbs] giving an average of 9.5 [lbs] (McMaster Part#: 84835A13).

## COMMENTS

While the spring plunger was slightly under the desired force, it was the closest available. We could have increased the detent angle to make the spring plunger meet the specifications or moved the spring plunger closer the cup such that more displacement was required to allow the ratchet wheel to move freely.

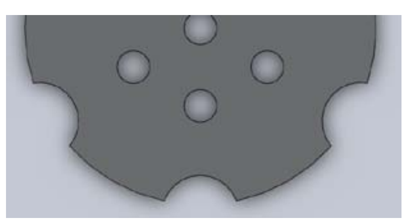


**Figure 19: The ratchet wheel we manufactured. When we designed the cup we did not realize that the indents should be counter sunk. This made the torque required to depress the spring plunger extremely high. To compensate for this, we filed down the circular indents into a counter sink.**

It can be shown that using semicircles as opposed to countersinks for the ratchet wheel detents increases the torque required to depress the plunger. To compensate for this we filed down the indents into countersinks.

## PAWLS

### OBJECTIVE

Determine the loads on the pawl, and verify if it would fail in some areas. There are two areas which are the most likely to fail. The first is the hook which engages with the ratchet wheel, and the second is the thin area surrounding the pin on the pawl.

### ASSUMPTIONS

- The pawls must exert a force to overcome the spring plunger and the compression spring at maximum compression. This is a conservative estimate, as this loading condition would only occur if the user tries to shift again when the spider has yet to shift.
- The pawls exert a force tangential to the cup. This is an assumption made to simplify computation, and is a conservative estimate. This force will be used throughout this subsection to give conservative estimates that will also simplify computation.
- Bearing forces will be small compared to the strength of the material and can be ignored in the analysis.

### GIVEN VARIABLES

- Input torque from the barrel cam through the compression springs is, $T_{bc}$ : 10 [in-lbs]
- The resistive force from the spring plunger is, $F_s$: 10.5 [lbs]
- Radius of the ratchet wheel pins from the center of the ratchet wheel, $r_p$: 0.44 [in]
- Outer radius of the ratchet wheel, $r_o$: .5 [in]
- The cross sectional area of the pawl through the line A-A is .08 [in^2]
- The cross sectional area of the pawls through the line B-B is .04 [in^2]
- Thickness of the pawls, top arm and ratchet wheel, t: 0.25 [in]

### ANALYSIS

First we will analyze the hook on the pawl and ensure it will not fail in shear. The pawl must exert a force on the rachet wheel which can overcome both the spring plunger and the torque produced by the compression springs in the cup.

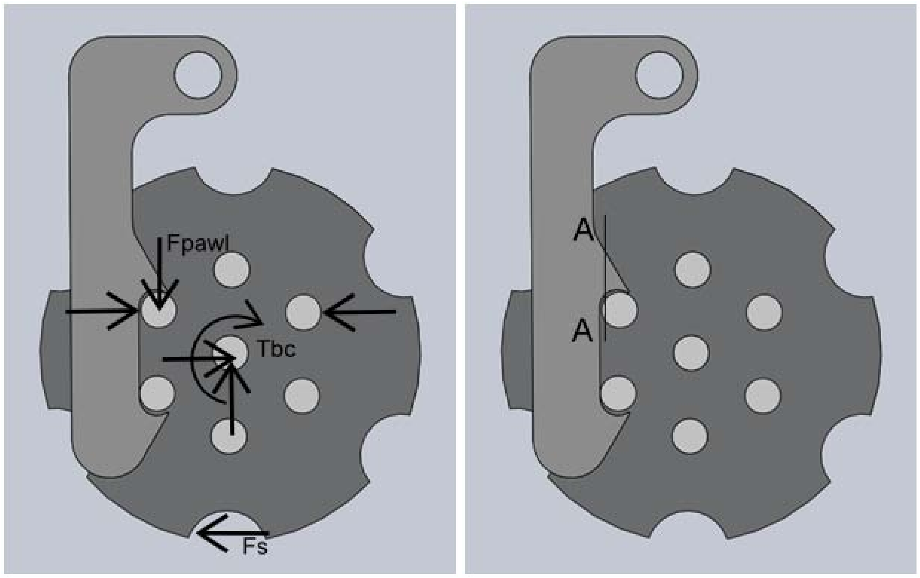



**Figure 20: A free body diagram of the forces acting on the ratchet wheel. For this analysis we need determine $F_{pawl}$, as such we will ignore the bearing forces and consider that the two torques produced by opposing pawls cancel each other. This leaves us with three salient forces, $F_{pawl}$ - the force which causes the ratchet wheel to rotate, $F_s$ - the resistive force from the spring plunger, and $T_{bc}$ - the torque from the barrel cam.**

Summing the moments about the center of the ratchet wheel,

$$F_{pawl} = \frac{T_{bc} + r_o F_s}{r_p} = 34.7\ [lbs]$$

Calculating the shear stress,

$$\tau = \frac{F_{pawl}}{A_{AA}} = 434\ [psi]$$

The yield strength of aluminum in shear is 4400 psi, demonstrating a factor of safety of about 10.

The other area where we might expect the pawl to fail is near the pin. The pin which holds the pawl must counter two forces during the shifting action. The pawl is connected to a torsion spring which ensures it is always engaged with the ratchet wheel. This torque must be countered by the ratchet wheel and in turn the pin. The pin must also resist the $F_{pawl}$we calculated earlier. We will consider both of these forces acting simultaneously on the pin and determine the maximum sizes torsion spring we can use to ensure the pawl does not fail in this area.

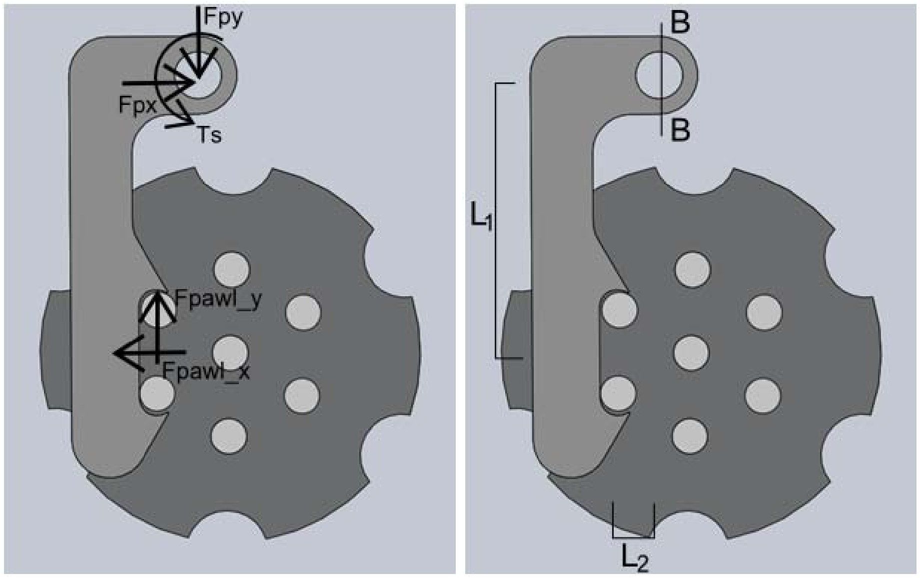



Figure 21 A free body diagram of the pawl. The area of interest is near the pin forces $F_{px}$ and $F_{py}$.

The magnitude of the bearing forces at the pin must be less than a certain value to prevent the pawl from shearing,

$\tau_{\max} = 4400\ [\text{psi}] = \frac{\text{F}_{\max}}{.04}$ $\qquad F_{max} = 176[lbs]$

Taking the moments about the pin,

$T_{ts} = L_1 F_{pawl_x} + L_2 F_{pawl_y} = 1.5\ F_{pawl_x} + .22\ (34.7) \quad [in\ lbs]$

Finding the maximum force on the pin as the sum of the components,

$\sqrt{F_{pawl_y}^2 + F_{pawl_x}^2} < F_{max} \qquad F_{pawl_x} < 172.6\ [lbs]$

Therefore the maximum torque the torsion spring for this pawl should produce is,

$$T_{ts} < 266.5\ [in\ lbs]$$

Searching McMaster-Carr, we found a torsion spring which fits the 1/4” pin we chose to use as the pin for the pawl (Part#: 9271K182). We chose a torsion spring with 180° deflection angle and maximum torque of 2.7 [in lbs]. This provides factor of safety of about 100.

## COMMENTS

This analysis only gives us an upper bound for the torsion spring we chose. Since the purpose of this torsion spring is only to make sure the pawl remains in contact with the ratchet wheel the lower bound is dependent on friction in the system. We were confident that the spring we chose would be able to overcome friction in the system, and therefore did not perform any analysis for the lower bound for the spring. Also we were limited to the torsion springs which we could purchase from McMaster since needed to have the appropriate diameter for our pin.

It should be noted that we used the same torsions springs (but with the opposite handedness) for the other pawl in the actuator. The second pawl encounters less stress that the pawl we analyzed in this section.

## HANDLE

### OBJECTIVE

Determine the force that is required by the user to move the actuator.

### ASSUMPTIONS

- There is no deformation in the handle and top arm. Since the pawl did not even yield and had a smaller cross sectional area, this assumption is valid.
- In the prototype, the handle is a long bolt that is connected to a block (handle shaft) that is connected to the top arms. This is a much weaker structure, as the long bolt will experience a very large force, which could shear the bolt. A later revision would space the bolts further apart to reduce the loads they experience. Since it was a late design decision which will not be carried forward, the handle is assumed to be an extension of the upper half of the top arm.

### GIVEN PROPERTIES

- Force from the pawls, $F_{pawls}$ : 34.7 [lbs]
- Distance between the pivot of pawl 1 and the top arm, $l_3$:i 1.07 [in]
- Distance between the pivot of top arm and point of application of force from the user, $l_4$: 6 [in]

### ANALYSIS

To find the required input torque, $T_{in}$, consider the free body diagram of the top arm and pawls.

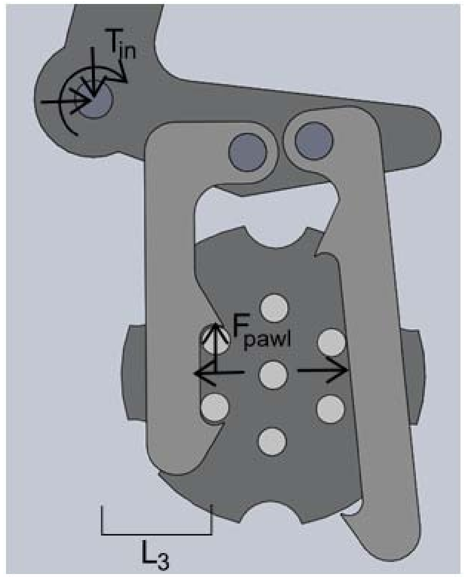



**Figure 22: A free body diagram of the top arm and pawls. $T_{in}$ is the input torque, and $F_{pawl}$ is the force which is required to turn the actuator. Assume that the horizontal forces on the pawls are negligible and cancel each other.**

$$T_{input} = F_{pawl} l_3 = 1.07 * 34.7 = 37.1\ [in\ lbs]$$

Assuming a lever arm of 6 inches for the top arm, the user needs to exert,

$$F_{user} = \frac{T_{input}}{l_4} = 6.2\ [lbs]$$

This seems reasonable.

### COMMENTS

The substantial force required suggests that it should not accidentally be actuated, which is a good safety feature for the gearbox.

## STRUCTURAL DESIGN

### MATERIAL SELECTION

Our material selection generally corresponded to available, machinable materials. We chose commercial gears from Boston Gear as they are inexpensive and readily available. Boston gear only offers the gears we chose in mild steel (for pinions) and cast iron (for large gears). *Fundamentals of Machine Component Design* (FMCD) references that steel is appropriate for pinions as it has relatively high strength while cast iron has less strength but greater surface durability. We computed guidelines for strength vs. density to determine how appropriate steel and cast iron are for gears. Applying the guideline shown below to Figure 3.12 in FMCD (Strength vs. Density), steel would be most appropriate (engineering ceramics are lighter, though more brittle, which is not appropriate for gears). Cast iron is less ideal in terms of strength to weight, however as mentioned previously it has greater surface durability. We recognize that this analysis is not ideal as a different material would likely be more appropriate with a variable pitch or outer diameter. However for simplicity we kept these constant. The spider analysis would be relatively similar as it uses teeth in bending to transmit torque.

$$m = \rho b \frac{\pi}{4} d^2 \qquad \sigma = \frac{F_t P}{bY} = \frac{F_t N}{bYd}$$

$$\frac{\sigma}{\rho} = \frac{\pi F_t N^2 P}{4m}$$

$$log\sigma = log\rho - C$$

Our shift fork must be strong enough to move the spider back and forth under bending stress, however it should be as light as possible. The load counteracted by the fork is small as it is only the friction between shaft and the spider. We assumed aluminum would be appropriate as it has a reasonable strength to weight ratio and a high strength is not required. It is also relatively easy to machine. According to FMCD, a guideline for minimum weight design against strength is $S^{2/3}/\rho$. Using Figure 3.12 in FMCD, it was apparent either titanium or aluminum would be most appropriate (engineering ceramics provided lighter weight). Similarly, forces on the barrel cam are relatively small and weight must be minimized, while a large volume is needed for the cam interface to function properly. Therefore aluminum would be most appropriate.

We chose steel for our layshaft because gear center to center distances must be maintained despite high loads. Based on the Figure 3.11 in FMCD, steel has the highest stiffness of available, machinable materials (it's surpassed by beryllium and tungsten, however these are not available).

We were concerned that in some cases we had similar metals sliding over each other without lubrication, yielding high amounts of friction. While we considered adding oilite bushings to every sliding interface, we determined that such a change would require excess machining time and would not provide significant improve in a demonstration prototype as our Gate 1 prototype operated successfully. Future redesign of this gearbox should include efforts to further reduce friction.

## GATE 1

### FABRICATION

We fabricated 9 components for our Gate 1 prototype over a time span of a week and a half. Each component was evaluated by the shop staff (Bruce Andruskiewicz) for manufacturability. Our materials were cast iron, mild carbon steel, and 6061 aluminum. All of these materials are relatively easy to machine, however we used carbide tools for most operations because of the higher available cutting speed to make the machining process more efficient. Also cast iron is an abrasive material and tends to wear out high speed steel cutting tools. The following chart lists the operations completed for our Gate 1 Prototype. The CNC milling operations incorporated programming time, producing tool paths that can be reused for Gate 2 improving the efficiency of our manufacturing system. All operations were completed by members of our team, except cutting grooves into the barrel cam, which required the 4 axis machining center.

| Operation | Machine | Machinist | Hours |
|---|---|---|---|
| **Layshaft** | | | |
| **Create Snap Ring Groove Tool** | Bench Grinder | Ryan | .25 |
| **Snap Ring Grooves/Shorten** | Lathe | David | 1.5 |
| **Second Keyway** | CNC Mill | David | 1.5 |
| **File Bearing Surfaces for Precise Fit** | Lathe | David | .25 |
| **Spider Interface on Cast Iron Gears** | | | |
| **Soft Jaws to hold gear** | CNC Mill | Ryan | .75 |
| **Interface Bosses/Snap Ring Recess** | CNC Mill | Ryan | 2.5 |
| **Steel Spider (Teeth on Single Side)** | | | |
| **Outer Diameter** | Lathe | Jay | 1.5 |
| **Inner Diameter/Shift Fork Groove** | Lathe | Ryan | 1 |
| **Soft Jaws to hold spider** | CNC Mill | Ryan | .75 |
| **Teeth** | CNC Mill | Ryan | 2 |
| **Keyway** | Broach | Ryan | .5 |
| **Key** | | | |
| **Size key** | Hack Saw/File | Eddie | .75 |
| **Shift Fork** | | | |
| **Create Profile** | CNC Mill | Ryan | 4 |
| **Pin Hole** | Manual Mill | Jay | .5 |
| **Barrel Cam** | | | |
| **Outer Diameter** | Lathe | Neil/Ryan | 1 |
| **Cam Surface** | 4-Axis Mill | Bruce | 2 |
| **Bearing Surfaces** | Lathe | Ryan | 1.5 |
| **Housing (3 Pieces)** | | | |
| **Outer Edges/Pin Holes** | Manual Mill | Jay | 2 |
| **Bearing Surfaces** | CNC Mill | David | 1 |
| **Press Bearings** | Arbor Press | Neil/Eddie | .25 |
| **Assembly** | | | |
| **General Assembly** | | Neil/Eddie | .5 |
| | | | |
| **Total** | | | 31 |

## PERFORMANCE

Our Gate 1 Prototype successfully moved a shift fork and spider linearly in order to both engage and disengage a gear, despite lacking any type of lubrication. This indicated our design was reasonable for continued analysis and improvement. Friction was relatively insignificant in the system though could be reduced by improving alignment for more precise manufacture. Slop in the barrel cam caused occasional jamming within the grooves, which was improved by modifying manufacture technique.

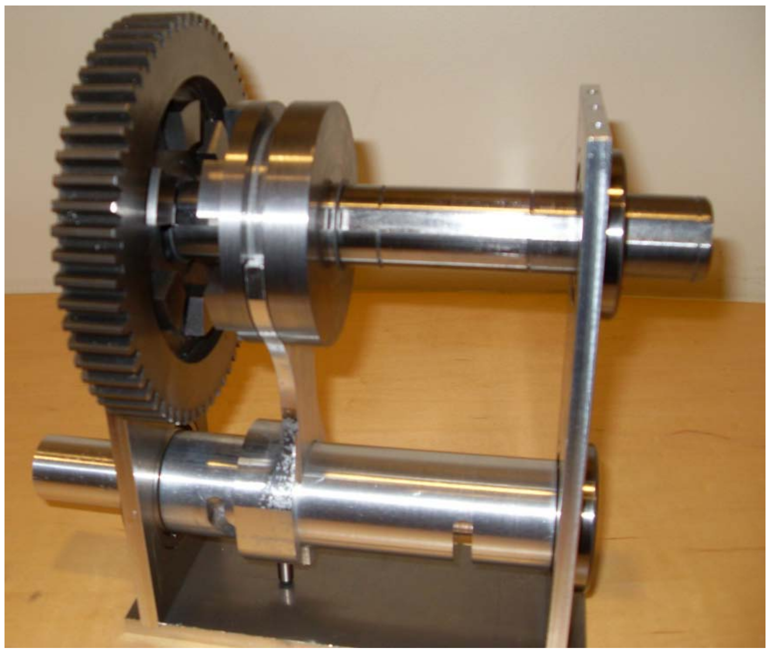


**Figure 23: Photograph of the gate 1 prototype. This prototype demonstrated the basic shifting concept with the capability to shift between all the gear settings, and engage and disengage a single gear.**

# GATE 2

## REDESIGN AND GATE 2 FABRICATION

Because our Gate 1 design worked well we reused four components from it and chose not to make any improvements to reduce friction (such as adding oilite bushings between layshaft gears and the layshaft) as these changes would significantly increase manufacturing time and would not provide a significantly greater learning experience. Instead we focused on designing and implementing an actuation system in a short time span by focusing on multiple prototype generation and redesign.

For Gate 2, we performed machining operations on 43 additional components (not including parts for actuator prototypes) over a week and a half. Our materials description from Gate 1 Fabrication still applies as we did not add new materials. We did however employ two more machining operations: lasercutting delrin parts for actuator prototyping (performed by David Gardner) and waterjet cutting aluminum parts for the final actuator (performed by Ryan Harris). Bruce Andruskiewicz produced a revised barrel cam for shifting in 60° increments on the 4-axis CNC machining center. Fortunately, we were able to reuse several programs for CNC milling operations decreasing machining time significantly.

| Operation | Machine | Machinist | Hours |
|---|---|---|---|
| **Input Shaft** | | | |
| **Snap Ring Grooves/Turn Down/Shorten** | Lathe | Ryan | 1.75 |
| **Output Shaft** | | | |
| **Snap Ring Grooves/Shorten** | Lathe | Ryan | .75 |
| **Spider Interface on Cast Iron Gear** | | | |
| **Interface Bosses/Snap Ring Recess** | CNC Mill | Ryan | 1.5 |
| **Steel Spiders (2)** | | | |
| **Cut to length** | Lathe | Ryan | .5 |
| **Teeth** | CNC Mill | Ryan | 2.5 |
| **Keyway** | Broach | Ryan | .5 |
| **Key (7)** | | | |
| **Size key** | Hack Saw/File | Ryan/Eddie | 1 |
| **Shift Fork** | | | |
| **Create Profile** | CNC Mill | Ryan | 2.5 |
| **Threaded Pin Hole** | Manual Mill | Neil | .5 |
| **Bolt-Pin (2)** | | | |
| **Turn Pin** | Lathe | ~~Ryan~~/Neil | 1 |
| **Shorten Bolt Head** | Sander | Neil | .25 |
| **Barrel Cam** | | | |
| **Outer Diameter** | Lathe | Neil | 1 |
| **Cam Grooves** | 4-Axis Mill | Bruce | 2 |
| **Bearing Surfaces** | Lathe | Ryan | .5 |
| **Actuator Key Pocket** | Manual Mill | David | .25 |
| **Housing (7 Pieces)** | | | |
| **Outer Edges/Pin Holes** | Manual Mill | Jay/Eddie/David/Ryan | 14 |
| **Bearing Surfaces** | CNC Mill | Ryan | 2.5 |
| **Press Bearings** | Arbor Press | Ryan/David | .25 |
| **Actuator Prototype** | | | |
| **Plastic Pieces** | Lasercutter | David | 2 |
| **Plastic Assembly** | | David/Jay | 3 |
| **Final Actuator** | | | |
| **Waterjet aluminum pieces** | Waterjet | Ryan | 2 |
| **Ream Holes** | Manual Mill/Drill | David/Jay | 5 |
| **Handle Block** | Manual Mill | Jay | 1 |
| **Spring-Plunger Block** | Manual Mill | Eddie | 1 |
| **Shorten 7 Dowel Pins** | Sander | Ryan | .75 |
| **Assembly** | | | |
| **General Assembly** | | Full Team | 1 |
| | | | |
| **Total** | | | 45.5 |

## GATE 2 PERFORMANCE

Our Gate 2 Prototype successfully demonstrated an actuation system capable of moving two shift forks in such a way that meshing occurs despite misalignment of the spider and gear. Two primary issues were evident: the high weight of the system ~25 lbs. and the inconsistency of our shifting mechanism.

The shifting mechanism would often fail to engage when under high friction, particularly when attempting to shift into reverse. Any actual implementation of this transmission in the baja car would require guaranteed shifts every time. Despite inadequacies in the shifting mechanism, this prototype represented success in our ability to machine precision components and in general feasibility of our concept with room for further analysis and improvement. Unfortunately this prototype is not capable of being run under a load and we were therefore not able to determine our success in sizing gears.

**Figure 24: Photograph of the gate 2 prototype. This prototype demonstrated the capability of a complex actuation system with a more complete representation of a baja gearbox. We successfully demonstrated a system to prevent meshing issues.**

## LESSONS LEARNED

### *Teamwork and Communication:*

Our team dynamic was derived from trust in each other's ability to perform quality mechanical design under tight time constraints. We would assign specific tasks with confidence that they would be completed and therefore we generally only had to meet once or twice weekly as a full team. To maximize output despite our emphasis on limiting group meetings we learned to assign tasks to pairs of people who would each push each other to develop models of the gearbox.

### *Mechanical Design:*

While we used an individual approach to maximize productivity, this technique generated difficulties in implementing revision control of new designs. Despite our best efforts to keep parts up to date, it was evident that there needs to be a steady stream of communication about the intricacies of parts: for tolerancing purposes the shift forks had a length of 4.003” however this detail was not made clear in drawings and our gear box did not assemble together properly without filing away the unintended material. Design for manufacture was our secondary lesson learned specifically the ability of reusing CNC programs to drastically reduce machining time and pinning together waterjet pieces for rapid production of complex components.

### *Fabrication:*

In preparing to machine several of our components, we learned that communicating with experienced machine shop staff generally saves significant time as their suggestions tend to increase the manufacturability of a part and decrease time taken to complete the part. We also developed skills in delegating specific manufacturing tasks to the group through daily email updates. These updates provided both the project status and next steps without requiring the full team to find time to meet together and allowing us to focus on getting our individual tasks done..

### *Redesign:*

It is evident that significant redesign will be necessary before our gearbox can be implemented in a Baja car. Most pressing is the need to reanalyze our gear design as our final analysis indicated that our gears would fail well before the necessary life requirement of 100 hours. Despite the apparent weakness in our gears we feel they are too heavy (15 lbs.) and take up too much volume. This contradiction that our gears are too small to withstand stress but too large for our design constraints indicates we should consider alternate materials such as hardened steel as opposed to mild steel and cast iron. Another potential improvement would be the use of helical gears allowing for greater torque transmission at a smaller diameter. As the diameter of our gears decreases, other components such as the spiders and shift forks can also become light and smaller.

The actuation system will require redesign as it currently miss-shifts and tends to bind under heavy frictional load (particularly going into reverse gear). An initial step in redesign includes replacing the compression springs in the cup with torsion springs as they are more appropriate for applying force over a rotational angle. The pawls also occasionally slip over pins without engaging indicating a need for stronger torsional springs. The detents in the cup for actuating the spring plunger should also be a countersink instead of a semicircle such that the spring can be depressed effectively. It is also unclear whether our current actuation design is most appropriate for our application and other potential options, such as electronic actuation or a simpler version of a double ratchet system, should be considered before finalizing an implementation.

While the gearbox prototype occasionally binds during shifting, it is an unlubricated system that should be tested with lubrication. In any case, we have developed ideas to reduce friction in the shifting mechanism. If an enclosed housing is designed, a liquid lubricant should be employed. There are currently similar metals acting as sliding interfaces, which causes scratching of both surfaces and uneven wear over time. Oilite bushings pressed into all sliding surfaces would eliminate scratching, provide lubrication, and allow for tighter tolerancing. The barrel cam was a source of friction as we were unable to machine the walls of the cam surface perpendicular to the axis of the barrel. Therefore the shift fork pin contacted the bearing surface at only one point, increasing binding. Determining a technique to machine a barrel cam to our specification would be useful.

The final redesign step would be designing a completely enclosed housing that can mount into the Baja car. This housing would likely need to be sealed to hold lubricant and prevent dirt from penetrating the system. Machining and designing an enclosure for this gearbox will likely be as large of a machining project as the demonstration prototype.

## SUMMARY AND CONCLUSIONS

Over the course of 2 months we successfully designed produced a demonstration prototype for a shifting mechanism that would be appropriate for a Baja car (relatively low weight, volume, and power and high reliability). For the power component of our project we sized gear reductions appropriate for the constraints of the SAE baja competition. Our transmission section considered a variety of elements including gear tooth and bearing loading, sliding and interfacing components, bending of a shift fork, and the geometric design of a complex actuation system. Structural design involved determining appropriate materials for the many components transmitting energy. Along with our design efforts was a significant machining portion including ~70 hours of machining components, many of which required CNC milling operations.

Though we successfully demonstrated the basic operation of a shifting mechanism and our ability to produce precision components, it is evident significant redesign will be required before this system is appropriate for an actual baja car. This redesign effort would focus on reducing the weight of the transmission significantly while using gears that are strong enough, increasing the reliability of shifts, and lubricating the system to make it more efficient. Completing this project was a significant accomplishment in both mechanical design and group communication and time management.

## APPENDIX

### CALCULATIONS CODE

#### BEARINGHIGHANALYSIS.M

```
%Bearing Force Calculator-High Gear
%In general you only have to change
the numbers with a D after them
%These D numbers represent
distances from an origin bearing
(furthest
%forward on a shaft)


%%%INPUTS%%%
EngineT = 13.75;
CVTReduction = 3;
%CVT radius
CVTR = .33;
%CVT distance to BearingI1
CVTD = 3;
%Pinion distance to BearingI1
GearD1 = .6875;
GearD2 = 3.1875;
GearD3 = 5.1875;
GearD4 = 6.5625;
%BearingI2 distance from BearingI1
```

```
BearingI2D = 5.875;

%distance of BearingL2 from
BearingL1
BearingL2D = 5.875;

%distance of output pinion from
BearingL1
GearL2D = 6.5625;
%Output shaft bearing distance from
BearingO1
BearingO2D = 1.375;
%Distance between output gear and
BearingO1
GearOD = .6875;
%Dist between output pinion
(reverse gear) to bearingO1
opinD=.6875;

%Gear Sizes
PinionR = (30/12)/2;

%LayshaftGear
GearLR = 5/2;

%LayshaftPinion
PinionLR = 2/2;

%Gear weights
Wa=.85;
Wb=.27;
Wc=.27;
Wd=3.21;
We=4.6;
Wf=3.21;
Wg=.48;
Wh=.48;
Wi=3.63;

phi=14.5*3.1415/180;
%reversegear and lay angle
rltheta = 74.04*3.1415/180;

%%%Calculations%%%5r
%%InputShaft%%
MaxT=EngineT*CVTReduction; %max
torque into system
Fcvt=MaxT/CVTR; %force is not
oriented in our coordinate system
correctly
Fcvtz=Fcvt*sin(3.14/4);
Fcvty=Fcvt*cos(3.14/4);
Ft=MaxT/(PinionR/12); %tangential
force due to gear on pinion
Fr=Ft*tan(phi);

%A,B --> these are for figuring out
y forces.
%from Fbi1+Fbi2=Fr, and
Fbi2*bearingI2D=GearD1*Fr
A=[1 1; 0 BearingI2D];
B=[Fr+Fcvty+(Wa+Wb+Wc)*cos(3.14/4);
(Wa*GearD1+Wb*GearD2+Wc*GearD3)*cos
(3.14/4)+GearD1*Fr-Fcvty*CVTD];
YI=inv(A)*B;

%C,D --> figuring out z forces
%from Fbi1+Fbi2=Fr-Fcvt, and
Fbi2*bearingI2D=GearD1*Ft+CVTD*Fcvt
C=[1 1; 0 BearingI2D];
D=[Ft+(Wa+Wb+Wc)*sin(3.14/4)-Fcvtz;
GearD1*(Ft+(Wa*sin(3.14/4)))+(GearD
2*Wb+GearD3*Wc)*sin(3.14/4)+CVTD*Fc
vtz];
ZI=inv(C)*D;

%Radial loads on input shaft
bearings
Fbi1=sqrt(YI(1)^2 + ZI(1)^2)
Fbi2=sqrt(YI(2)^2 + ZI(2)^2)


%%LayShaft%%
LTorque=Ft*GearLR;
Fgl1y=Ft;
Fgl1z=Fr;

%Layshaft and output shaft are at
an angle, so we have to break the
forces
%into components, and then sum up
vert/horiz forces.
Fto=LTorque/PinionLR;
Fro=Fto*tan(phi);
Fgl2y=Fto*cos(rltheta)+Fro*cos(3.14
/4 -rltheta);
Fgl2z=Fto*sin(rltheta)-
Fro*sin(3.14/4 - rltheta);

%E,F are for y forces
E=[1 -1; 0 BearingL2D];
F =[(Fgl1y-Fgl2y
+(Wd+We+Wf)*cos(3.14/4)) ;
(Fgl2y*GearL2D - Fgl1y*GearD1-
(Wd*GearD1+We*GearD2+Wf*GearD3+Wg*G
earD4)*cos(3.14/4))];
YL=inv(E)*F;

%G,H are for z forces
G=[1 1; 0 BearingL2D];
```

```
H=[Fgl1z+Fgl2z+(Wd+We+Wf+Wg)*sin(3.14/4);
(Fgl2z+Wg*sin(3.14/4))*GearD4+(Fgl1z+Wd*sin(3.14/4))*GearD1+(Wf*GearD3+We*GearD2)*sin(3.14/4)];
ZL=inv(G)*H;

%Radial loads on lay shaft bearings
Fbl1=sqrt(YL(1)^2 +ZL(1)^2)
Fbl2=sqrt(YL(2)^2 + ZL(2)^2)


%%OutputShaft%%
Fgoy=Fgl2y;
Fgoz=Fgl2z;
%I,J are for Y forces
I=[1 1 ; 0 BearingO2D];
J=[Fgoy+(Wh+Wi)*cos(3.14/4);Fgoy*GearOD+(Wh*cos(3.14/4)*opinD)+(Wi*cos(3.14/4)*GearOD)];
YO=inv(I)*J;

%K,L are for z forces
K=[1 1; 0 BearingO2D];
L=[Fgoz-(Wi+Wh)*sin(3.14/4); (Fgoz-Wi*sin(3.14/4))*GearOD+Wh*opinD*sin(3.14/4)];
ZO=inv(K)*L;

Fbo1=sqrt(YO(1)^2 + ZO(1)^2)
Fbo2=sqrt(YO(2)^2 + ZO(2)^2)
```

## BEARINGLOWANALYSIS.M

```
%Bearing Force Calculator-High Gear
%In general you only have to change the numbers with a D after them
%These D numbers represent distances from an origin bearing (furthest
%forward on a shaft)


%%%INPUTS%%%
EngineT = 13.75;
CVTReduction = 3;
%CVT radius
CVTR = .33;
%CVT distance to BearingI1
CVTD = 3;
%Pinion distance to BearingI1
GearD1 = .6875;
GearD2 = 3.1875;
GearD3 = 5.1875;
GearD4 = 6.5625;
%BearingI2 distance from BearingI1
BearingI2D = 5.875;

%distance of BearingL2 from BearingL1
BearingL2D = 5.875;

%distance of output pinion from BearingL1
GearL2D = 6.5625;
%Output shaft bearing distance from BearingO1
BearingO2D = 1.375;
%Distance between output gear and BearingO1
GearOD = .6875;
%Dist between output pinion (reverse gear) to bearingO1
opinD=.6875;

%Gear Sizes
PinionR = (30/12)/2;

%LayshaftGear
GearLR = 5/2;

%LayshaftPinion
PinionLR = 2/2;

%Gear weights
Wa=.85;
Wb=.27;
Wc=.27;
Wd=3.21;
We=4.6;
Wf=3.21;
Wg=.48;
Wh=.48;
Wi=3.63;

phi=14.5*3.1415/180;
%reversegear and lay angle
```

```
rltheta = 74.04*3.1415/180;

%%%Calculations%%%
%%InputShaft%%
MaxT=EngineT*CVTReduction; %max
torque into system
Fcvt=MaxT/CVTR; %force is not
oriented in our coordinate system
correctly
Fcvtz=Fcvt*sin(3.14/2);
Fcvty=Fcvt*cos(3.14/2);
Ft=MaxT/(PinionR/12); %tangential
force due to gear on pinion
Fr=Ft*tan(phi);

%A,B --> these are for figuring out
y forces.
%from Fbi1+Fbi2=Fr, and
Fbi2*bearingI2D=GearD1*Fr
A=[1 1; 0 BearingI2D];
B=[Fr+Fcvty-(Wa+Wb+Wc)*cos(3.14/4);
(Wa*GearD1+Wb*GearD2+Wc*GearD3)*cos
(3.14/4)+GearD1*Fr-Fcvty*CVTD];
YI=inv(A)*B;

%C,D --> figuring out z forces
%from Fbi1+Fbi2=Fr-Fcvt, and
Fbi2*bearingI2D=GearD1*Ft+CVTD*Fcvt
C=[1 1; 0 BearingI2D];
D=[Ft-Fcvtz+(Wa+Wb+Wc)*sin(3.14/4);
GearD2*(Ft+Wb*sin(3.14/4))+(Wa*Gear
D1+Wc*GearD3)*sin(3.14/4)+CVTD*Fcvt
z];
ZI=inv(C)*D;

%Radial loads on input shaft
bearings
Fbi1=sqrt(YI(1)^2 + ZI(1)^2)
Fbi2=sqrt(YI(2)^2 + ZI(2)^2)


%%LayShaft%%
LTorque=Ft*GearLR;
Fgl1y=Ft;
Fgl1z=Fr;


%Layshaft and output shaft are at
an angle, so we have to break the
forces
%into components, and then sum up
vert/horiz forces.
```

BEARINGREVERSEANALYSIS.M

```
Fto=LTorque/PinionLR;
Fro=Fto*tan(phi);
Fgl2y=Fto*cos(rltheta)+Fro*cos(3.14
/2 -rltheta);
Fgl2z=Fto*sin(rltheta)-
Fro*sin(3.14/2 - rltheta);

%E,F are for y forces
E=[1 -1; 0 BearingL2D];
F =[Fgl1y-
Fgl2y+(Wd+We+Wf)*cos(3.14/4);
(Fgl2y*GearL2D -Fgl1y*GearD1)-
(Wd*GearD1+We*GearD2+Wf*GearD3+Wg*G
earD4)*cos(3.14/4)];
YL=inv(E)*F;

%G,H are for z forces
G=[1 1; 0 BearingL2D];
H=[Fgl1z+Fgl2z+(Wd+We+Wf+Wg)*sin(3.
14/4);
(Fgl2z+Wg*sin(3.14/4))*GearD4+(Fgl1
z+We*sin(3.14/4))*GearD2+(GearD1*Wd
+GearD3*Wf)*sin(3.14/4)];
ZL=inv(G)*H;

%Radial loads on lay shaft bearings
Fbl1=sqrt(YL(1)^2 +ZL(1)^2)
Fbl2=sqrt(YL(2)^2 + ZL(2)^2)


%%OutputShaft%%
Fgoy=Fgl2y;
Fgoz=Fgl2z;

%I,J are for Y forces
I=[1 1 ; 0 BearingO2D];
J=[Fgoy+(Wh+Wi)*cos(3.14/4);Fgoy*Ge
arOD+(Wh*cos(3.14/4)*opinD)+(Wi*cos
(3.14/4)*GearOD)];
YO=inv(I)*J;

%K,L are for z forces
K=[1 1; 0 BearingO2D];
L=[Fgoz-(Wi+Wh)*sin(3.14/4); (Fgoz-
Wi*sin(3.14/4))*GearOD+Wh*opinD*sin
(3.14/4)];
ZO=inv(K)*L;

Fbo1=sqrt(YO(1)^2 + ZO(1)^2)
Fbo2=sqrt(YO(2)^2 + ZO(2)^2)


%Bearing Force Calculator-High Gear
%In general you only have to change
the numbers with a D after them
```

```
%These D numbers represent
distances from an origin bearing
(furthest
%forward on a shaft)


%%%INPUTS%%%
EngineT = 13.75;
CVTReduction = 3;
%CVT radius
CVTR = .33;
%CVT distance to BearingI1
CVTD = 3;
%Pinion distance to BearingI1
GearD1 = .6875;
GearD2 = 3.1875;
GearD3 = 5.1875;
GearD4 = 6.5625;
%BearingI2 distance from BearingI1
BearingI2D = 5.875;

%distance of BearingL2 from
BearingL1
BearingL2D = 5.875;

%distance of output pinion from
BearingL1
GearL2D = 6.5625;
%Output shaft bearing distance from
BearingO1
BearingO2D = 1.375;
%Distance between output gear and
BearingO1
GearOD = .6875;
%Dist between output pinion
(reverse gear) to bearingO1
opinD=.6875;

%Gear Sizes
PinionR = (30/12)/2;

%LayshaftGear
GearLR = 5/2;

%LayshaftPinion
PinionLR = 2/2;

%Gear weights
Wa=.85;
Wb=.27;
Wc=.27;
Wd=3.21;
We=4.6;
Wf=3.21;
Wg=.48;
Wh=.48;
Wi=3.63;

phi=14.5*3.1415/180;
%reversegear and lay angle
rltheta = 74.04*3.1415/180;

%%%Calculations%%%
%%InputShaft%%
MaxT=EngineT*CVTReduction; %max
torque into system
Fcvt=MaxT/CVTR; %force is not
oriented in our coordinate system
correctly
Fcvtz=Fcvt*sin(3.14/2);
Fcvty=Fcvt*cos(3.14/2);
Ft=MaxT/(PinionR/12); %tangential
force due to gear on pinion
Fr=Ft*tan(phi);

%A,B --> these are for figuring out
y forces.
%from Fbi1+Fbi2=Fr, and
Fbi2*bearingI2D=GearD1*Fr
A=[1 1; 0 BearingI2D];
B=[Fr+Fcvty-(Wa+Wb+Wc)*cos(3.14/4);
(Wa*GearD1+Wb*GearD2+Wc*GearD3)*cos
(3.14/4)+GearD1*Fr-Fcvty*CVTD];
YI=inv(A)*B;

%C,D --> figuring out z forces
%from Fbi1+Fbi2=Fr-Fcvt, and
Fbi2*bearingI2D=GearD1*Ft+CVTD*Fcvt
C=[1 1; 0 BearingI2D];
D=[Ft-Fcvtz+(Wa+Wb+Wc)*sin(3.14/4);
GearD2*(Ft+Wb*sin(3.14/4))+(Wa*Gear
D1+Wc*GearD3)*sin(3.14/4)+CVTD*Fcvt
z];
ZI=inv(C)*D;

%Radial loads on input shaft
bearings
Fbi1=sqrt(YI(1)^2 + ZI(1)^2)
Fbi2=sqrt(YI(2)^2 + ZI(2)^2)


%%LayShaft%%
LTorque=Ft*GearLR;
Fgl1y=Ft;
Fgl1z=Fr;


%Layshaft and output shaft are at
an angle, so we have to break the
forces
```

```
%into components, and then sum up
vert/horiz forces.
Fto=LTorque/PinionLR;
Fro=Fto*tan(phi);
Fgl2y=Fto*cos(rltheta)+Fro*cos(3.14
/2 -rltheta);
Fgl2z=Fto*sin(rltheta)-
Fro*sin(3.14/2 - rltheta);

%E,F are for y forces
E=[1 -1; 0 BearingL2D];
F =[Fgl1y-
Fgl2y+(Wd+We+Wf)*cos(3.14/4);
(Fgl2y*GearL2D -Fgl1y*GearD1)-
(Wd*GearD1+We*GearD2+Wf*GearD3+Wg*G
earD4)*cos(3.14/4)];
YL=inv(E)*F;

%G,H are for z forces
G=[1 1; 0 BearingL2D];
H=[Fgl1z+Fgl2z+(Wd+We+Wf+Wg)*sin(3.
14/4);
(Fgl2z+Wg*sin(3.14/4))*GearD4+(Fgl1
z+We*sin(3.14/4))*GearD2+(GearD1*Wd
+GearD3*Wf)*sin(3.14/4)];
ZL=inv(G)*H;

%Radial loads on lay shaft bearings
Fbl1=sqrt(YL(1)^2 +ZL(1)^2)
Fbl2=sqrt(YL(2)^2 + ZL(2)^2)

%%OutputShaft%%
Fgoy=Fgl2y;
Fgoz=Fgl2z;

%I,J are for Y forces
I=[1 1 ; 0 BearingO2D];
J=[Fgoy+(Wh+Wi)*cos(3.14/4);Fgoy*Ge
arOD+(Wh*cos(3.14/4)*opinD)+(Wi*cos
(3.14/4)*GearOD)];
YO=inv(I)*J;

%K,L are for z forces
K=[1 1; 0 BearingO2D];
L=[Fgoz-(Wi+Wh)*sin(3.14/4); (Fgoz-
Wi*sin(3.14/4))*GearOD+Wh*opinD*sin
(3.14/4)];
ZO=inv(K)*L;

Fbo1=sqrt(YO(1)^2 + ZO(1)^2)
Fbo2=sqrt(YO(2)^2 + ZO(2)^2)
```

## SPIDER_BEARING_STRESS.M

```
%Bearing stress calculations

T = 200; % [ft-lbs]
h = 0.25; % height of the teeth [in]
n = 1; % number of teeth
r = 1.25; %radius of  contact patch from center [in]
tau_b = 90e3; % maximum bearing stress [psi]
t = (12*T)/(h*n*r*tau_b)
```

## SPIDER_KEY.M

```
% Spider keyway calculations

Sy = 32e3; % yield strength [ksi]
T = 12*200; % torque transmitted through the key [in-lbs]
d = 1; %diameter of the shaft [in]
L = T*8/(0.58*Sy*(d^2)) %length of the keyway [in]
```

## SPIDER_SHEAR_STRESS.M

```
%% Shear Stress Calculation.
```

```
%What is the minimum outer radius such that the dog teeth don't shear.

T = 200; % Torque applied to the spider [ft-lbs]
r_inner = 0.5; %inner radius [in]
tau_max = 90e3; %Torsional shear strength of ASTM Class 60 Cast Iron [psi]

r = linspace(r_inner+0.001,1.6);
tau = 24 .* T .* r ./ (pi .* (r.^4 - r_inner.^4));
plot(r, tau./1000,'r','linewidth',4)
xlabel('Radius of the spider [in]');
ylabel('Maximum shear stress [ksi]');
title('Radius required of the spider to withstand shear stress')
```

## BAJACAR.M

```
% Baja Car dynamics with ODE45
% 14.09.2006 Brian Bingham
% 16.09.2009 Jay Gorasia. Added event and consolidated code to 1 file
% 30.09.2009 Jay Gorasia. Added baja car dynamics

function bajacar
% Initial Condition
% The vector holds x,y,x_dot,y_dot
x0 = zeros(4,1);
x0(2) = 0;                          % [m]
v0 = 0.1;                           % initial speed [m/s]
theta0 = 0.1;                        % initial gradient [degress]
x0(3) = v0*cos(theta0/180*pi);      % [m/s]
x0(4) = v0*cos(theta0/180*pi);      % [m/s]

% Time Bounds
t0 = 0;
tF = 40;                            % [s]

% ODE45 Call
options = odeset('Events', @events);
[tt,yy] = ode45(@proj2d,[t0 tF],x0, options);

dd = sqrt(yy(:,1).^2+yy(:,2).^2); %distance [m]
vv = sqrt(yy(:,3).^2+yy(:,4).^2); %speed [m/s]

% Plot the results
clf
subplot(2,2,1);
plot(tt,dd.*3.2808399)
grid on
title('Baja car movement')
xlabel('T (s)')
ylabel('Distance (ft)')

subplot(2,2,2);
plot(tt,vv.*2.236936292)
title('Baja car speed')
xlabel('T (s)')
```

```
ylabel('Vx (mph)')

subplot(2,2,3);
R_w = 0.2921;                                % Tire radius [m]
omega = vv./R_w;                             % Wheel speed [rad/s]
t_wheel = 0.95*0.7*5000./omega*0.737562121;  %Wheel torque [ft-lbf]
omega = omega/(2*pi)*60;                     % Wheel speed [RPM]

plot(tt,omega);
title('Angular velocity of wheels')
xlabel('Time (s)')
ylabel('Omega (RPM)')

subplot(2,2,4);
plot(tt,t_wheel)
title('Torque at wheels')
xlabel('Time (s)')
ylabel('Torque (ft-lbf)')
axis([0 40 0 500])

end

function [value,isterminal,direction] = events(t,x0)
value = x0(2); % Extract the current height.
isterminal = 1; % Stop the integration if height crosses zero.
direction = -1; % But only if the height is decreasing.
end

function dx = proj2d(t,x)
% ODE45 function for baja car
% Let x = {sx,sy,vx,vy}

%Engine Input Power
%Assume engine is producing power to give maximum torque (6.7 hp=5000W)
%At this power, engine speed is 2600RPM = 272.3 rad/s
%and torque is 13.75 lbf ft = 18.64 Nm
P_m = 5000; %[W]

%Terrain input
grad = 0.1/180*pi;

% Baja Car Constants
m = 280;                        % Mass of car and 70kg person [kg]
g = 9.81;                       % m/s^2
e_cvt = .7;                     % CVT efficiency
e_gearbox = 0.95;               % Gearbox efficiency
P_w = P_m *e_cvt*e_gearbox;     % Wheel power [W]
R_w = 0.2921;                   % Tire radius [m]

v = sqrt(x(3)^2+x(4)^2);        %Speed [m/s]

% Viscous Drag force
C_d = 2;                % Flat flat perpendicular to flow
rho = 1.204;            % air density at 25 celcius kg/m^3
A = 1.1;                % Frontal area [m^2]
```

```
F_d = 0.5*C_d*rho*A*(v^2); % [N]

% Weight of car
W = m*g;

% Assume the tires do not slip
w_tire = v/R_w; % Angular velocity of the wheel
T_car = P_w / w_tire;
F_car = T_car/(R_w);

%Rotation matrix
R = [cos(grad) -sin(grad); sin(grad) cos(grad)];

%Forces in frame of car
F1 = [F_car-sign(x(3))*F_d-sin(grad)*W; 0];

%Forces in absolute reference frame
F = R*F1; %Rotate relative frame
ddx = F./m; %acceleration

%Pack output vector
dx = zeros(4,1);
dx(1) = x(3);
dx(2) = x(4);
dx(3) = ddx(1);
dx(4) = ddx(2);
end
```

## DRAWINGS

Attached are drawings specifying our Gate 1 and Gate 2 prototypes. We divided the Gate 2 Prototype into the following subassemblies:

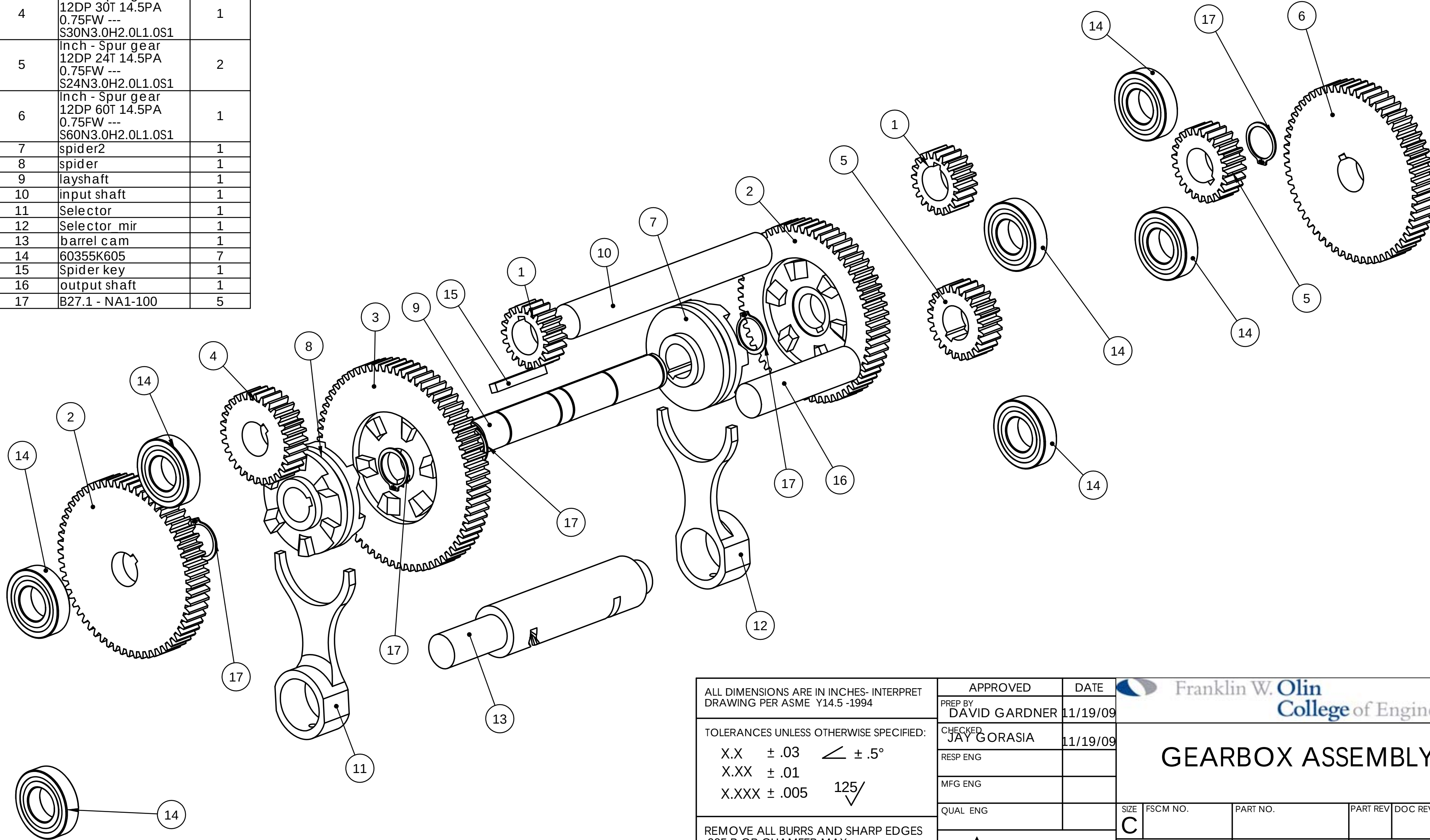
- Final Assembly
- Actuator
- Cup
- Layshaft
- Input Shaft
- Barrel Cam and Shifters

| REV | ZONE | DESCRIPTION | APPROVED | DATE |
|---|---|---|---|---|
| - | | INITAL RELEASE | | |

NOTES, UNLESS OTHERWISE SPECIFIED

1. UNSPECIFIED FILLET RADII .005-.010.
2. MATERIAL: AL BAR 6061-T651
3. THREADS PER ANSI B1.1.

| ITEM NO. | PART NUMBER | QTY. |
|---|---|---|
| 1 | Inch - Spur gear 12DP 20T 14.5PA 0.75FW --- S20N3.0H2.0L1.0S1 | 2 |
| 2 | Inch - Spur gear 12DP 60T 14.5PA 0.75FW --- S60N3.0H2.0L1.0S1 | 2 |
| 3 | GD70 | 1 |
| 4 | Inch - Spur gear 12DP 30T 14.5PA 0.75FW --- S30N3.0H2.0L1.0S1 | 1 |
| 5 | Inch - Spur gear 12DP 24T 14.5PA 0.75FW --- S24N3.0H2.0L1.0S1 | 2 |
| 6 | Inch - Spur gear 12DP 60T 14.5PA 0.75FW --- S60N3.0H2.0L1.0S1 | 1 |
| 7 | spider2 | 1 |
| 8 | spider | 1 |
| 9 | layshaft | 1 |
| 10 | input shaft | 1 |
| 11 | Selector | 1 |
| 12 | Selector_mir | 1 |
| 13 | barrel cam | 1 |
| 14 | 60355K605 | 7 |
| 15 | Spider key | 1 |
| 16 | output shaft | 1 |
| 17 | B27.1 - NA1-100 | 5 |




ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| | APPROVED | DATE |
|---|---|---|
| PREP BY | DAVID GARDNER | 11/19/09 |
| CHECKED | JAY GORASIA | 11/19/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

◆ = (CRITICAL DIMENSION)

Franklin W. Olin College of Engineering

# GEARBOX ASSEMBLY

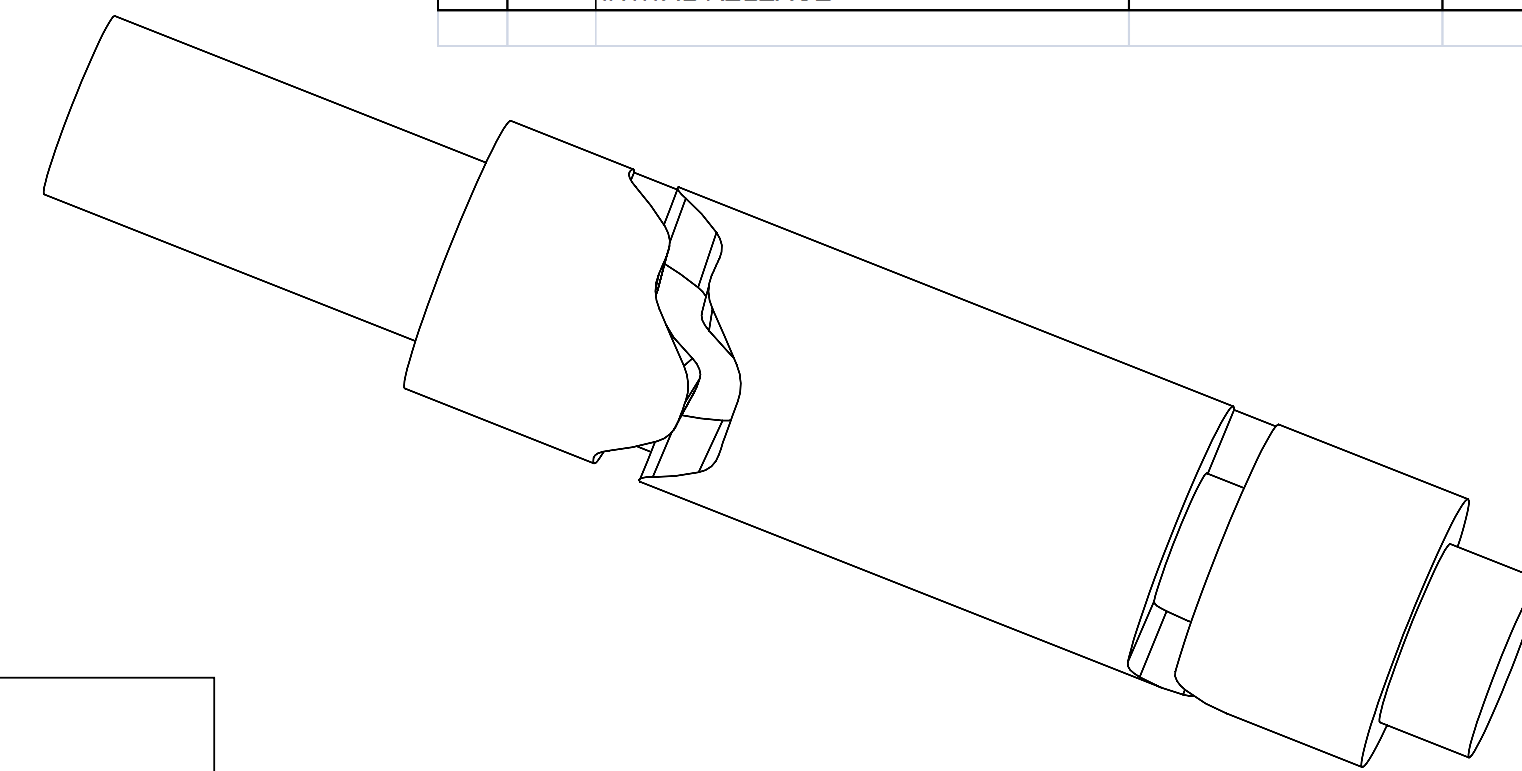
| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |

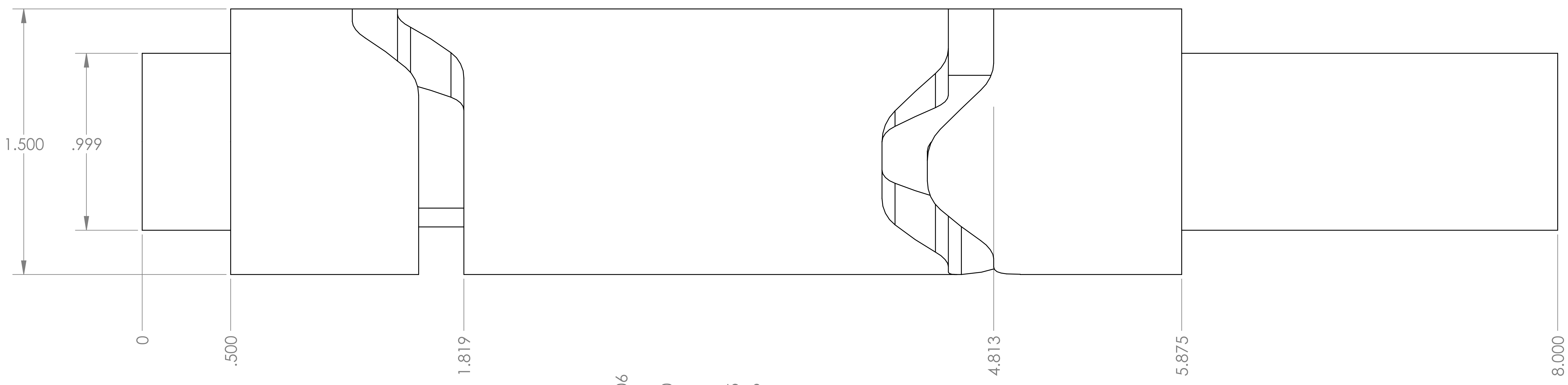
SCALE 1:8 WT SHEET 1 OF 1

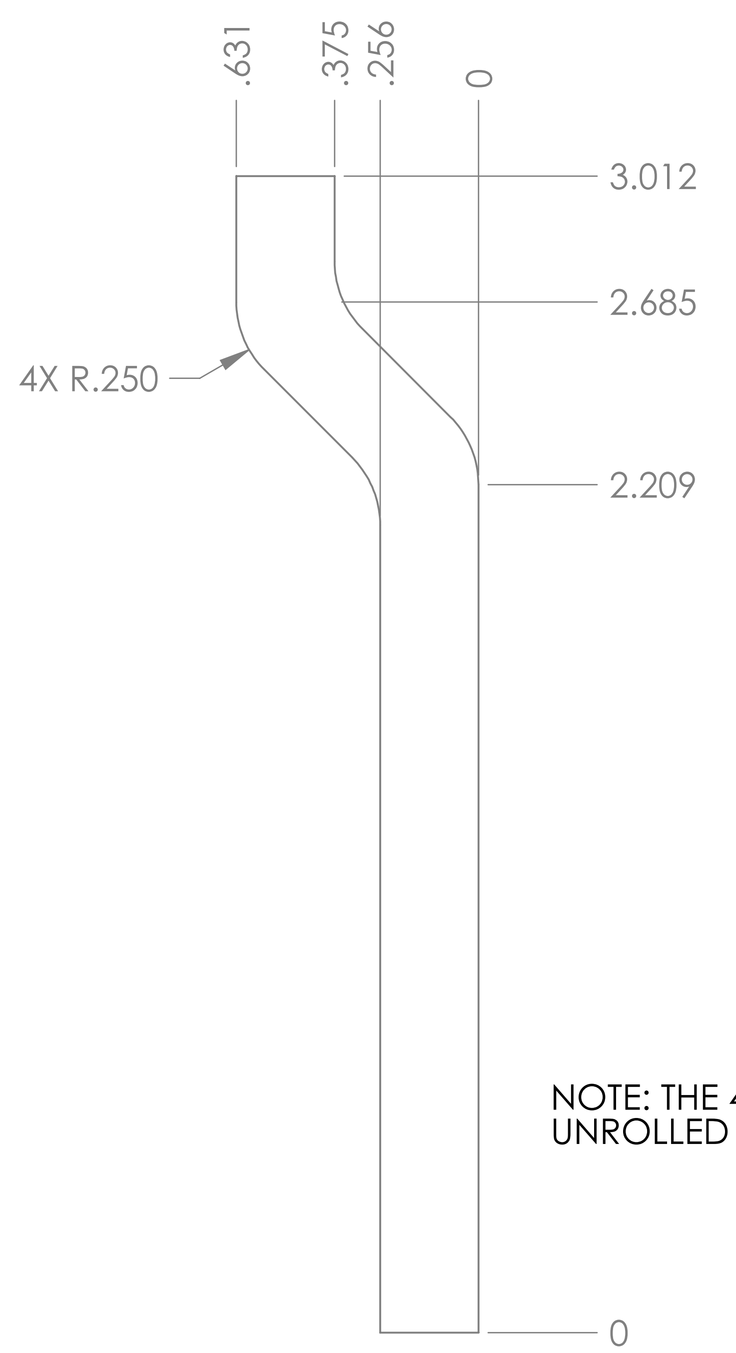
NOTES, UNLESS OTHERWISE SPECIFIED

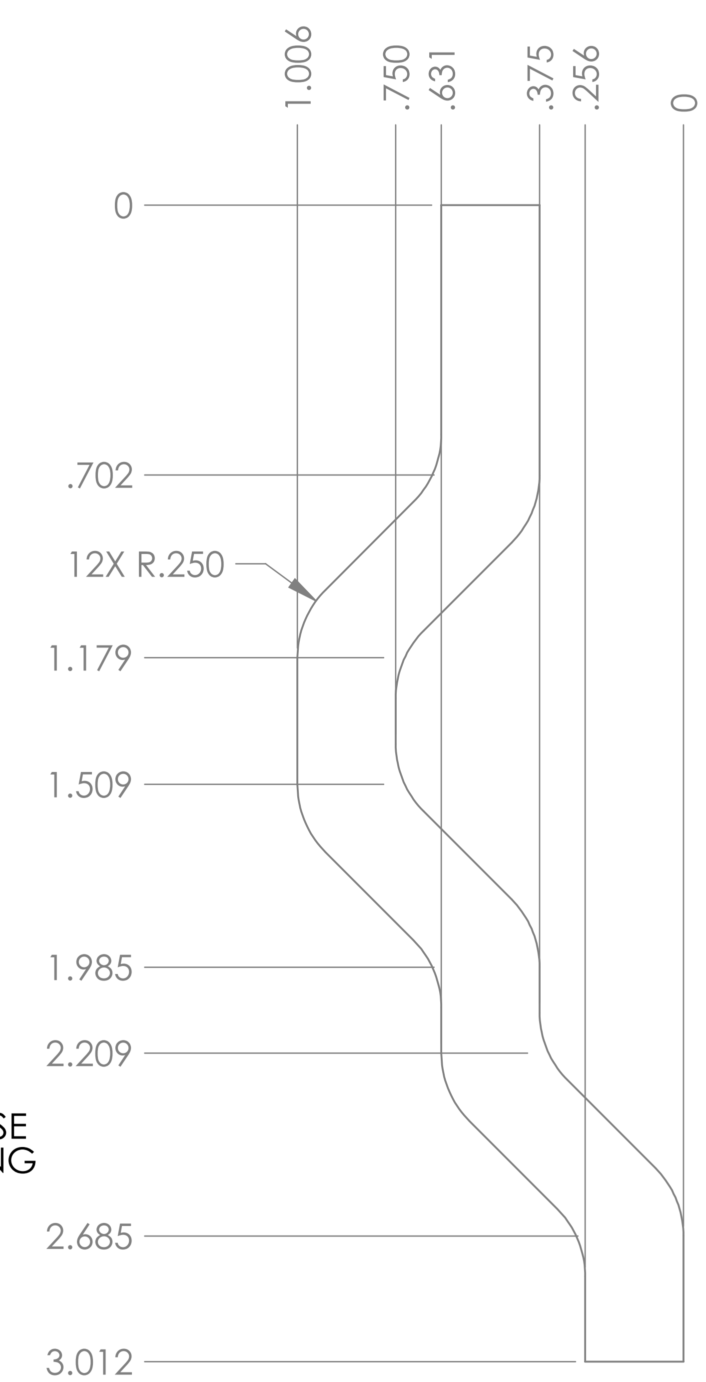
1. UNSPECIFIED FILLET RADII .005-.010.
2. MATERIAL: AL BAR 6061-T651
3. THREADS PER ANSI B1.1.

| REV | ZONE | DESCRIPTION | APPROVED | DATE |
|---|---|---|---|---|
| - | | INTITAL RELEASE | | |






NOTE: THE 4 AXIS CNC USES THESE UNROLLED PROFILES FOR CUTTING

QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:

X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| | APPROVED | DATE |
|---|---|---|
| PREP BY | N. PAULSON | 11/19/09 |
| CHECKED | | |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

♦ = (CRITICAL DIMENSION)

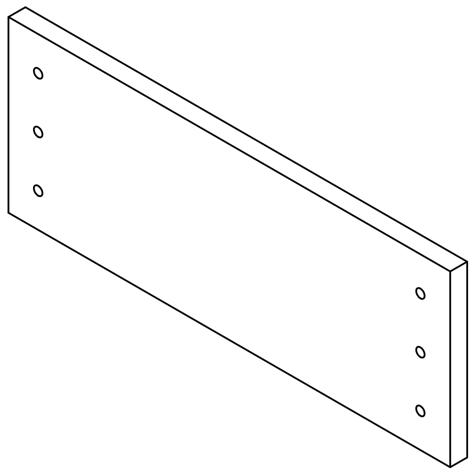
Franklin W. Olin College of Engineering

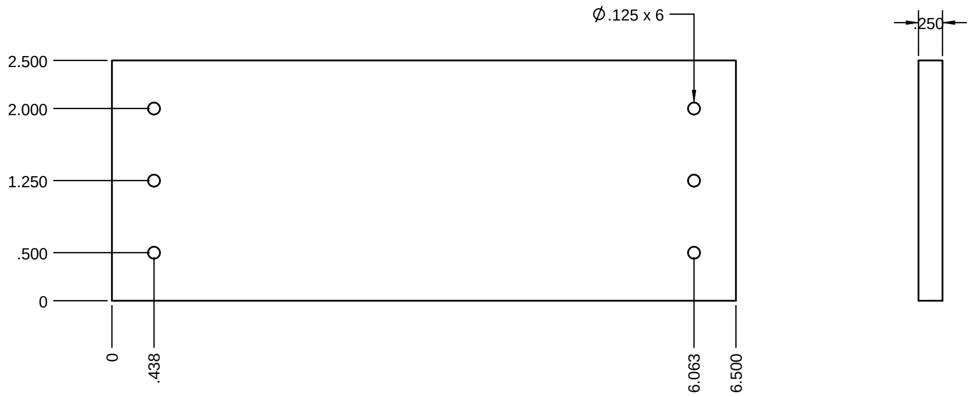
# BARREL CAM

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 1:1 | | WT | SHEET 1 OF 1 | |

Notes:
The holes are press fit for .125" dowels
The tolerance for these holes is .5 thousandths

The plate is .25" stock



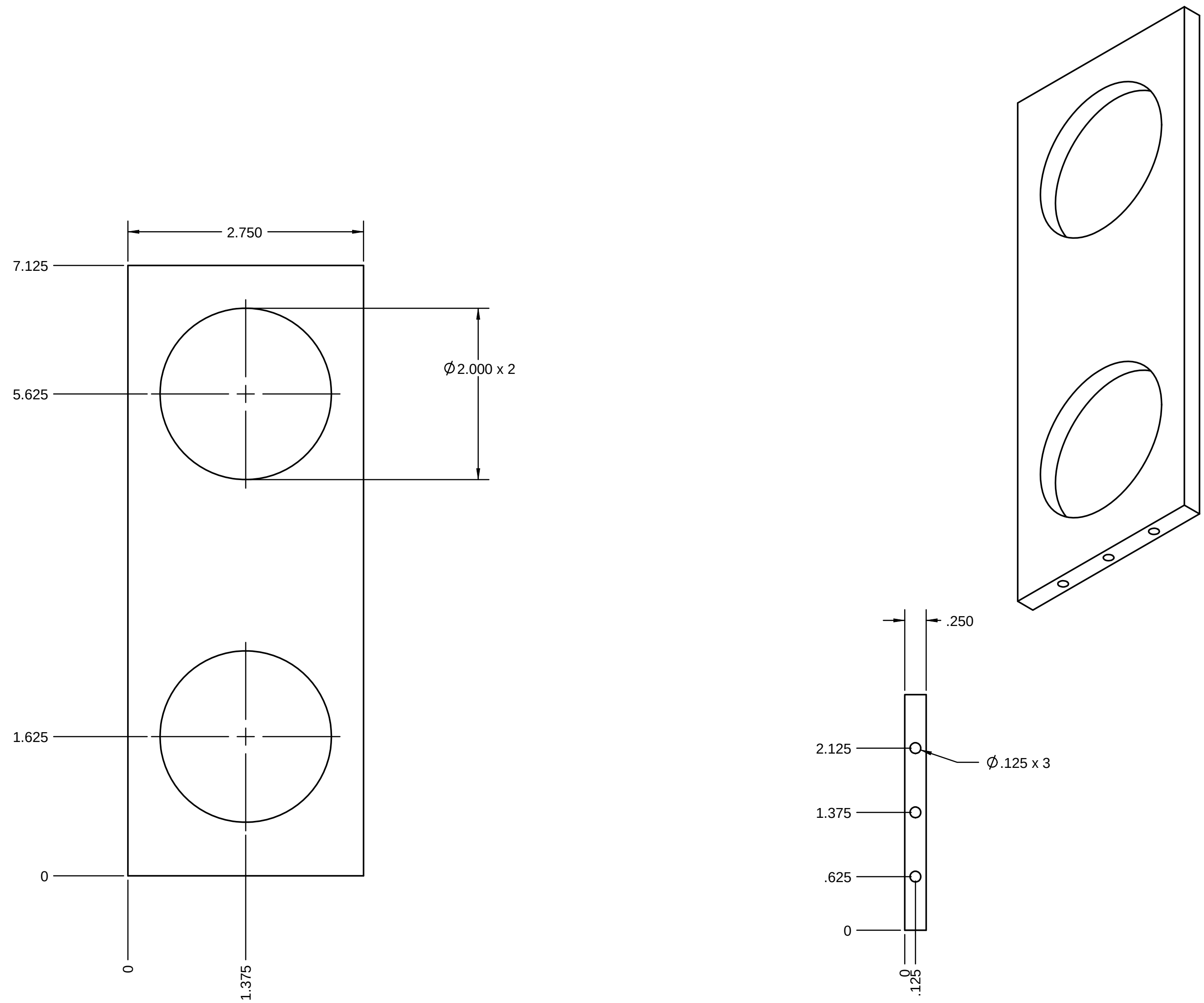



QUANTITY:1

| | | | |
|---|---|---|---|
| ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994 | APPROVED | DATE | Franklin W. Olin College of Engineering |
| | PREP BY J. GORASIA | 11/6/09 | |
| TOLERANCES UNLESS OTHERWISE SPECIFIED: X.X ± .03; X.XX ± .01; X.XXX ± .005; ∠ ± .5°; 125√ | CHECKED | | GATE 1 BOTTOM |
| | RESP ENG | | |
| | MFG ENG | | |
| | QUAL ENG | | SIZE C; FSCM NO.; PART NO.; PART REV; DOC REV |
| REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX | ⧫ = (CRITICAL DIMENSION) | | SCALE 1:1; WT; SHEET 1 OF 1 |

Note:
Dowel holes on the bottom are close fit.
The tolerance for these holes is .3 thousandths

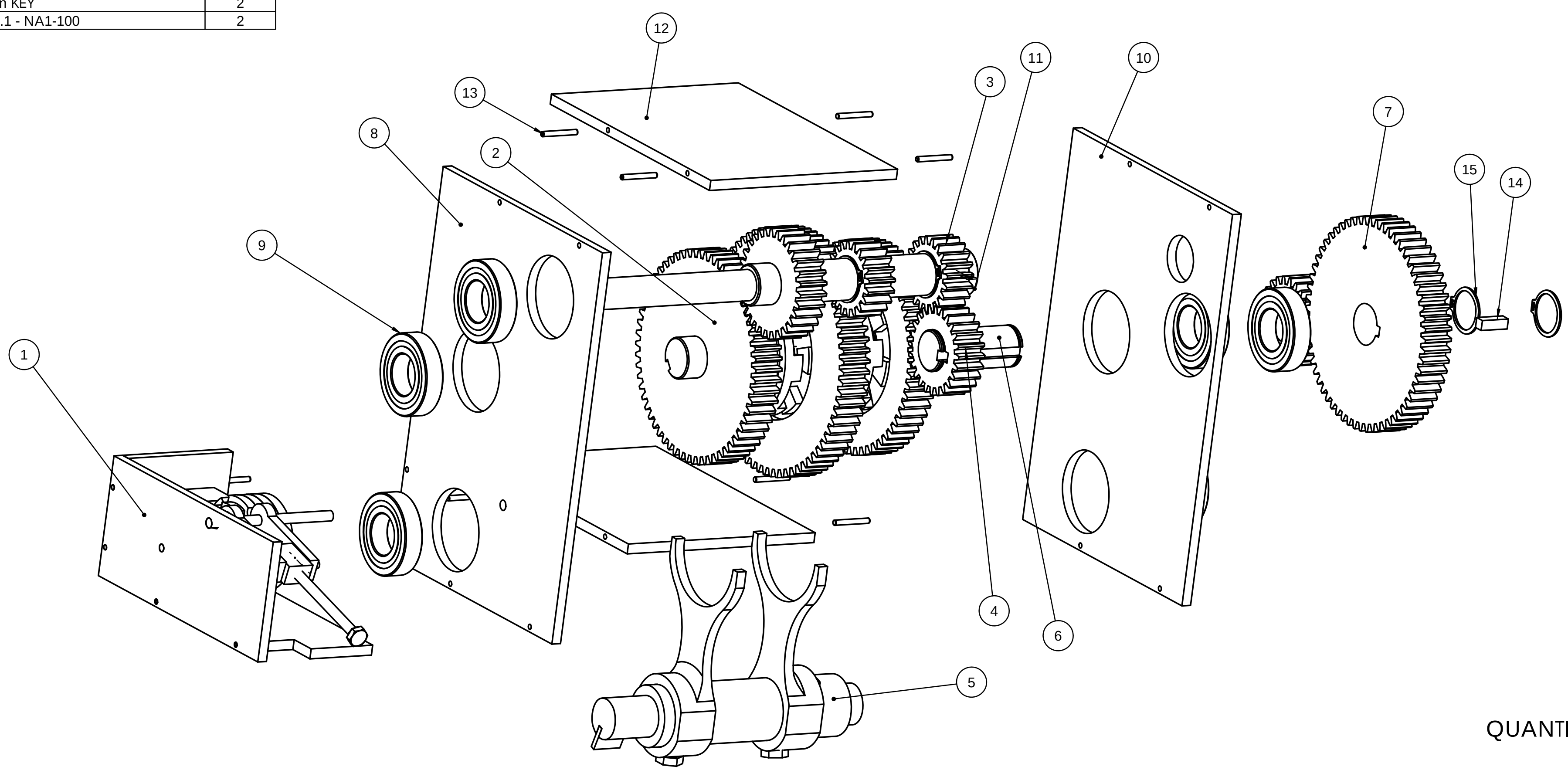
Bearing holes (side) are press fit.
Tolerance is 1 thousandths




QUANTITY: 2

| | | | |
|---|---|---|---|
| ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994 | APPROVED | DATE | Franklin W. Olin College of Engineering |
| TOLERANCES UNLESS OTHERWISE SPECIFIED: X.X ± .03; X.XX ± .01; X.XXX ± .005; ∠ ± .5°; 125√ | PREP BY J. GORASIA | 11/6/09 | GATE 1 SIDE HOUSING |
| | CHECKED | | |
| | RESP ENG | | |
| | MFG ENG | | SIZE C · FSCM NO. · PART NO. · PART REV · DOC REV |
| REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX | QUAL ENG | | |
| | ⧫ = (CRITICAL DIMENSION) | | SCALE 1:1 · WT · SHEET 1 OF 1 |

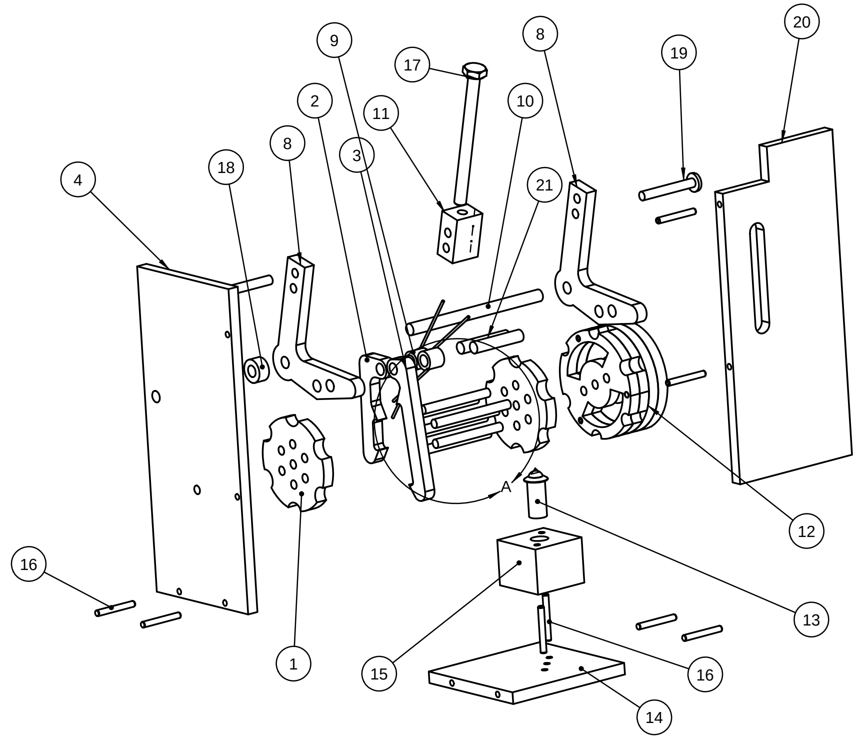
| ITEM NO. | PART NUMBER | QTY. |
|---|---|---|
| 1 | ACTUATOR | 1 |
| 2 | LAYSHAFT | 1 |
| 3 | INPUT SHAFT | 1 |
| 4 | INCH - SPUR GEAR 12 DP 24T 14.5 PA 0.75 FW -- S24N3.0H2.0L1.0S1 | 2 |
| 5 | BARREL CAM AND SHIFTERS | 1 |
| 6 | outputshaftmachining | 1 |
| 7 | INCH - SPUR GEAR 12DP 60T 14.5PA 0.75FW ---S60N3.0H2.0L1.0S | 1 |
| 8 | INPUT HOUSING | 1 |
| 9 | 60355K605 | 6 |
| 10 | OUTPUT HOUSING | 1 |
| 11 | 60355K601 | 1 |
| 12 | TOP HOUSING | 2 |
| 13 | DPM 0.125x1 | 8 |
| 14 | .75in KEY | 2 |
| 15 | B27.1 - NA1-100 | 2 |

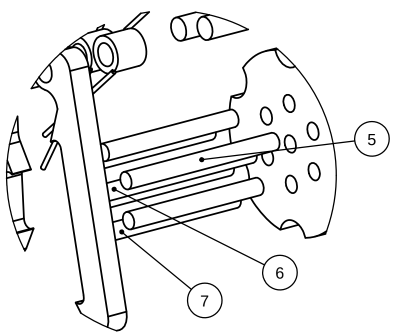



QUANTITY: 1

| | | | |
|---|---|---|---|
| ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994 | APPROVED | DATE | Franklin W. Olin College of Engineering |
| TOLERANCES UNLESS OTHERWISE SPECIFIED: X.X ± .03; X.XX ± .01; X.XXX ± .005; ∠ ± .5°; 125√ | PREP BY J. GORASIA | 12/13/09 | GATE 2 FINAL ASSEMBLY |
| | CHECKED R. HARRIS | 12/14/09 | |
| | RESP ENG | | |
| | MFG ENG | | |
| REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX | QUAL ENG | | SIZE C; FSCM NO.; PART NO.; PART REV; DOC REV |
| | ◆ = (CRITICAL DIMENSION) | | SCALE 1:2; WT; SHEET 1 OF 1 |

| ITEM NO. | PART NUMBER | DESCRIPTION | QTY. |
|---|---|---|---|
| 1 | RATCHET WHEEL | | 2 |
| 2 | PAWL1 SIMPLE | | 1 |
| 3 | PAWL 2 SIMPLE | | 1 |
| 4 | FRONT PLATE | | 1 |
| 5 | 3_16DOWELPIN | 3/16"X2" STEEL DOWEL PIN | 2 |
| 6 | 3_16DOWELPIN | 3/16"X2.25" STEEL DOWEL PIN | 1 |
| 7 | 3_16DOWELPIN | 3/16"X1.75" STEEL DOWEL PIN | 4 |
| 8 | TOP ARM | | 2 |
| 9 | TORSION SPRING 2 | MCMASTERR CARR PART #: 9271K182 | 2 |
| 10 | 1_4 DOWEL PIN | 1/4""X3" STEEL DOWEL PIN | 1 |
| 11 | HANDLE SHAFT | | 1 |
| 12 | CUP ASSEMBLY | | 1 |
| 13 | SPRING PLUNGER 2 | MCMASTER CARR PART: 84835A13 | 1 |
| 14 | OUTSIDE SIDE HOUSING | | 1 |
| 15 | PLUNGER OFFSET BLOCK | | 1 |
| 16 | DPM 0.125x1 | 1/8" X1" STEEL DOWEL PIN | 8 |
| 17 | HFBOLT 0.25-20x3x0.75-N | 1/4-20 X 3.0 HEX HEAD BOLT | 1 |
| 18 | TOP ARM SPACER | | 1 |
| 19 | SL-BHMS 0.19-24x1.375x1-N | #10-24 X 1.375 MACHINE SCREW | 2 |
| 20 | OUTSIDE BOTTOM HOUSING | | 1 |
| 21 | 1_4 DOWEL PIN | 1/4"X1.25" STEEL DOWEL PIN | 2 |



DETAIL A
SCALE 1 : 1

QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:

X.X ± .03 ∠ ± .5°

X.XX ± .01

X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/14/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

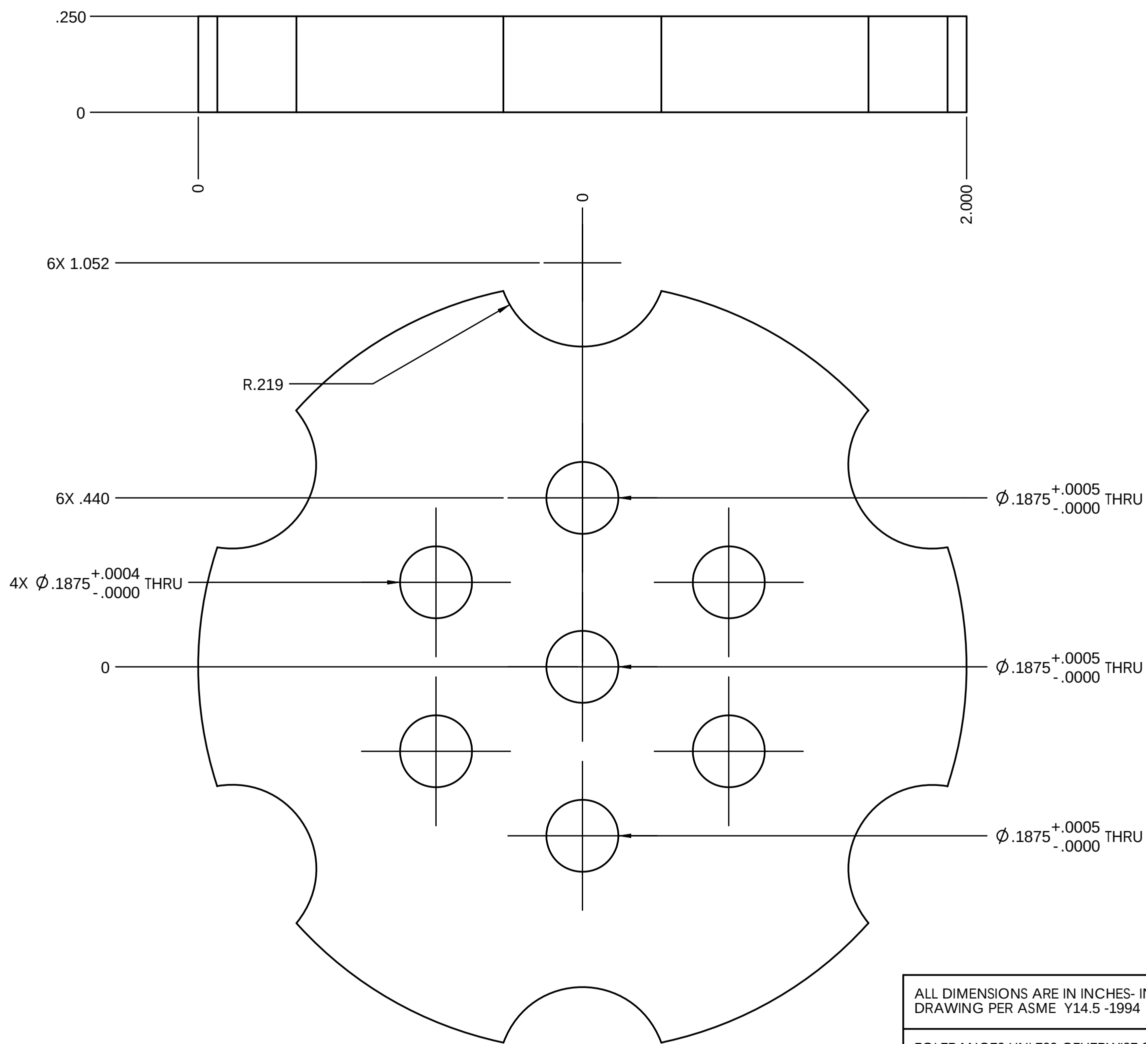
♦ = (CRITICAL DIMENSION)

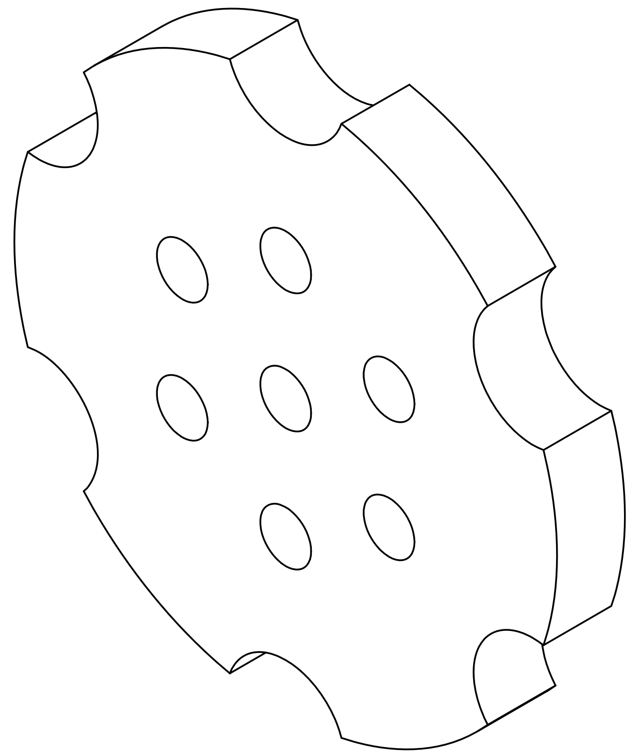
Franklin W. Olin College of Engineering

ACTUATOR

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 1:2 | | WT | SHEET 1 | OF 1 |

NOTES:
1. USE 0.25" 6061T6 ALUMINUM PLATE STOCK
2. CUT OUTSIDE PROFILE USING WATERJET, THEN DRILL AND REAM HOLES



QUANTITY: 2

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03   ∠ ± .5°
X.XX ± .01
X.XXX ± .005   125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| | APPROVED | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/13/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

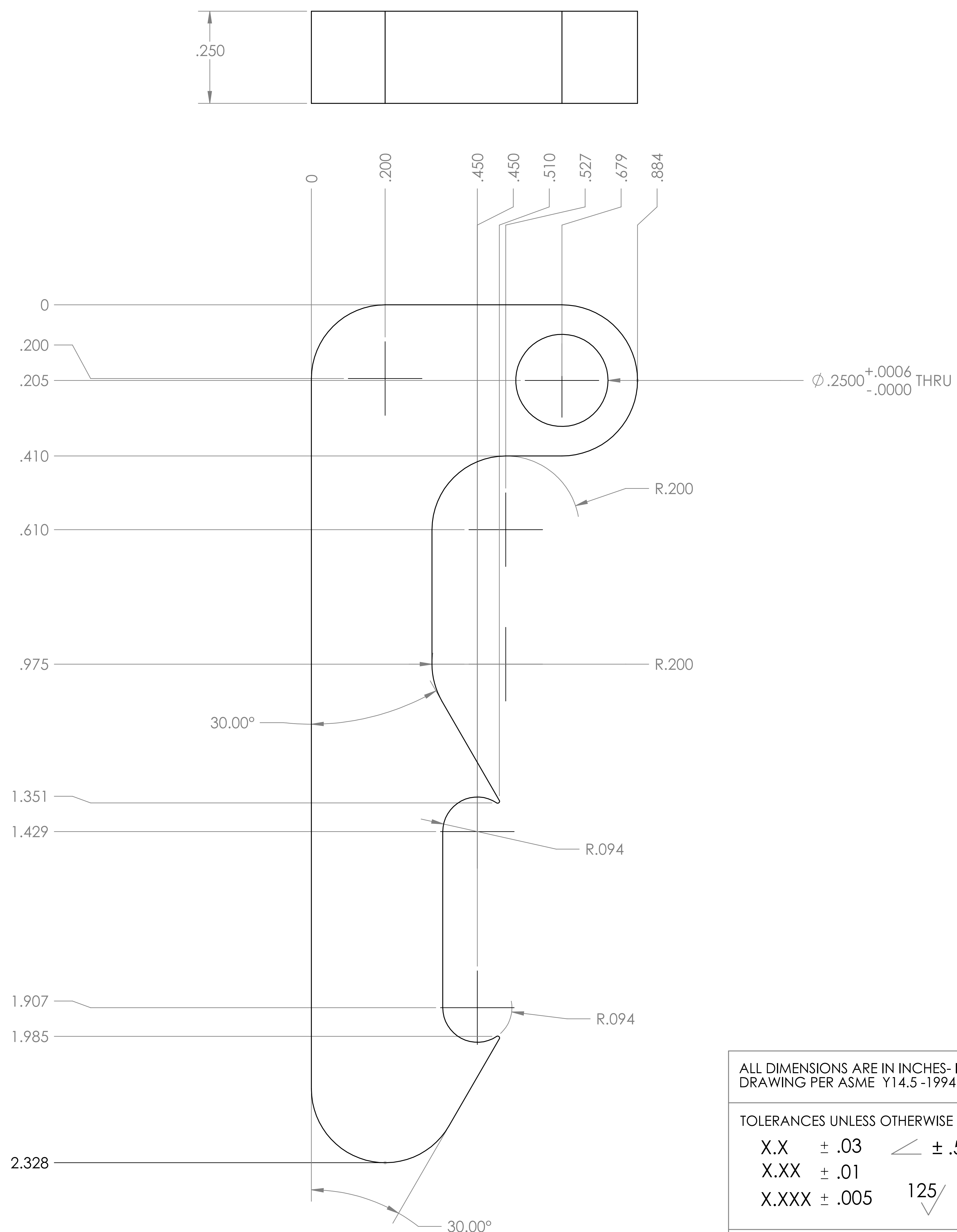
♦ = (CRITICAL DIMENSION)

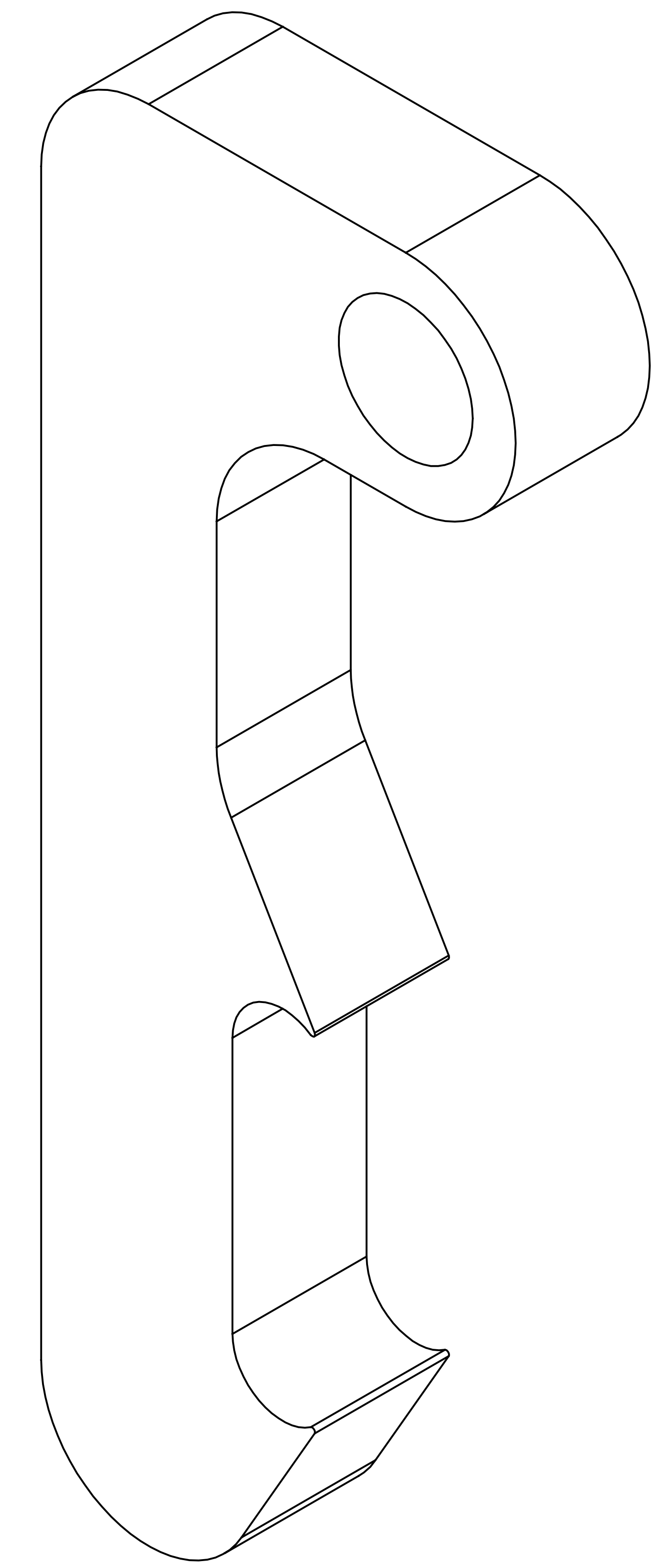
Franklin W. Olin College of Engineering

RATCHET WHEEL

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 4:1 | | WT | SHEET 1 | OF 1 |

NOTES:
1. USE 0.25" 6061T6 ALUMINUM PLATE STOCK
2. CUT OUTSIDE PROFILE USING WATERJET, THEN DRILL AND REAM HOLES



QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/13/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

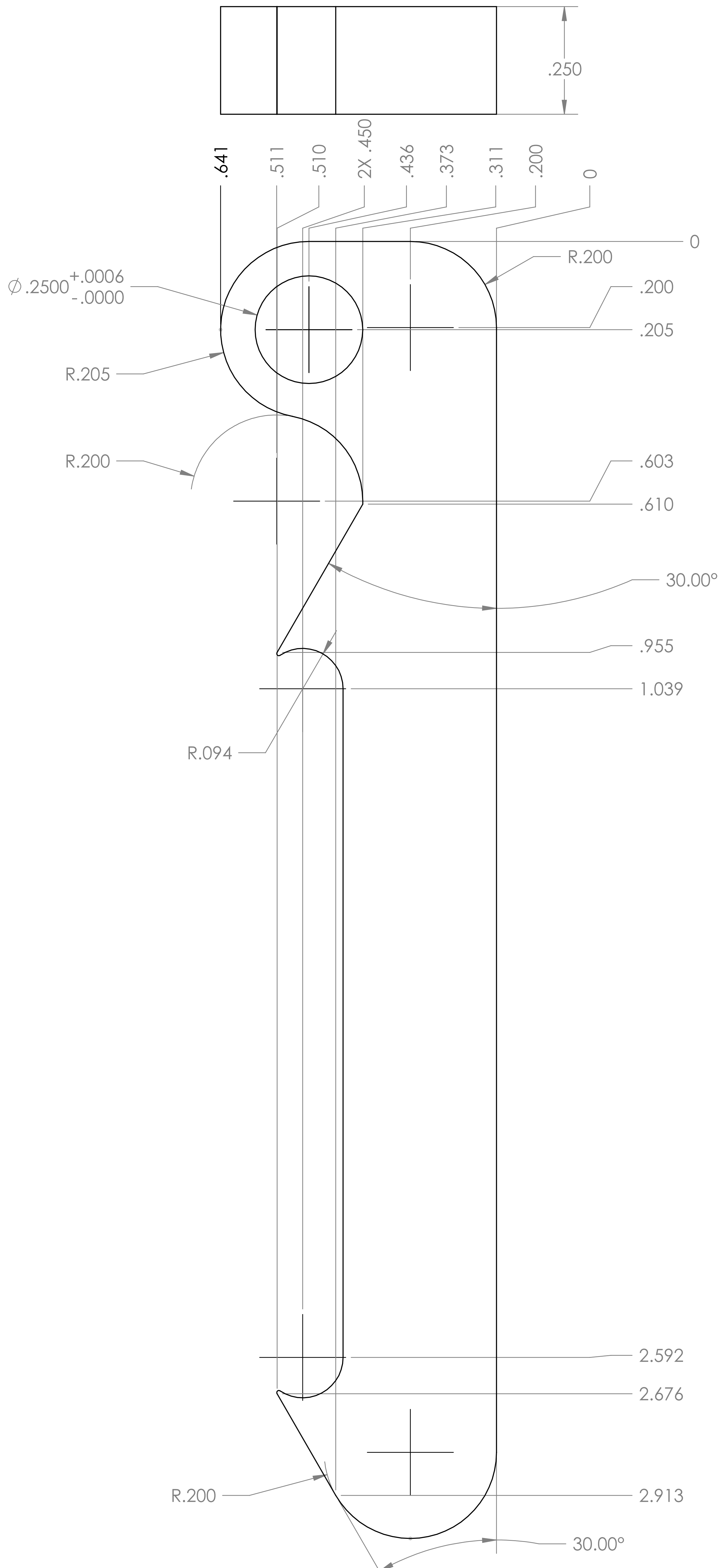
◆ = (CRITICAL DIMENSION)

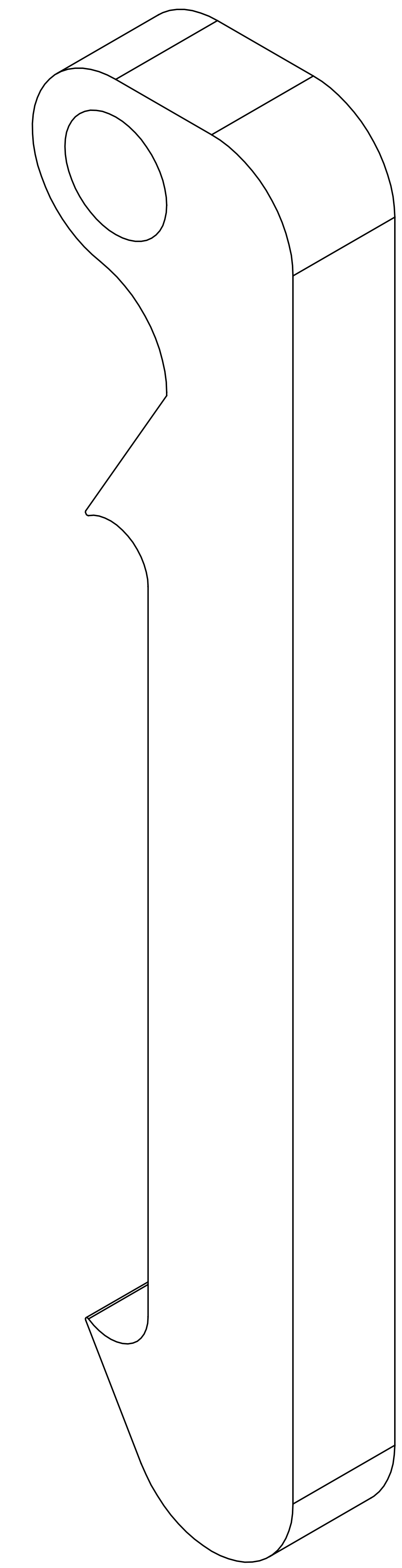
Franklin W. Olin College of Engineering

PAWL 1 SIMPLE

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | A |
| SCALE 4:1 | | WT 0.02 | SHEET 1 | OF 1 |

NOTES:
1. USE 0.25" 6061T6 ALUMINUM PLATE STOCK
2. CUT OUTSIDE PROFILE USING WATERJET, THEN DRILL AND REAM HOLES

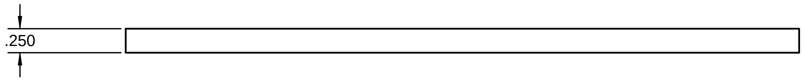


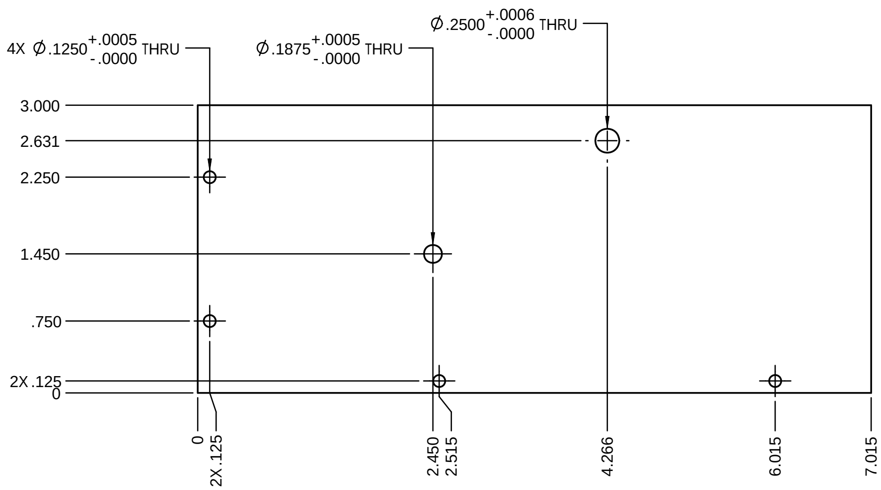



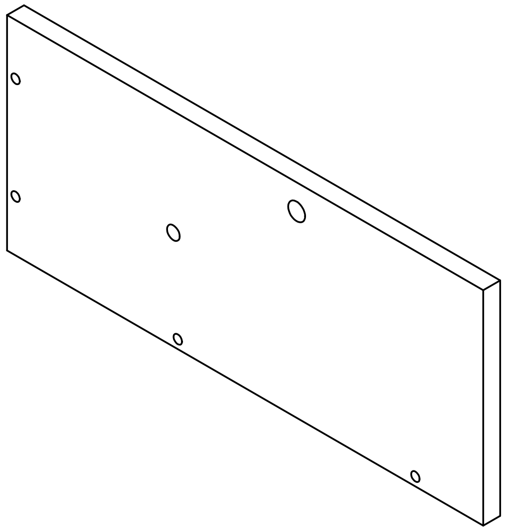
QUANTITY: 1

| | | | |
|---|---|---|---|
| ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994 | APPROVED | DATE | Franklin W. Olin College of Engineering |
| TOLERANCES UNLESS OTHERWISE SPECIFIED: X.X ± .03; X.XX ± .01; X.XXX ± .005; ∠ ± .5°; 125 | PREP BY J. GORASIA | 12/13/09 | PAWL 2 SIMPLE |
| | CHECKED R. HARRIS | 12/14/09 | |
| | RESP ENG | | |
| | MFG ENG | | |
| | QUAL ENG | | SIZE C · FSCM NO. · PART NO. · PART REV · DOC REV A |
| REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX | ⧫ = (CRITICAL DIMENSION) | | SCALE 4:1 · WT 0.03 · SHEET 1 OF 1 |

NOTES:
1. USE 1/4" 6061-T6 ALUMINUM STOCK



QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/13/09 |
| CHECKED | E. BYUN | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

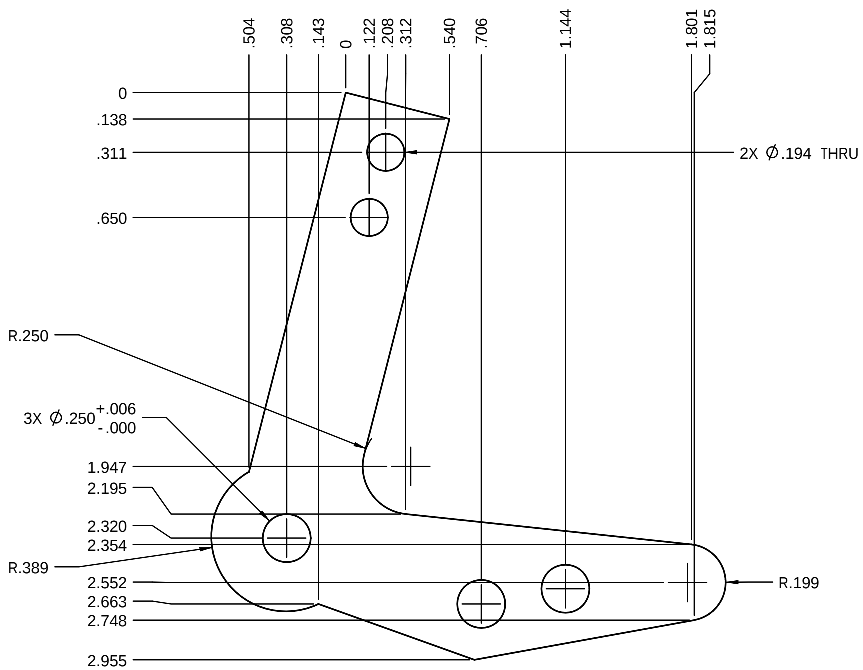
◆ = (CRITICAL DIMENSION)

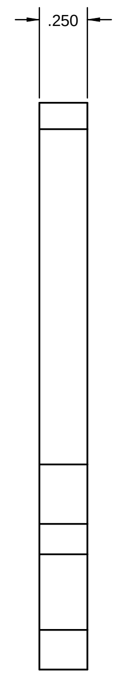
Franklin W. Olin College of Engineering

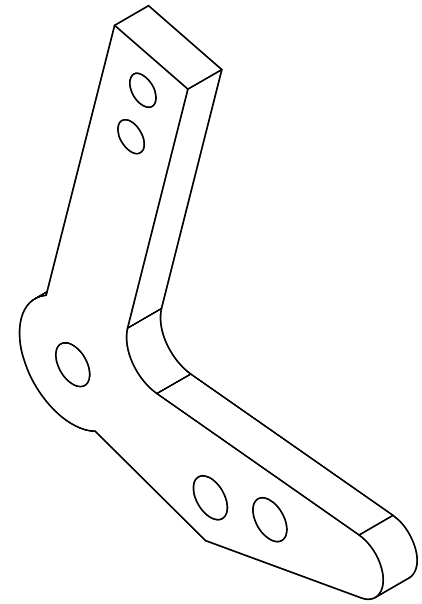
FRONT PLATE

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 1:1 | | WT | SHEET 1 OF 1 | |

NOTES:
1. USE 0.25" 6061T6 ALUMINUM PLATE STOCK
2. CUT OUTSIDE PROFILE USING WATERJET, THEN DRILL AND REAM HOLES



QUANTITY: 2

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03   ∠ ± .5°
X.XX ± .01
X.XXX ± .005   125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/13/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

♦ = (CRITICAL DIMENSION)

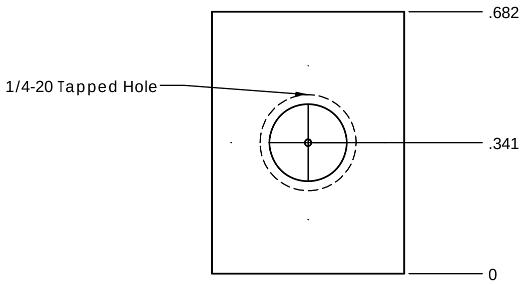
Franklin W. Olin College of Engineering

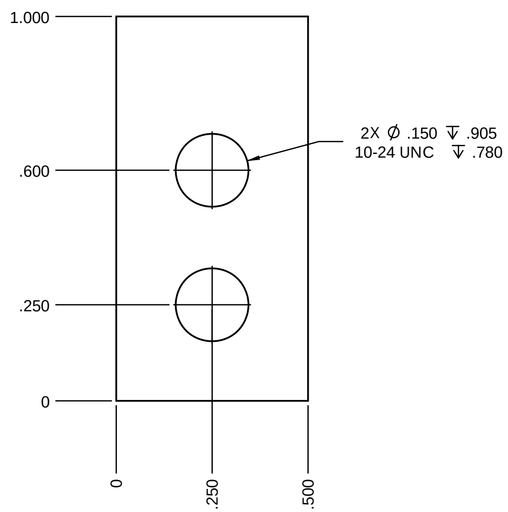
TOP ARM

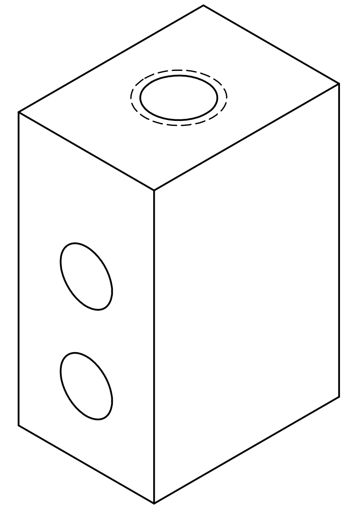
SIZE C | FSCM NO. | PART NO. | PART REV | DOC REV

SCALE 2:1 | WT | SHEET 1 OF 1

NOTES:
1. USE ALUMINUM 6061 T6



QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03  ∠ ± .5°
X.XX ± .01
X.XXX ± .005  125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/14/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

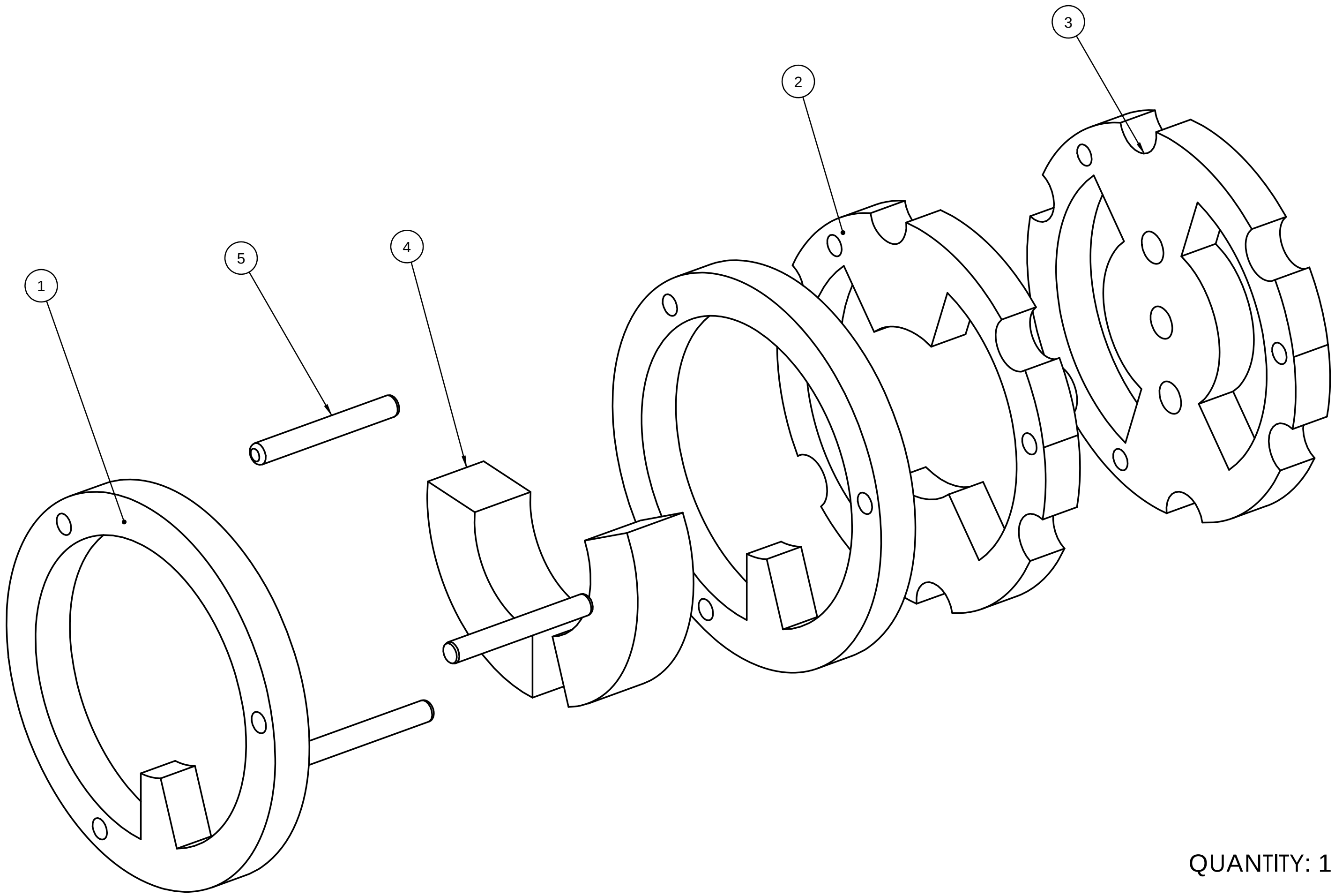
⧫ = (CRITICAL DIMENSION)

Franklin W. Olin College of Engineering

HANDLE SHAFT

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 4:1 | | WT | SHEET 1 OF 1 | |

| ITEM NO. | PART NUMBER | DESCRIPTION | QTY. |
|---|---|---|---|
| 1 | CUP PLATE 1 | | 2 |
| 2 | CUP PLATE 2 | | 1 |
| 3 | CUP PLATE 3 | | 1 |
| 4 | 13.32 SPRING | MCMASTER CARR PART #:9657K37 | 2 |
| 5 | DPM 0.125x1 | 1/8"x1" STEEL DOWEL PIN | 3 |




QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| | APPROVED | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/14/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

◆ = (CRITICAL DIMENSION)

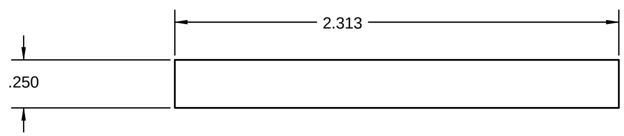
Franklin W. Olin College of Engineering

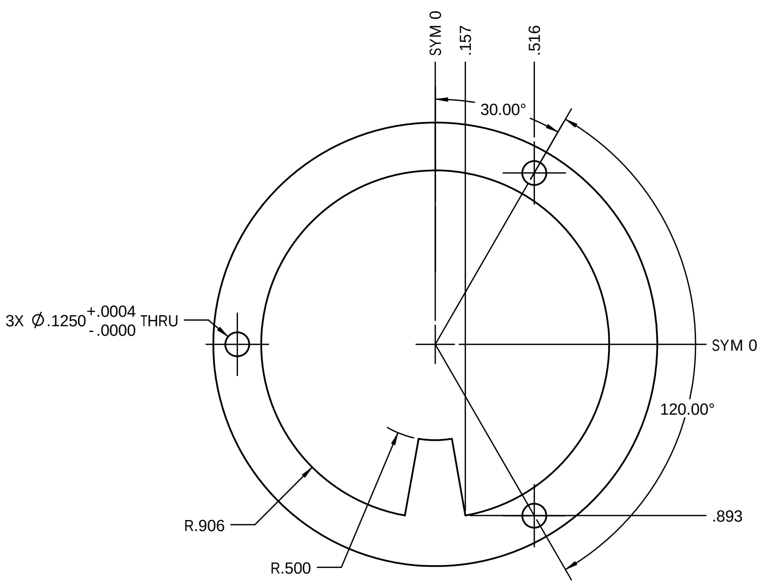
CUP ASSEMBLY

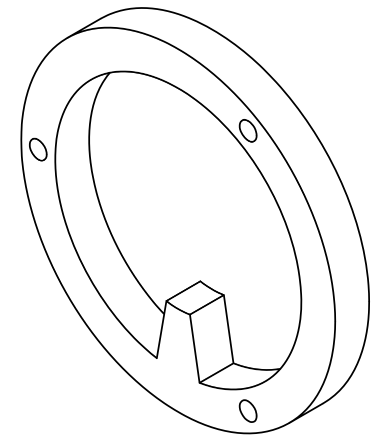
SIZE C | FSCM NO. | PART NO. | PART REV | DOC REV

SCALE 2:1 | WT | SHEET 1 OF 1

NOTES:
1. USE 0.25" 6061T6 ALUMINUM PLATE STOCK
2. CUT OUTSIDE PROFILE USING WATERJET, THEN DRILL AND REAM HOLES



QUANTITY: 2

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/13/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

◆ = (CRITICAL DIMENSION)

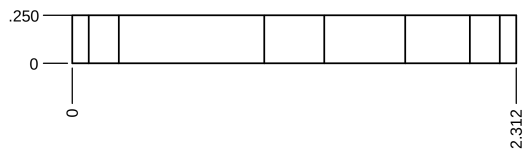
Franklin W. Olin College of Engineering

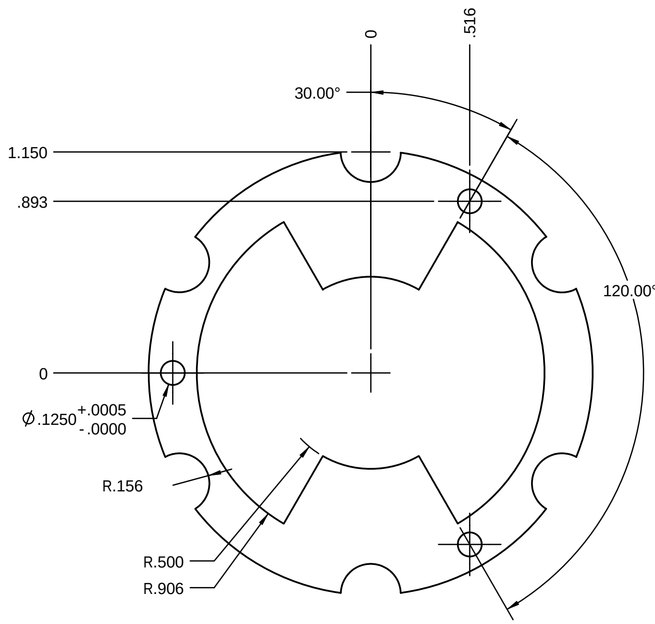
CUP PLATE 1

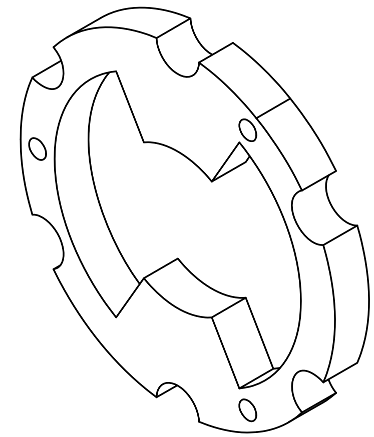
| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |

SCALE 2:1 | WT | SHEET 1 OF 1

NOTES:
1. USE 0.25" 6061T6 ALUMINUM PLATE STOCK
2. CUT OUTSIDE PROFILE USING WATERJET, THEN DRILL AND REAM HOLES



QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/13/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

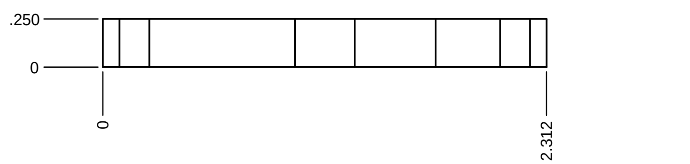
◆ = (CRITICAL DIMENSION)

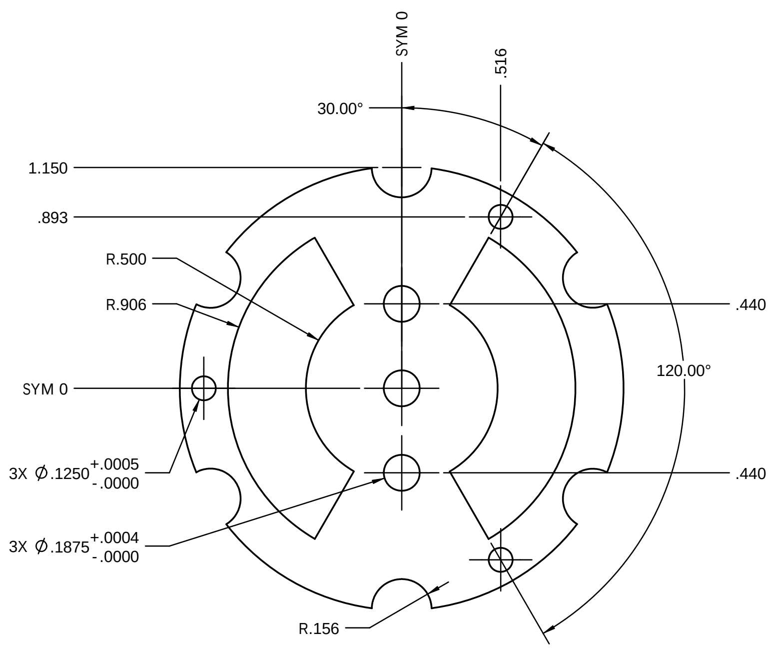
Franklin W. Olin College of Engineering

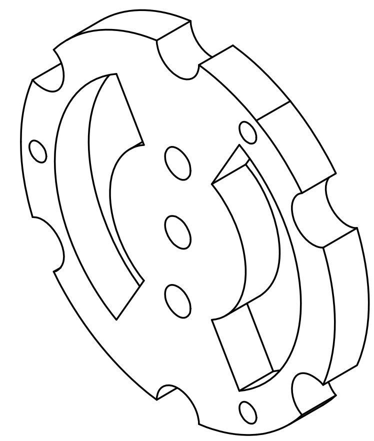
CUP PLATE 2

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 2:1 | | WT | SHEET 1 OF 1 | |

NOTES:
1. USE 0.25" 6061T6 ALUMINUM PLATE STOCK
2. CUT OUTSIDE PROFILE USING WATERJET, THEN DRILL AND REAM HOLES



QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/13/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

♦ = (CRITICAL DIMENSION)

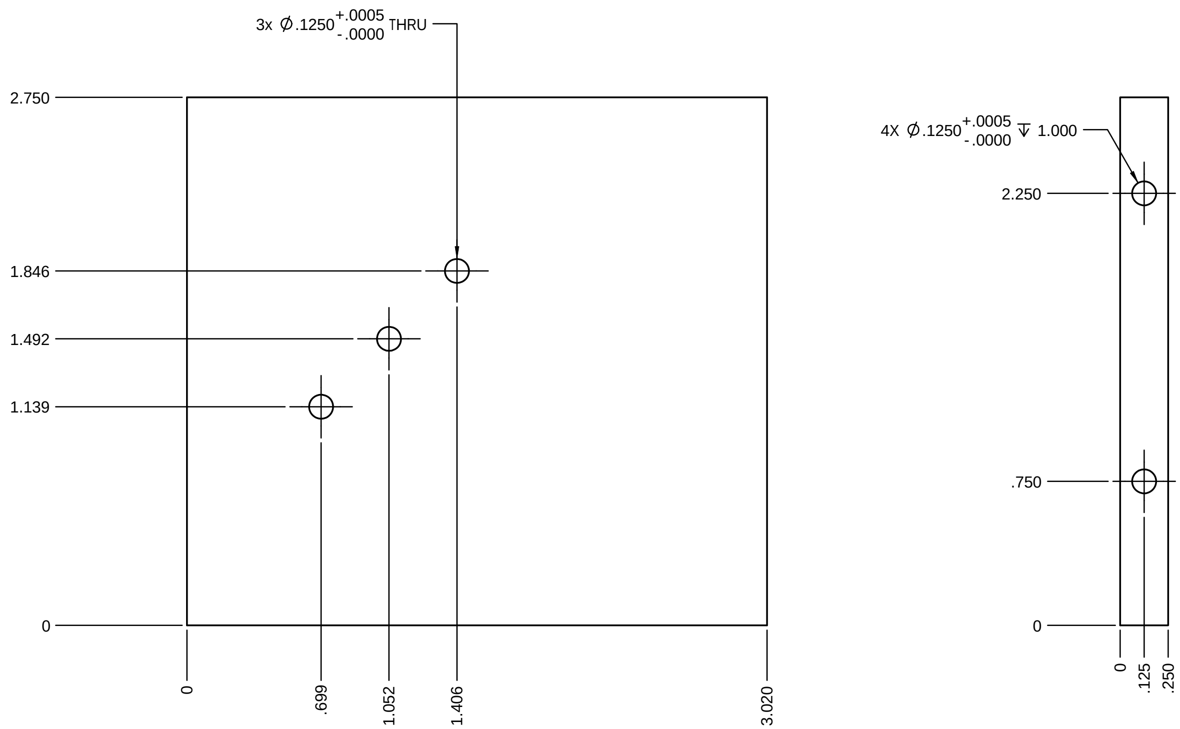
Franklin W. Olin College of Engineering

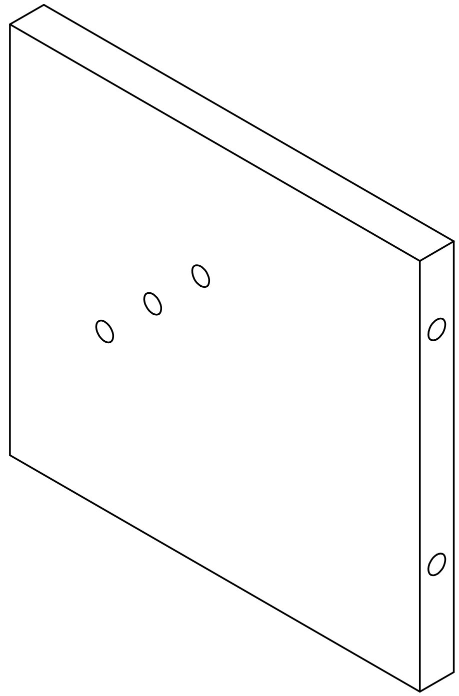
CUP PLATE 3

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |

SCALE 2:1 | WT | SHEET 1 OF 1

NOTES:
1. USE 1/4" 6061-T6 ALUMINUM STOCK



QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| | APPROVED | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/13/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

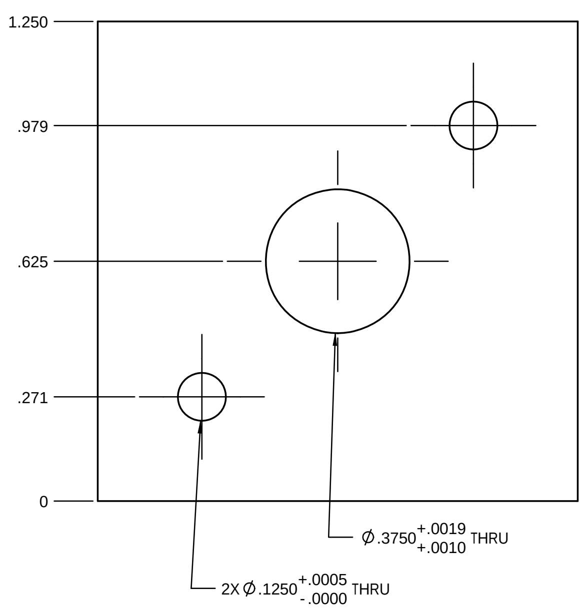
⧫ = (CRITICAL DIMENSION)

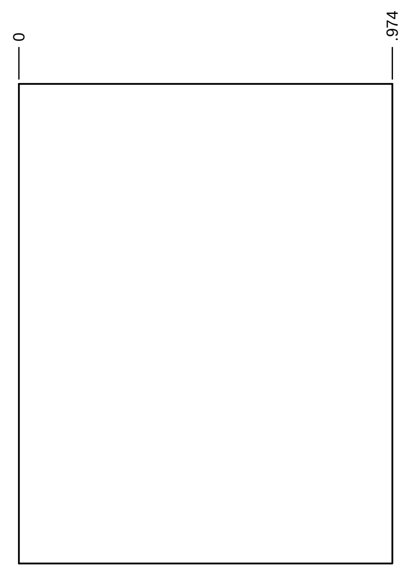
Franklin W. Olin College of Engineering

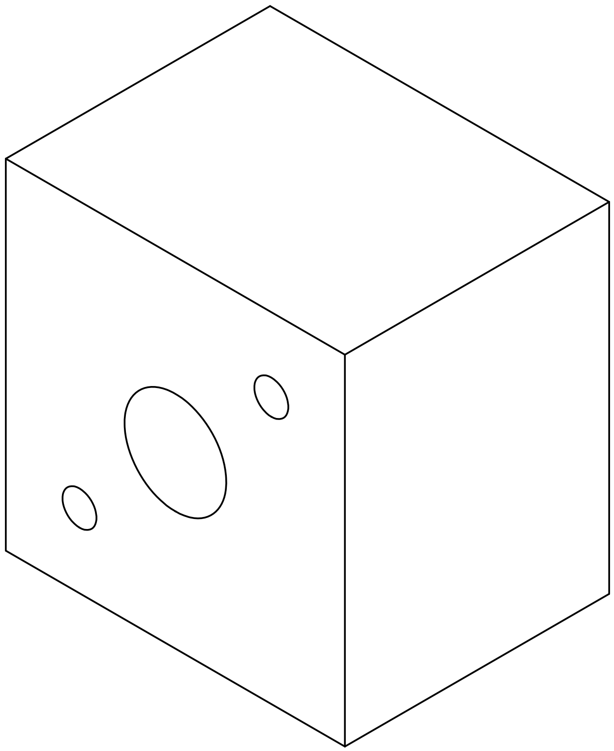
OUTSIDE SIDE HOUSING

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 2:1 | | WT | SHEET 1 OF 1 | |

NOTES:
1. USE ALUMINUM 6061-T6



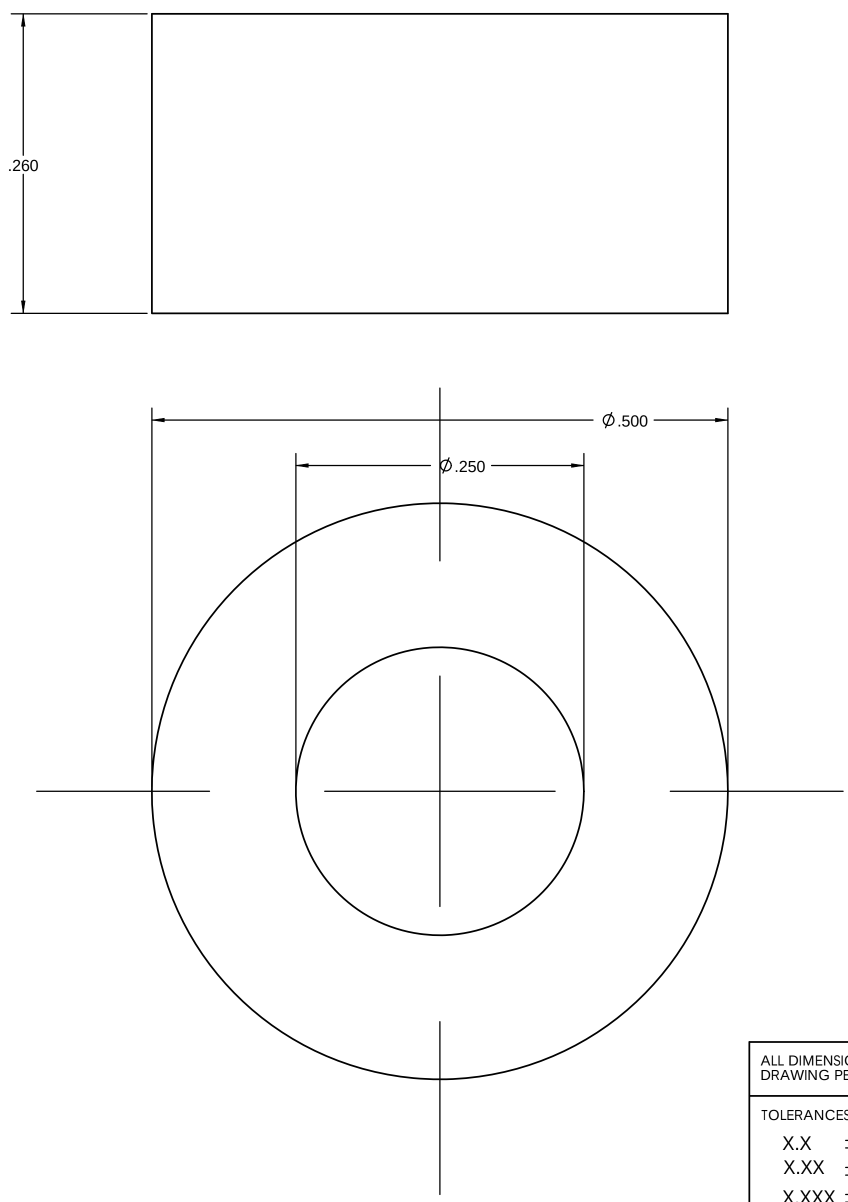



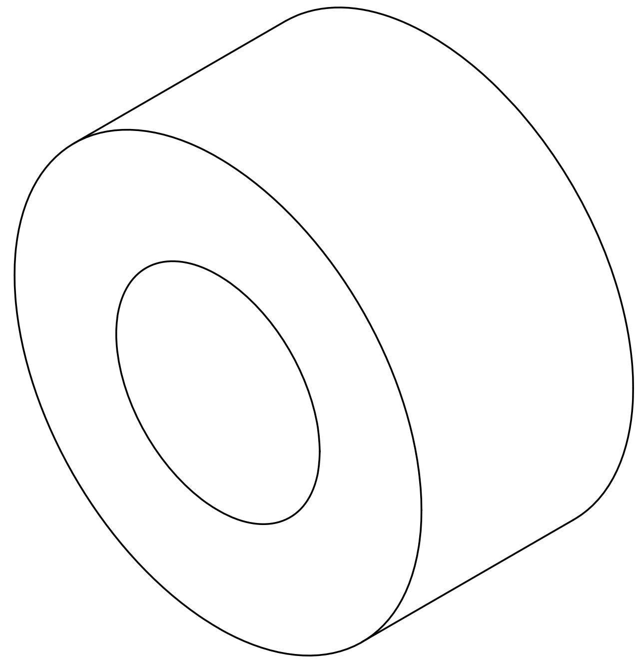



QUANTITY: 1

| | | | |
|---|---|---|---|
| ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994 | APPROVED | DATE | Franklin W. Olin College of Engineering |
| | PREP BY J. GORASIA | 12/13/09 | |
| TOLERANCES UNLESS OTHERWISE SPECIFIED: X.X ± .03, X.XX ± .01, X.XXX ± .005, ∠ ± .5°, 125√ | CHECKED R. HARRIS | 12/14/09 | PLUNGER OFFSET BLOCK |
| | RESP ENG | | |
| | MFG ENG | | |
| | QUAL ENG | | SIZE C; FSCM NO.; PART NO.; PART REV; DOC REV |
| REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX | ⧫ = (CRITICAL DIMENSION) | | SCALE 4:1; WT; SHEET 1 OF 1 |

NOTES:
1. USE NYLON 6 STOCK OR PURCHASE FEMALE UNTHREADED SPACERS AND CUT TO SIZE

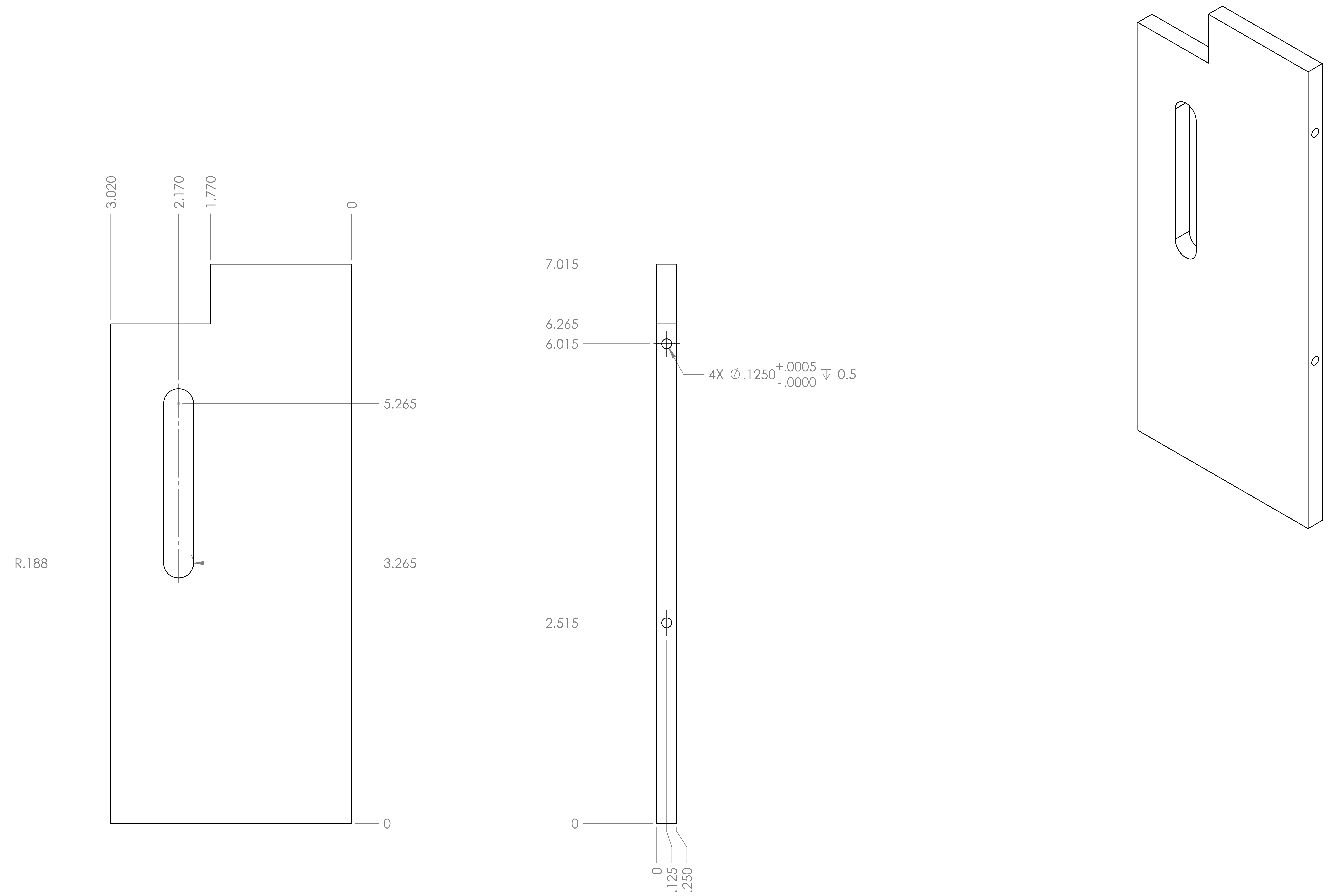


QUANTITY: 1

| | | | |
|---|---|---|---|
| ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994 | APPROVED | DATE | Franklin W. Olin College of Engineering |
| | PREP BY J. GORASIA | 12/13/09 | |
| TOLERANCES UNLESS OTHERWISE SPECIFIED: X.X ± .03, X.XX ± .01, X.XXX ± .005, ∠ ± .5°, 125√ | CHECKED R. HARRIS | 12/14/09 | TOP ARM SPACER |
| | RESP ENG | | |
| | MFG ENG | | |
| | QUAL ENG | | SIZE C · FSCM NO. · PART NO. · PART REV · DOC REV |
| REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX | ◆ = (CRITICAL DIMENSION) | | SCALE 12:1 · WT · SHEET 1 OF 1 |

NOTES:
1. USE 1/4" 6061-T6 ALUMINUM STOCK




QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| | APPROVED | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/13/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

◆ = (CRITICAL DIMENSION)

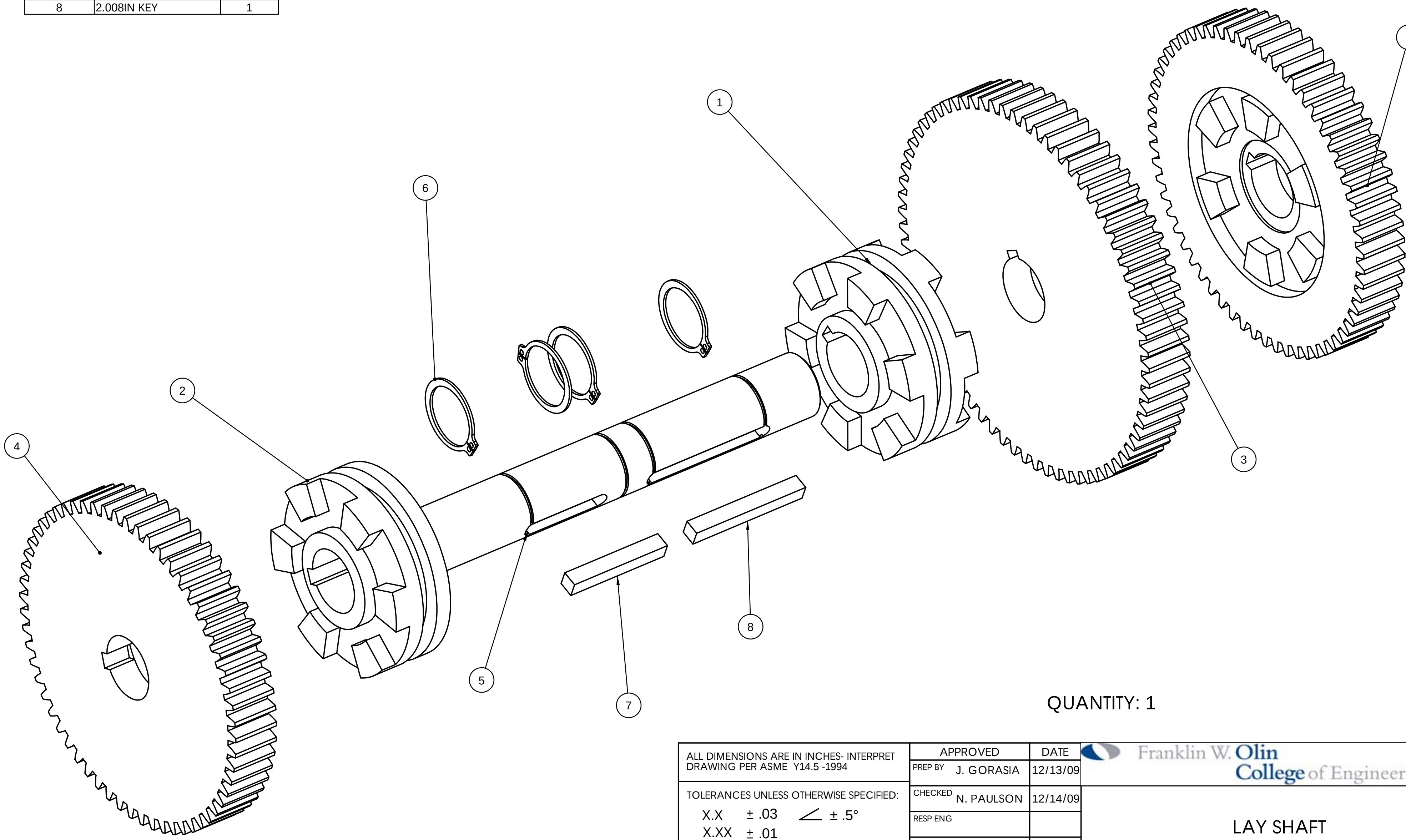
Franklin W. Olin College of Engineering

OUTSIDE BOTTOM HOUSING

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |

| SCALE | 1:1 | WT | SHEET 1 | OF 1 |
|---|---|---|---|---|

| ITEM NO. | PART NUMBER | QTY. |
|---|---|---|
| 1 | SPIDER | 1 |
| 2 | SPIDER2 | 1 |
| 3 | GD70 | 1 |
| 4 | GD60 | 2 |
| 5 | LAYSHAFTMACHINING | 1 |
| 6 | B27.1 - NA1-100 | 4 |
| 7 | 1.712INCH KEY | 1 |
| 8 | 2.008IN KEY | 1 |




QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:

X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| | APPROVED | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/13/09 |
| CHECKED | N. PAULSON | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

♦ = (CRITICAL DIMENSION)

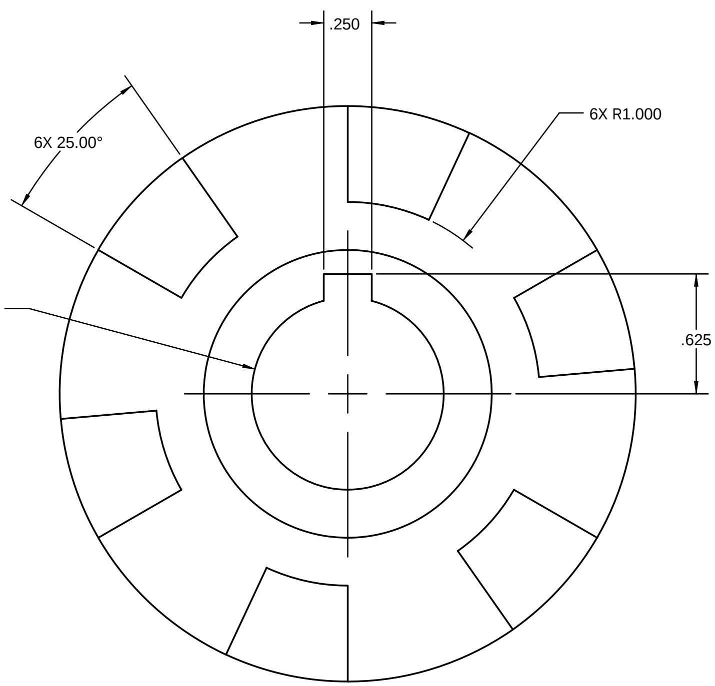
Franklin W. Olin College of Engineering

LAY SHAFT

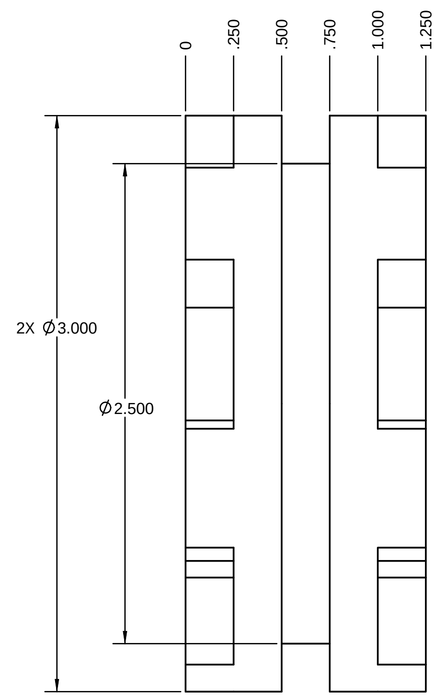
SIZE C | FSCM NO. | PART NO. | PART REV | DOC REV

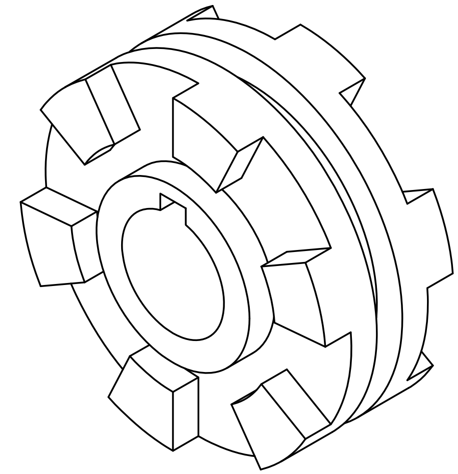
SCALE 1:1 | WT | SHEET 1 OF 1

NOTE
1. USE 1018 MILD STEEL STOCK




NOTE: USE SAME DIMENSIONS FOR REVERSE SIDE



SCALE 3:2

QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | N. PAULSON | 12/14/09 |
| CHECKED | J. GORASIA | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

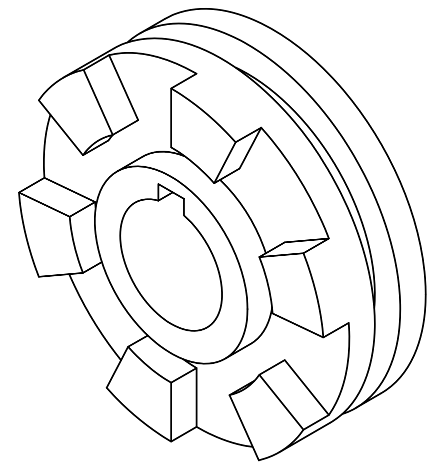
♦ = (CRITICAL DIMENSION)

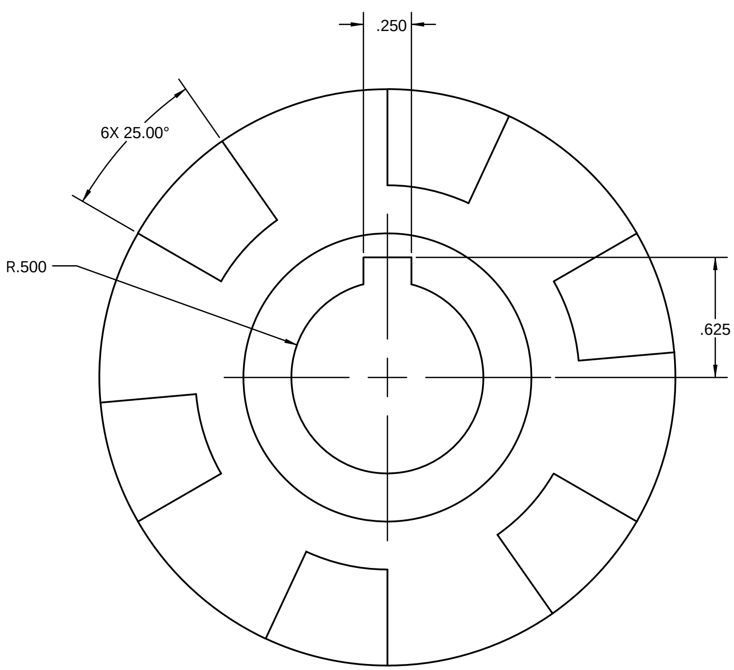
Franklin W. Olin College of Engineering

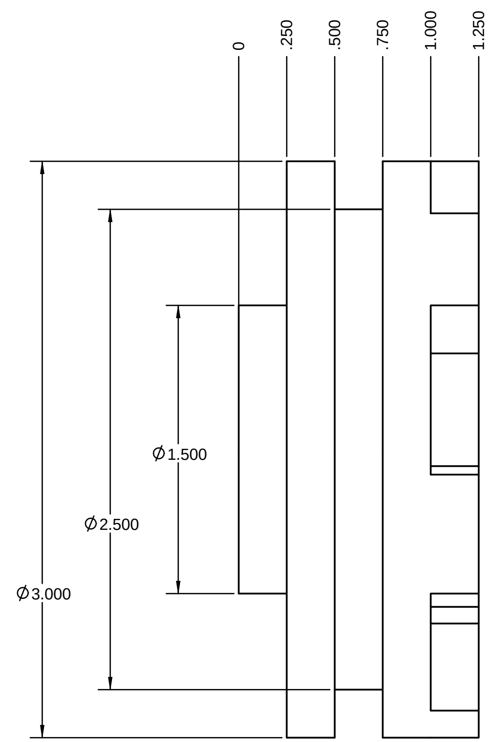
SPIDER

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 2:1 | | WT | SHEET 1 OF 1 | |

NOTE
1. USE 1018 MILD STEEL STOCK






QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03
X.XX ± .01
X.XXX ± .005
∠ ± .5°
125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| | APPROVED | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 11/6/09 |
| CHECKED | N. PAULSON | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

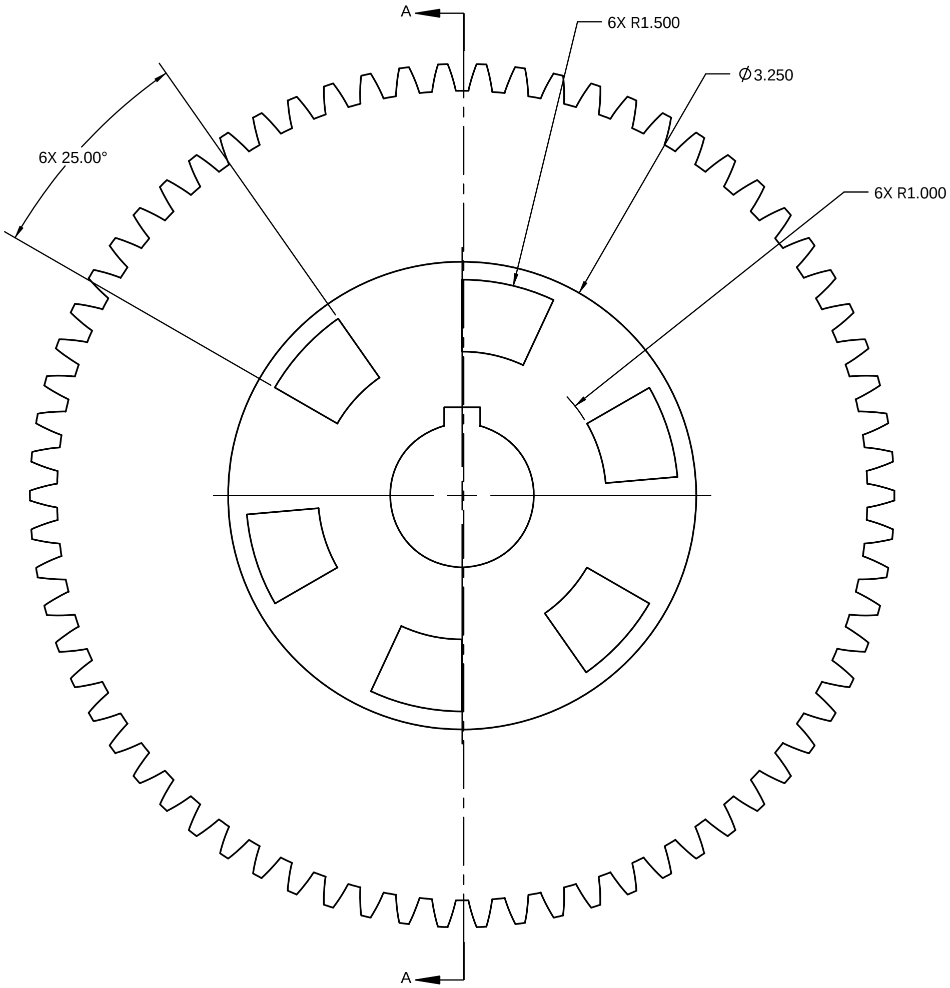
⧫ = (CRITICAL DIMENSION)

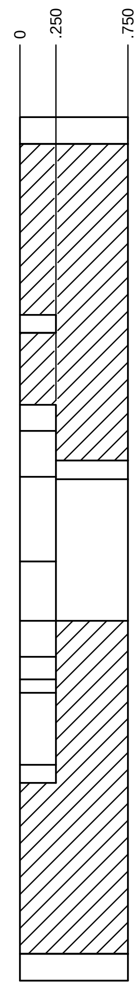
Franklin W. Olin College of Engineering

# SPIDER 2

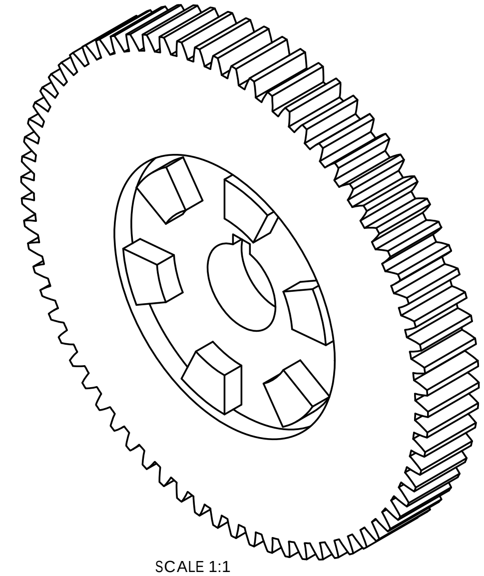
| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 2:1 | | WT | SHEET 1 | OF 1 |

NOTES:
1. USE 3/4 INCH GD70 CAST IRON SPUR GEARS




SECTION A-A



SCALE 1:1

QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03
X.XX ± .01
X.XXX ± .005
∠ ± .5°
125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | N. PAULSON | 12/14/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

◆ = (CRITICAL DIMENSION)

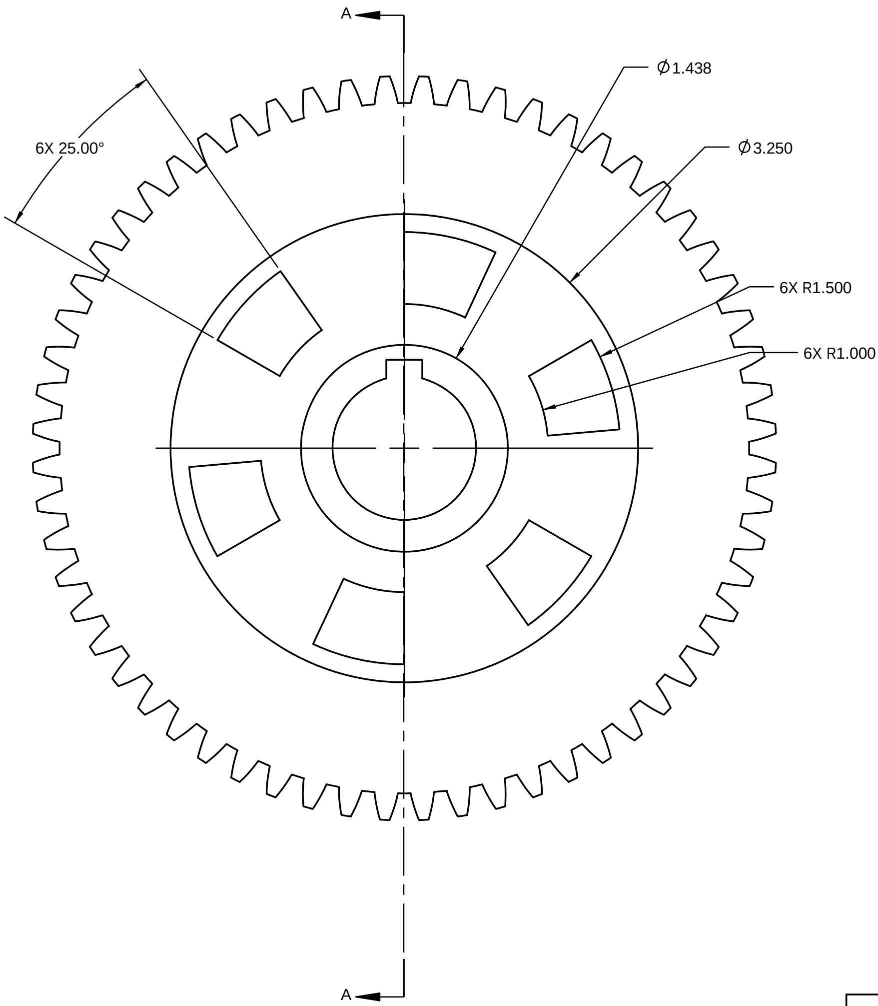
Franklin W. Olin College of Engineering

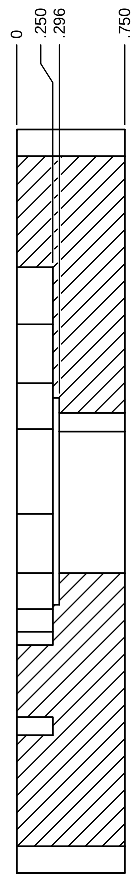
GD70

SIZE C | FSCM NO. | PART NO. | PART REV | DOC REV

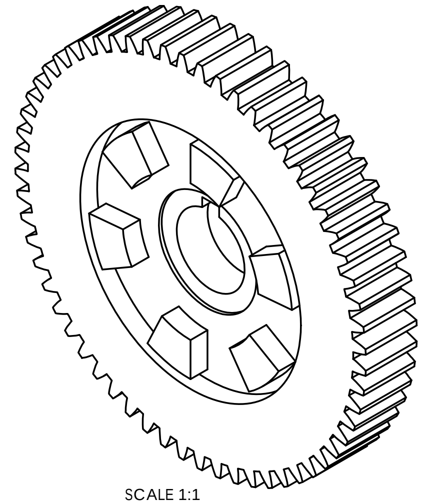
SCALE 3:2 | WT | SHEET 1 OF 1

NOTES:
1. USE 3/4 INCH GD60 CAST IRON SPUR GEARS




SECTION A-A



SCALE 1:1

QUANTITY: 2

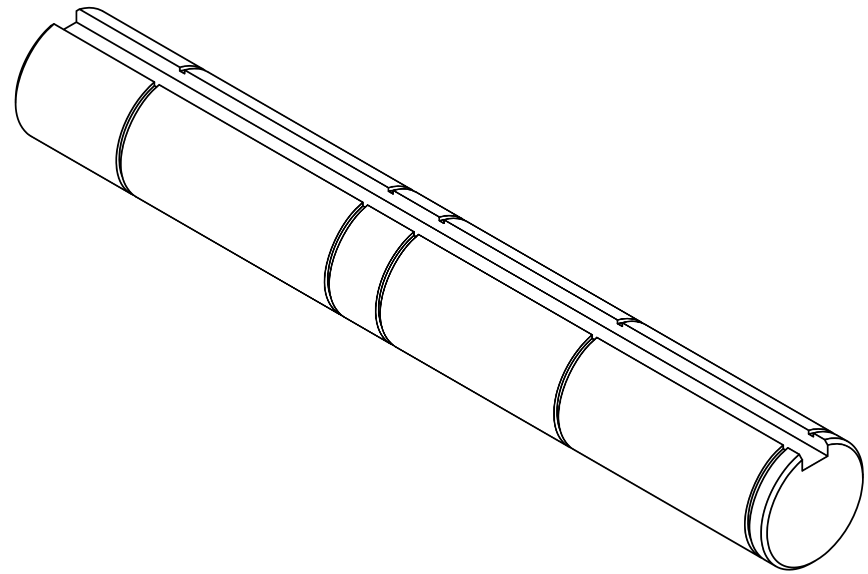
| | | |
|---|---|---|
| ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994 | APPROVED | DATE |
| TOLERANCES UNLESS OTHERWISE SPECIFIED:<br>X.X ± .03 ∠ ± .5°<br>X.XX ± .01<br>X.XXX ± .005 125√ | PREP BY N. PAULSON | 12/14/09 |
| | CHECKED R. HARRIS | 12/14/09 |
| | RESP ENG | |
| | MFG ENG | |
| | QUAL ENG | |
| REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX | ◆ = (CRITICAL DIMENSION) | |

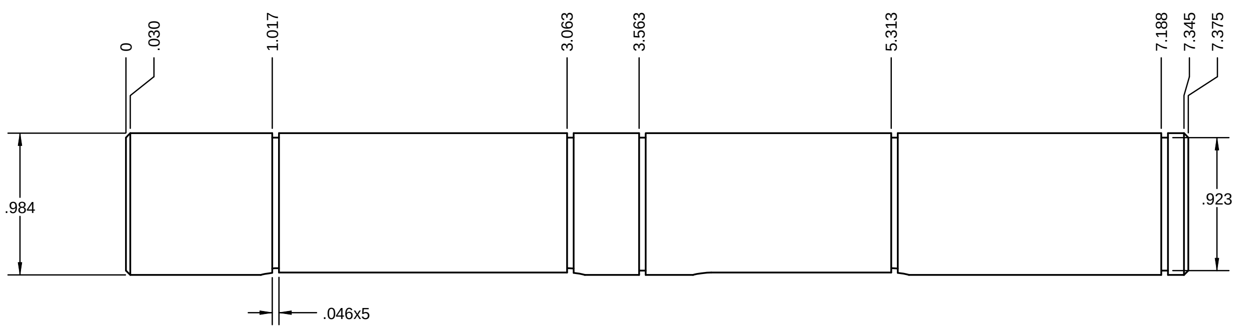
Franklin W. Olin College of Engineering

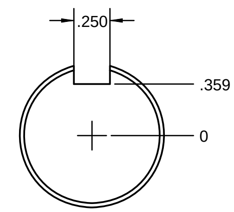
GD60

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 3:2 | | WT | SHEET 1 | OF 1 |

NOTES:
1. USE 1 INCH DIAMETER 1045 UNHARDENED STEEL SHAFT






QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 11/6/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

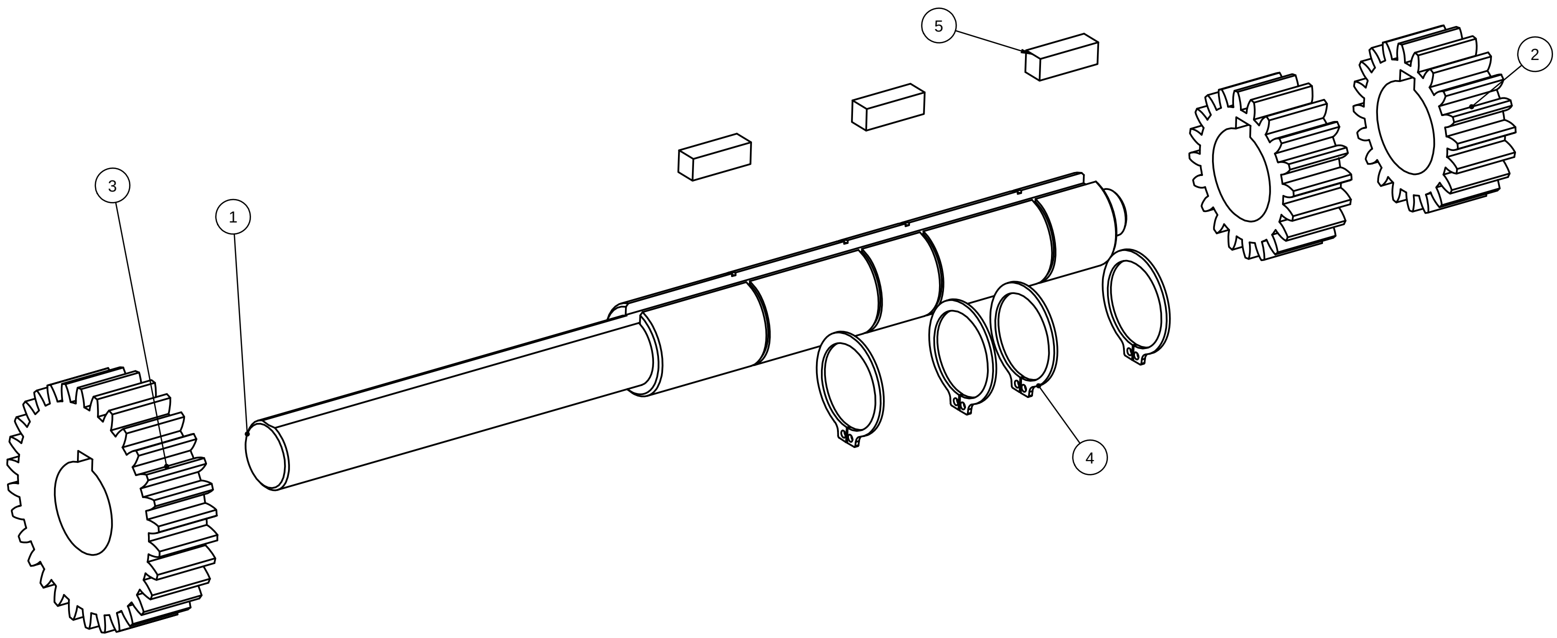
⧫ = (CRITICAL DIMENSION)

Franklin W. Olin College of Engineering

LAYSHAFTMACHINING

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 3:2 | | WT | SHEET 1 OF 1 | |

| ITEM NO. | PART NUMBER | DESCRIPTION | QTY. |
|---|---|---|---|
| 1 | INPUTSHAFTMACHINING | | 1 |
| 2 | INCH - SPUR GEAR 12DP 20T 14.5PA 0.75FW ---S20N3.0H2.0L1.0S1 | | 2 |
| 3 | INCH - SPUR GEAR 12DP 30T 14.5PA 0.75FW ---S30N3.0H2.0L1.0S1 | | 1 |
| 4 | 1 INCH EXTERNAL RETAINING RING | | 4 |
| 5 | .75 INCH LONG KEY | .25 X .25 INCH UNDERSIZED STEEL | 3 |




QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/13/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

◆ = (CRITICAL DIMENSION)

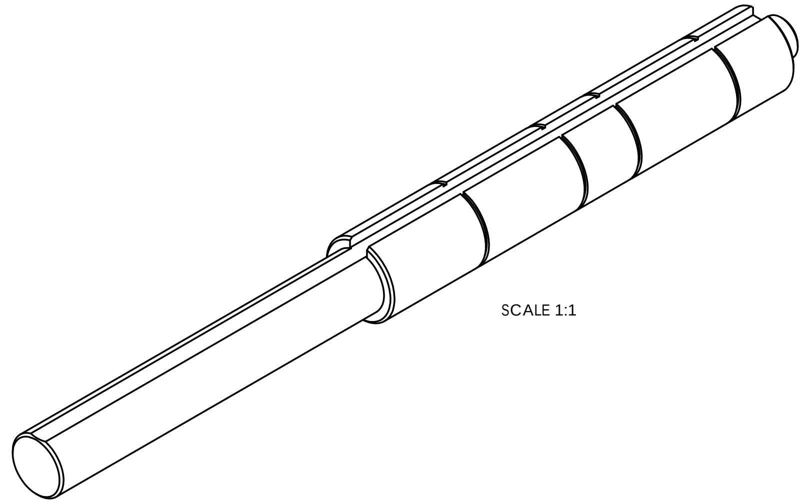
Franklin W. Olin College of Engineering

INPUT SHAFT

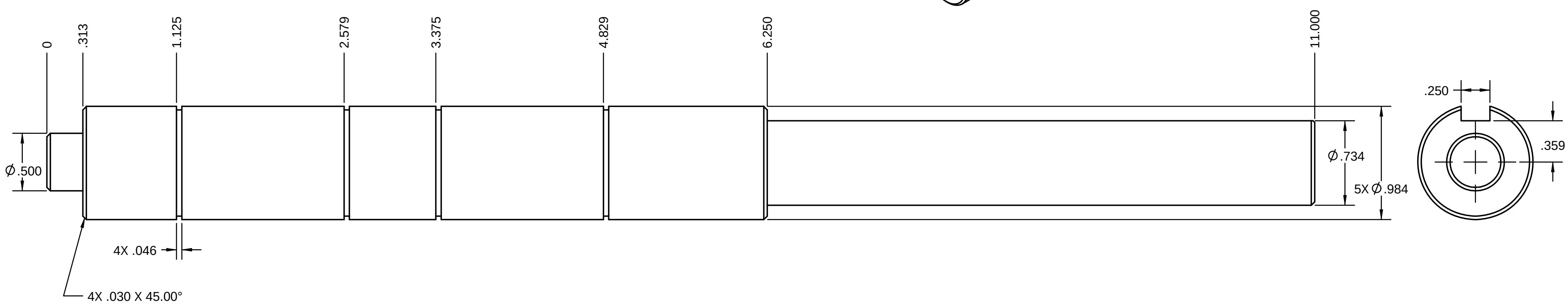
SIZE C | FSCM NO. | PART NO. | PART REV | DOC REV

SCALE 1:1 | WT | SHEET 1 OF 1

NOTES:
1. USE 1 INCH DIAMETER 1045 UNHARDENED STEEL SHAFT



SCALE 1:1




QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:

| | | |
|---|---|---|
| X.X | ± .03 | ∠ ± .5° |
| X.XX | ± .01 | |
| X.XXX | ± .005 | 125√ |

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| | APPROVED | DATE |
|---|---|---|
| PREP BY | N. PAULSON | 12/14/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

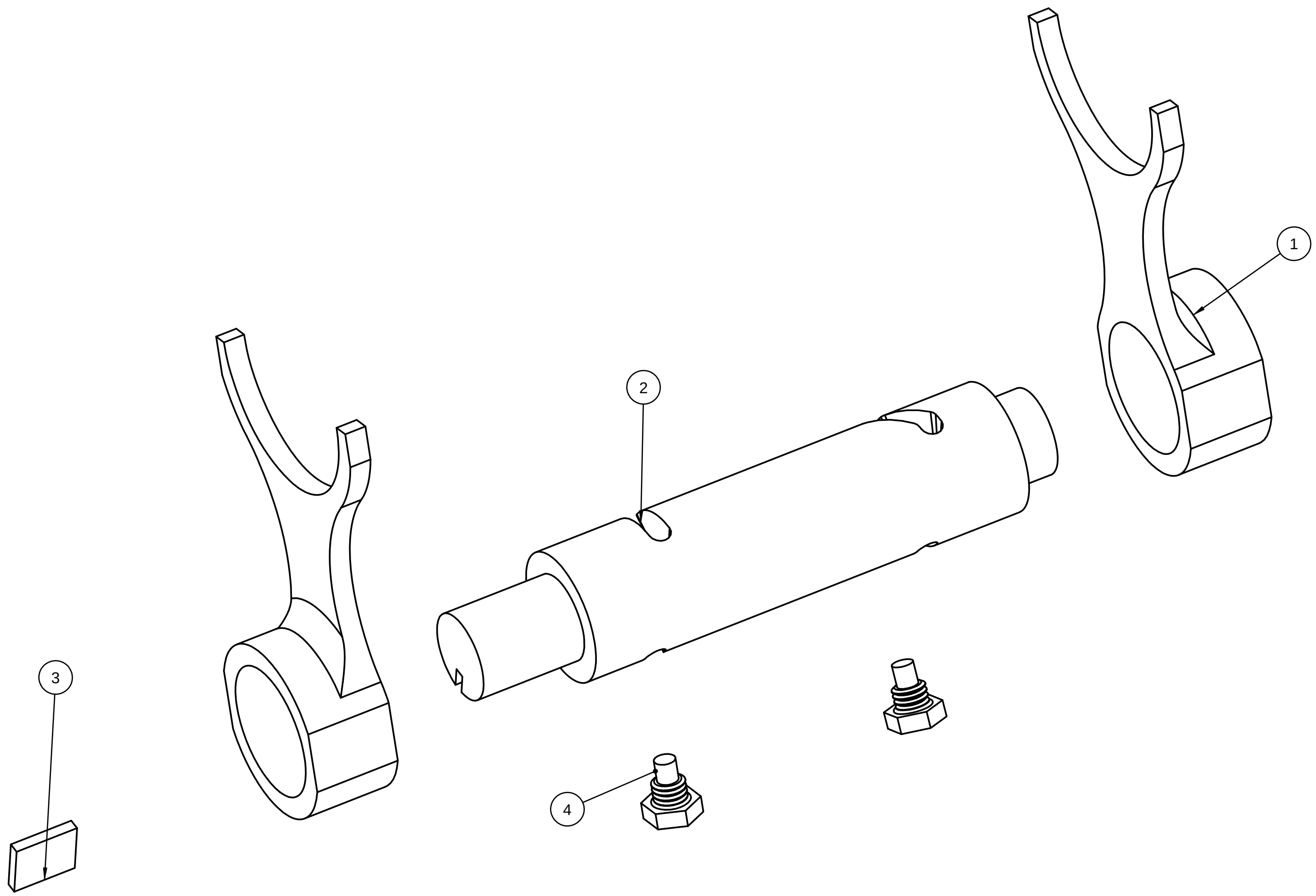
♦ = (CRITICAL DIMENSION)

Franklin W. Olin College of Engineering

INPUTSHAFTMACHINING

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 3:2 | | WT | SHEET 1 OF 1 | |

| ITEM NO. | PART NUMBER | QTY. |
|---|---|---|
| 1 | SELECTOR | 2 |
| 2 | BARREL CAM | 1 |
| 3 | BARREL CAM NUB | 1 |
| 4 | PIN-BOLT | 2 |




QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | N. PAULSON | 12/14/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

♦ = (CRITICAL DIMENSION)

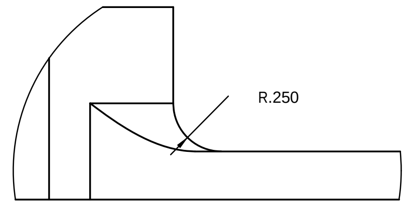
Franklin W. Olin College of Engineering

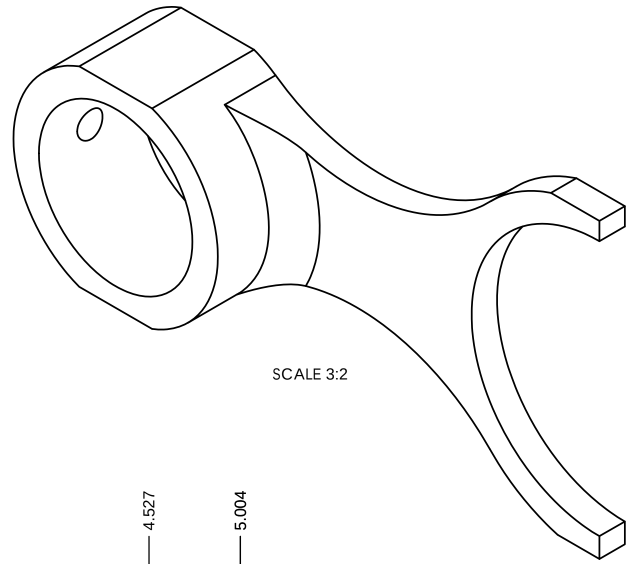
BARREL CAM AND SHIFTERS

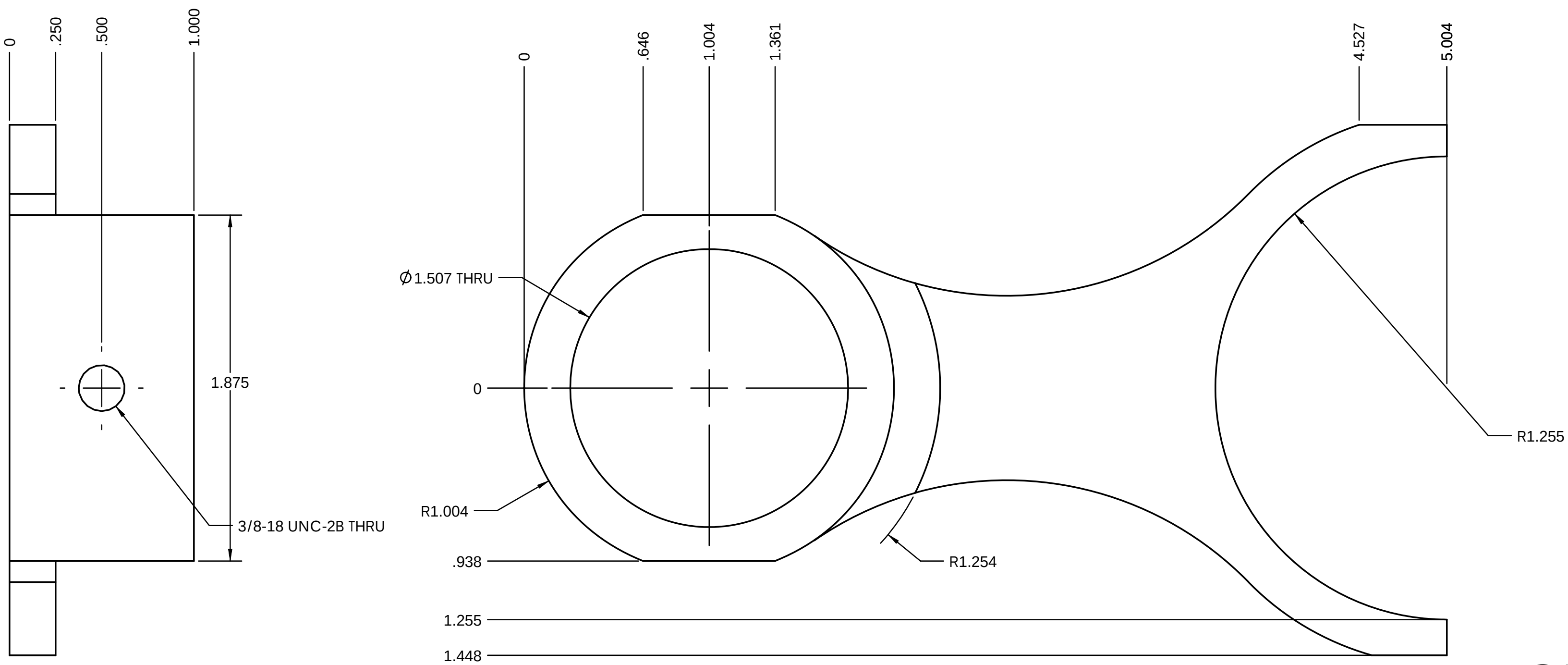
SIZE C | FSCM NO. | PART NO. | PART REV | DOC REV

SCALE 1:1 | WT | SHEET 1 OF 1

NOTES:
1. USE ALUMINUM 6061 T6 STOCK



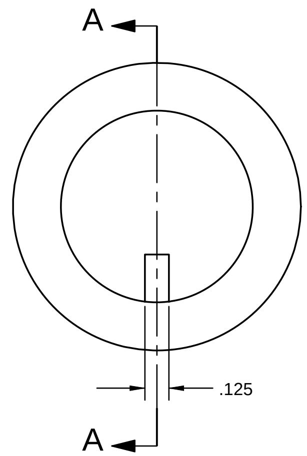



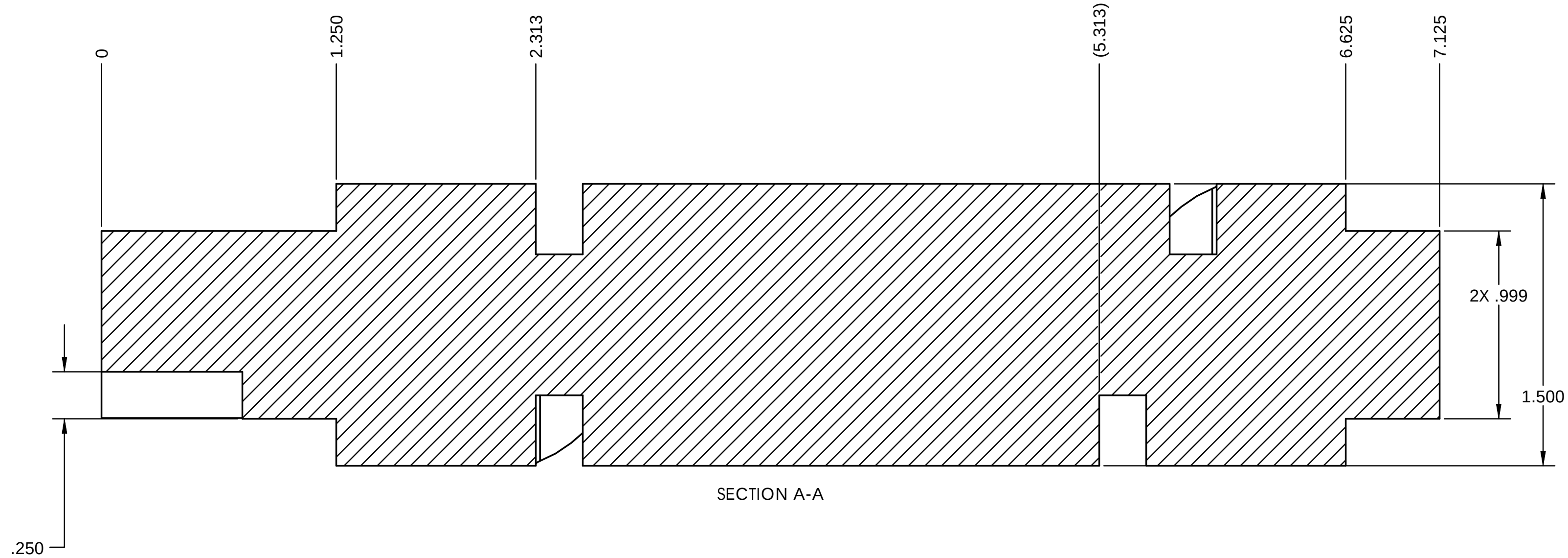




QUANTITY: 2

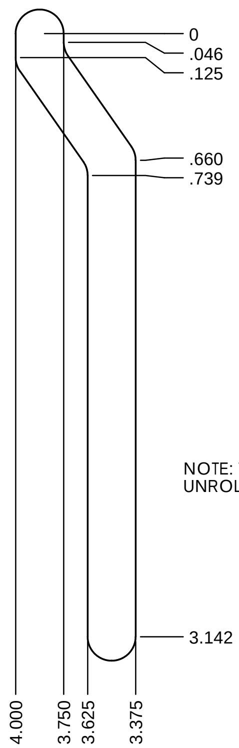
| | | | |
|---|---|---|---|
| ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994 | APPROVED | DATE | Franklin W. Olin College of Engineering |
| TOLERANCES UNLESS OTHERWISE SPECIFIED: X.X ± .03; X.XX ± .01; X.XXX ± .005; ∠ ± .5°; 125√ | PREP BY N. PAULSON | 12/14/09 | SELECTOR |
| | CHECKED R. HARRIS | 12/14/09 | |
| | RESP ENG | | |
| | MFG ENG | | |
| REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX | QUAL ENG | | SIZE C; FSCM NO.; PART NO.; PART REV; DOC REV |
| | ⧫ = (CRITICAL DIMENSION) | | SCALE 2:1; WT; SHEET 1 OF 1 |

MATERIAL: ALUMINUM 6061 T6

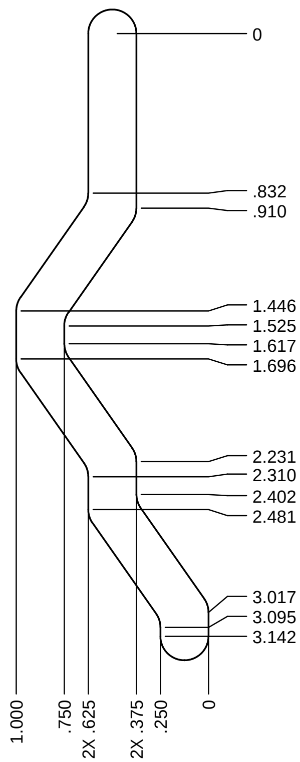


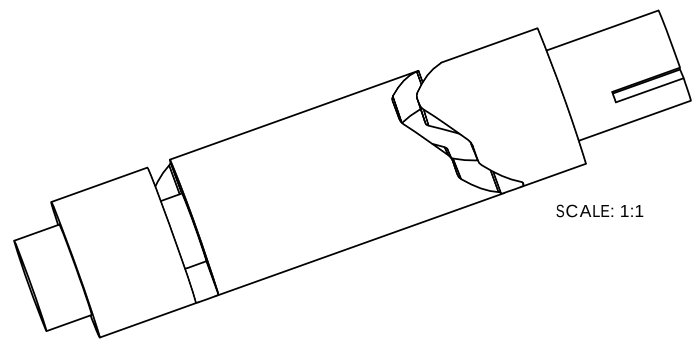




SECTION A-A




NOTE: THE 4 AXIS CNC USES THESE UNROLLED PROFILES FOR CUTTING



SCALE: 1:1

QUANTITY: 1

ALL CORNERS .125 INCH RADIUS

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | DATE |
|---|---|
| PREP BY N. PAULSON | 12/14/09 |
| CHECKED R. HARRIS | 12/14/09 |
| RESP ENG | |
| MFG ENG | |
| QUAL ENG | |

◆ = (CRITICAL DIMENSION)

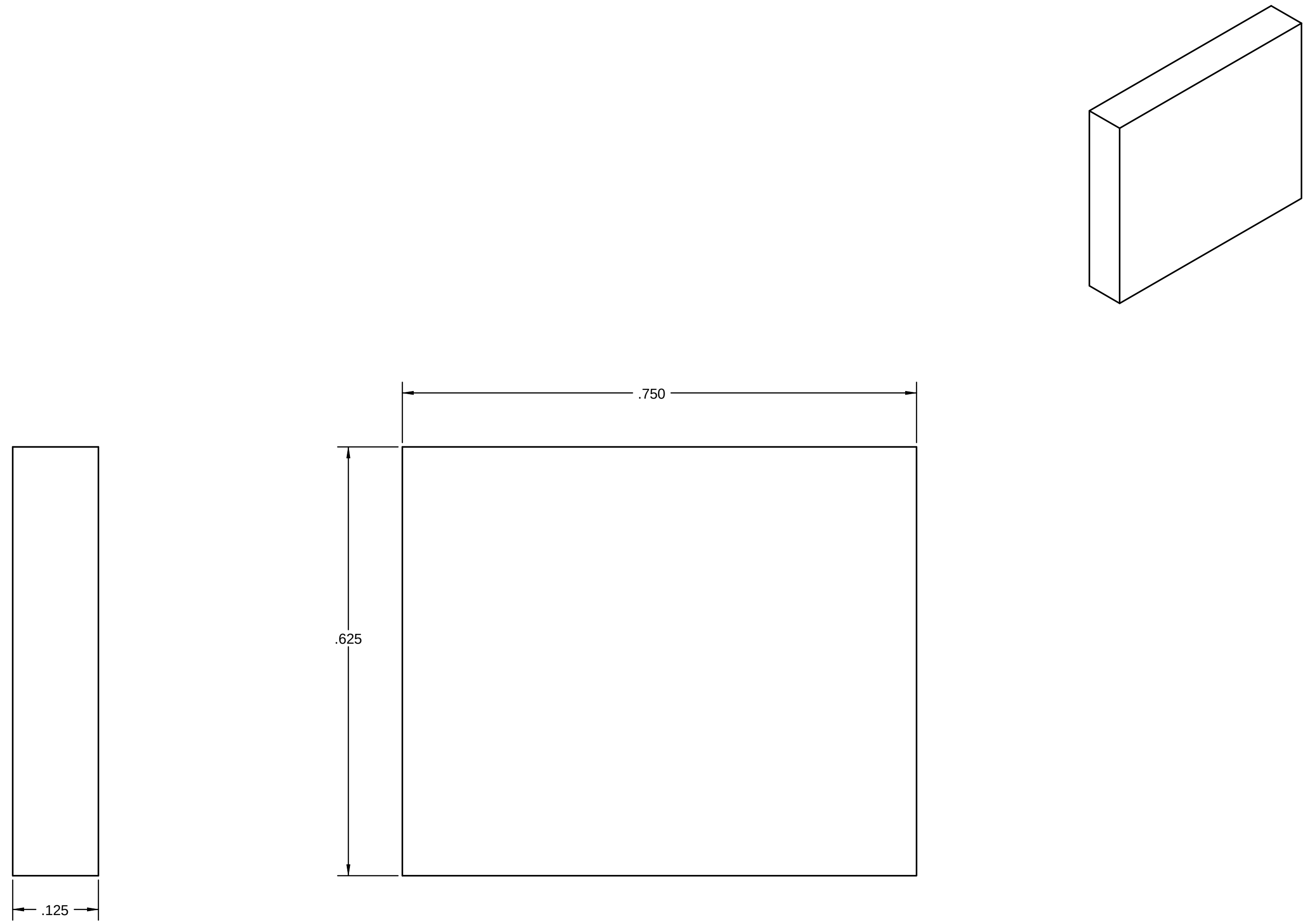
Franklin W. Olin College of Engineering

# BARREL CAM

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |

SCALE 2:1 | WT | SHEET 1 OF 1

NOTES:
1. USE 1018 MILD STEEL STOCK




QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:

X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
| --- | --- | --- |
| PREP BY | N. PAULSON | 12/14/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

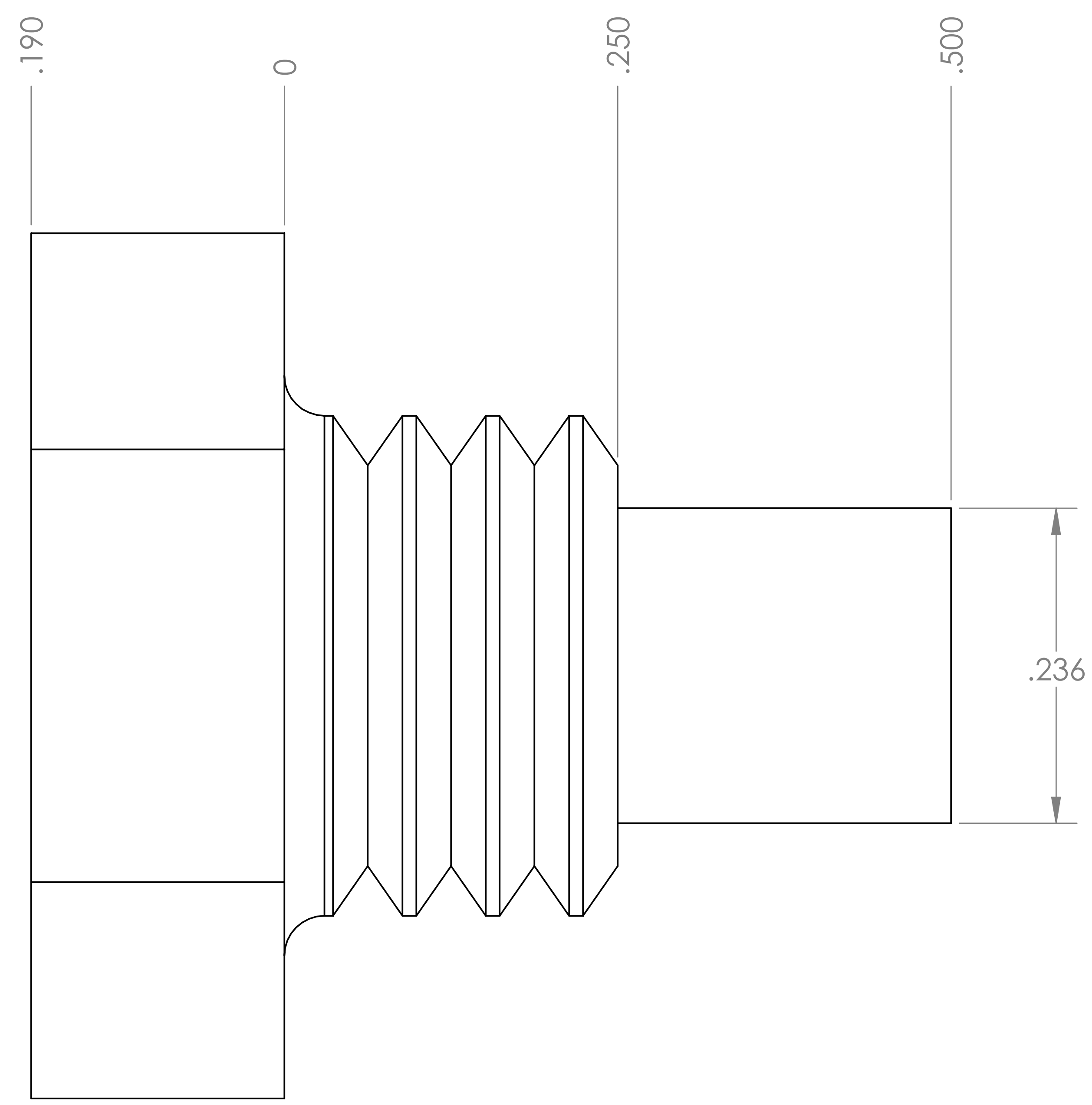
⧫ = (CRITICAL DIMENSION)

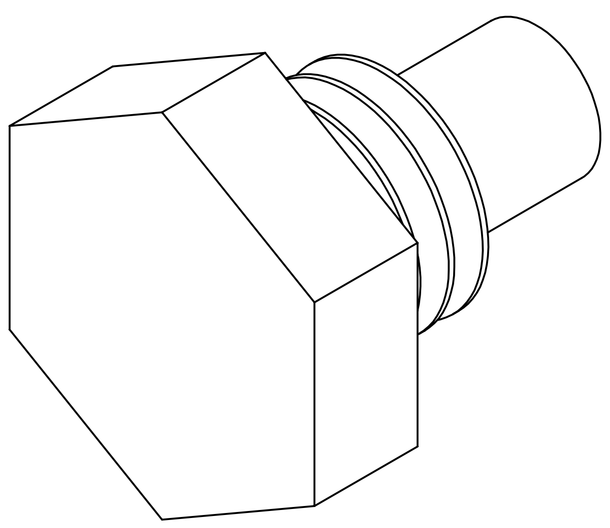
Franklin W. Olin College of Engineering

BARREL CAM NUB

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
| --- | --- | --- | --- | --- |
| C | | | | |
| SCALE 8:1 | | WT | SHEET 1 OF 1 | |

NOTE
1. MACHINE FROM A ASTM A193 3/8-16 UNC-2B HEX HEAD BOLT



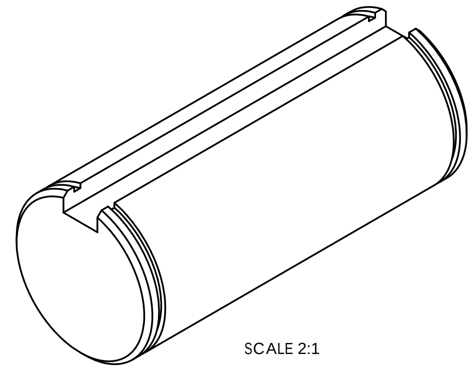



SCALE 4:1

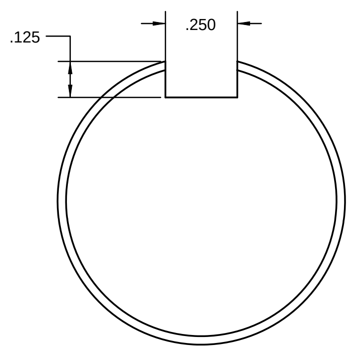
QUANTITY: 2

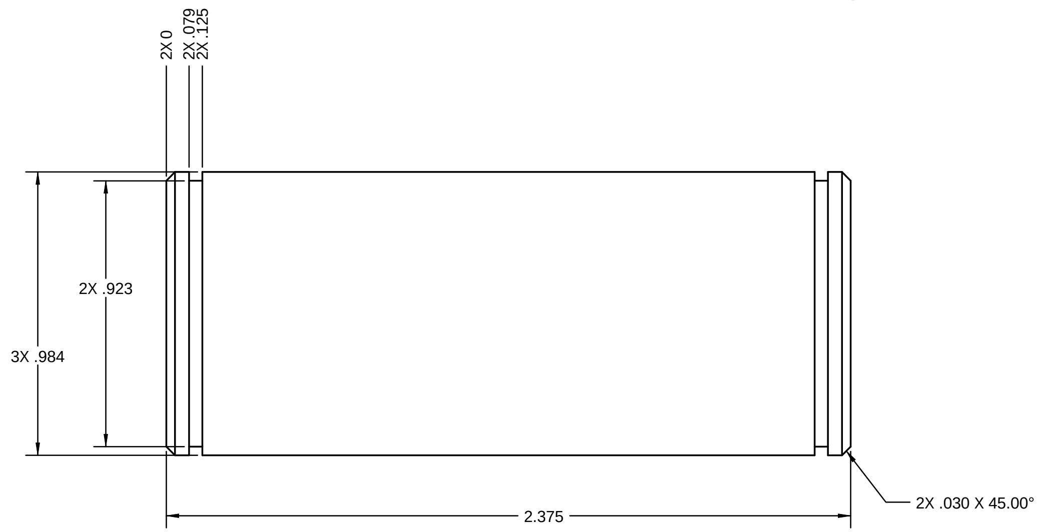
| | | | |
|---|---|---|---|
| ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994 | APPROVED | DATE | Franklin W. Olin College of Engineering |
| TOLERANCES UNLESS OTHERWISE SPECIFIED: | PREP BY N. PAULSON | 12/14/09 | BOLT-PIN |
| X.X ± .03 ∠ ± .5° | CHECKED R. HARRIS | 12/14/09 | |
| X.XX ± .01 | RESP ENG | | |
| X.XXX ± .005 125√ | MFG ENG | | SIZE C, FSCM NO., PART NO., PART REV, DOC REV |
| REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX | QUAL ENG | | SCALE 8:1, WT, SHEET 1 OF 1 |
| | ◆ = (CRITICAL DIMENSION) | | |

NOTES:
1. USE 1 IN DIAMETER 1045 UNHARDENED STEEL SHAFT



SCALE 2:1

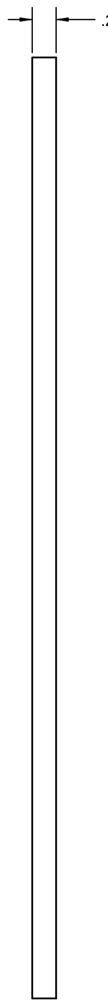


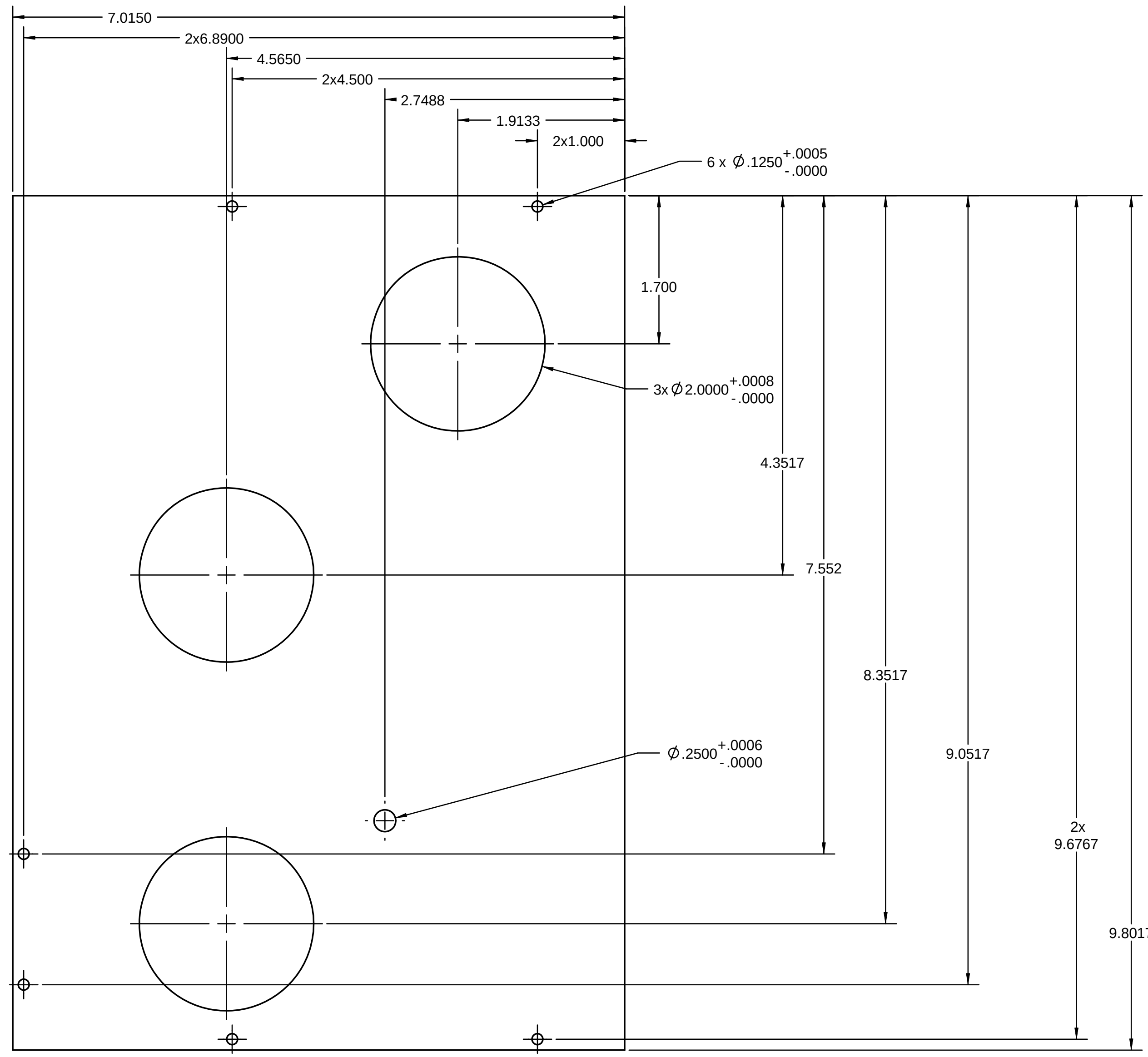




QUANTITY: 1

| | | | |
|---|---|---|---|
| ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994 | APPROVED | DATE | Franklin W. Olin College of Engineering |
| TOLERANCES UNLESS OTHERWISE SPECIFIED: | PREP BY N. PAULSON | 12/14/09 | |
| X.X ± .03 ∠ ± .5° | CHECKED J. GORASIA | 12/14/09 | OUTPUT SHAFT |
| X.XX ± .01 | RESP ENG | | |
| X.XXX ± .005 125√ | MFG ENG | | |
| | QUAL ENG | | SIZE C · FSCM NO. · PART NO. · PART REV · DOC REV |
| REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX | ♦ = (CRITICAL DIMENSION) | | SCALE 3:1 · WT · SHEET 1 OF 1 |

NOTES:
1. USE 1/4" 6061-T6 ALUMINUM STOCK

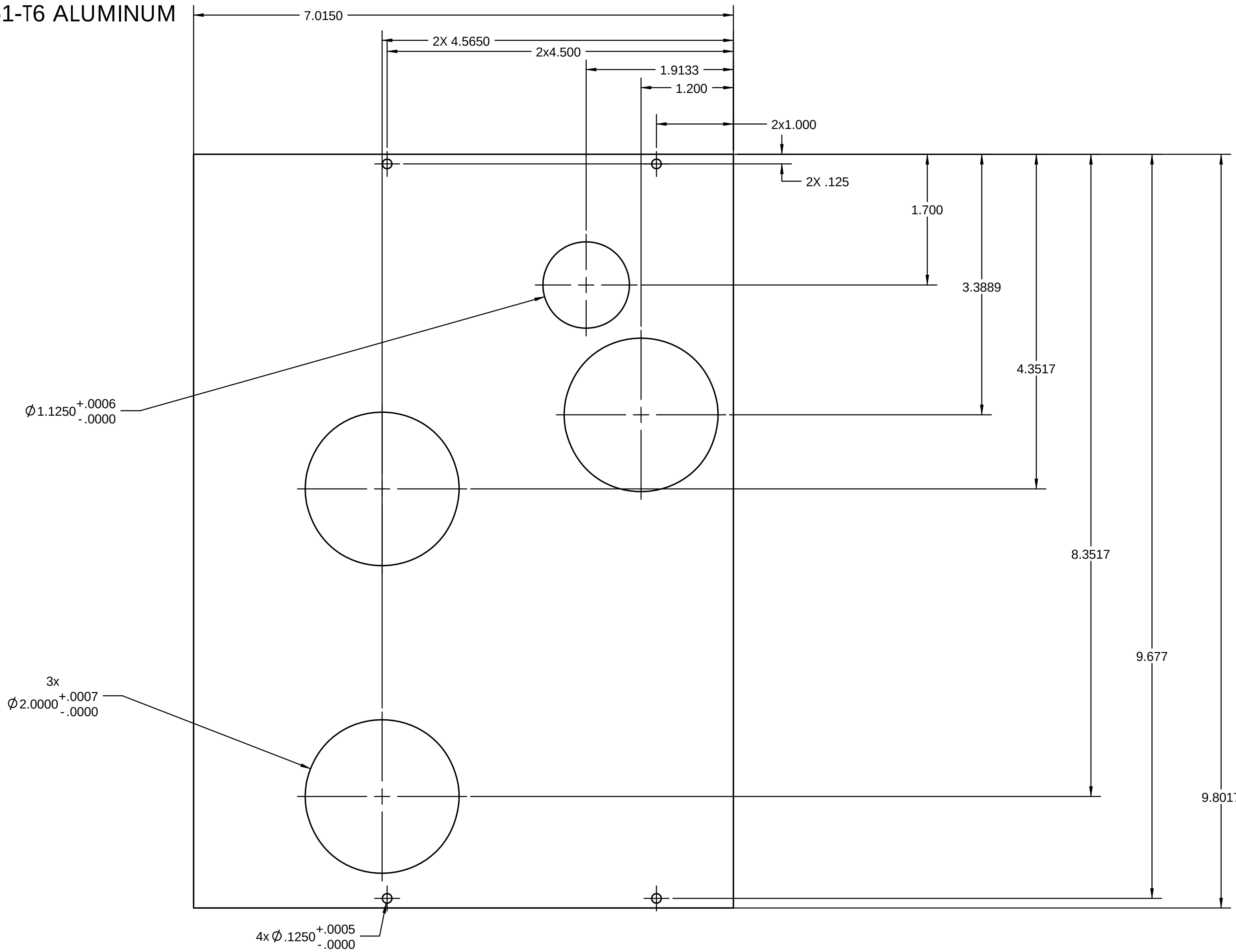



QUANTITY: 1

| ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994 | APPROVED | DATE | Franklin W. Olin College of Engineering |
|---|---|---|---|
| TOLERANCES UNLESS OTHERWISE SPECIFIED: | PREP BY EDDIE BYUN | 12/13/09 | |
| X.X ± .03 ∠ ± .5° | CHECKED J. GORASIA | 12/14/09 | INPUT HOUSING |
| X.XX ± .01 | RESP ENG | | |
| X.XXX ± .005 125√ | MFG ENG | | |
| | QUAL ENG | | SIZE C · FSCM NO. · PART NO. · PART REV · DOC REV |
| REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX | ⧫ = (CRITICAL DIMENSION) | | SCALE 1:1 · WT · SHEET 1 OF 1 |

NOTES:
1. USE 1/4" 6061-T6 ALUMINUM STOCK




QUANTITY: 1

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03  ∠ ± .5°
X.XX ± .01
X.XXX ± .005  125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | EDDIE BYUN | 12/13/09 |
| CHECKED | J. GORASIA | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

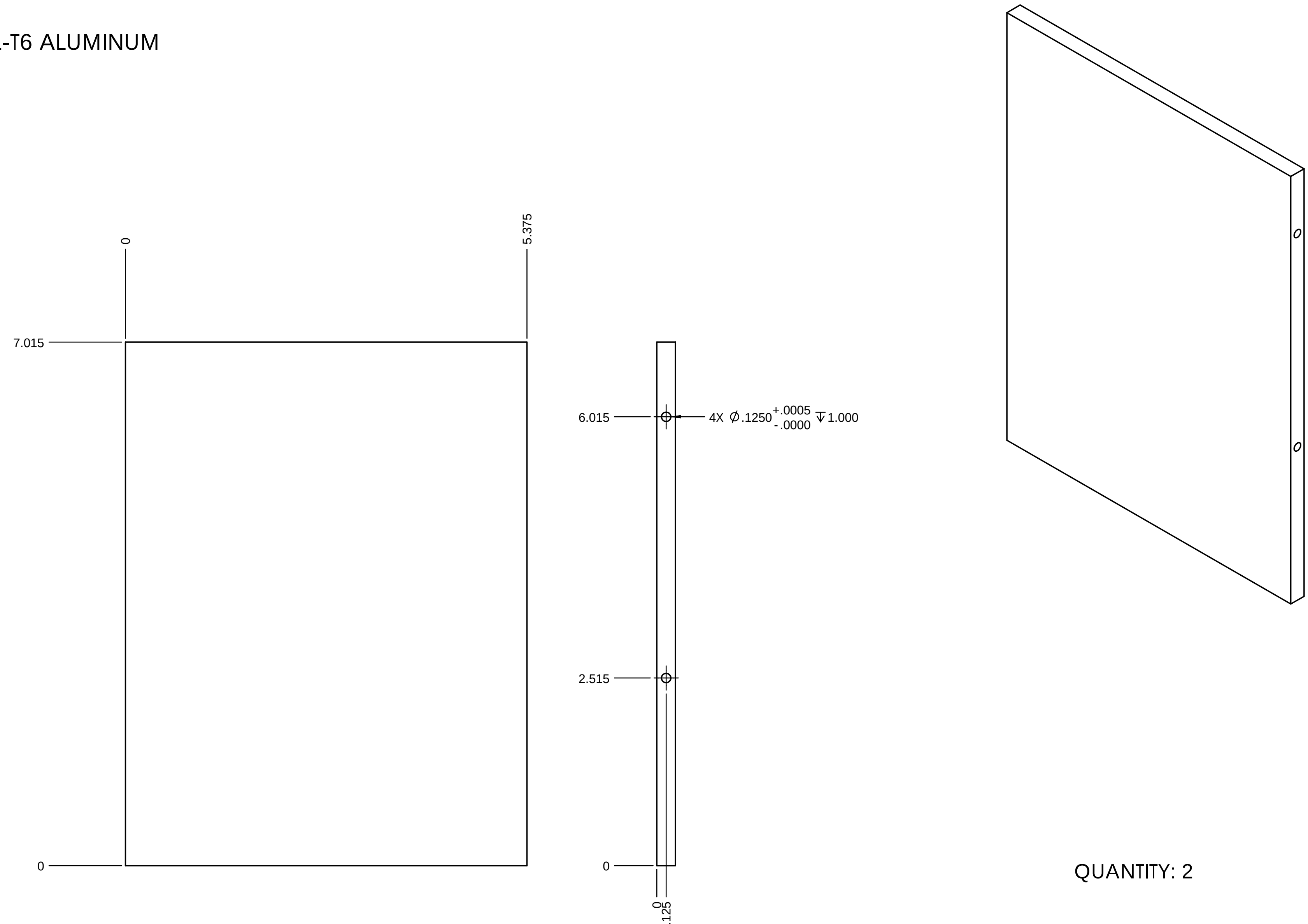
♦ = (CRITICAL DIMENSION)

Franklin W. Olin College of Engineering

OUTPUT HOUSING

| SIZE | FSCM NO. | PART NO. | PART REV | DOC REV |
|---|---|---|---|---|
| C | | | | |
| SCALE 1:1 | | WT | SHEET 1 OF 1 | |

NOTES:
1. USE 1/4" 6061-T6 ALUMINUM STOCK




QUANTITY: 2

ALL DIMENSIONS ARE IN INCHES- INTERPRET DRAWING PER ASME Y14.5 -1994

TOLERANCES UNLESS OTHERWISE SPECIFIED:
X.X ± .03 ∠ ± .5°
X.XX ± .01
X.XXX ± .005 125√

REMOVE ALL BURRS AND SHARP EDGES .005 R OR CHAMFER MAX

| APPROVED | | DATE |
|---|---|---|
| PREP BY | J. GORASIA | 12/14/09 |
| CHECKED | R. HARRIS | 12/14/09 |
| RESP ENG | | |
| MFG ENG | | |
| QUAL ENG | | |

⧫ = (CRITICAL DIMENSION)

Franklin W. Olin College of Engineering

TOP HOUSING

SIZE C | FSCM NO. | PART NO. | PART REV | DOC REV

SCALE 1:1 | WT | SHEET 1 OF 1