GROUP 4 // BEVEL'S ADVOCATE

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ABSTRACT

With head-turning styling, seamlessly smooth acceleration, and that indescribable but incredibly potent diwheel allure, the Bevel's Advocate is one of the most desirable rides on the market.[1] At an applied rider torque of 26 lb-ft, converting to 5.5 lb-ft[2] on each wheel, this diwheel accelerates from 0 to 5 mph in under 10 seconds. Helping to make Bevel's Advocate's sublime driving dynamics possible is a rigid, open chassis that features advanced hardened steel and Thayer machine shop manufacturing and welding techniques. The suspension is derived from hoop spring between the offset drive wheel several inches above the center of mass and the front guide wheel, and the custom made rear differential maintains smooth turning for tight maneuverability. Standard aluminum bike brakes are effective at bringing the Bevel's Advocate down from extreme diwheel speeds, and the anti-gerbilling mechanisms provide both stylistic additions as well balance and over-correction protection.

Sporting a 135° reclined rider angle, and optimized for riders from 5'2" to 6'4" Bevel's Advocate is balanced at equilibrium with and without a rider. At 31.5" in width this diwheel has a 0° turning radius. With five speed shifting, acceleration at the start line is smooth, and avoids unnecessary torqueing on the drive mechanism by allowing the rider to build momentum and speed as they shift into higher gears, instead of always starting in their race gear.

As the champion of the 2015 ENGS 146 DiWheel Competition, as well as the winner of the Fabrication, Fit and Finish award, Bevel's Advocate lived up to performance expectations. The team's extra effort to fine tune and fix initial design weaknesses in the days leading up to the race paid off when it really counted. The bevel is in the details.

Built with durability in mind, extra gussets and strong steel penetrating welds are the backbone of the frame design. Innovative new challenges like shifting and largest rider height difference drove our design beyond prior year's reigning champions. Finally, good camaraderie and excellent design from our fellow competitors inspired and challenged Bevel's Advocate to go where no diwheel has gone before.

6 teams. 25 blossoming engineers. 4 incredible TAs. 1 amazing professor. 5 weeks.
4 challenges. Unlimited access to McMaster Carr.
And 1 goal – one diwheel to rule them all.

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[1] Left Lane. http://www.leftlanenews.com/new-car-buying/ferrari/458-italia/#.

[2] Actual values calculated for Bevel's Advocate were 315 in/lbs rider input, and 65 in/lbs, as compared to the 2015 Ferrari 458 Italia at 398 lb-ft at 6,000 rpm.

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INTRODUCTION

Figure 1: From concept to final fabrication, on the left is our fully rendered SolidWorks model, and on the right is the manufactured DiWheel.

The Diwheel Competition has a storied three year tradition at Thayer in general and in ENGS 146: Computer-Aided Mechanical Engineering Design specifically. Countless young engineers are inspired to take the class based primarily on watching the current students putter around in their diwheels in the days leading up to the competition. With these high expectations in place, we, as Bevel's Advocate, knew we had a lot to live up to. In addition, due to the repetition of the competition from the last two years, the rules were changed this year to add new twists and challenges. In the past two years, the challenge had been a simple relay race around a course, but this year that was restructured as three agility challenges and a drag race. The agility section consisted of a "figure-eight" challenge, a balance-beam challenge, and an upside-down plunger filled with water challenge. The drag race was a simple down and back, with a rider exchange in the middle. Therefore, this year, diwheels would have to be built both for speed and agility. In terms of scoring, both the agility section and the speed section were equally weighted, with the winning diwheel the one that is able to perform across both sections. In terms of other constraints, we used the same 42" hoops as the previous two years and our design had to be less than 32" wide to fit through doors. Also, we had to use steel tubing (square and c-channel) for the frame. Lastly, this year, a differential was a necessary design choice.

It is with these design constraints in place that we set off to maximize both speed and maneuverability. In our initial high-level design brainstorming, after tossing out radical ideas like having a high seat placement, we settled on a recumbent seat design that would be neutrally balanced both with and without a rider. We chose a recumbent position to maximize rider comfort and minimize need for upper body movement. By examining previous diwheels, we saw that this was a successful and elegant solution to the problem of the naturally unbalanced diwheel. In thinking about our required differential, we decided to pursue a bevel gear differential to advance the state of the art, and provide a tight turning radius. In designing the frame, we prioritized an open cockpit similar to the state of the art in order to maximize ease of rider switching. When searching for our truly innovative feature, we decided to go where no other diwheel had been before- shifting.

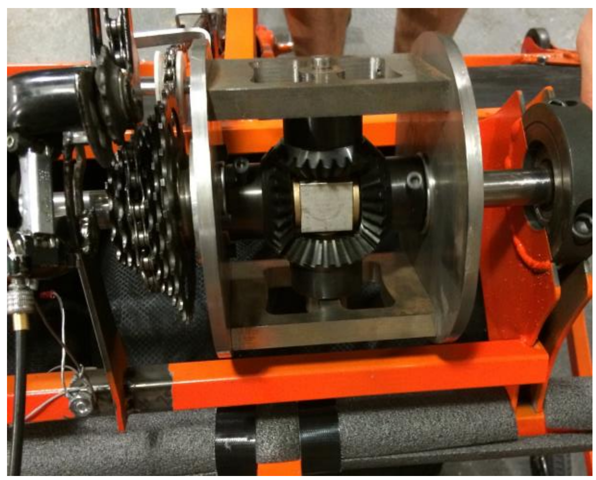
Thus, we designed our Diwheel to accommodate and utilize a bike derailleur to shift gears. In this way, we were able to solve the problem of poor acceleration while still maintaining a high top speed. Lastly, throughout all of these design choices, we kept in mind both manufacturing and assembly. By continually prioritizing design for manufacturing and assembly (DFMA), we set ourselves a goal of completing fabrication several days before the competition to allow us to troubleshoot any problems with the design.

In the following sections, we will discuss individually each component of the diwheel, covering both design (and simulations where applicable) and manufacturing for each one. We will then discuss the days leading up to the competition, where we assembled and tweaked our Diwheel to perfection. Finally, we will conclude with a discussion of the competition results and a post-mortem of our Diwheel performance.

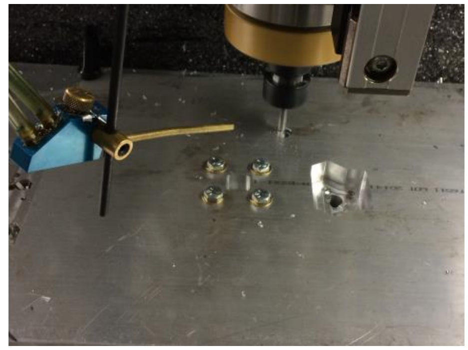
DISCUSSION // DESIGN & MANUFACTURING

The Differential

The differential made a big *diff*erence in the performance of our diwheel in the competition events, and was the first part of our design we completed. The ENGS 146 state of the art from prior years was a spur gear differential that allowed turning on a dime, but the automotive state of the art is a bevel gear differential. Thus, we decided that to improve upon last year's design, we would design and execute on a bevel gear differential. The theory behind a differential is that when a vehicle with parallel wheels turns, the outer wheel needs to move faster to prevent slippage on the inner wheel. This was important for us because we needed to be able to turn on a dime for the figure-8 challenge and for general maneuverability. Key factors that we kept in mind while designing were: reduction of friction, ease of assembly and disassembly, correct gear meshing based on tolerances, maintaining alignment, and ability to manufacture. For reduction of friction, we placed bushings in the gears and bearings in the differential side plates (after initially trying bushings and experiencing alignment issues). For ease of assembly, we made sure to continually visualize the assembly process when adding each additional part to the assembly in SolidWorks. By keeping order of assembly in mind throughout our design process, we were not surprised that it worked, when it came time to actually assemble the differential. For correct gear meshing, Hunter thoroughly researched bevel gear options on McMaster Carr, and designed a flexible gear mesh configuration in SolidWorks that allowed us to quickly swap gear sizes before we had arrived on our final version. After assembly, we carefully adjusted set screws and alignment of the various other components to allow the gears to mesh within the desired



Differential and drive shaft stack-up, including cassette tolerance. For maintaining alignment, we placed a "spider block" in the middle of our differential that aligned the drive shafts with each other and kept them rotating concentrically. The spider block was fixed on the "spider shaft" with retaining rings. Finally, for manufacturability, we leveraged our extensive machine shop experience to visualize each part as a series of operations. By bringing our experience to bear, we had a great sense of both what was possible and what was easy to do in Thayer's machine shop.



CNC Milling the Drive Wheel

CNC operations on the mill were essential to manufacturing our differential. In particular, we used CNC extensively to program hole patterns, circular profiles and pockets, and used CAM to cut well-designed, bevel shaped, speed holes in our ½" thick spacer plates - which were essential to the aesthetic appeal of our diwheel. After manufacturing the pieces, we assembled and troubleshot the assembly to find areas of friction. One key modification was replacing the bushings in the side plates of the differential with bearings to allow for vertical forces which were causing misalignment of the independent drive shafts. We chose to make this modification because after assembling our differential we found the interface between the shafts and the sidewalls to be a key point of friction, due to small misalignments of the shaft from horizontal. To further mitigate this friction we made our spider block rigid in the axial direction by placing washers and retaining rings. These two interventions created a significant difference, and after some wear in period our differential was very smooth.

After the initial assembly, we found that the differential was unnecessarily heavy, and we CNC cut speed holes in the flat sidewalls. We eventually laser cut a plastic case to isolate the differential interior from debris and contain grease to lubricate the bevel gears.

One last detail, to drive the differential, we needed to mount a cassette from one of our bicycles to our differential casing. After removing the cassette we measured the diameter of the external threads from the bicycle wheel and used a thread pitch gauge to confirm that the threads were a 1.75"-24 thread. Using the lathe we cut test threads and ultimately an attachment piece that mounted the cassette to the differential while allowing the bearing to fit comfortably in the case without interference and a through hole for the drive shaft.

In the submission package is a differential motion study demonstrating the gear meshing and rotation of the shafts and drive wheels in the bearings.

Drive Shaft and Wheels

In parallel with manufacturing the differential box itself, we designed and manufactured the drive shafts and drive wheels. The main piece of design we had to do was transferring the rotation from the differential to the drive shafts and then subsequently to the drive wheels. To tackle the first part, we brainstormed a few options. We could cut key slots in the gear and the shaft and insert a key, but this would still allow the shaft to move axially. Next we considered a through-hole through the shaft, through which we would insert a massive set screw. However, this would have weakened the shaft in the most important part (center) and was difficult to get the alignment correct. Ultimately, we settled on machining two flats on the drive shaft, which would secure the gear to the shaft. We initially worried that this wouldn't be strong enough, but after seeing that this is how the mill cutting bits are secured in their holders, our fears were assuaged. Next, we had to secure the end of the drive shaft to the drive wheel, to be able to transfer the rotational motion to the hoops. The key here was again fixturing the assembly in both the axial and concentric directions. In terms of concentrically fixturing the drive wheel, we initially implemented a cross-like design in both the end of the drive shaft and the drive wheel, however after critically thinking about how to machine this, realized that this was infeasible. We eventually ended up switching to a simple square design that was very easy to machine and sufficiently secure. In terms of securing it in the axial direction, we simply tapped into the end of the shaft and placed a "hub cap" over the drive wheel that essentially acted as a giant washer for a hex screw to secure the wheel to the shaft.

Drive wheel bevel pattern and hubcap fastening bolts.

Jig for welding the shaft collar gusset assemblies

During testing and driving, we found that the left wheel consistently loosened and began to wobble, because the rotational force in the wheel was acting in the direction of loosening the screw. Ideally, we would have used a left handed screw and tap to solve this issue, but instead periodically tightened our screw and used Loctite. The right hand side on the other hand was self-tightening and did not present any problems.

Welding the gusset shaft collar sub-assemblies to the base of the diwheel frame. The vertical aluminum shaft was used as a jig for rigid alignment during welding.

To add a bit of artistic and thematic flair to the drive wheels, we designed a "bevel-gear" pattern that we used CAM to machine into the drive wheels. We used a SolidWorks static study (included in the Appendix) to ensure that the pattern cutout wouldn't affect the structural integrity of the diwheel.

An essential part of manufacturing and running CAM is deciding on the best tools and fixtures for each part. Of note, for the aluminum drive wheels and the welding jig we used a single flute ¼” end mill which allowed us to run at faster feeds and speeds while maintaining good chips and heat dispersion.

Alignment and rigidity of the drive shafts and differential are important to optimizing the performance of the drive system, and reducing unwanted friction. In parallel with the design and manufacture of the drive mechanism, we designed the attachment to the frame, and rigidity of the system. Here, we should give fair credit to the Green Party team from last year for their bearing/shaft collar design, as we used their basic design. We placed bearings on the shaft for optimal rolling, then purchased shaft collars that were sized to perfectly encase them, then designed and plasma cut gussets that cradled the shaft collars, and welded the top half of the shaft collar directly to the gussets. However, after identifying that the alignment of the drive shaft attachments on the frame was a crucial part of the design, we added our own style to the process by designing and using several welding jigs. The first used CAM to machine a jig, pictured to the right, that allowed us to space the plates on either side of the shaft collar. Furthermore, we used a long aluminum rod when welding all four gusset-shaft collar subassemblies to the rest of the frame to align the welds and hold the frame rigid to resist the bending caused by heating of the steel tube. All of these jigs allowed us to be confident in our alignment, and this paid dividends in how smoothly our assembled differential ended up running.

The Frame & Anti-Gerbilling

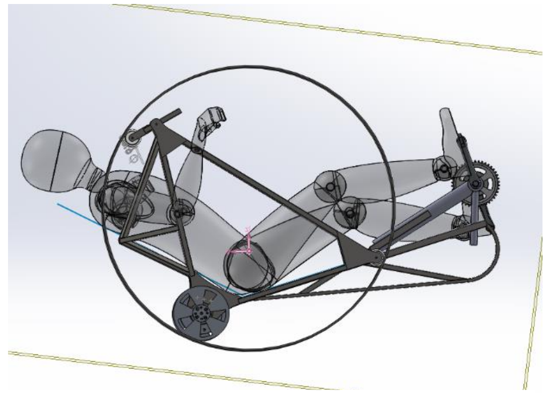
The design of the frame was driven by several goals and constraints. Between the 42” diameter hoops, the 32” maximum width constraint, and the amount of pre-ordered square and c-channel steel tubing, we had a rough idea of the overall size and materials. We also learned from past diwheel designs that an open cockpit was achievable, but would reduce lateral stability and consequently increase the likelihood of derailment. This was a challenge we were willing to take on. Other design-driving factors for the frame included the framework and space needed for our recumbent-style seat, attachments for the drive axle and derailleur components, and of course the anti-gerbilling system to ensure maximum safety for the rider.

Testing our rear anti-gerbilling mechanism, we decided that it would very rarely be used in competitive conditions.

The first step in designing the frame was determining the optimal angle for pedaling. With the design goal of having a recumbent seating component, we looked at several recumbent bicycles and determined a

135° chest-to-knee angle would provide sufficient pedaling power and comfort to the rider. This angle determined the locations of the frame cross-bars, which serve as the underlying framework for the seat component.

Recumbent Bicycle


Early Frame Design with Dummy in Recumbent Position

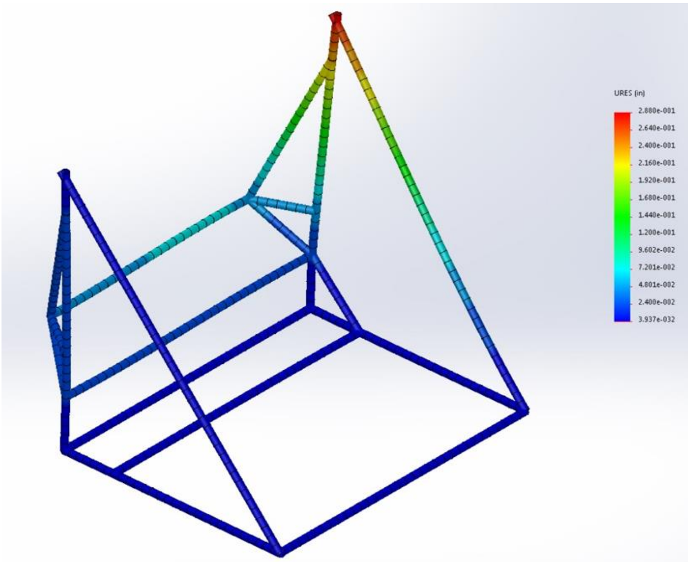
As suggested by Professor Diamond and previous diwheelers, we opted for a three-point contact design between the frame and the hoop. Because of this, the triangular shape of the frame sides was an obvious choice as the most efficient method to achieve this. In conjunction with the gussets on each corner of the triangles, this portion of the frame was grossly overbuilt to ensure negligible deformation in the plane of the hoops.

Many of the other aspects of the overall frame design were inspired by the previous diwheel champion, *The Green Party.* This includes the rough locations for the cross beams. As seen in the above SolidWorks rendering, the cross-bracing is concentrated in the bottom and back of the frame to provide an open cockpit and therefore fast rider changes. Their placement also aids in the attachment of various components such as the seat, the anti-gerbilling components, and the gussets that attach the frame to the drive axle (discussed in The Differential section).

Jeff top is 6'4" tall and Hilary bottom is 5'2" tall. Our diwheel easily accomadates all riders in between without any mechanical adjustment.

While we designed the overall diwheel to balance perfectly, we decided to incorporate a front and back bumper made from C-channel tubing as an anti-gerbilling system in order to inspire confidence in the rider. These were positioned such that they provide ample protection for the rider from over-tipping, but high enough that they would not regularly drag along the ground and dramatically increase the driving resistance. The curved portions of the bumpers were achieved by making many cuts along C-channel steel tubes and manually bending them into the correct position with the guidance of 1:1 scale drawings.

Along with the anti-gerbilling function of the front members of the frame, they also served as the main connection to the bike pedals. Due to the high amount of variable torque experienced by the pedals, we knew we needed to provide ample support to keep them firmly connected to the frame. This was achieved by clamping the original bike tubing between two c-channel members that were welded to the front of the frame. The front anti-gerbilling bumper was used as an additional support that created a strong triangular structure below the pedals, providing vertical rigidity.



Static Load Study of Frame: Lateral Deformation

One concern with an open cockpit design was the lateral rigidity of the top. With too much lateral deflection in these sections, derailment would become a possibility. To test this, we performed the finite element analysis seen on the right. Even with an extreme lateral load (200 lbs.), the top experienced less than 3” of deflection, which we determined to be sufficiently small to avoid derailment. This result drove our later decision to add gussets in key locations to increase rigidity. This analysis was also useful when we decided to eliminate the top back support triangular members, because they would be difficult angular cuts to make and they did not provide significant lateral stability.

The fabrication process of the frame was kept in mind throughout the design phase. We knew alignment would be essential for a smooth ride with minimal resistance and few occurrences of derailment. To guarantee the angular alignment of the side frame triangles we plasma cut interior guide gussets, seen in the photo at right. These, combined with tack welding key frame components together over plotter paper with 1:1 scale drawings (another useful tip from *The Green Party*), resulted in reduced misalignment due to the welding deformation and ultimately precise angles within the triangles. As discussed at length in the differential section, the lateral alignment of the frame was achieved through the pre-alignment of the drive axle gussets.

Inner gussets welded first to ensure proper alignment and accurate angles in frame.

The fabrication of the frame went smoothly due to our mindfulness in the design phase, but it did present challenges that we did not predict. Our largest problem had to do with the trapezoidal back cross-brace. Initially, this component was designed with very complex cut angles that we could either not ensure accuracy on or could not accomplish altogether. Because we had already begun welding when we came to this realization, we were required to make several quick design iterations and reached a solution that required only simple 90 degree cut angles. Another roadblock we ran into, despite our best efforts to avoid it in the design phase, was alignment. Due to very small deformations in the welding process, the vertical members in the back of the frame were not perfectly parallel. This resulted in a spacing that was so wide that when we went to attach the back trapezoidal cross-brace, it could not reach both sides. This proved to be an easy fix, as we simply cut the cross-brace in half and added 2” of square tubing to elongate the piece. Perfect alignment on the anti-gerbilling bumpers proved challenging as well. The cut-and-bend strategy for forming the curves did not provide enough control to exact the shape we wanted. Because their dimensions were not integral to the overall design, however, we were able to bend and cut adjustments to the parts in order to fit them to the frame. In the final product, several of these slight misalignments are hardly noticeable and the components which relied heavily on perfect alignment were fabricated with jigs to meet this requirement.

Steering & Braking

Maneuverable steering was an essential sub-assembly of the diwheel to compete in the figure-eight, balance beam, and plunger competitions. Achieving easy to manipulate steering meant optimizing important interactions between the drive train and braking mechanisms. Incorporating steering into the diwheel is fairly limited to braking steering, manipulating the diwheel to the right or to the left by braking that side of the drive assembly. Our team's innovation beyond prior year's braking mechanisms was implementing a disc braking system in which we stopped the drive wheel attached to our drive axle instead of applying the braking force directly to the hoop wheel. This design, innovative based on primary years, ended up being a common mechanism among this year's diwheels after we were the first ones to implement it. The primary advantage of this design is that the brakes are statically fixed directly to the frame in a location that they are not exposed to lateral forces when the hoops derail. We intentionally avoided this because it was the key failure we identified inspecting the brakes from the Green Party diwheel. Furthermore, beyond the value of rigidity of the braking mechanism, braking on the smaller diameter drive wheels was more responsive and allowed for more precision than braking the larger hoops.

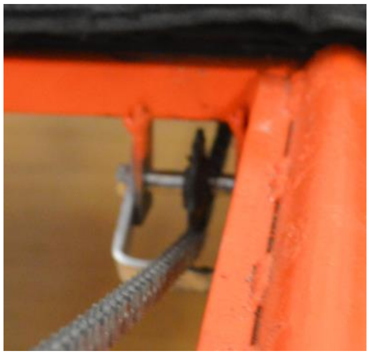
Brakes attached to handle offset from the frame.

Brake mounted to the corner of the frame and positioned on the rim of the drive wheel.

For our brakes, handle bars and shifters we used the handlebars off of one of our bikes, which we welded directly to our frame. One other detail, we kept the brake handles loose enough that with some force they could be rotated – this helped us accommodate riders of many different sizes, as larger riders naturally placed their arms outside of the handles, whereas smaller riders gripped through the handles. Finally, as with many of the details in our diwheel we spent extra time assembling our brakes making sure to clean and grease the wires before inserting them into the cable housing, and adjusting them so that the brake pads were spaced about ½” away from the drive wheel on either side minimizing the travel distance of the break pad before engaging with the wheel, this added to the responsiveness of our braking system.

The Drive Train & Derailleur

An early goal of the project was to shift gears. After attempting to ride the Green Party Diwheel from the last year it was clear that it was hard to get started and that the rider was jolted backwards when they tried to accelerate from a dead stop. This makes the diwheel hard to ride and creates a steep learning curve as a beginner tries to balance. However, if the diwheel were in an easier gear, the diwheel would be slower, but hopefully easy to ride. Therefore, shifting was the ideal solution as it allowed easy acceleration while maintaining a high top speed. Knowing that we could always add a chain tensioner if the derailleur did not function on race day, we decided to give it a shot and hopefully have time to make it work for the drag race competition.



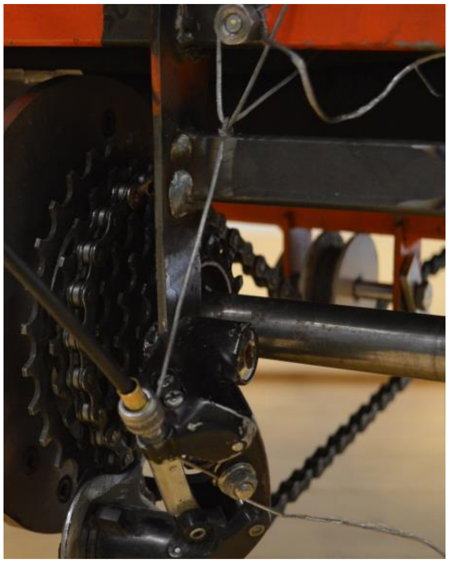
Front Chain Guide Sprocket

One initial issue was where to run the chain through the frame. We designed mounts for sprockets that could slide along an axis and spin to guide the chain under our frame. When we actually assembled these mechanisms we ran into issues when the chain periodically jumped off the sprocket and grinded on the aluminum that the sprocket slid on. To combat this, we added sheet metal bent guides to eliminate the derailments and keep our chain moving as we planned, as well as machined a delrin guide wheel with flanges to keep the chain aligned.

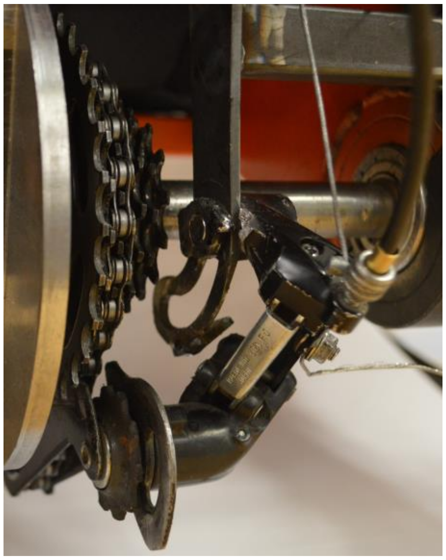
Derailleur in Easiest Gear and Back Chain Guide Sprocket

Designing the shifting relied on a couple of relatively simple concepts: The location of the derailleur, the tension in the shifting cable, the high shifting adjustment, and the low shifting adjustment. Because it was hard to tell exactly where the chain would run and where the derailed should sit, we cut a mount with a 3/8” arced slot out of 1/8” steel plate. The position was set at 3 cm away from the axis of rotation for the drive wheels. Then the mount itself was positioned at about 1 cm away from the lowest gear on the cassette. Overall, this provided versatility to move the derailleur to function as smoothly as possible. Finally, to make the installation of the shifting mechanism easier we cut apart and reused the handlebars from one of our bikes, which simplified the design and installation time significantly.

After completing the frame and assembling the drive mechanism, we installed the derailleur and ran the cable. Then we adjusted the tension and set the upper and lower limits. However, unfortunately the derailleur dragged on the ground when we rode it. To fix this issue we moved the derailleur to the top of the slot, added a cable to pull the first component of the derailleur into better tension, took out a few links of chain, and ground down parts of the derailleur to make it lower profile. In the end we have 5 fully functioning gears that provide a smooth ride, an easy startup and a high top speed.



The Arced Slot Used for Adjusting the Derailleur Position at its Highest Position.



Cable Used to Pull the Derailleur into Better Tension with the Chain.

Guide Wheels

Two important specifications for the guide wheels were the frame placement and low friction.

Acetyl guide wheel on lower back gusset.

The axial design of the wheels was based on skateboard wheels, which have been optimized for speed and low friction. Inset on either side of each acetyl wheel were mini high-precision stainless steel flanged ball bearings with ¼” ID, ½” OD and 3/16” wide. We originally planned to use steel shafts with retaining rings for the guide wheel axles, but shifted the design to incorporate a screw and nut assembly instead, which did not significantly reduce friction, but did facilitate the ease of assembly and disassembly, and allowed us to hand tighten the bolt to keep the wheel from wobbling, but not tighten it too much to cause drag between the gussets and the acetyl wheel.

Placement of the guide wheels was driven by the frame design and the reclined position we chose for the rider. We found that placing the guide wheels closer to each other around the arc created shorter arc distances between the points of contact with the hoop, and a much larger unsupported arc of hoop in the front. Keeping the guide wheels closer together minimized deflection and kept our diwheel from derailing. We chose three points of contact with the hoop as the optimal number for a sufficiently constrained, but not over-constrained system. Our analysis of hoop deflection which guided these design decisions is documented in Appendix B, FEA Study 3: The Hoop.

Top spring loaded guide wheel assembly.

Side view of the diwheel to show the placement of the supports in the hoop.

Different from prior years, we placed the top, spring loaded, guide wheel lever arm with the wheel facing the rear of the assembly which contributed to the shortening of the arcs between the guide wheels. We also added extra length to the lever arm to facilitate the mechanical advantage of the lever, making it easier to place the guide wheel within the hoop.

Aligning the gusset holes during welding was an important part of the fabrication of the wheel. Misalignment of the holes would have led to unnecessary friction due to skew of the guide wheel shaft. We built a small jig to align the holes during welding to ensure straight alignment.

The acetyl wheels were machined on the lathe out of a 2” diameter stock using a boring operation for the press fit for the bearing and the small offset for the bearing flange. A through hole was drilled based on the shaft dimensions, but acetyl was left as a spacer in the center of the two bearings. After cutting the guide wheels to the interior width of the hoop c-channel, a 60° chamfer was lathed on both sides of each wheel to minimize friction and wear as the guide wheel fit into the interior groove of the c-channel hoop. This small addition paid off on race day when our diwheel did not derail. The ball bearings were press fit to complete the assembly.

The Seat

Although the frame itself provided seating support for the rider, we wanted our diwheel to offer a pleasureable riding experience. To achieve this, we purchased a simple camping chair that provided ample padding and was flexible enough to accomplish the necessary 135° sitting angle. We added additional c-channel foam padding to the back cross-braces for even more comfort, although the optimal sitting position for the rider was found to only touch the bottom cross-brace for lumbar support. This position offered the best balance point for the overall driving of the diwheel. Finally, to secure the seat and provide a rigid bottom to the component, we placed a 3/8” thick plywood sheet beneath the camping chair. The plywood sheet set into the square shape made by the bottom of the frame, sitting atop the four large gussets. The gussets acted as excellent fixture points, and we simply bolted the plywood to these. We then bolted the chair itself to the plywood so that it would not move around no matter how enthusiastic the rider was.

The Finished Seat, Stencil and All

The Plunger

The key design criterion for the plunger challenge was the ease of driving the diwheel, and not necessarily the complexity of the plunger holder. Therefore, we focused on designing a rockin’ drive mechanism and smooth differential for a fly ride, and implemented a simple plunger attachment to hold the plunger and provide some counter-swinging ballast to the base of the plunger through a set screw weight.

If we were to compete again in the plunger challenge our team would shift the chain to an even lower gear and start further back from the starting line to build momentum before facing the challenging hill between the grass and the path.

Trouble-Shooting

As we mentioned earlier, our goal was to finish fabrication several days before the competition so that we would be able to troubleshoot any problems with our Diwheel, and finesse the small details. We pushed hard on the manufacturing timeline to make this a reality. One problem that we discovered after differential fabrication, but before full frame fabrication, was that we had initially designed our drive train such that the differential was centered in the overall assembly, leaving the sprocket offset from center by a few inches. However, this offset did not line up with the smaller front pedal offset, which would have meant that the chain would not have run properly. After initially thinking that we would just weld the central pedal assembly a couple inches off center, we realized that we could instead modify the drive shafts such that one was shorter and one was longer, so that the back sprocket was positioned correctly on center. The next change that we made after assembly was grinding down unnecessary material at the bottom of the derailleur, as it was dragging on the ground. When we set up the derailleur, the screw that was keeping it in the correct position sheared, so we had to use wire to rig it into the right position. However, this added stress combined with the cantilever of the differential mounting plate meant that the lever was bending out of shiftable position. Therefore, we cut and welded in a small lateral support, which was able to provide the necessary rigidity. Lastly, we took advantage of our time to practice riding our Diwheel extensively, which had the intended effect of wearing in our components and showing us which screws tended to loosen with use. We used this to populate a checklist of all the final tasks that we should do right before the race. For example, the screw that secured the drive wheel to the drive shaft on the left side kept coming loose, because it was a right-hand thread on the left hand side. Therefore, we made sure to tighten this down before and between competition events.

Decorations and Aesthetics

Thanks to our accelerated fabrication process, we were left with abundant time before the competition to not only mechanically assess areas for optimization on our diwheel, but also spice it up aesthetically. We first applied to the frame three layers of primer and three layers of Farm Equipment Orange to give the car a zany yet playful mood. Since the camping chair, pedals, hoops, and handles were all black, we decided to alternate between the two colors to match the diwheel's tiger-like maneuverability. The plywood got hit with the black spray paint to match the rest of the seat component and we used the vinyl-cutting machine to produce die cut stickers with various inspirational phrases that were placed on the frame and plywood surface. Since the differential received a disproportionate amount of design time, we decided to throw some flair its way and laser-etched a signature Bevel's Advocate design into the acrylic casing. Finally, we created a stencil by laser cutting acrylic sheet to create the official Bevel's Advocate logo, featured on the team uniforms and the front and back of the seat. While these modifications may seem gratuitous, they serve a subtle yet important purpose. Like racing car stripes, they add to the overall aesthetic appeal and make you excited to ride in it.

CONCLUSION

After much planning, preparation, and hard work, we not only met our design goals, but exceeded them. By finishing manufacturing days ahead of other teams, twirking our design to squeeze out all the functionality that we could, and by practicing driving our diwheel for several days, we were able to enter Race Day prepared and excited. Therefore, we were able to execute in all facets of the competition, proudly earning the overall competition victory and the coveted prize for “Best Fabrication Fit and Finish.” To wit, we finished with a robust, maneuverable, fast, shiftable, and aesthetically beautiful diwheel that was optimized for this specific competition. In terms of specific results, we scored 6 of 10 points in the figure-eight event, 3 out of 10 in the upside-down plunger event (made unintentionally difficult by the initial incline, and an event that saw widespread water-loss), and a full 10 out of 10 on the balance beam event (a common result for teams, as the ability to drive straight was luckily one that came along with a drivable diwheel). Finally, in the elimination-style drag races, we were able to outlast several formidable opponents and come out on top. Among the class, catastrophic failures were common in this section, as many teams were pushing the Diwheel beyond how they had practiced. However, due to our preparation and robust design, we were able to avoid any catastrophic mechanical failures during the competition and won all three races, including the finals against last year’s champion, to take the speed and therefore overall title.

In terms of analyzing our design and where we could improve, we see a few areas that merit further examination. First, most of the diwheel is very overbuilt, but one area that is not is the junction between the drive wheel and the drive shaft. A single 10-32 screw is holding this junction together, and the one on the left side kept coming unscrewed due to its right hand threading. Therefore, we would advise using left hand thread, or even using two screws here instead of just one, as this redundancy would prove valuable. In addition, we did not load test the diwheel for users above 180 pounds, and when (after the competition) a user weighing approximately 250 pounds drove the diwheel, that screw failed and the drive wheel was ejected. We also did not test the diwheel for large shock forces. We found this out the hard way when on race day, one group member attempted to ride the diwheel up a slope, lost momentum, and fell back in an awkward position. The result was that a couple welds loosened up and broke. Luckily, the broken welds were in a spot where the bar was able to be strapped back on for the duration of the competition, and the welds were recompleted the next day for the safety of any future riders. Therefore, in the future, we would attempt to stress test the diwheel so that we could identify any troublesome areas before race day.

However, despite those few small failures, our diwheel performed admirably on race day, exceeding our expectations. The diwheel is an intriguing technology and mechanical concept, so it’s natural to ponder its commercial viability. However, McMaster parts alone cost us nearly $700 (see the Bill of Materials in the Appendix for details), and this doesn't even include the steel (tubing, c-channel, sheet) stock that we used for our frame. Add this to the approximately 500 person-hours that our team spent designing and manufacturing our diwheel, and the picture starts to look not great. Even at a low hourly rate of $15/hour, this puts the overall cost of the diwheel at more than $8000. Granted, not all of those hours are manufacturing labor hours, nor would the next 100 diwheels each take nearly as long as the first one did, but this is still an incredibly expensive, relatively ineffective form of two-wheeled transportation. However, when assessed as an educational experience for aspiring mechanical engineers, the diwheel passes with flying colors. We had a blast throughout the whole process and loved the project from start to finish. With that, Bevel's Advocate is signing off, Keep Calm and Diwheel On!!

APPENDICES

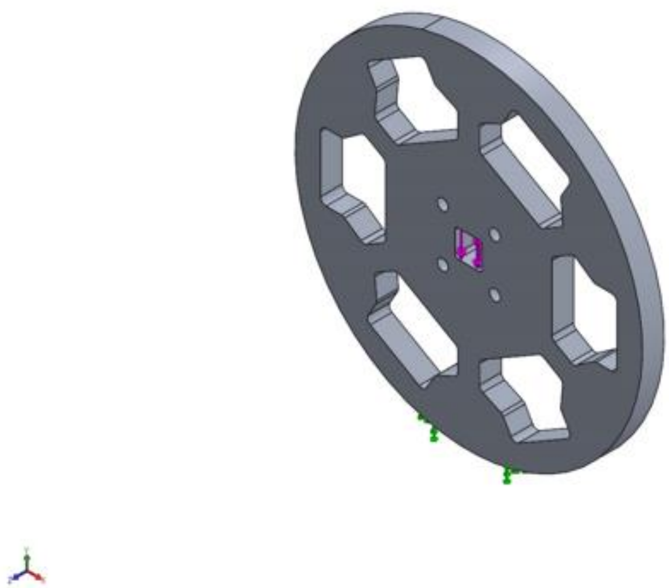
APPENDIX A: Bill of Materials

Subassembly	Part	McMaster Number	Unit Cost	Unit Order	Needed from Unit	Total in Unit	Total Cost for Part(s)	Description
Differential	Diff Side Plates	8910K12	$16.10	2	6	6	$32.20	Low-Carbon Steel Rectangular Bar 1/4", 6" Width
Differential	Diff Flat Plates	8910K702	$21.28	1	8	12	$14.19	Low-Carbon Steel Rectangular Bar 1/2", 3" Width
Differential	Diff Case Screws	91251A345	$10.96	1	12	100	$1.32	Black-Oxide Alloy Steel Socket Head Cap Screw 10-32 Thread, 3/4" Length
Differential	Diff Shaft Bearing	6384K67	$13.47	2	1	1	$26.94	Steel Ball Bearing, Plain Double Shielded for 3/4" Shaft Diameter, 1-5/8" OD
Differential	Bevel Gear	6529K22	$37.01	4	1	1	$148.04	Steel Plain Bore 20 Degree Angle Miter Gear, 12 Pitch, 24 Teeth, 2" Pitch Diameter, 1/2" Bore
Differential	Spider Shaft	1346K11	$6.76	1	5	12	$2.82	Steel Drive Shaft 3/8" OD, 12" Length
Differential	Spider Block	9008K14	$4.40	1	1	6	$0.73	Multipurpose 6061 Aluminum
Differential	Spider Shaft Bushings	6338K415	$0.85	4	1	1	$3.40	SAE 841 Bronze Flanged-Sleeve Bearing for 3/8" Shaft Diameter, 1/2" OD, 1/2" Length
Differential	Spider Shaft Retaining Rings	93576A110	$9.16	1	2	10	$1.83	Light Duty Spiral External Retaining Ring 18-8 Stainless Steel, for 3/8" Shaft Diameter
Differential	Spider Shaft Shims	97022A216	$3.18	1	2	10	$0.64	Type 316 Stainless Steel Round Shim 0.002" Thick, 3/8" ID, 5/8" OD
Differential	Acrylic Diff Casing	8560K259	$23.21	1	90	576	$3.63	Optically Clear Cast Acrylic Sheet 1/8" Thick, 24" x 24"
Drive Train	Shaft Collar	8386K18	$25.87	4	1	1	$103.48	Extra-Grip Two-Piece Clamp-on Shaft Collar for 1-3/4" Diameter
Drive Train	Shaft Bearing	6384K69	$13.78	4	1	1	$55.12	Plain Double Shielded for 3/4" Shaft Diameter, 1-3/4" OD
Drive Train	Drive Shaft Retaining Rings	97633A250	$12.30	1	8	100	$0.98	Black-Finish Steel External Retaining Ring for 3/4" Shaft Diameter
Drive Train	Drive Shaft	1346K33	$36.90	1	32	32	$36.90	Steel Drive Shaft 3/4" OD, 36" Length
Drive Train	Drive Shaft attempt 2	1346K33	$36.90	1	16	32	$18.45	Steel Drive Shaft 3/4" OD, 36" Length
Drive Train	Drive Wheels	8975K106	$40.88	2	1	1	$81.76	Multipurpose 6061 Aluminum 1/2" Thick, 10" Width, 1 ft Length
Drive Train	Hub Caps	8975K87	$8.51	1	1	1	$8.51	Multipurpose 6061 Aluminum 1/4" Thick, 3" Width, 1 ft Length
Drive Train	Hub Cap Screws	92865A544	$7.54	1	8	50	$1.21	Medium-Strength Grade 5 Zinc-Plated Steel Cap Screw 1/4"-20 Fully Threaded, 1-1/4" Long
Drive Train	Hub Cap Nuts	95462A029	$4.40	1	8	100	$0.35	Grade 5 Steel Hex Nut Zinc Plated, 1/4"-20 Thread Size, 7/16" Wide, 7/32" High
Drive Train	Idler Shaft (Screws)	92865A544	$7.54	1	2	50	$0.30	Medium-Strength Grade 5 Zinc-Plated Steel Cap Screw 1/4"-20 Fully Threaded, 1-1/4" Long
Drive Train	Idler Shaft (Nuts)	95462A029	$4.40	1	2	100	$0.09	Grade 5 Steel Hex Nut Zinc Plated, 1/4"-20 Thread Size, 7/16" Wide, 7/32" High
Frame	Frame Square-Tube	*stock provided*	–	–		–		*3/4" Steel Tubing*
Frame	Frame C-Tube	*stock provided*	–	–		–		*3/4" Steel C-Channel Tubing*
Frame	Frame Gussets	*stock provided*	–	–		–		*1/8" Steel Sheet*
Frame	C-Channel Foam Padding	4339T8	$10.30	1	6	6	$10.30	APPX- Weather-Resistant EPDM Foam Tube, 3/4" OD, 1/2" ID, 6' Length
Guide Wheels	Acetal Wheels (stock)	8576K32	$27.10	1	4	12	$9.03	Wear- and Water- Resistant Delrin Acetal Resin, 2.5" OD
Guide Wheels	Acetal Wheel Bearings	57155K323	$5.70	8	1	1	$45.60	Mini High-Precision Stainless Steel Ball Bearing - ABEC-5 Flanged Shielded, 1/4" ID, 1/2" OD, 3/16" Wide
Guide Wheels	Guide Wheel Shaft (Screws)	92865A544	$7.54	1	6	50	$0.90	Medium-Strength Grade 5 Zinc-Plated Steel Cap Screw 1/4"-20 Fully Threaded, 1-1/4" Long
Guide Wheels	Guide Wheel Shaft (Nuts)	95462A029	$4.40	1	6	100	$0.26	Grade 5 Steel Hex Nut Zinc Plated, 1/4"-20 Thread Size, 7/16" Wide, 7/32" High
Guide Wheels	Guide Wheel Springs	9433K47	$6.87	1	2	3	$4.58	Precision Stainless Steel Extension Spring, 2.50" Length, .500" OD, .063" Wire Diameter
Seat	REI Stadium Seat	–	$19.99	1	1	1	$19.99	http://www.rei.com/product/882237/mountain-summit-gear-stadium-seat
Seat	Plywood Base	*stock provided*						3/8" Plywood
Misc.	Bike Parts	–	$40.00	1	1	1	$40.00	Various bike parts from junked bikes (Rear Sprocket, Pedals, Derailleur, Idler Sprockets)
		Total:	**$810.18**				$673.55	
Not Used	Diff Shaft Bushing	6338K429	$2.06	4	1	1	$8.24	SAE 841 Bronze Flanged-Sleeve Bearing
Not Used	Gear Set Screw	91251A352	$5.92	1	2	25	$0.47	Black-Oxide Alloy Steel Socket Head Cap Screw 10-32 Thread, 1-3/4" Length
Not Used	Guide Wheel Springs	1832K33	$9.79	1	2	6	$3.26	Spring-Tempered Steel Cot Extension Spring 3.0" Length, .750" OD, .105" Wire Diameter
Not Used	Guide Wheel Springs	1832K35	$10.83	1	2	6	$3.61	Spring-Tempered Steel Cot Extension Spring 3.0" Length, .750" OD, .105" Wire Diameter
Not Used	Chain Guide Sprocket	6793K118	$9.46	1	1	1	$9.46	Steel Machinable-Bore Sprocket for ANSI Number 35 Roller Chain, 3/8" Pitch, 11 Teeth
Not Used	Acetal Wheel Retaining Rings	97633A130	$7.82	1	16	100	$1.25	Black-Finish Steel External Retaining Ring for 1/4" Shaft Diameter
Not Used	Acetal Wheel Axle	1886K1	$4.88	1	11	12	$4.47	Black-Oxide Coated Steel Shaft, 1/4" OD

APPENDIX B: Simulations

FEA Study 1: The Drive Wheel

To ensure that cutting the bevel gear speed hole pattern in our drive wheels would not compromise the integrity of the wheel and cause failure of our design we conducted an FEA study on the drive wheel. From this study we concluded that with the shown spacing of the bevel gear design our drive wheels have a factor of safety of 9 based on a 100 lbf load.



Model name: DriveWheel_STATIC_TEST
Current Configuration: Simplify_1

Solid Bodies

Document Name and Reference	Treated As	Volumetric Properties	Document Path/Date Modified
Split Line2 	Solid Body	Mass:0.746723 kg Volume:0.000276564 m^3 Density:2700 kg/m^3 Weight:7.31789 N	P:\15spring\engs146\workspace\DiWheel Groups\Bevel's Advocate\Diff Case\DriveWheel_STATIC_TEST.SLDPRT Jun 06 18:26:37 2015

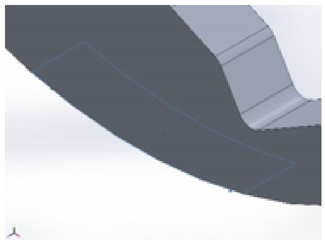
Study Properties

Study name	Static - 0deg
Analysis type	Static
Mesh type	Solid Mesh
Thermal Effect:	On
Thermal option	Include temperature loads
Zero strain temperature	298 Kelvin
Include fluid pressure effects from SolidWorks Flow Simulation	Off
Solver type	FFEPlus
Inplane Effect:	Off
Soft Spring:	Off
Inertial Relief:	Off
Incompatible bonding options	Automatic
Large displacement	Off
Compute free body forces	On
Friction	Off
Use Adaptive Method:	Off
Result folder	SolidWorks document (c:\users\d36106r\appdata\local\temp)

Material Properties

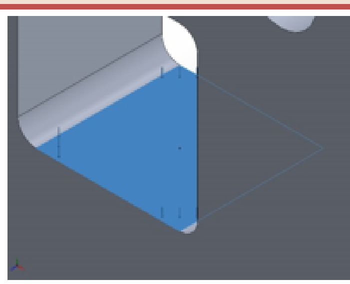
Model Reference	Properties	Components
	Name: 6061 Alloy Model type: Linear Elastic Isotropic Default failure criterion: Unknown Yield strength: 5.51485e+007 N/m^2 Tensile strength: 1.24084e+008 N/m^2 Elastic modulus: 6.9e+010 N/m^2 Poisson's ratio: 0.33 Mass density: 2700 kg/m^3 Shear modulus: 2.6e+010 N/m^2 Thermal expansion coefficient: 2.4e-005 /Kelvin	SolidBody 1(Split Line2)(DriveWheel)
Curve Data:N/A		

Loads and Fixtures

Fixture name	Fixture Image	Fixture Details
Fixed-2		**Entities:** 1 face(s) **Type:** Fixed Geometry

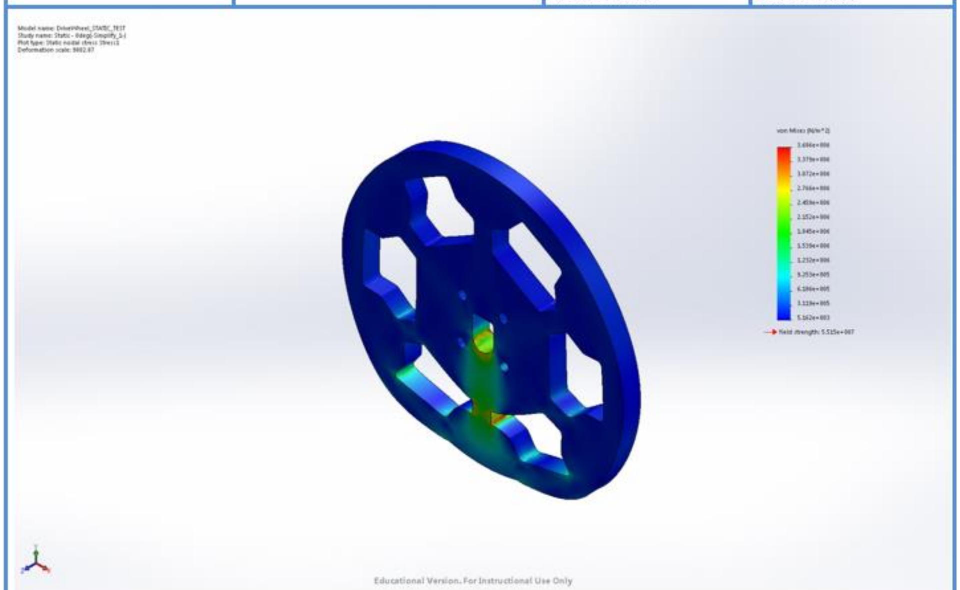
Resultant Forces

Components	X	Y	Z	Resultant
Reaction force(N)	-0.00846505	444.844	-0.00181782	444.844
Reaction Moment(N.m)	0	0	0	0

Load name	Load Image	Load Details
Force-1		**Entities:** 1 face(s) **Type:** Apply normal force **Value:** 100 lbf

Study Results

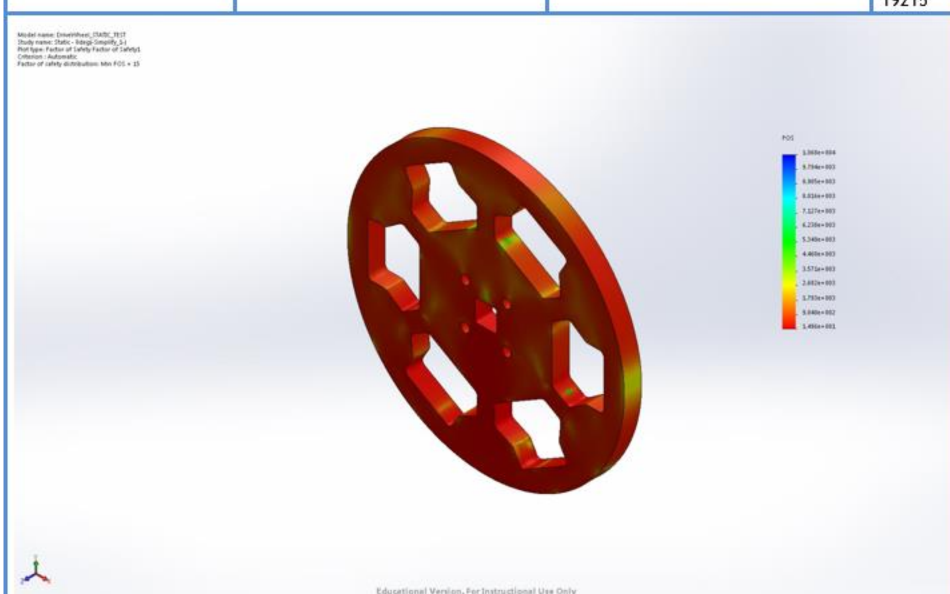
Name	Type	Min	Max
Stress1	VON: von Mises Stress	5162.03 N/m^2 Node: 19215	3.68567e+006 N/m^2 Node: 13337



DriveWheel_STATIC_TEST-Static - 0deg-Stress-Stress1

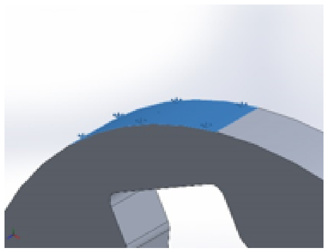
Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 mm Node: 216	0.00207637 mm Node: 18012

Name	Type	Min	Max
Factor of Safety1	Automatic	14.963 Node: 13337	10683.5 Node: 19215



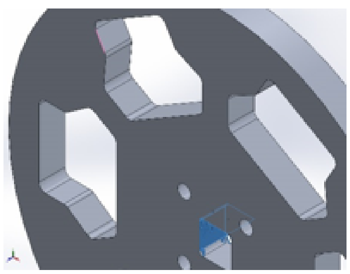
DriveWheel_STATIC_TEST-Static - 0deg-Factor of Safety-Factor of Safety1

Loads and Fixtures

Fixture name	Fixture Image	Fixture Details
Fixed-2		Entities: 1 face(s) Type: Fixed Geometry

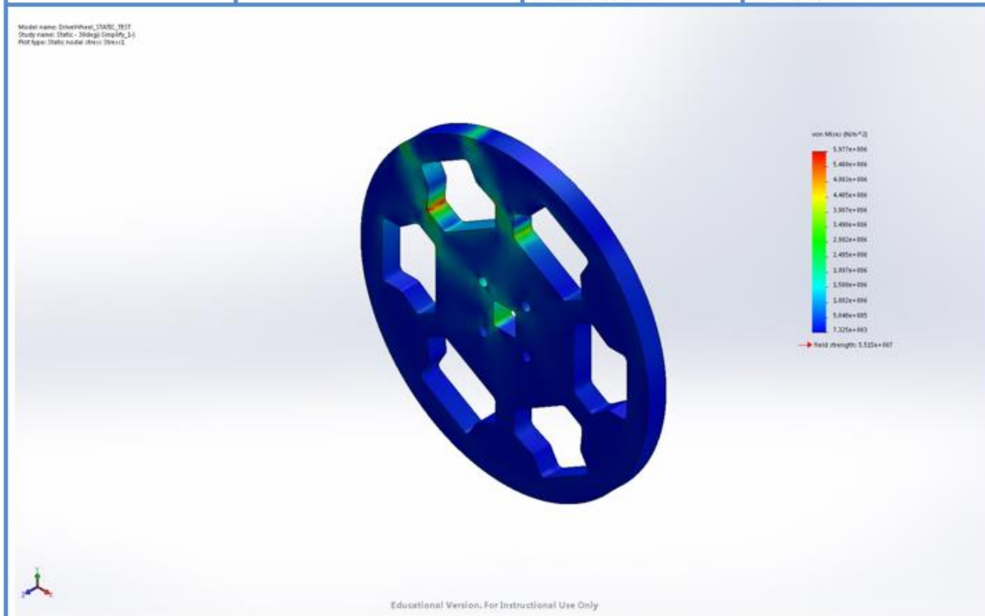
Resultant Forces

Components	X	Y	Z	Resultant
Reaction force(N)	222.366	-385.237	0.0195328	444.808
Reaction Moment(N.m)	0	0	0	0

Load name	Load Image	Load Details
Force-1		Entities: 3 face(s) Reference: Edge< 1 > Type: Apply force Values: ---, ---, 100 lbf

Study Results

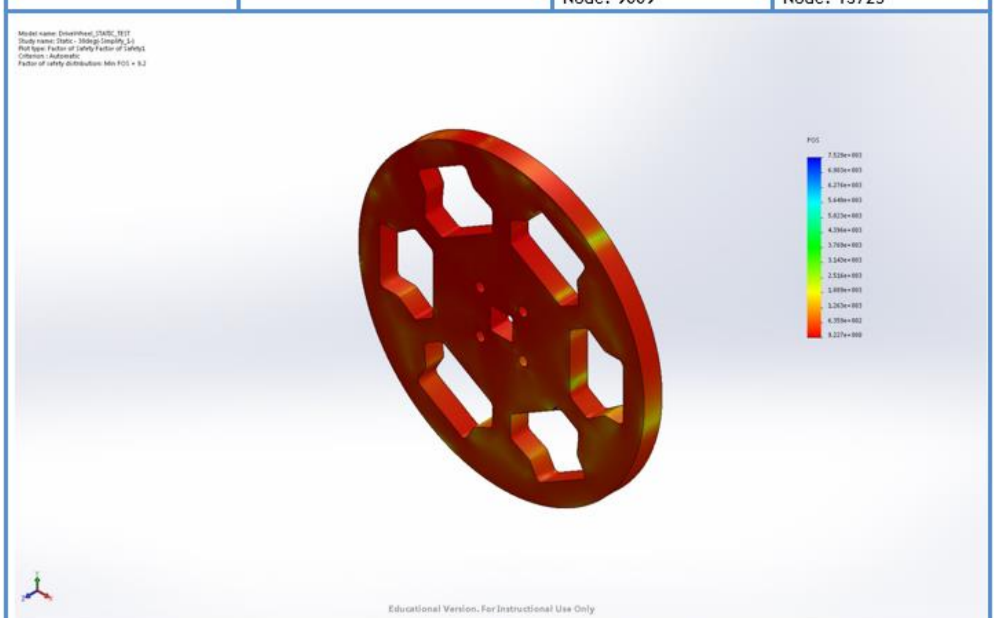
Name	Type	Min	Max
Stress1	VON: von Mises Stress	7324.63 N/m^2 Node: 13725	5.97701e+006 N/m^2 Node: 9009



DriveWheel_STATIC_TEST-Static - 30deg-Stress-Stress1

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 mm Node: 85	0.0024658 mm Node: 9961

Name	Type	Min	Max
Factor of Safety1	Automatic	9.22677 Node: 9009	7529.18 Node: 13725

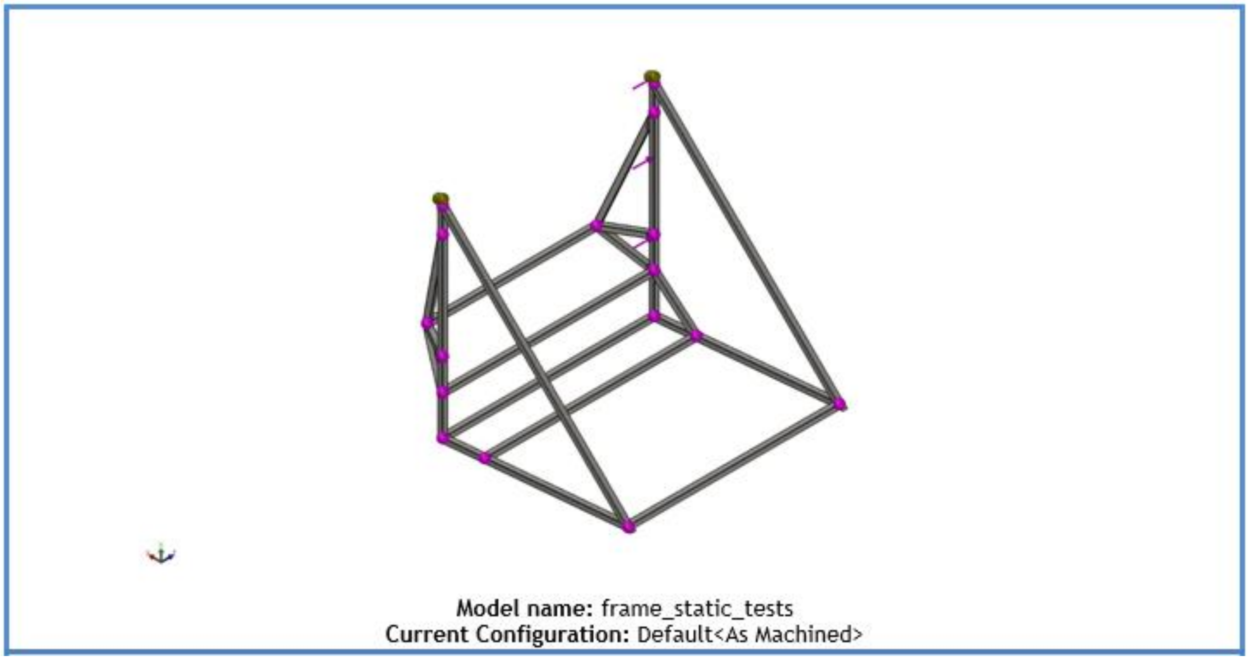


DriveWheel_STATIC_TEST-Static - 30deg-Factor of Safety-Factor of Safety1

FEA Study 2: The Frame

The frame shown in the study is from an earlier design which we tested and iterated on to come to our final frame design. One of our concerns was deflection of the top most triangular corners under lateral load due to decreased bracing from the open cockpit design. Two studies were performed, one with vertical loads, and the second with a singularly horizontal load as a worse-case scenario. Our conclusions are articulated in the frame section of the report, but briefly we found that the open cockpit was a viable design with deflections under 0.4 inches.

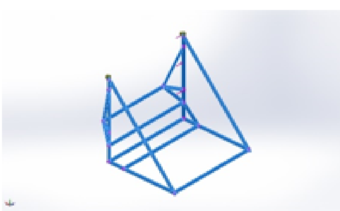
Model Information



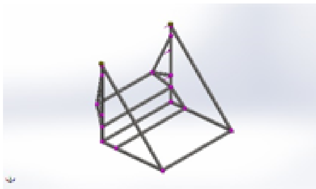
Study Properties

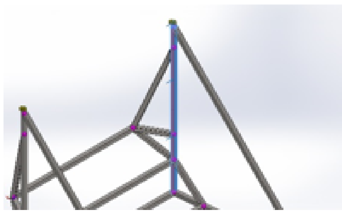
Study name	Static Lateral
Analysis type	Static
Mesh type	Mixed Mesh
Thermal Effect:	On
Thermal option	Include temperature loads
Zero strain temperature	298 Kelvin
Include fluid pressure effects from SolidWorks Flow Simulation	Off
Solver type	Direct sparse solver
Inplane Effect:	Off
Soft Spring:	Off
Inertial Relief:	Off
Incompatible bonding options	Automatic
Large displacement	Off
Compute free body forces	On
Friction	Off
Use Adaptive Method:	Off
Result folder	SolidWorks document (c:\users\d37668n\appdata\local\temp)

Material Properties

Model Reference	Properties	Components
	Name: AISI 1020 Model type: Linear Elastic Isotropic Default failure criterion: Unknown Yield strength: 3.51571e+008 N/m^2 Tensile strength: 4.20507e+008 N/m^2 Elastic modulus: 2e+011 N/m^2 Poisson's ratio: 0.29 Mass density: 7900 kg/m^3 Shear modulus: 7.7e+010 N/m^2 Thermal expansion coefficient: 1.5e-005 /Kelvin	SolidBody 1(Trim/Extend10[1])(frame_static_tests), SolidBody 2(Structural Member1[1])(frame_static_tests), SolidBody 3(Structural Member1[14])(frame_static_tests), SolidBody 4(Structural Member1[6])(frame_static_tests), SolidBody 5(Trim/Extend10[2])(frame_static_tests), SolidBody 6(Trim/Extend3)(frame_static_tests), SolidBody 7(Structural Member1[2])(frame_static_tests), SolidBody 8(Trim/Extend7)(frame_static_tests), SolidBody 9(Structural Member1[4])(frame_static_tests), SolidBody 10(Structural Member2[2])(frame_static_tests), SolidBody 11(Structural Member1[7])(frame_static_tests), SolidBody 12(Trim/Extend11[1])(frame_static_tests), SolidBody 13(Structural Member1[3])(frame_static_tests), SolidBody 14(Structural Member1[9])(frame_static_tests), SolidBody 15(Trim/Extend11[2])(frame_static_tests), SolidBody 16(Structural Member1[5])(frame_static_tests), SolidBody 17(Structural Member1[8])(frame_static_tests), SolidBody 20(Trim/Extend15[1])(frame_static_tests)
Curve Data:N/A		

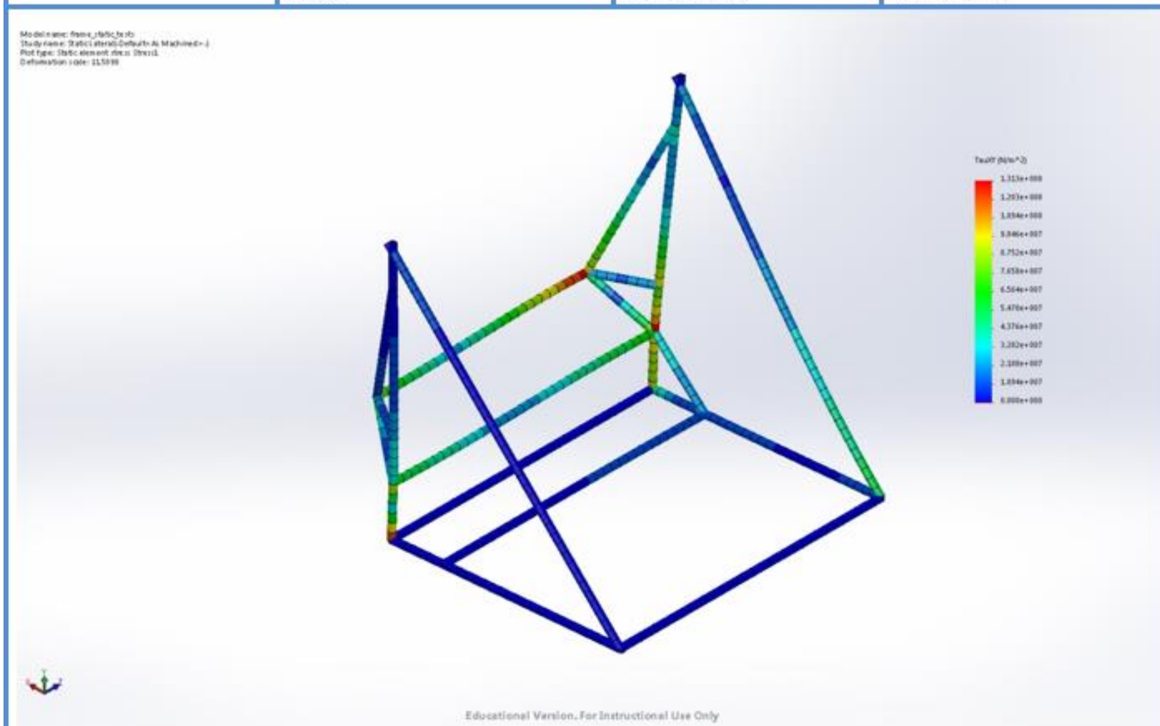
Loads and Fixtures (Lateral)

Fixture name	Fixture Image	Fixture Details
Fixed-1		**Type:** Fixed Geometry

Load name	Load Image	Load Details
Force-1		**Entities:** 1 Beam (s) **Reference:** Face< 1 > **Type:** Apply force **Values:** ---, ---, 200 lbf **Moments:** ---, ---, --- lbf.in

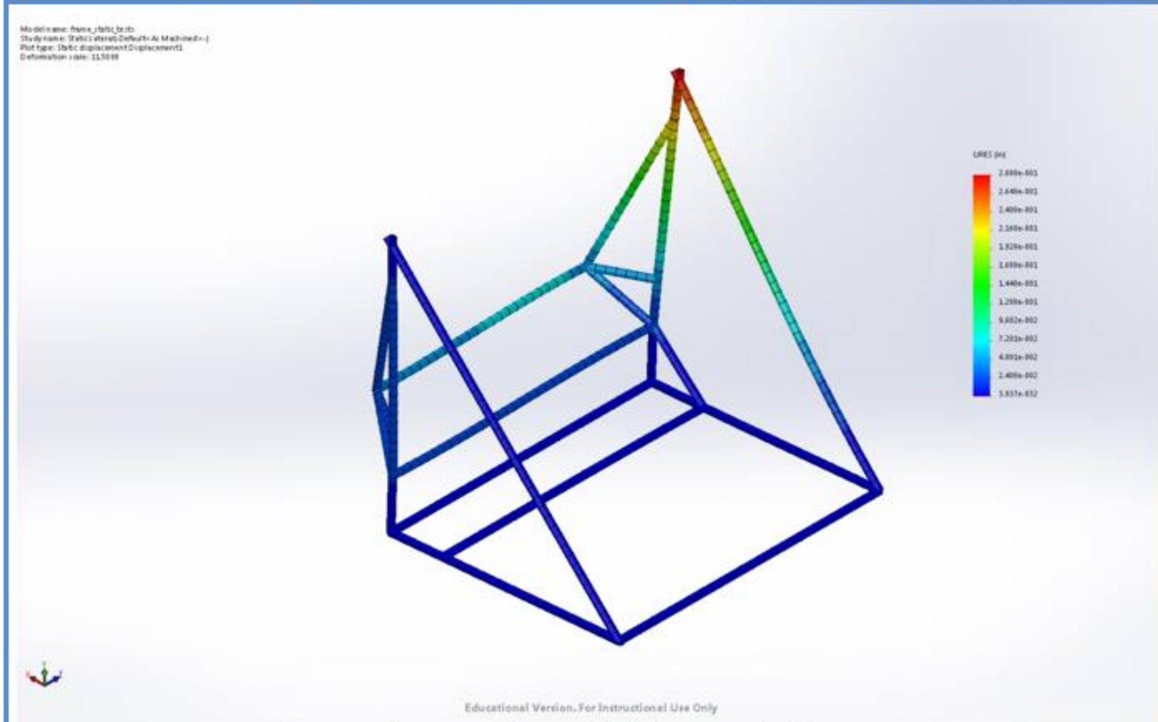
Study Results (Lateral)

Name	Type	Min	Max
Stress1	TXY: Shear in Y Dir. on YZ Plane	0 N/m^2 Element: 73	1.3128e+008 N/m^2 Element: 33



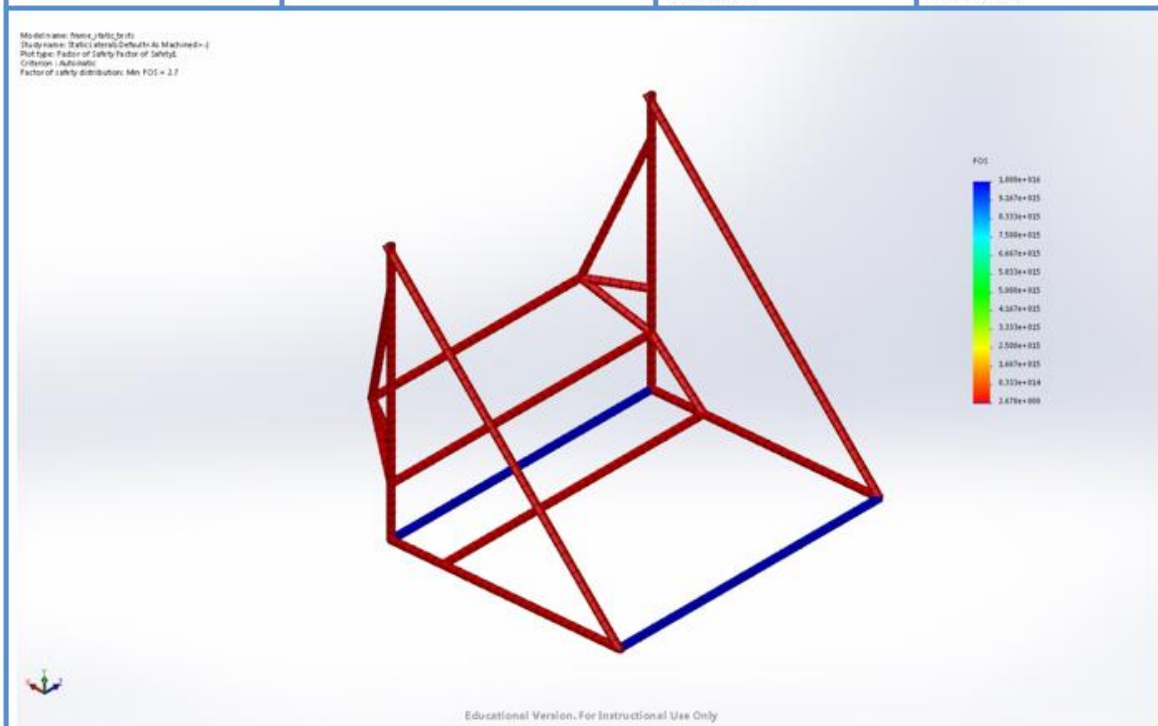
frame_static_tests-Static Lateral-Stress-Stress1

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 in Node: 41	0.28805 in Node: 9



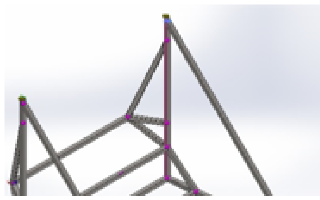
frame_static_tests-Static Lateral-Displacement-Displacement1

Name	Type	Min	Max
Factor of Safety1	Automatic	2.67803 Node: 34	1e+016 Node: 75



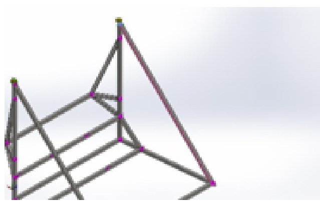
frame_static_tests-Static Lateral-Factor of Safety-Factor of Safety1

Loads and Fixtures (Vertical)

Fixture name	Fixture Image	Fixture Details
Reference Geometry-1		Reference: Face< 1 > Type: Use reference geometry Translation: 0.12, -0.12, 0.12 Rotation: ---, ---, --- Units: in, rad

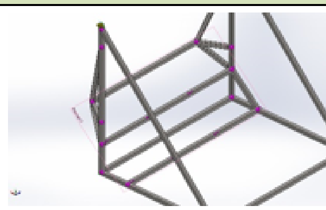
Resultant Forces

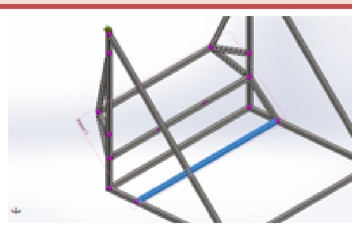
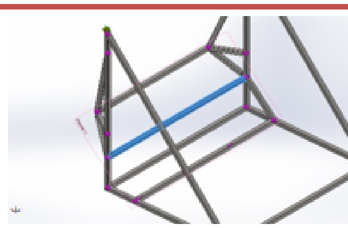
Components	X	Y	Z	Resultant
Reaction force(N)	-140.248	226.797	58949.3	58949.9
Reaction Moment(N.m)	0	0	0	1e-033

Fixture name	Fixture Image	Fixture Details
Reference Geometry-2		Reference: Face< 1 > Type: Use reference geometry Translation: 0.12, -0.12, 0.12 Rotation: ---, ---, --- Units: in, rad

Resultant Forces

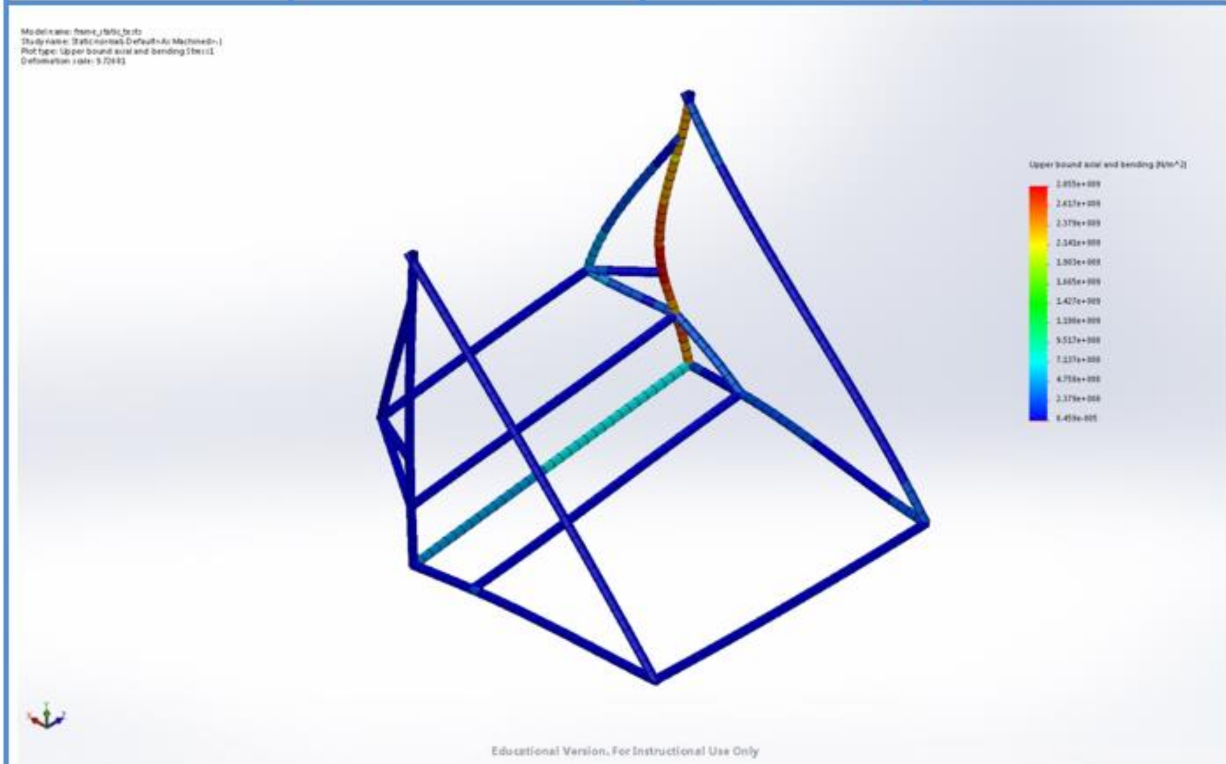
Components	X	Y	Z	Resultant
Reaction force(N)	30.2484	-250157	-171.632	250157
Reaction Moment(N.m)	0	0	0	1e-033

Fixture name	Fixture Image	Fixture Details
Reference Geometry-3		Entities: 1 plane(s) Reference: Plane11 Type: Use reference geometry Translation: 0.2, 0.2, 0.2 Rotation: ---, ---, --- Units: in, rad

Load name	Load Image	Load Details
Force-1		Entities: 1 plane(s), 1 Beam (s) Reference: Plane11 Type: Apply force Values: ---, ---, 120 lbf Moments: ---, ---, --- lbf.in
Force-2		Entities: 1 plane(s), 1 Beam (s) Reference: Plane11 Type: Apply force Values: ---, ---, 80 lbf Moments: ---, ---, --- lbf.in

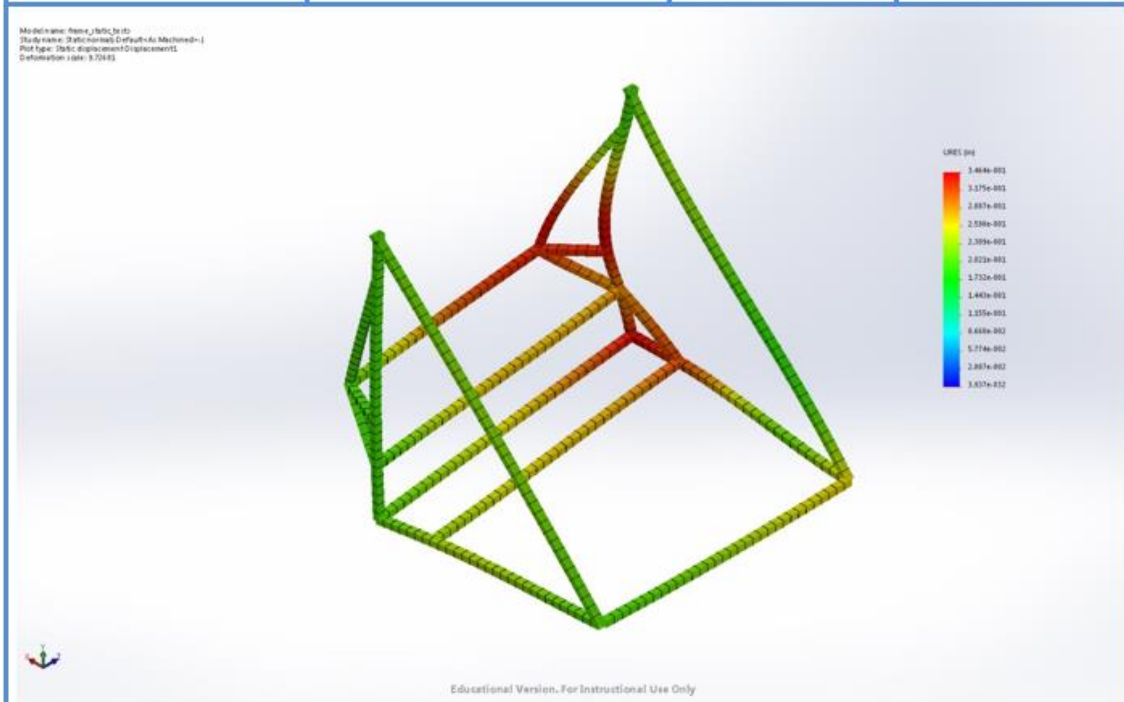
Study Results (Vertical)

Name	Type	Min	Max
Stress1	TXY: Shear in Y Dir. on YZ Plane	0 N/m^2 Element: 439	2.85497e+009 N/m^2 Element: 28



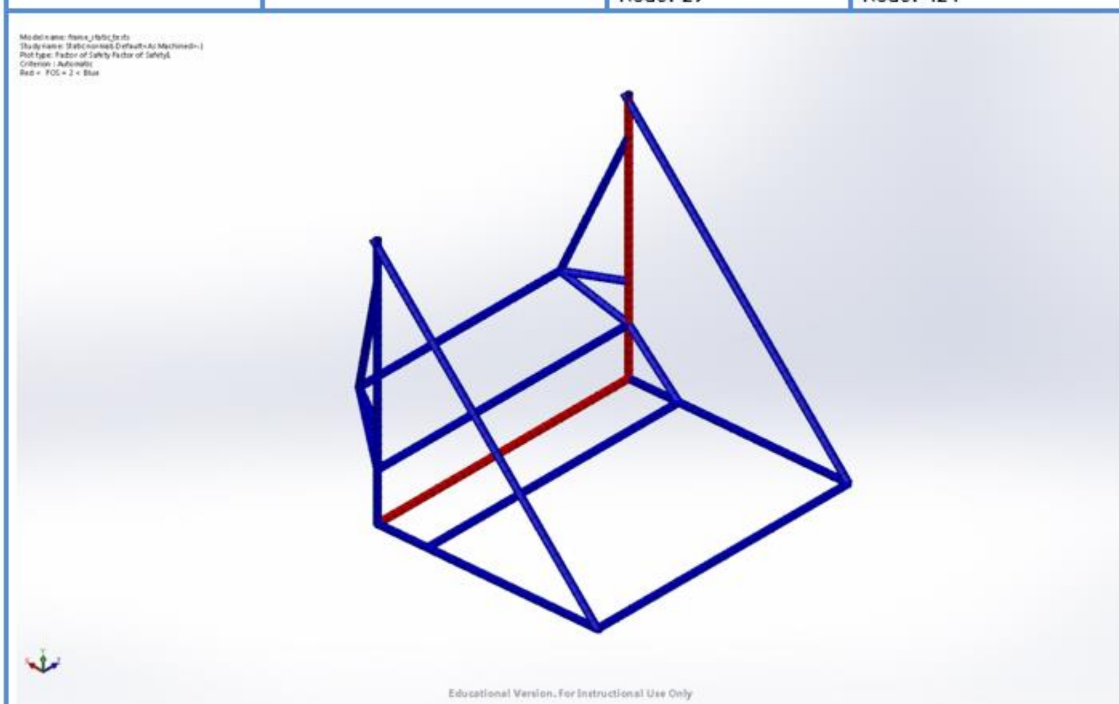
frame_static_tests-Static normal-Stress-Stress1

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 in Node: 433	0.34641 in Node: 41



frame_static_tests-Static normal-Displacement-Displacement1

Name	Type	Min	Max
Factor of Safety1	Automatic	0.36943 Node: 29	1.24688e+013 Node: 424



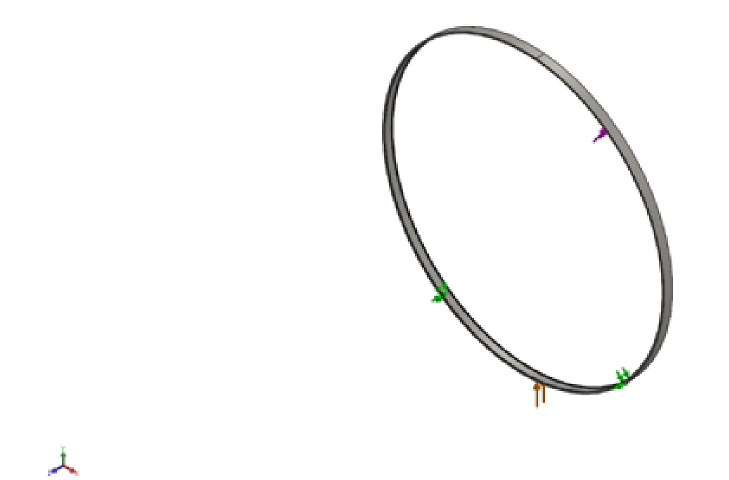
frame_static_tests-Static normal-Factor of Safety-Factor of Safety1

FEA Study 3: The Hoop

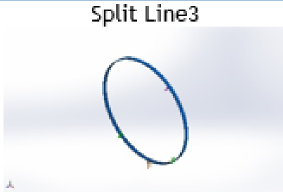
Based on the placement of our drive wheels and guide wheels we wanted to know the deflection of the hoop, because in the past a significant challenge for teams during the competition was derailment. This study shows that there is some deflection of the hoop, but we accounted for it with the spring constant and guide wheel hoop tensioning mechanism.

In initial hoop static studies we found much larger deflections with the hoop fixed, but when the fixture was changed to a translational fixture, which we felt was more realistic to actual operation, we found the results shown below.

Model Information



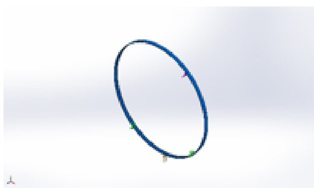
Model name: hoop
Current Configuration: Default

Solid Bodies			
Document Name and Reference	**Treated As**	**Volumetric Properties**	**Document Path/Date Modified**
Split Line3 	Solid Body	Mass:2.63829 kg Volume:0.00033396 m^3 Density:7900 kg/m^3 Weight:25.8552 N	P:\15spring\engs146\work space\DiWheel Groups\Bevel's Advocate\frame\hoop.SLD PRT Jun 07 12:08:42 2015

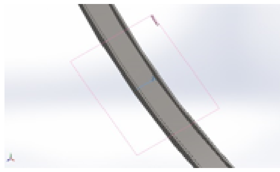
Study Properties

Study name	Static 2
Analysis type	Static
Mesh type	Solid Mesh
Thermal Effect:	On
Thermal option	Include temperature loads
Zero strain temperature	298 Kelvin
Include fluid pressure effects from SolidWorks Flow Simulation	Off
Solver type	FFEPlus
Inplane Effect:	Off
Soft Spring:	Off
Inertial Relief:	Off
Incompatible bonding options	Automatic
Large displacement	On
Compute free body forces	Off
Friction	Off
Use Adaptive Method:	Off
Result folder	SolidWorks document (c:\users\d37213q\appdata\local\temp)

Material Properties

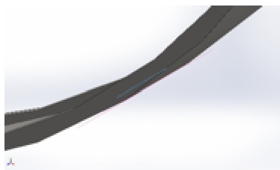
Model Reference	Properties	Components
	Name: AISI 1020 Model type: Linear Elastic Isotropic Default failure criterion: Unknown Yield strength: 3.51571e+008 N/m^2 Tensile strength: 4.20507e+008 N/m^2 Elastic modulus: 2e+011 N/m^2 Poisson's ratio: 0.29 Mass density: 7900 kg/m^3 Shear modulus: 7.7e+010 N/m^2 Thermal expansion coefficient: 1.5e-005 /Kelvin	SolidBody 1(Split Line3)(hoop)
Curve Data:N/A		

Loads and Fixtures

Fixture name	Fixture Image	Fixture Details
Reference Geometry-1		Entities: 1 edge(s), 1 plane(s) Reference: Plane1 Type: Use reference geometry Translation: 1, 1, 1 Units: in

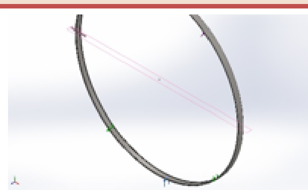
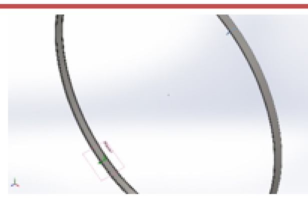
Resultant Forces

Components	X	Y	Z	Resultant
Reaction force(N)	-16.2242	-17.7537	0.00592041	24.0504
Reaction Moment(N.m)	0	0	0	0

Fixture name	Fixture Image	Fixture Details
Reference Geometry-2		Entities: 1 edge(s), 1 plane(s) Reference: Plane2 Type: Use reference geometry Translation: 1, 1, 1 Units: in

Resultant Forces

Components	X	Y	Z	Resultant
Reaction force(N)	3.09928	-0.257999	-0.0059967	3.11001
Reaction Moment(N.m)	0	0	0	0

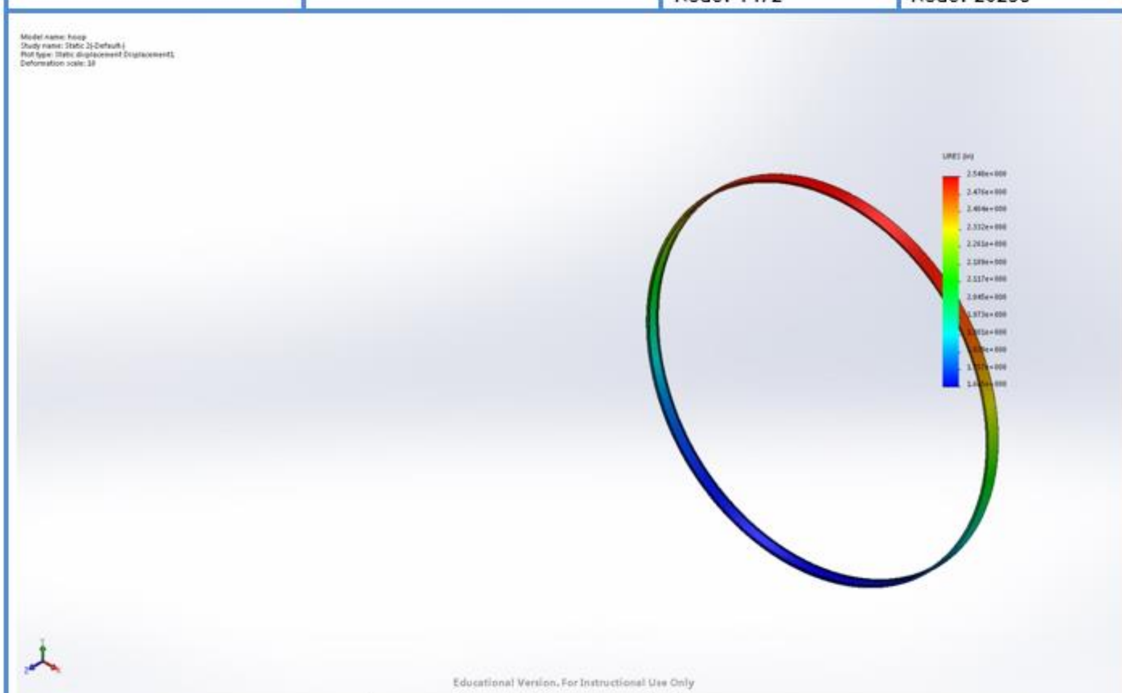
Load name	Load Image	Load Details
Force-4		Entities: 1 edge(s), 1 plane(s) Reference: Top Plane Type: Apply force Values: ---, ---, 200 lbf
Force-5		Entities: 1 edge(s), 1 plane(s) Reference: Plane1 Type: Apply force Values: ---, ---, 5 lbf

Study Results

Name	Type	Min	Max
Stress1	VON: von Mises Stress	32401.1 N/m^2 Node: 25060	7.49815e+007 N/m^2 Node: 31854

hoop-Static 2-Stress-Stress1

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	1.68548 in Node: 1472	2.54814 in Node: 20256

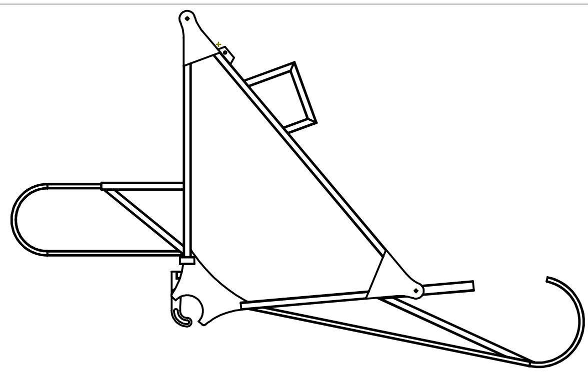


hoop-Static 2-Displacement-Displacement1

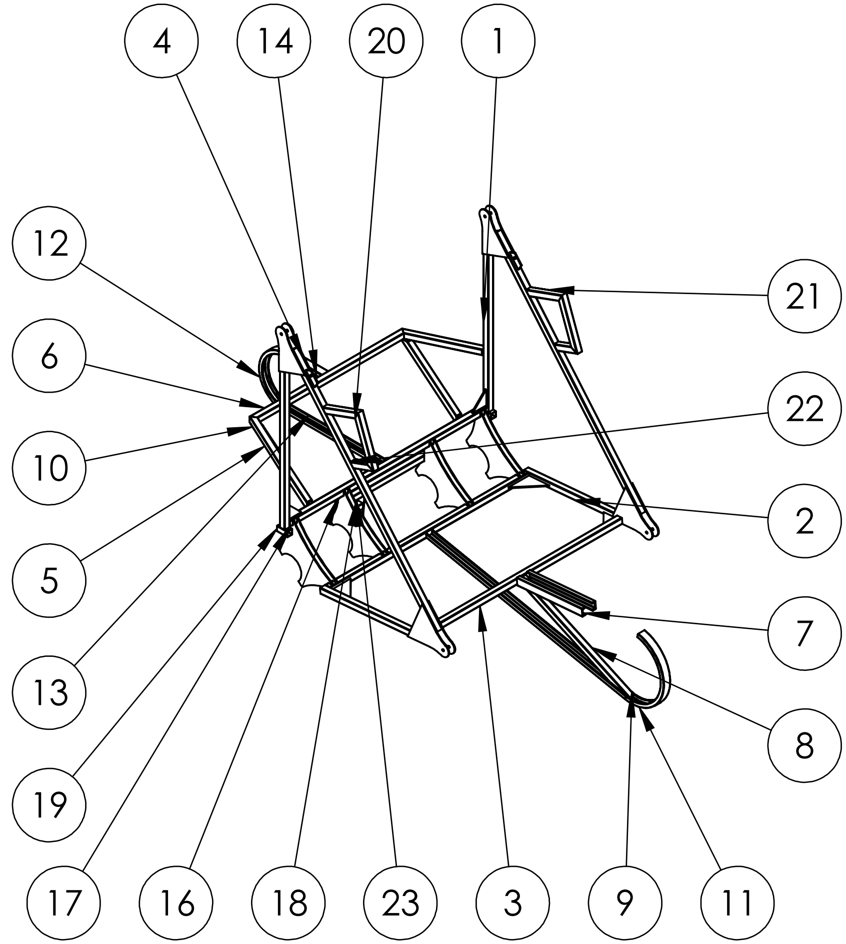
Motion Analysis

See full diwheel motion study titled “BA Diwheel Motion Study” and differential motion study titled “BA Differential Motion Study” in the submission folder on ThayerFS

APPENDIX C: Engineering Drawings



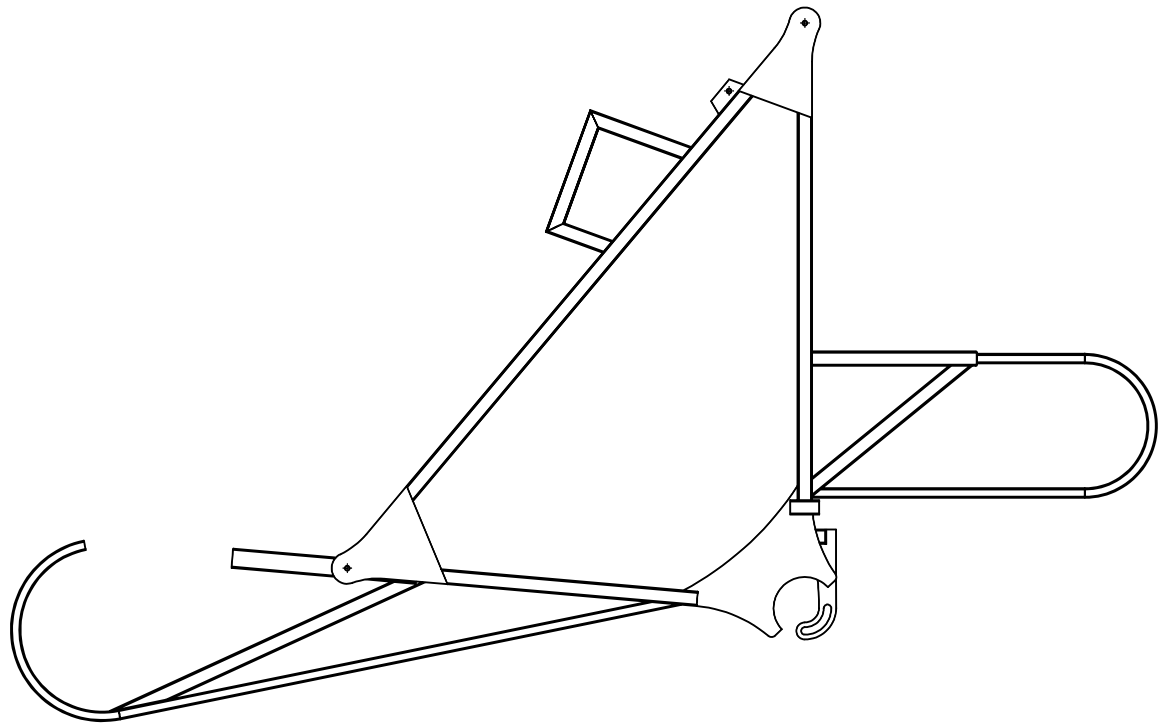
ITEM NO.	QTY.	LENGTH	ANGLE1	ANGLE2	TYPE
1	2	22.85	0.00	0.00	Square
2	2	16.29	0.00	0.00	Square
3	1	29	0.00	0.00	Square
4	2	33.06	0.00	0.00	Square
5	2	9.46	32.50	25.00	Square
6	1	22.48	32.50	32.50	Square
7	2	9.63	0.00	0.00	C
8	1	17.73	76.87	-	Square
9	1	30	0.00	16.53	C
10	2	10.76	39.22	50.78	Square
11	1	14.14	0.00	0.00	C
12	1	11	0.00	0.00	C
13	1	14.2	0.00	0.00	C
14	1	5.63	0.00	0.00	C
15	1	0.38	0.00	0.00	Square
16	1	12.85	0.00	0.00	Square
17	1	58.5	0.00	0.00	Square
18	1	29.25	0.00	0.00	Square
19	2	1.5	0.00	0.00	Square
20	2	6.75	45.00	45.00	Square
21	2	5.61	20.00	45.00	Square
22	2	3.43	45.00	20.00	Square
23	1	3.11	0.00	0.00	Square



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES:	DRAWN			
	CHECKED			FRAME
ANGULAR: MACH± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS:			SIZE A · DWG. NO. · REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:5 · WEIGHT: · SHEET 5 OF 5

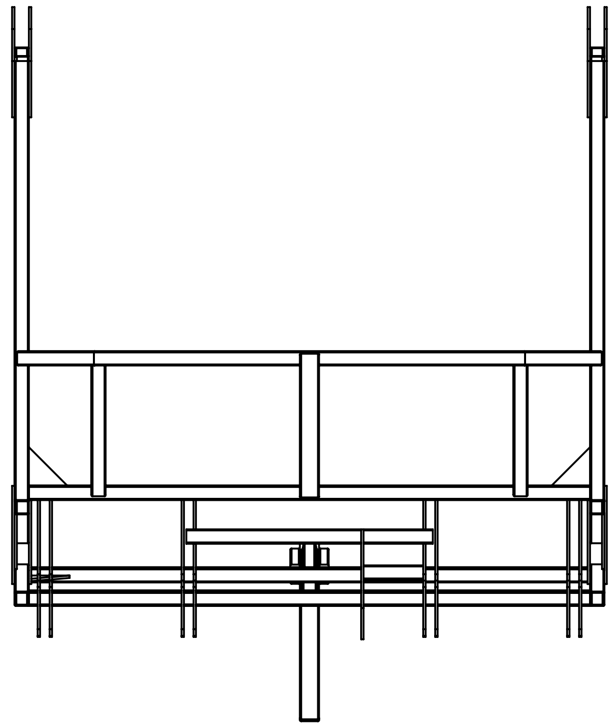
5 4 3 2 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN			Frame Side
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS:			SIZE A · DWG. NO. · REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:20 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

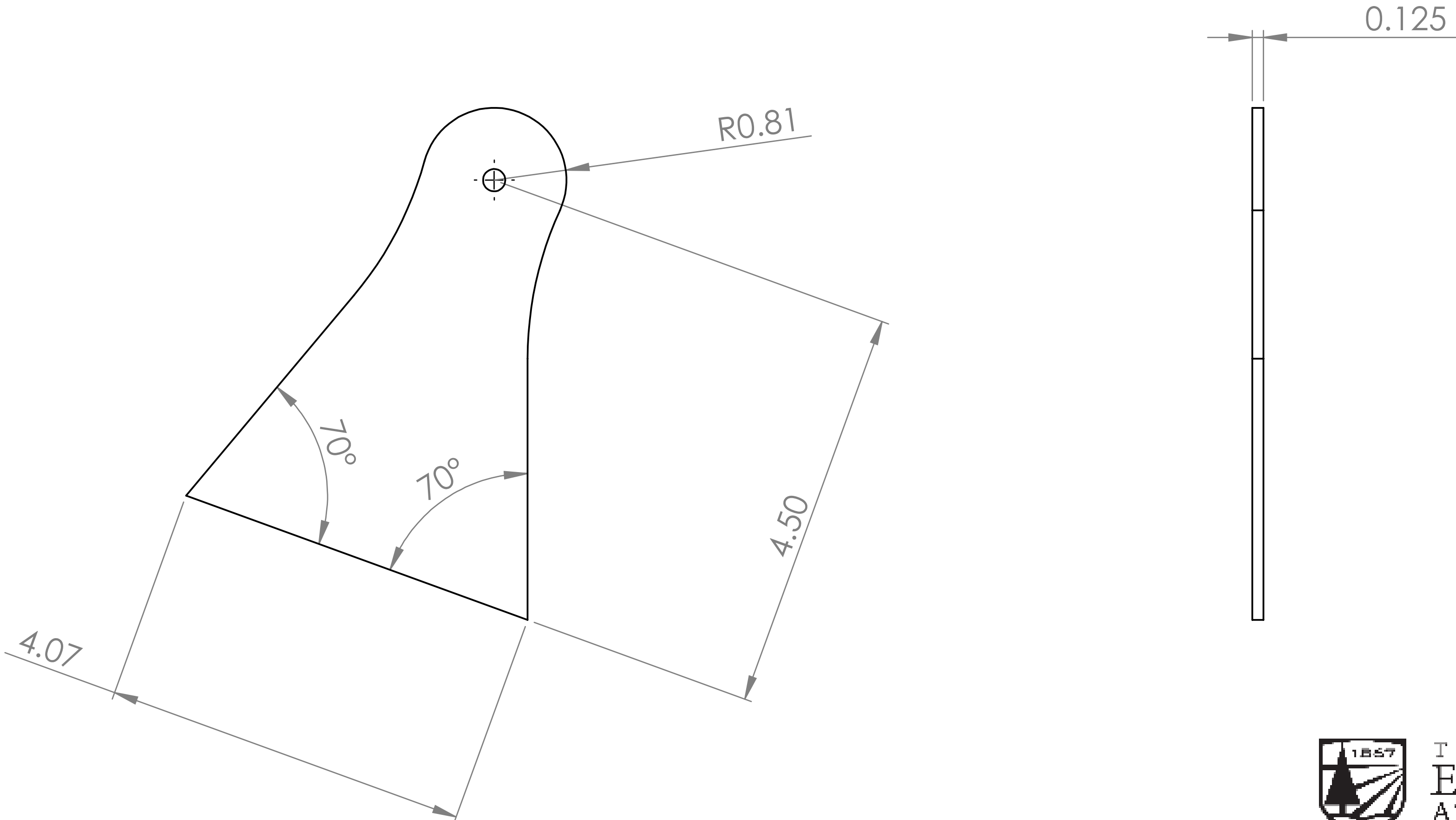
TITLE:

Frame Back

SIZE	DWG. NO.	REV
A		

SCALE: 1:20 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:

ANGULAR: MACH ± 30'
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
1/8" Steel Stock

FINISH
As Plasma Cut

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

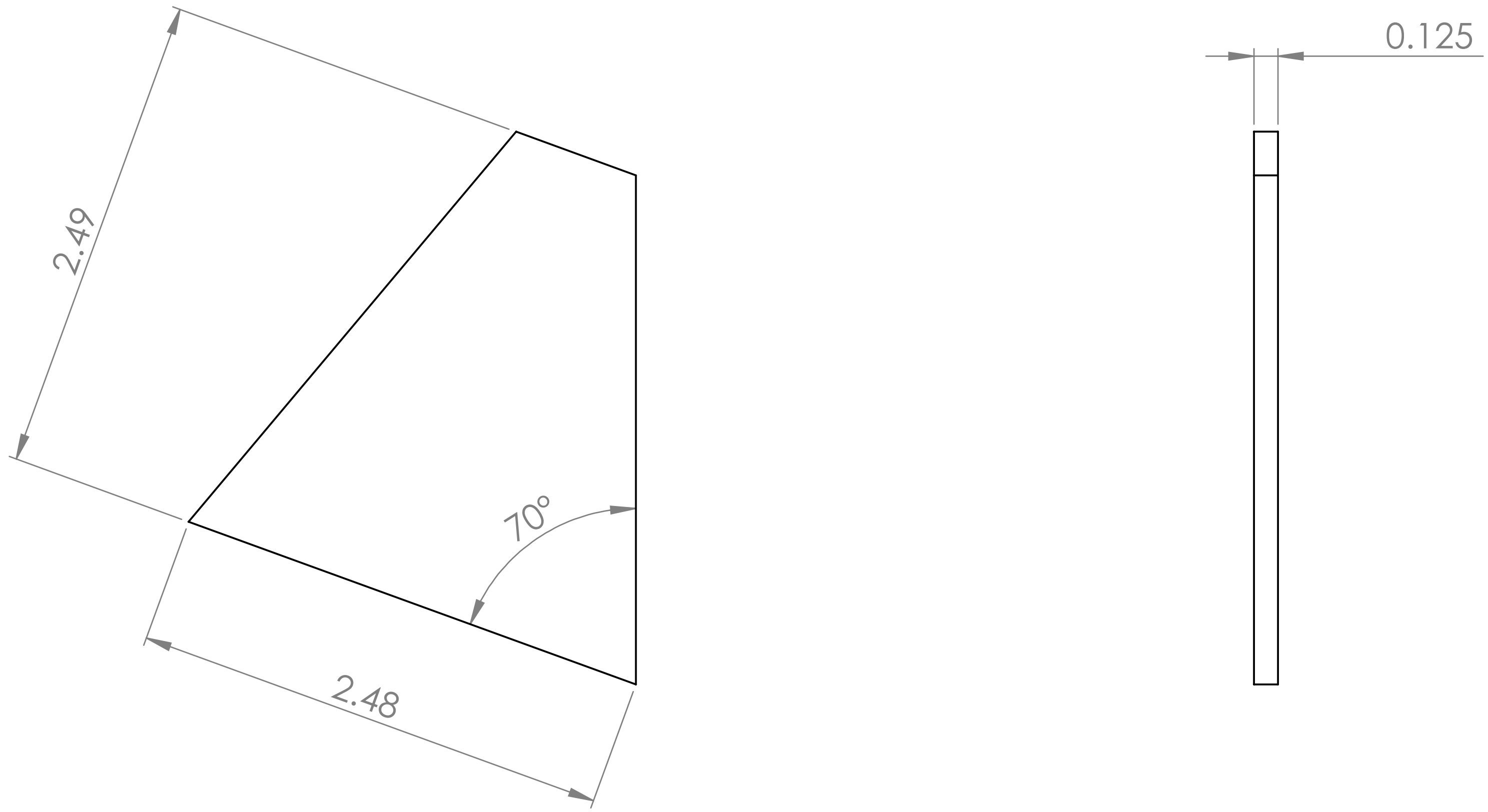
TITLE:

Top Outside Gusset

SIZE A | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

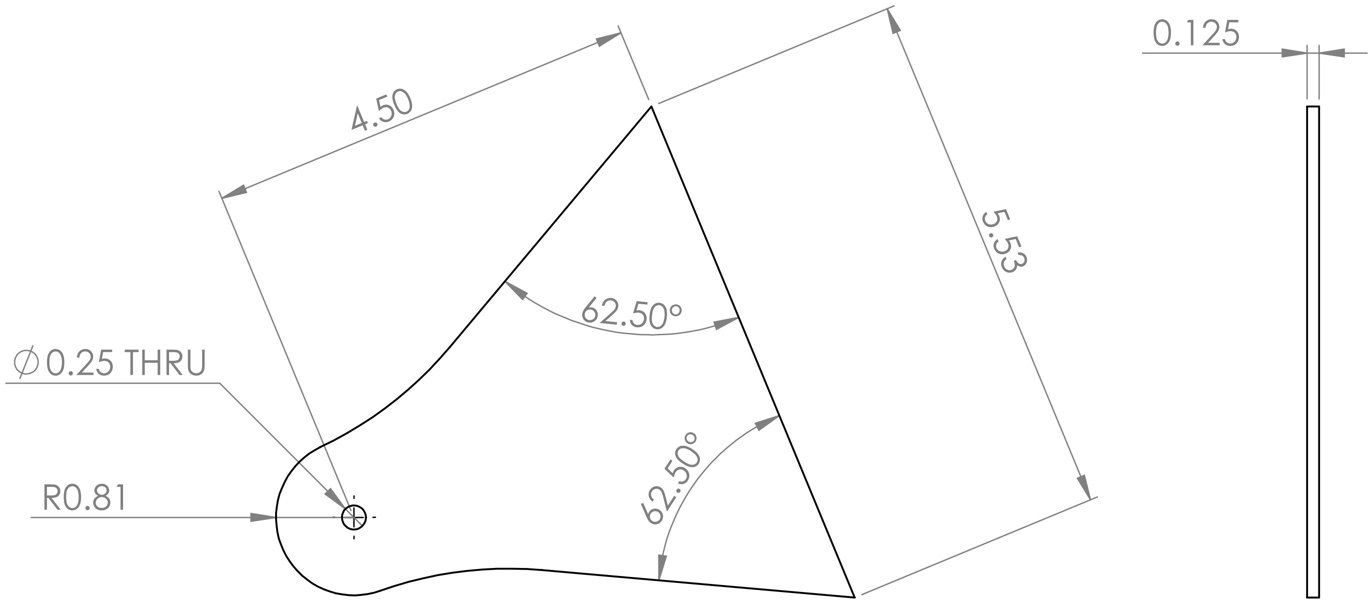
5 4 3 2 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN			Top Inside Gusset
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL 1/8" Steel Stock	COMMENTS:			SIZE A · DWG. NO. · REV
FINISH As Plasma Cut				
DO NOT SCALE DRAWING				SCALE: 1:1 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:

ANGULAR: MACH ± 30'
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL 1/8" Steel Stock

FINISH As Plasma Cut

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

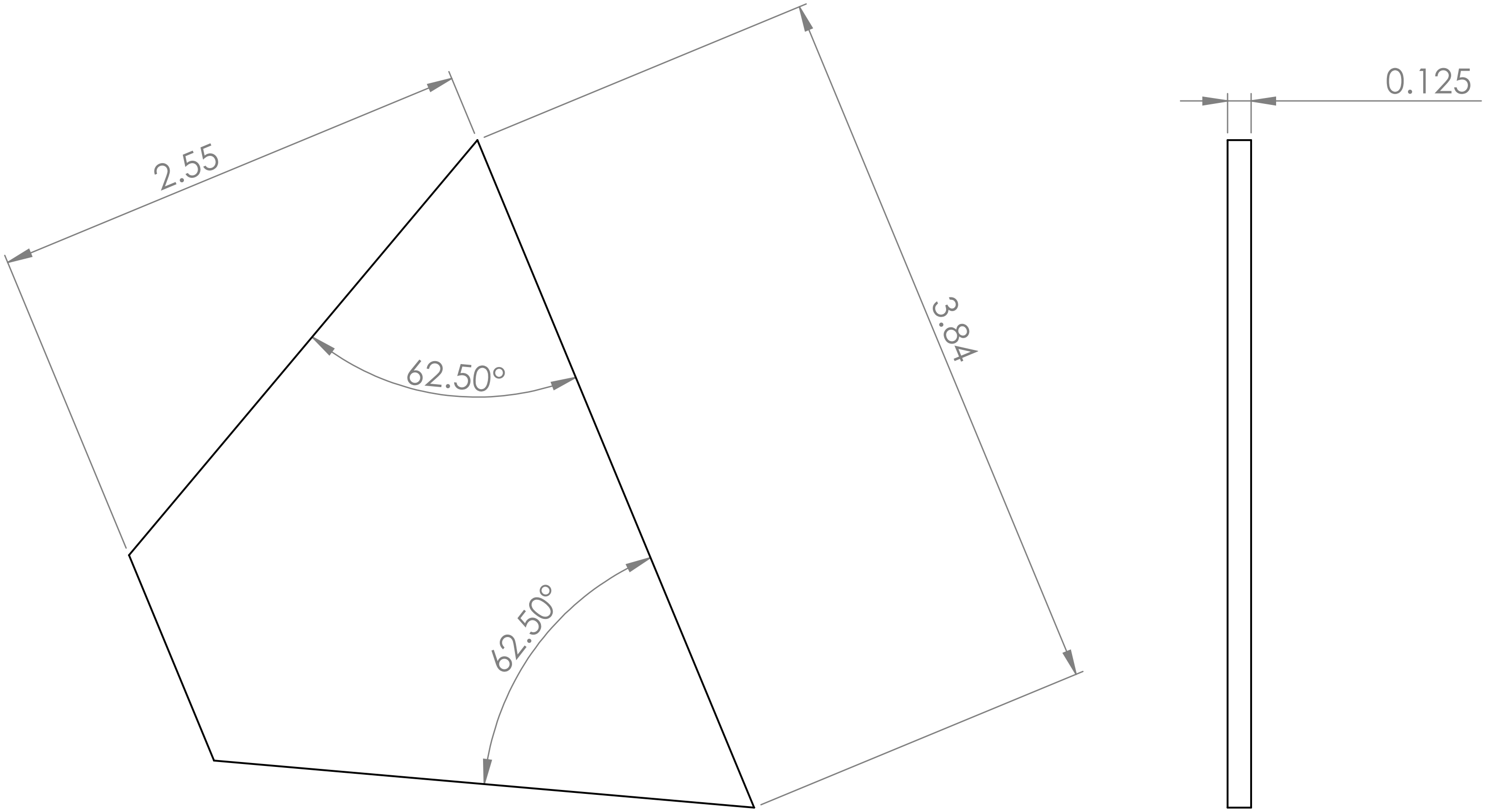
TITLE:

Bottom Outer Gusset

SIZE A | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN			Bottom Inside Gusset
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL 1/8" Steel Stock	COMMENTS:			SIZE A · DWG. NO. · REV
FINISH As Plasma Cut				
DO NOT SCALE DRAWING				SCALE: 1:1 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1

THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL 1/8" Steel Stock	COMMENTS:		
FINISH As Machined			
DO NOT SCALE DRAWING			

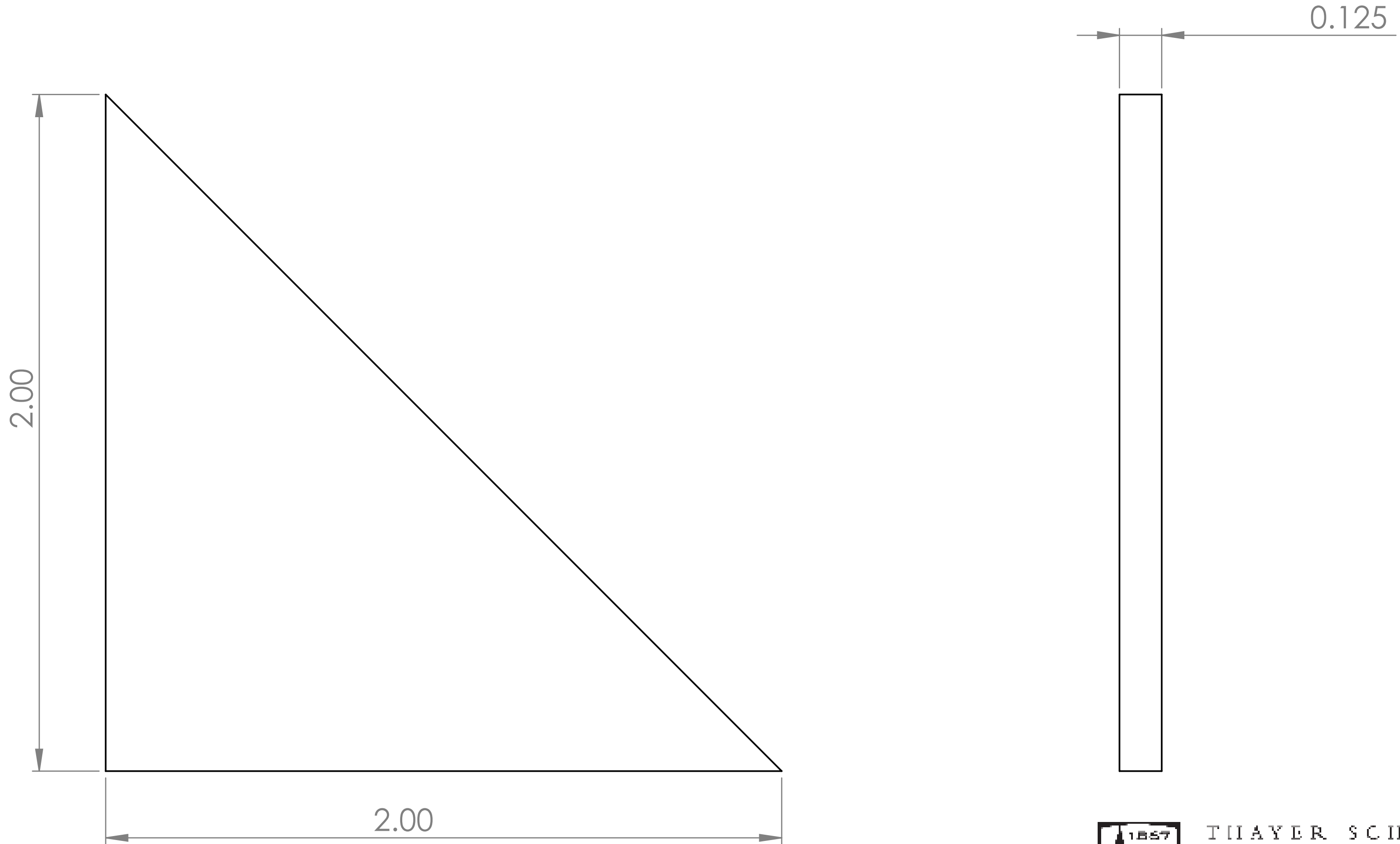
TITLE:

Back Frame Gusset

SIZE	DWG. NO.	REV
A		

SCALE: 1:1	WEIGHT:	SHEET 1 OF 1

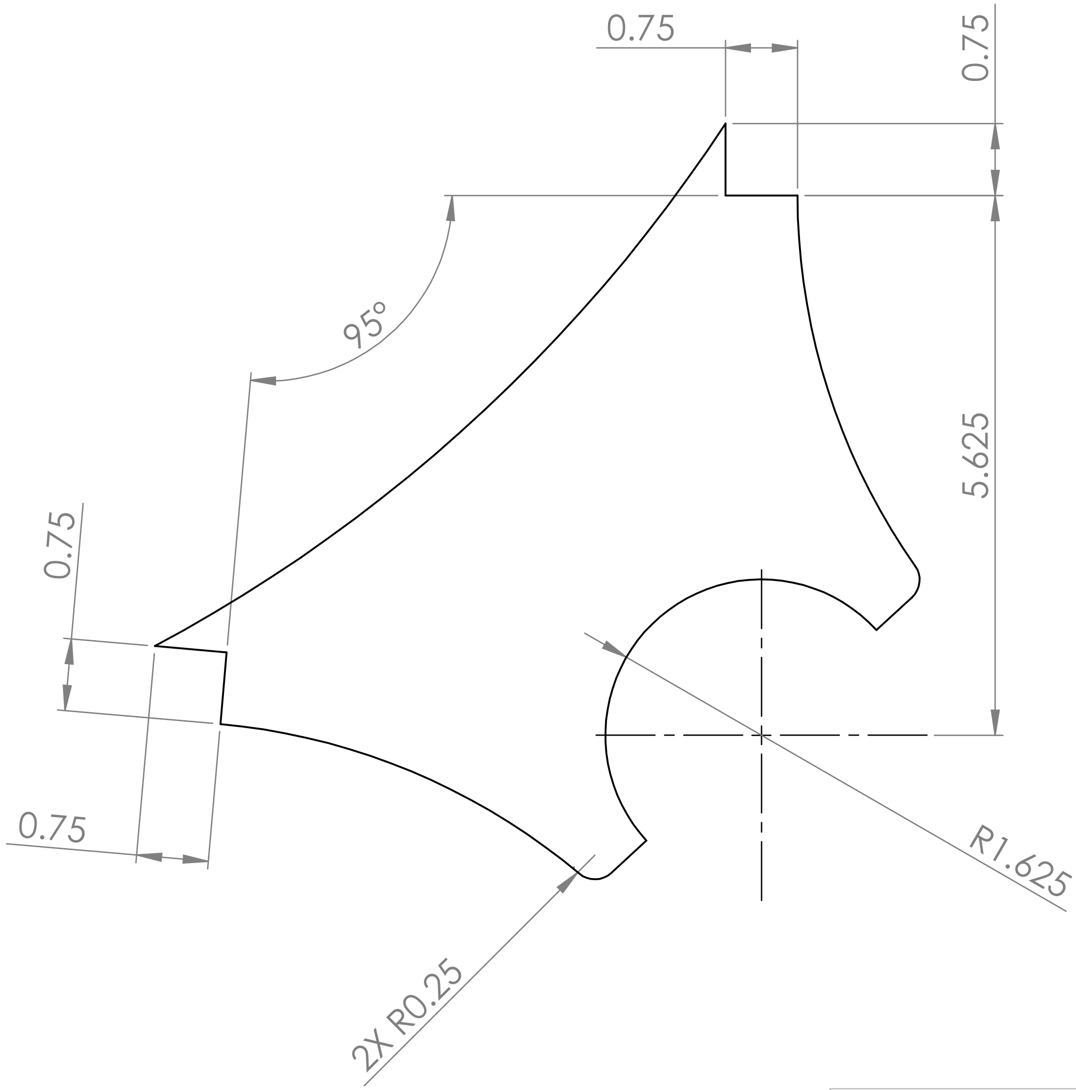
5 4 3 2 1

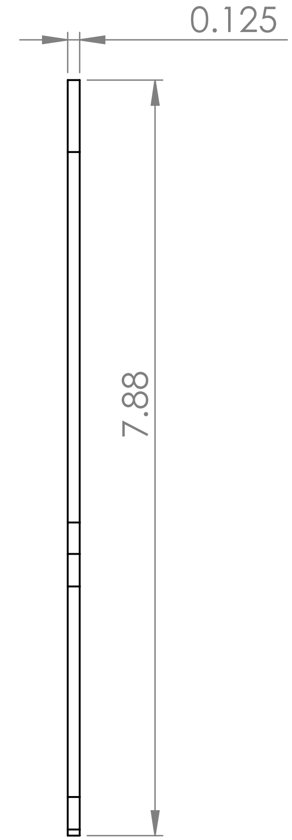


THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN			Seat Back Gusset
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL 1/8" Steel Stock	COMMENTS:			SIZE A	DWG. NO.	REV
FINISH As Plasma Cut				
DO NOT SCALE DRAWING				SCALE: 2:1	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1





THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL 1/8" Steel Stock	COMMENTS:		
FINISH As Plasma Cut			
DO NOT SCALE DRAWING			

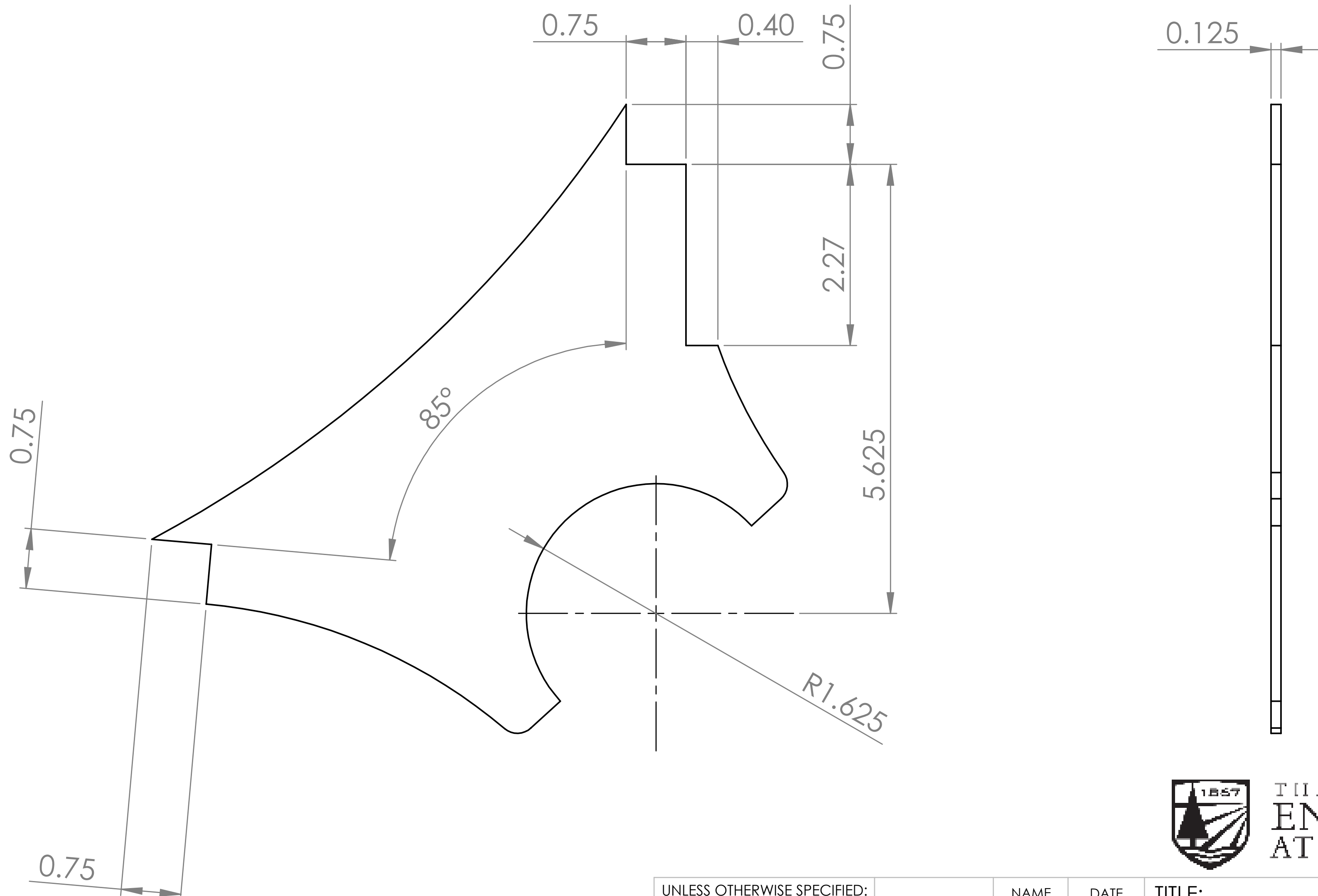
TITLE:

Drive Shaft Outer Gusset

SIZE	DWG. NO.	REV
A		

SCALE: 1:2	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

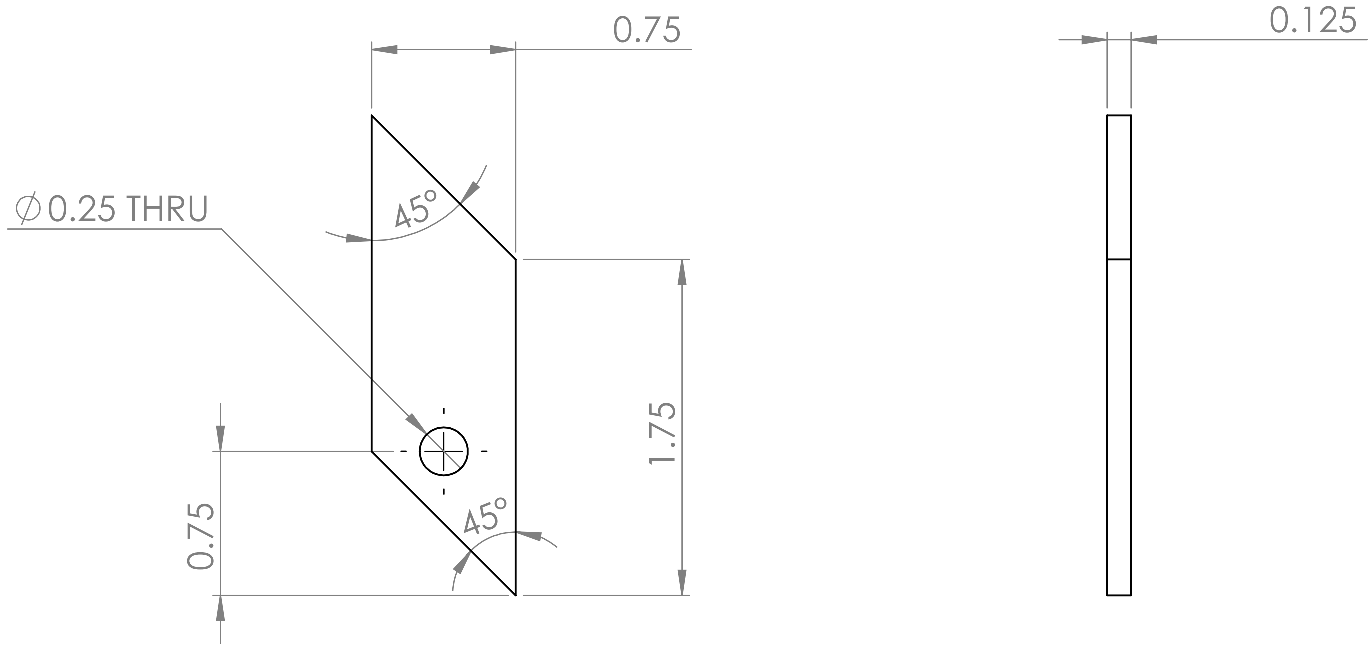
UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL 1/8" Steel Stock	COMMENTS:		
FINISH As Plasma Cut			
DO NOT SCALE DRAWING			

TITLE:

Drive Shift Inner Gusset

SIZE	DWG. NO.	REV
A		

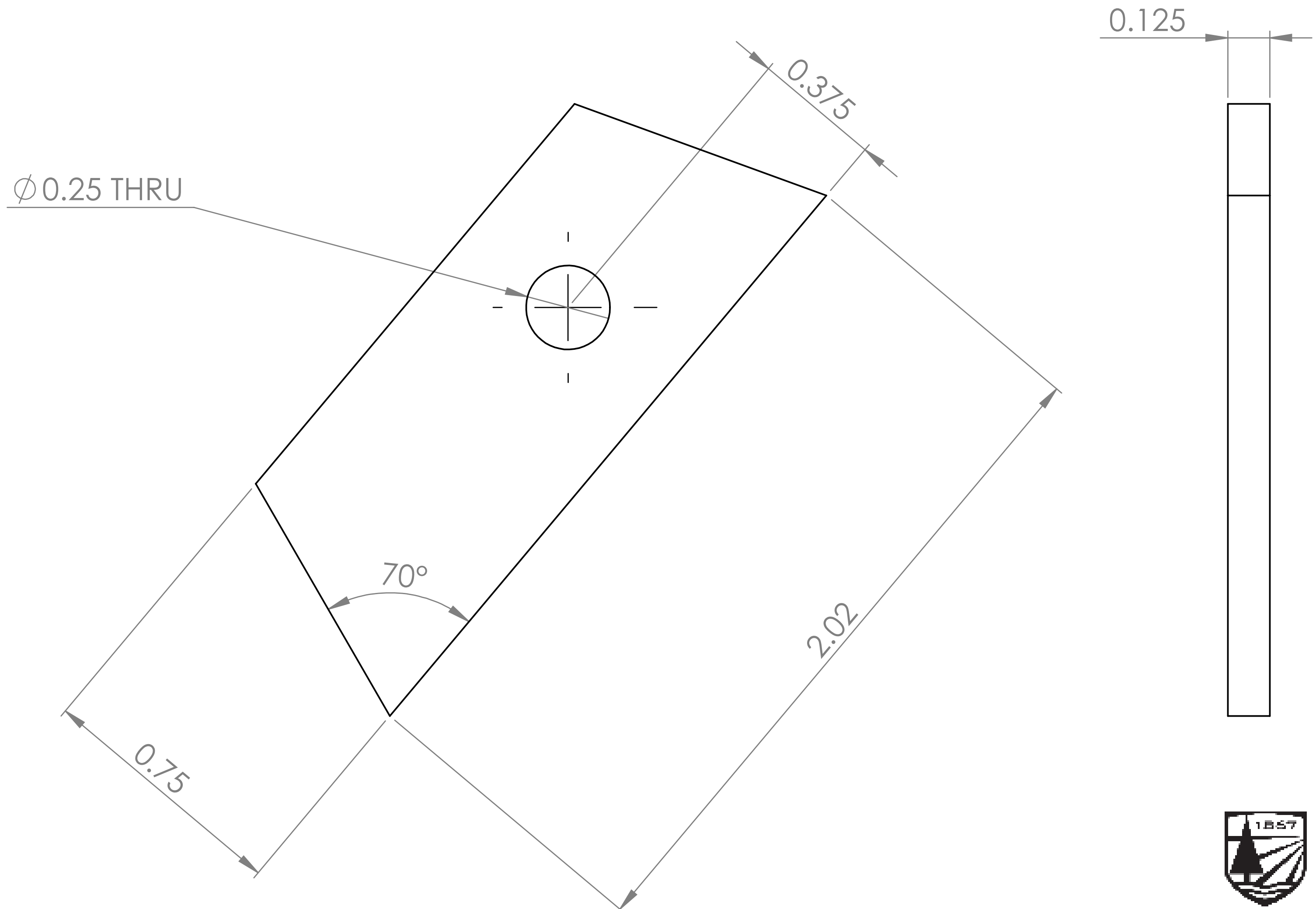
SCALE: 1:2	WEIGHT:	SHEET 1 OF 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN			Spring Attachment
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL 1/8" Steel Stock	COMMENTS:			SIZE A / DWG. NO. / REV
FINISH As Plasma Cut				
DO NOT SCALE DRAWING				SCALE: 1:1 / WEIGHT: / SHEET 1 OF 1

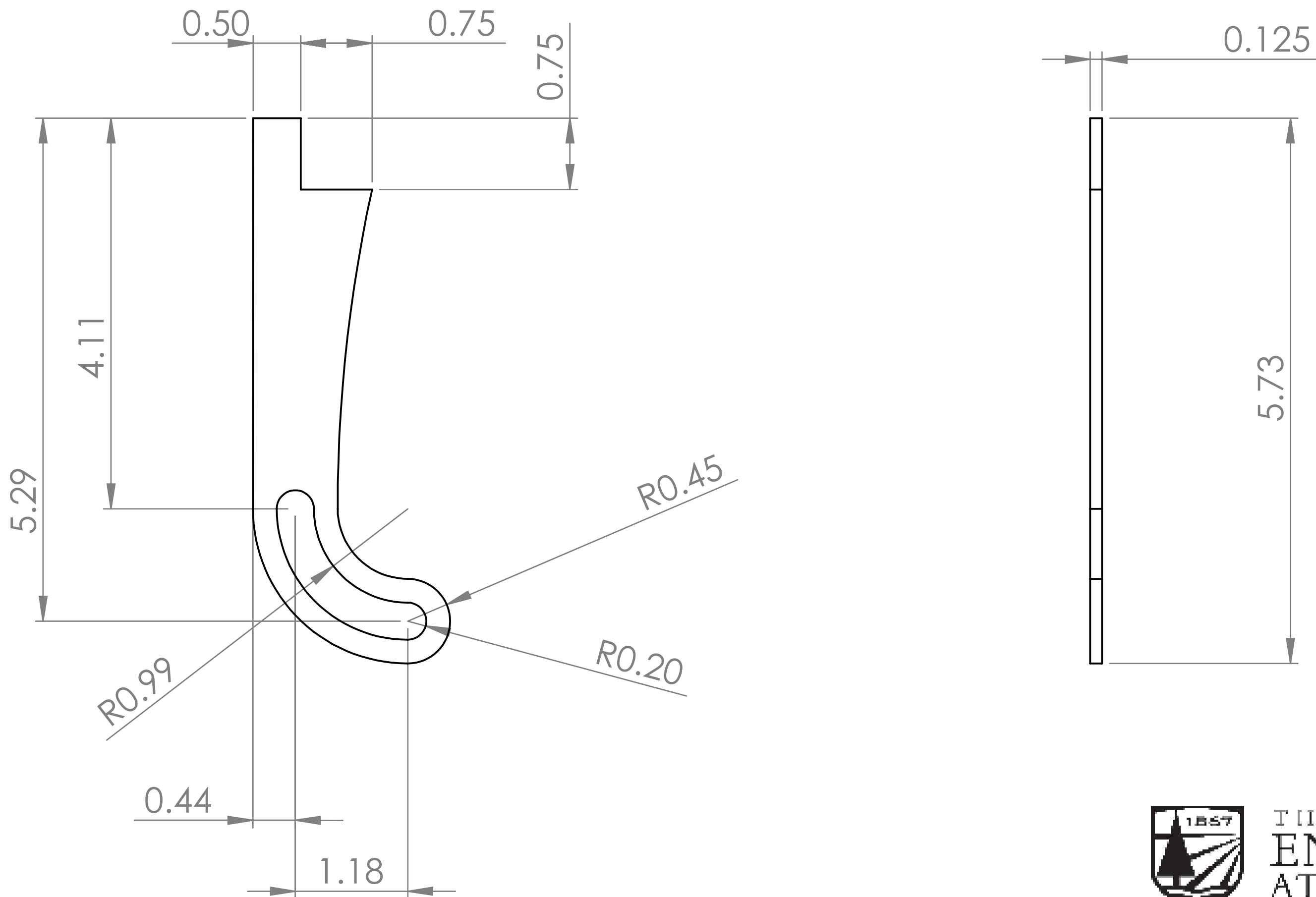
5 4 3 2 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN			Spring Lower Attachment
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL 1/8" Steel Stock	COMMENTS:			SIZE A	DWG. NO.	REV
FINISH As Plasma Cut				
DO NOT SCALE DRAWING				SCALE: 2:1	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1

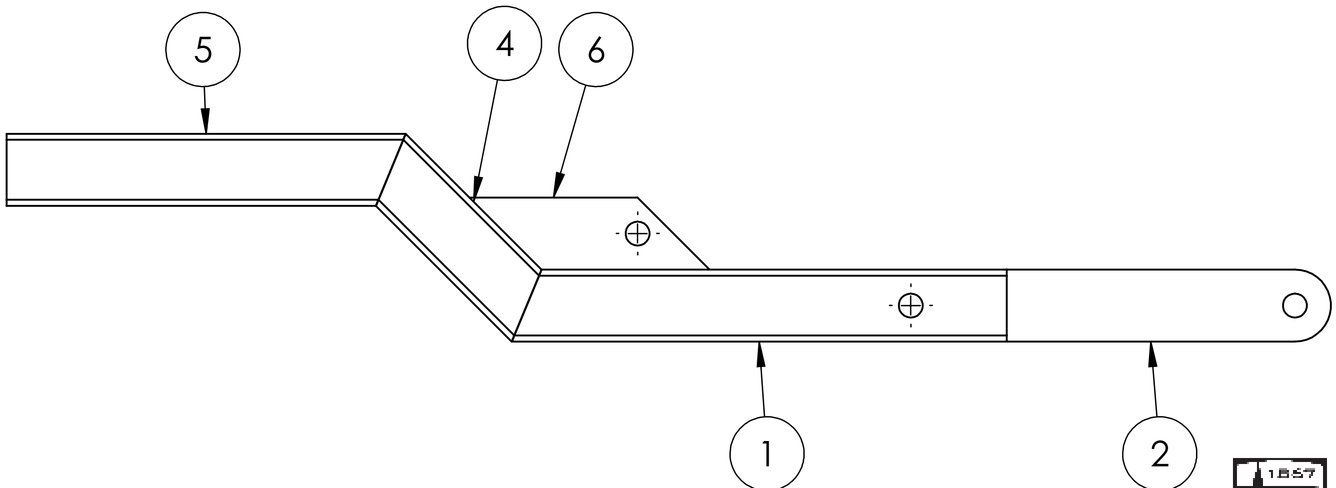


THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN			Derailleur Mount
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL 1/8" Steel Stock	COMMENTS:			SIZE A · DWG. NO. · REV
FINISH As Plasma Cut				
DO NOT SCALE DRAWING				SCALE: 1:1 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1

ITEM NO.	QTY.	DESCRIPTION	LENGTH
1	1		6.16
2	1	Wheel Connector Gusset	
3	1	Wheel Connector Gusset (Not Shown)	
4	1		2.31
5	1		4.16
6	1	Spring Connector Gusset	



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

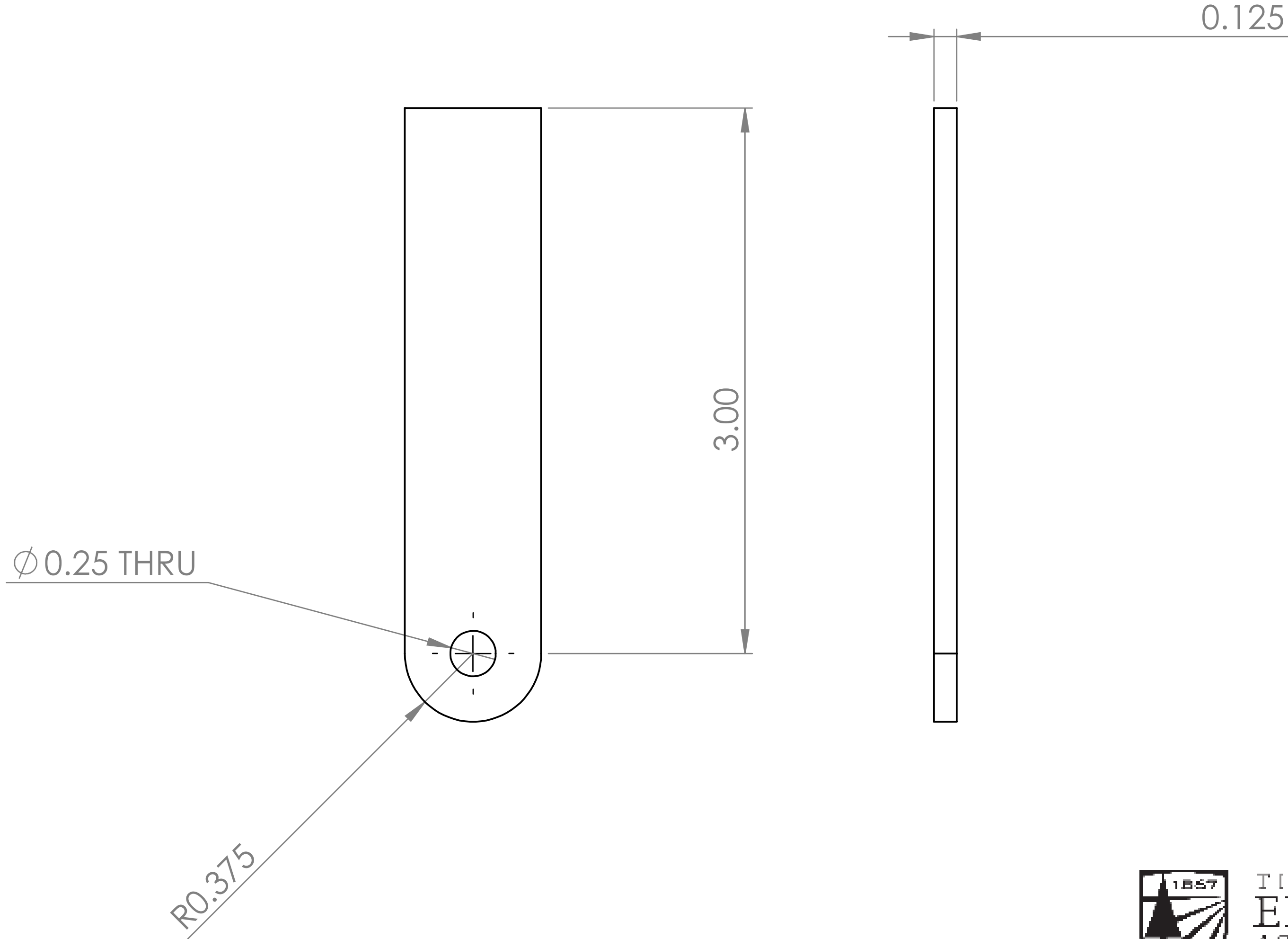
TITLE:

Spring Lever Arm

SIZE A | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:

ANGULAR: MACH ± 30'
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.003

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
1/8" Steel Stock

FINISH
As Plasma Cut

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

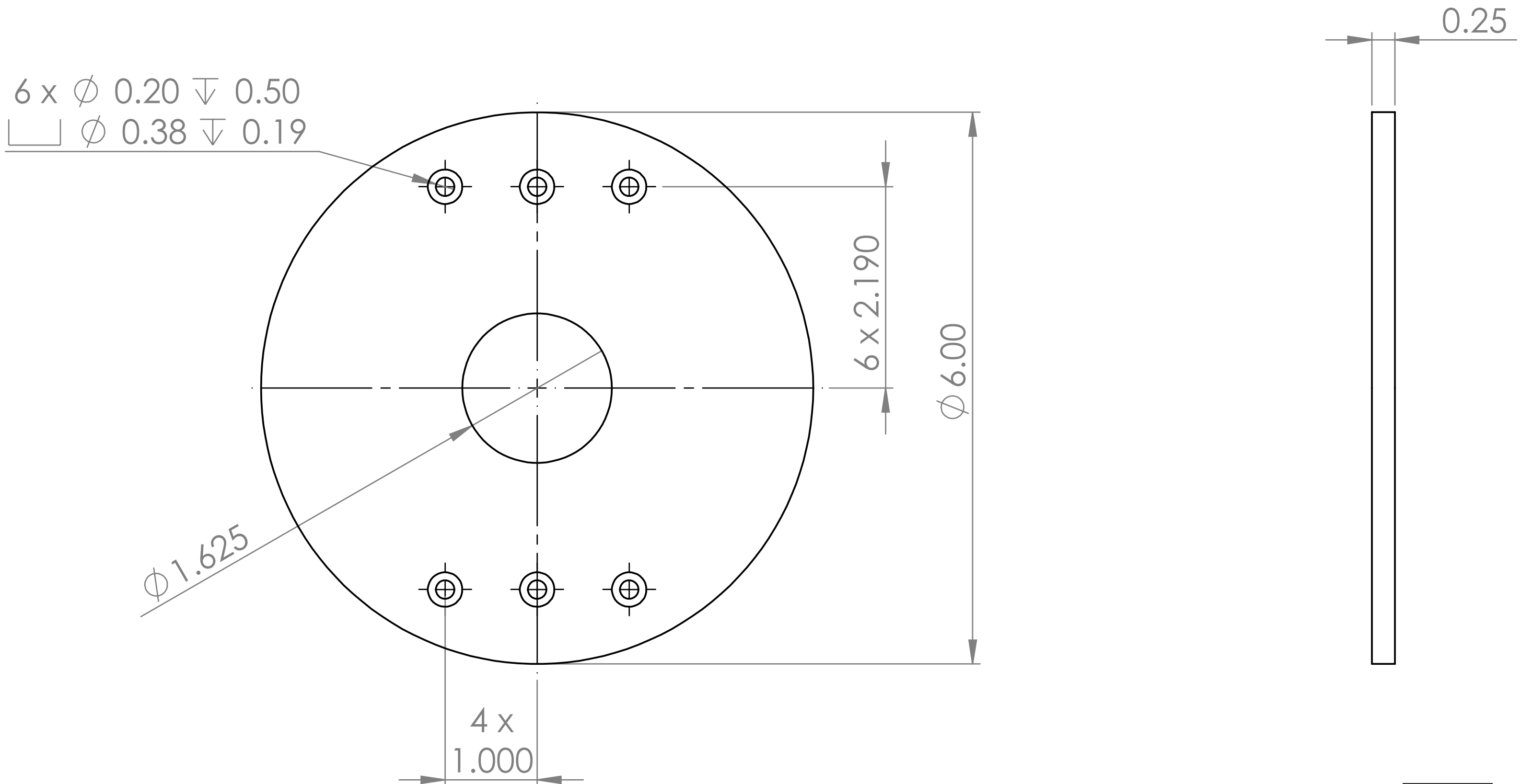
TITLE:

Lever Arm Gusset

SIZE A | DWG. NO. | REV

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1

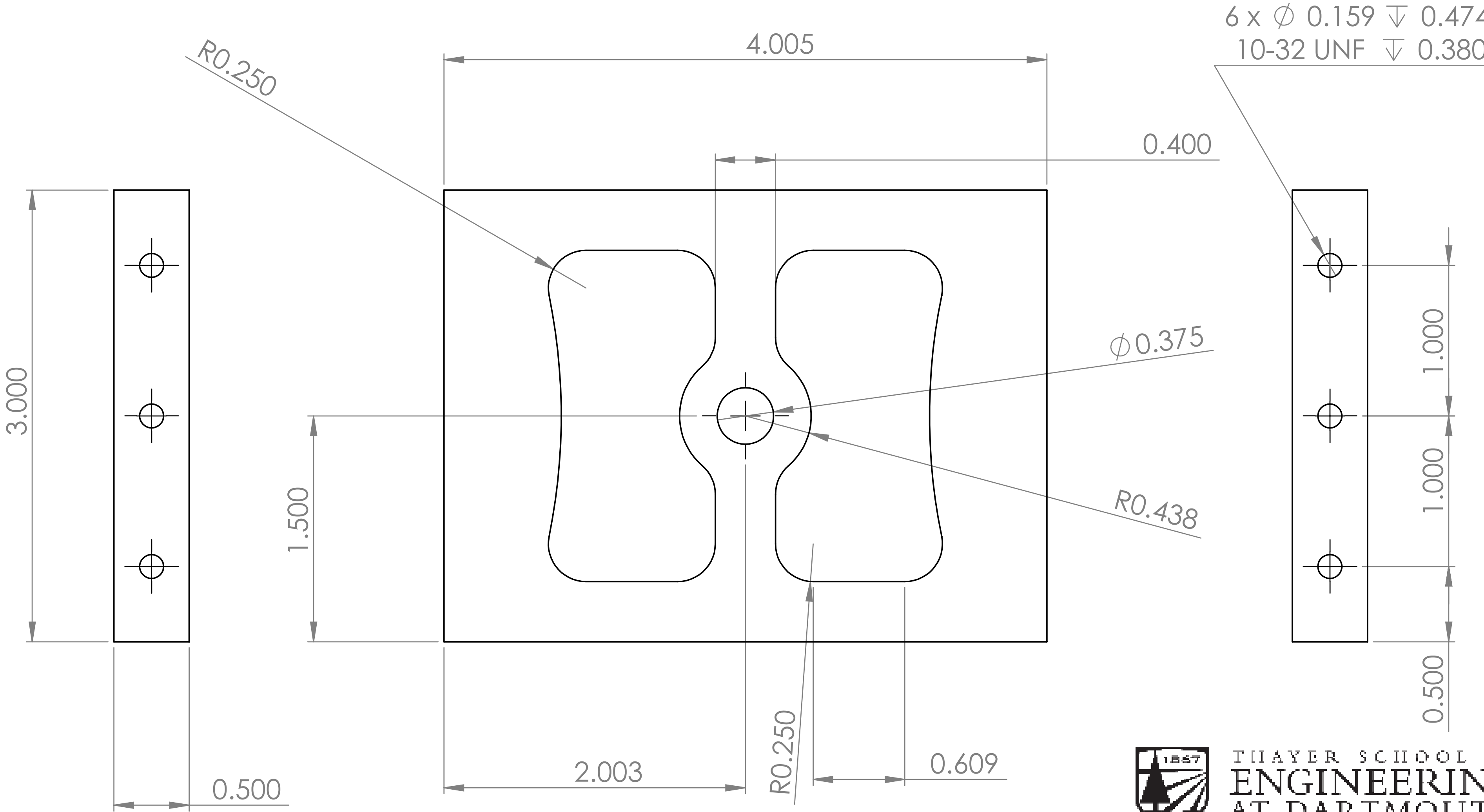
5 4 3 2 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN			Diff Case Siding
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL 1/4" Steel Stock	COMMENTS:			SIZE A · DWG. NO. · REV
FINISH As Machined				
DO NOT SCALE DRAWING				SCALE: 1:2 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL 1/2" Steel Stock	COMMENTS: Inner "speed hole" cuts made later using CAM; cuts down weight significantly.		
FINISH As Machined			
DO NOT SCALE DRAWING			

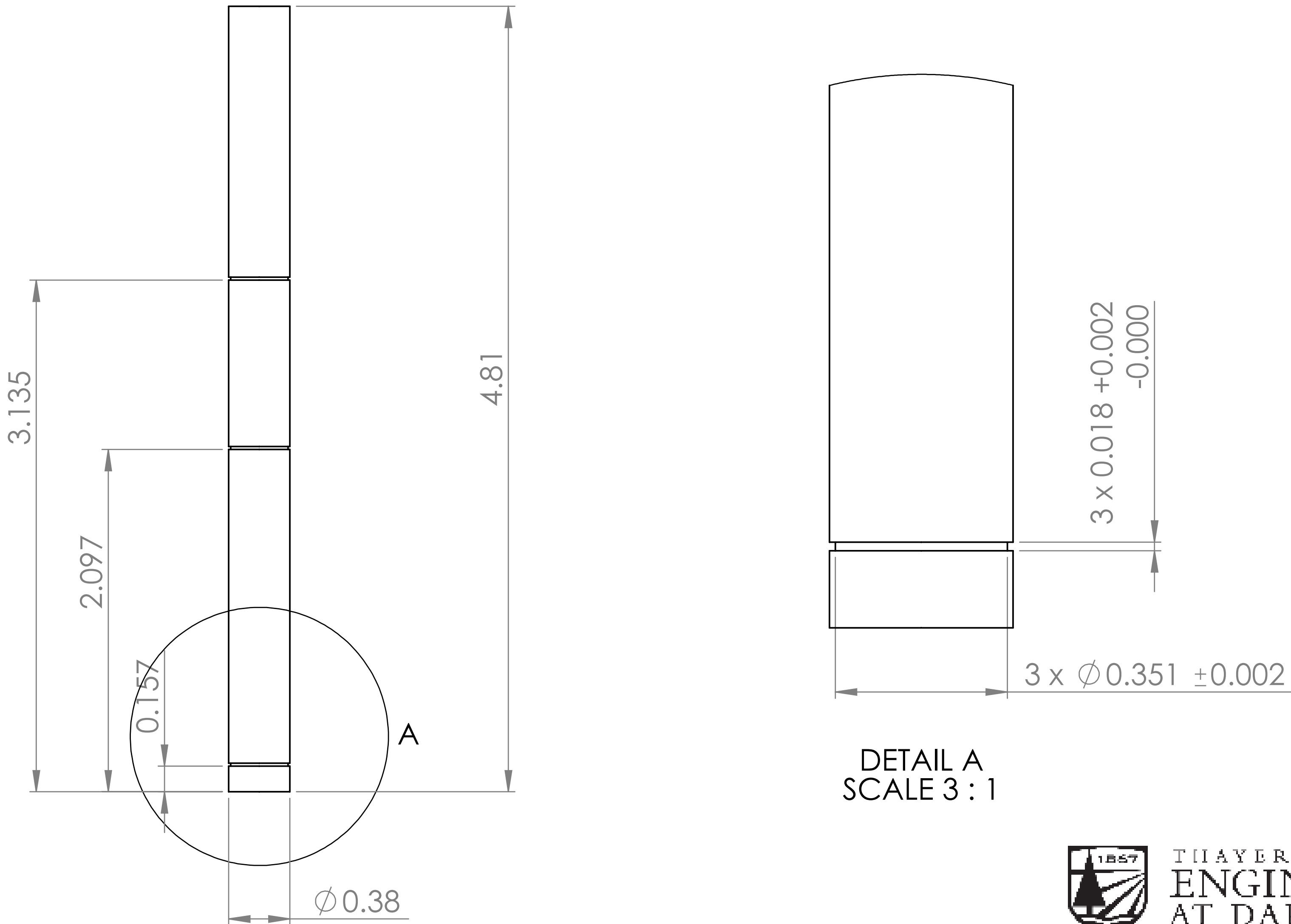
TITLE:

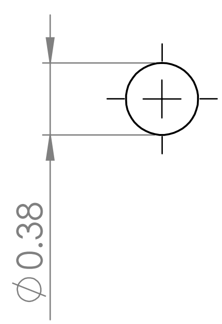
Case Spacers

SIZE	DWG. NO.	REV
A		

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

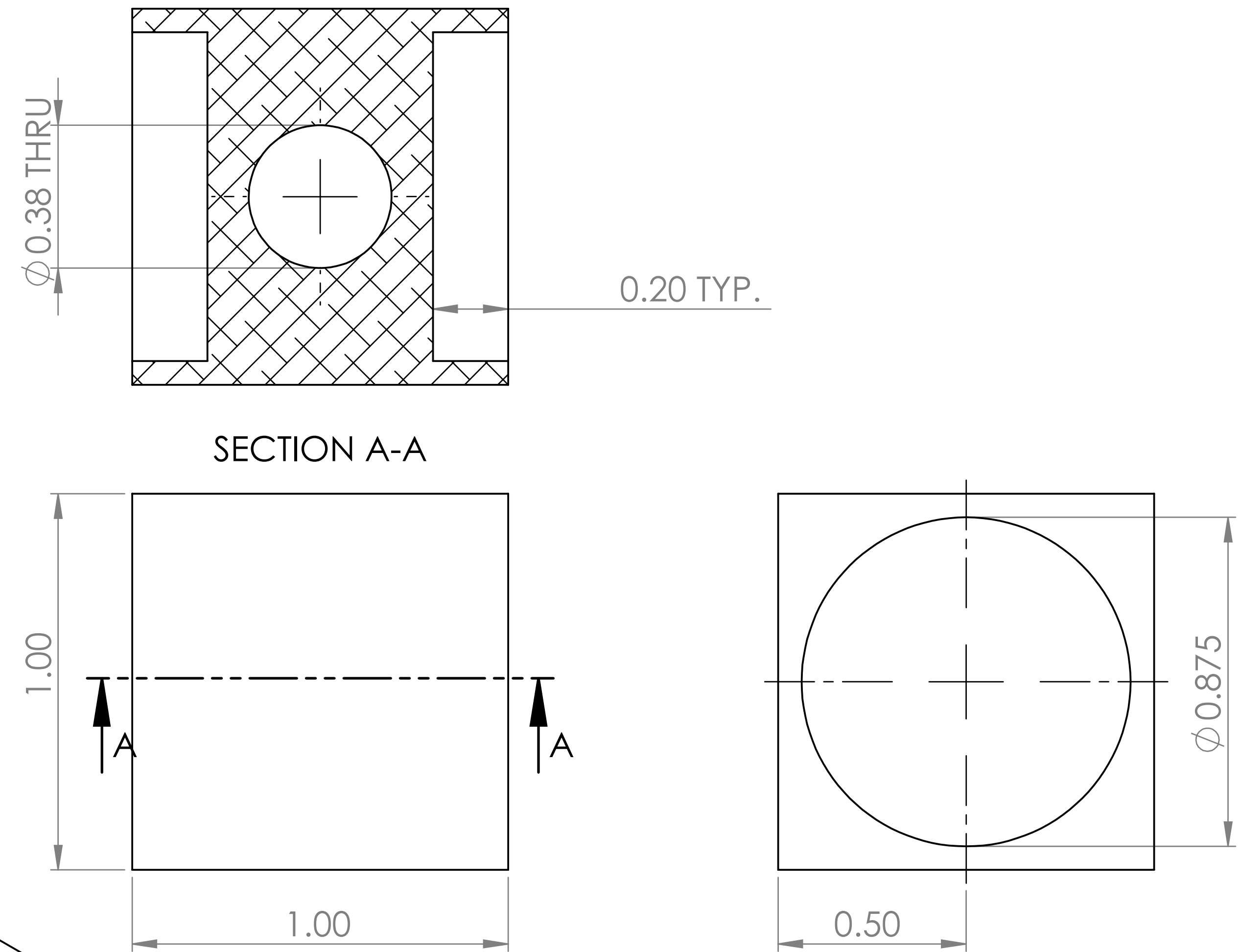


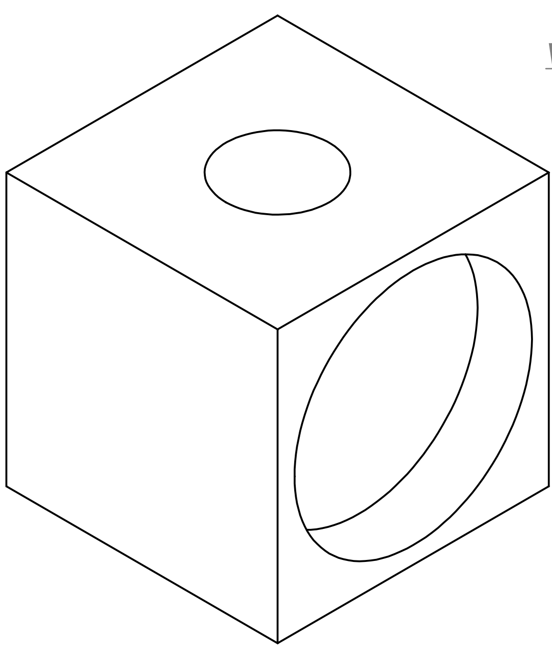


THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN			Spider Shaft
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL 3/8" Steel Shaft	COMMENTS:			SIZE A — DWG. NO. — REV
FINISH As Machined				
DO NOT SCALE DRAWING				SCALE: 1:1 — WEIGHT: — SHEET 1 OF 1

5 4 3 2 1

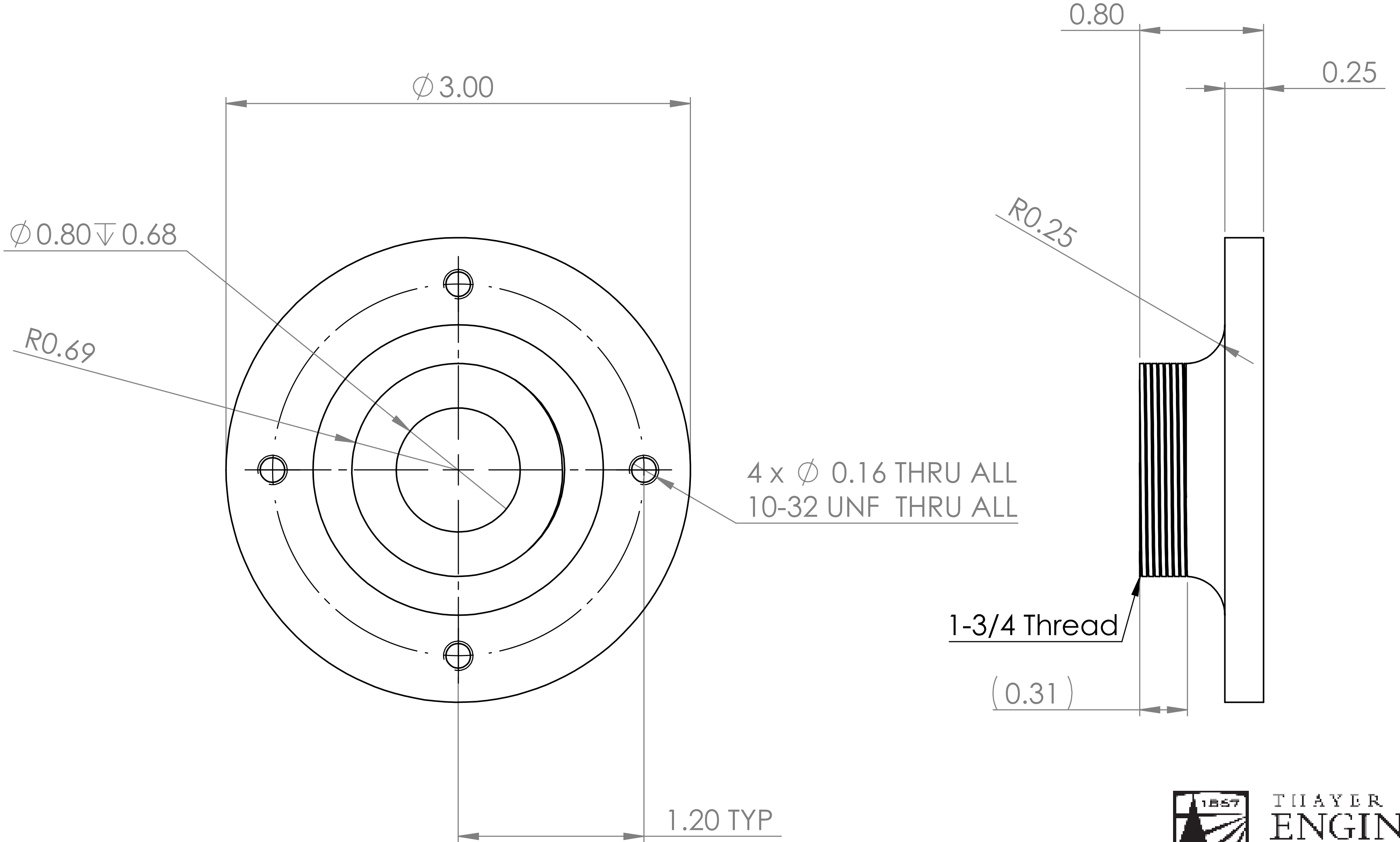




THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN			Spider Block
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL 6061 Aluminum	COMMENTS:			SIZE A	DWG. NO.	REV
FINISH As Machined				
DO NOT SCALE DRAWING				SCALE: 2:1	WEIGHT:	SHEET 1 OF 1

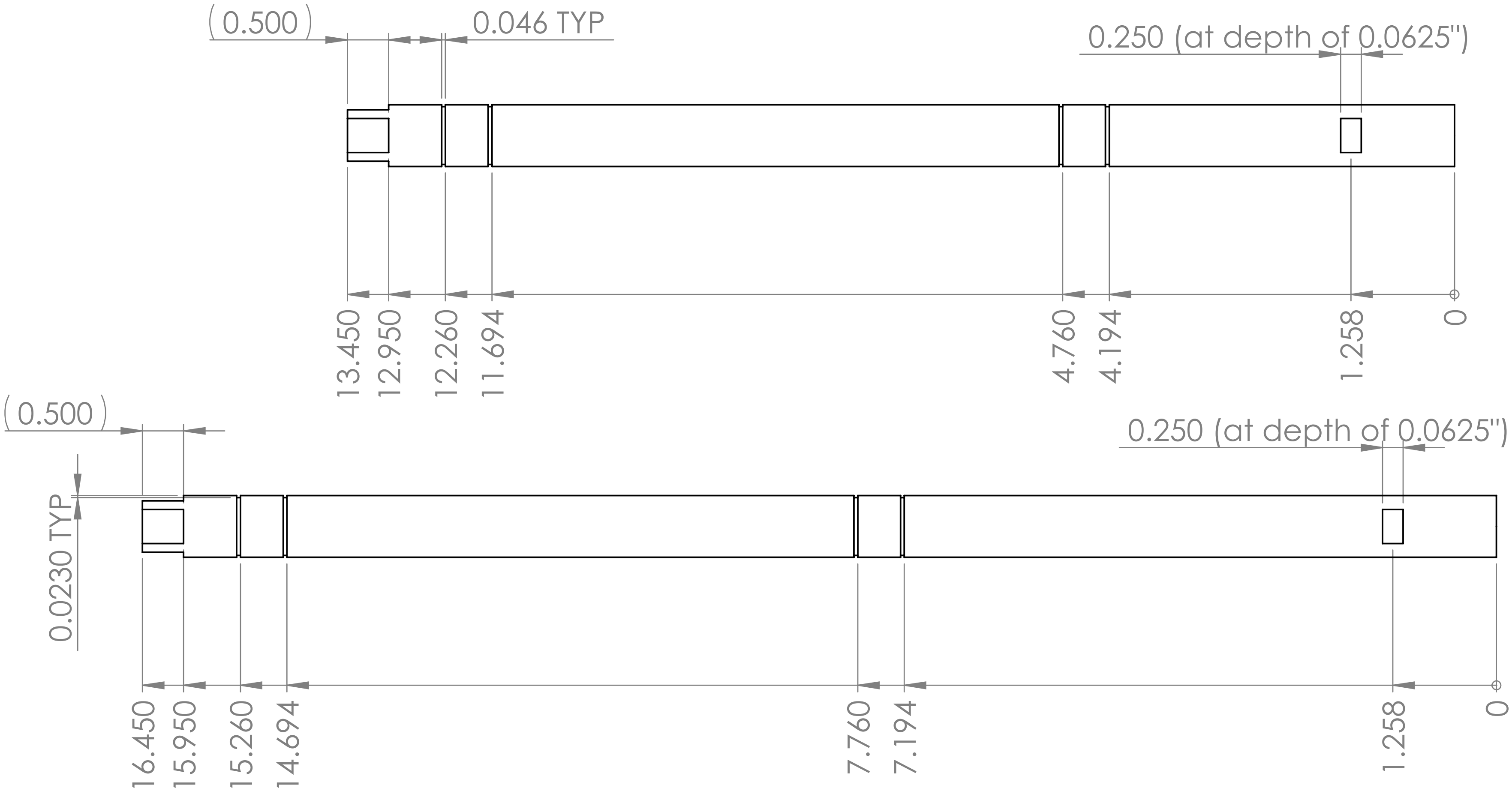
5 4 3 2 1

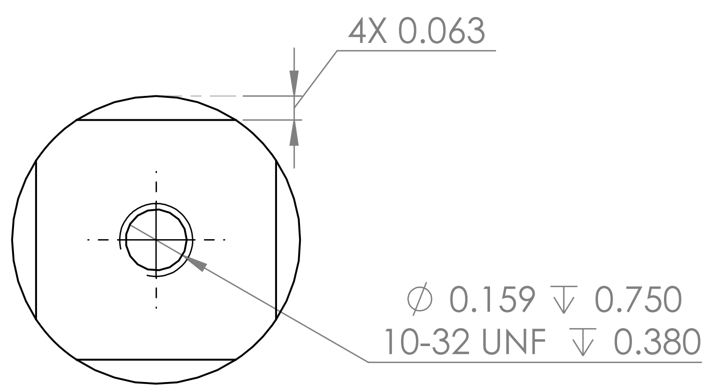


THAYER SCHOOL OF
ENGINEERING
AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN			Free Wheel Connector
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL 6061 Aluminum	COMMENTS:			SIZE A · DWG. NO. · REV
FINISH As Machined				
DO NOT SCALE DRAWING				SCALE: 1:1 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1



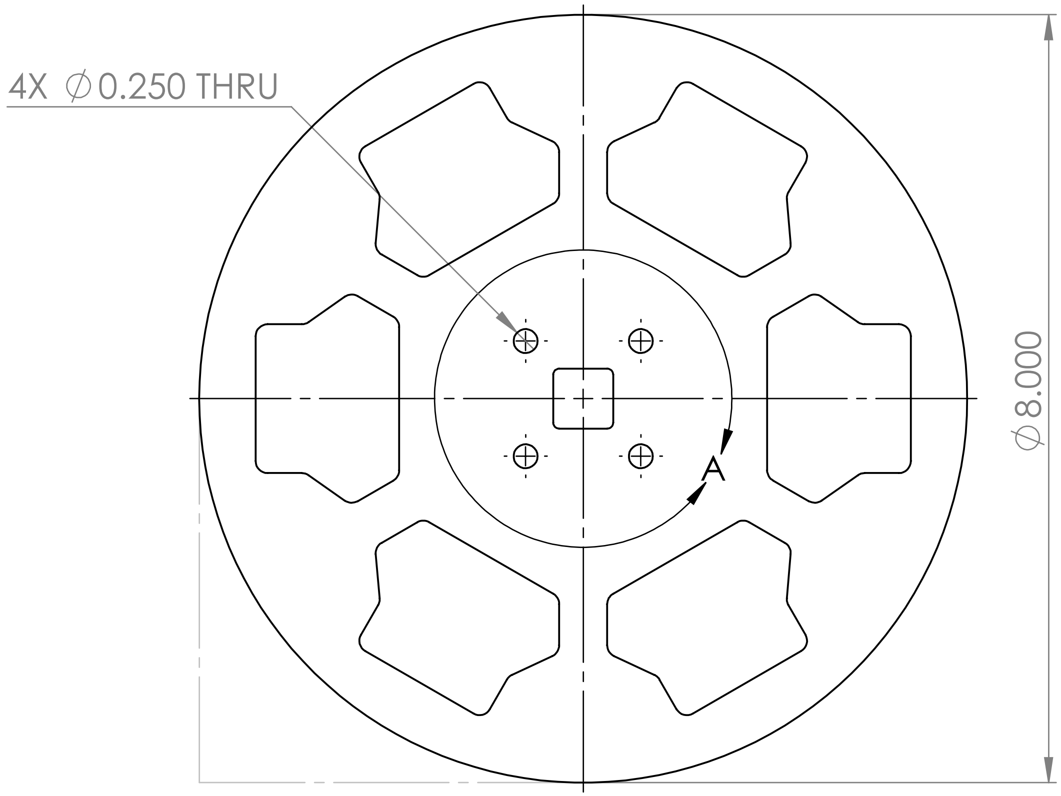


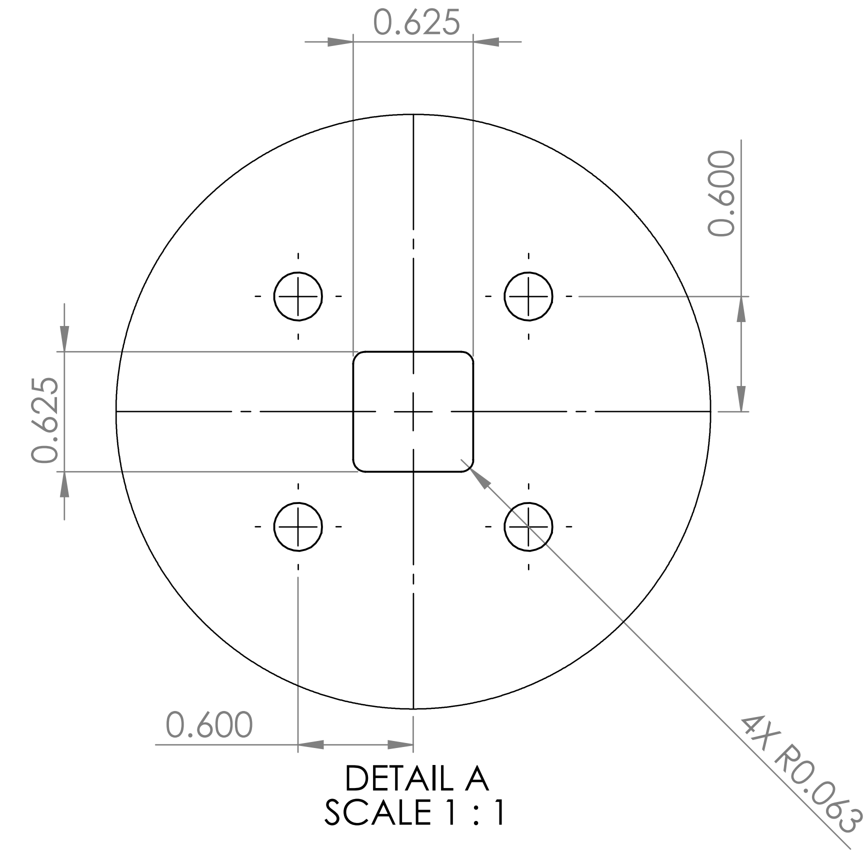
THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL 3/4" Steel Shaft	COMMENTS:		
FINISH As Machined			
DO NOT SCALE DRAWING			

TITLE: Drive Shaft		
SIZE A	DWG. NO.	REV
SCALE: 1:2	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1



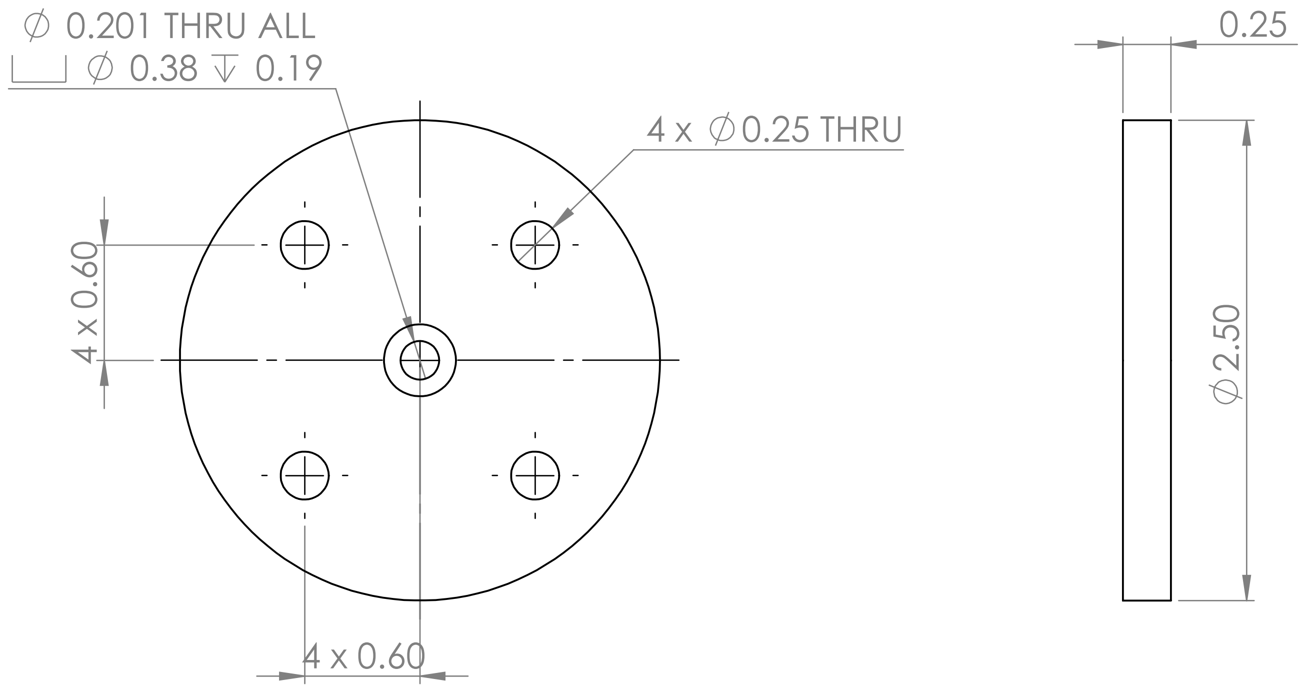


DETAIL A
SCALE 1 : 1

THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN			Drive Wheel
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL 1/2" 6061 Aluminum	COMMENTS: Relied heavily on CAM due to the complex bevel shapes.			SIZE A	DWG. NO.	REV
FINISH As Machined				
DO NOT SCALE DRAWING				SCALE: 1:5	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1



THAYER SCHOOL OF ENGINEERING AT DARTMOUTH

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ± 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.003	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL 1/4" 6061 Aluminum	COMMENTS:		
FINISH As Machined			
DO NOT SCALE DRAWING			

TITLE:

Hub Cap

SIZE A | DWG. NO. | REV

SCALE: 2:1 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1