Formula SAE Powertrain
Phase 4: Performance Validation and Path Forward

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Advisor:
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December 10th 2010

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Introduction:

Team FSAE Powertrain has made tremendous strides in the final phase of the Senior Design Process. As this phase, and the semester, comes to an end, our team has complete manufacturing of the majority of components and hammered out a very clear path forward for future work in all project aspects.

Our group's work on the powertrain for the 2010-11 FSAE car was divided into 3 major project aspects, with some additional breakdowns under them.

1. Drivetrain
    a. Differential
    b. Axles
    c. Hubs
2. Air System
    a. Air Intake
    b. Exhaust
3. Cooling

The wiring harness for the 2010-11 car will be taken on by William "Jay" Kistler as an Honors project and will be completed early in winter session 2011.

As we manufactured our way through phase 4 our list of tangible deliverables which will be completed by the end of the semester along with the report, poster, presentation, and other various paperwork required by the MEEG401 syllabus, had to change slightly. The final list is as follows…

1. 1 limited slip differential, complete with custom housing and mounting fixtures ready for placement in the 2010-11 car
2. 2 axles ready to be installed in 2010-11 car
3. 2 rear drive hubs ready for interface with axles, suspension components, brake components, and wheels, ready to be installed in 2010-11 car
4. 2 aluminum spacers ready for use in the rear hub assemblies of the 2010-11 car
5. 1 prototyped air intake with expandable plenum and runner tube that will assist in the design of the final 2010-11 air intake and can be saved for design work in future years
6. Air filter, throttle body, fuel injector, rubber gasket, throttle position sensor, and mass flow sensor to accompany the future air intake on the 2010-11 car
7. 1 exhaust maintube, including O_2 sensor, ready for installation between the head and muffler of the 2010-11 car
8. 1 radiator for the 2010-11 car
9. 1 radiator pull fan for the 2010-11 car
10. Plans for 2 radiator ducts—one leading up to the radiator and connecting the radiator to the puller fan to be given to the Chassis team for incorporation into their sidepod design

The following report will be broken down into the three main aspects of our project. All other powertrain componentry and systems other than those touched on in the introduction and elaborated on throughout the report are out of the scope of our project.

Basic Project Criteria:

There are a few constraints/wants/metrics that apply to all of the aspects of our project and are therefore discussed in detail in the majority of the following report sections in such a way that is pertinent to the system being discussed. This basic criteria as well as more specific project requirements were assembled from interactions with our costumer, Mr. Paul Schwarz, our advisor, Dr. Steve Timmins, and current veteran SAE club members. Much information was also gathered from the FSAE rule book to help set the bounds of our project.

Cost Constraint: The cost of the whole project may not exceed $1,000. *See the Budget section near the end of this report for breakdown.*

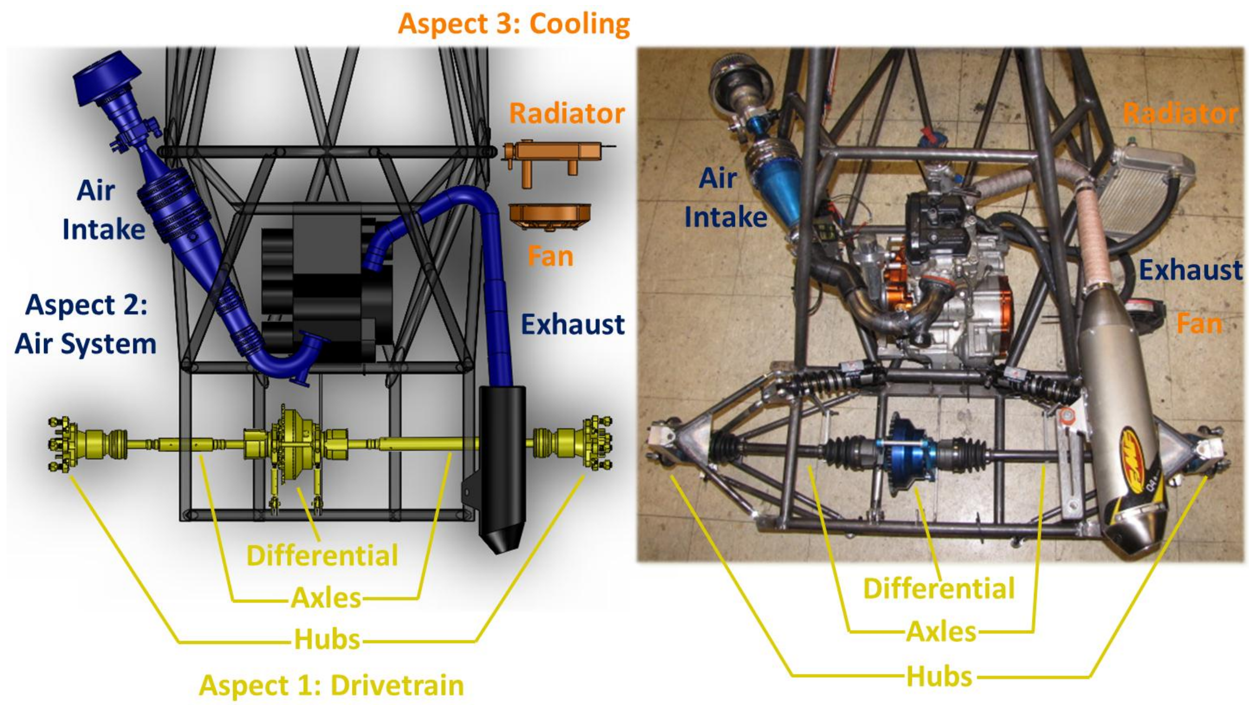
Weight Want: Weight reduction is a want that applies to every aspect of our project. This reduction allows the vehicle to retain its maximum power to weight ratio.

Durability / Reliability Want: It is imperative that we develop robust designs that will allow the 2010-11 car to finish competition and serve as a baseline for future years optimization.

Aesthetics/Neatness/Workmanship Want: Finally our customer, Paul Schwarz, strongly expressed that all system attributed to our team be neat and well manufactured. Many previous UD SAE cars were very hastily assembled only weeks (or even days) before competition. This resulted in sloppy assemblies and a negative aesthetic quality.

Performance: While the focus this year was on creating durable designs there was a desire to have everything perform well to allow the SAE club to not only finish, but excel at competition.

Semester Success:



As shown above by the comparison of *Solidworks* to what now exists in real life the semesters efforts were largely successful. The only deliverable not installed in the new car are the wheel hubs, which still need to be splined and heat treated by a professional company. They are shown separately above.

The remainder of the report is broken down into our 3 project aspects.

Project Aspect 1: Drivetrain (Differential, Axles, and Hubs)

The first aspect of our project is the drivetrain. The drivetrain in any vehicle is responsible for taking the power/torque produced by the engine and transmitting it to the wheels to propel the car. In the 2010-11 FSAE race car this will be accomplished by transferring torque from the engine to the differential assembly (Fig 1 Yellow) via a chain and sprocket (not shown). Power will be transmitted from the differential out through the axles (Fig 1 Blue) and into the hub assembly (Fig 1 Green) which will turn the rear driving wheels of the vehicle.

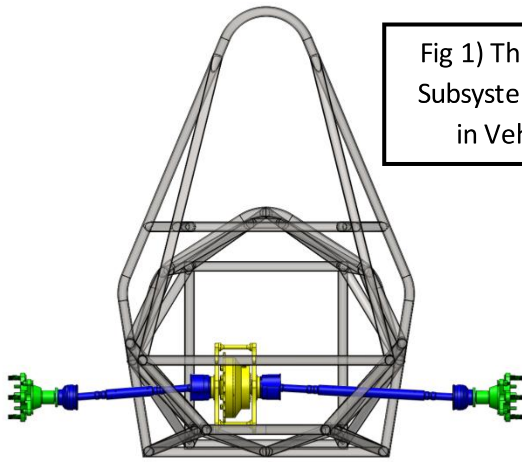
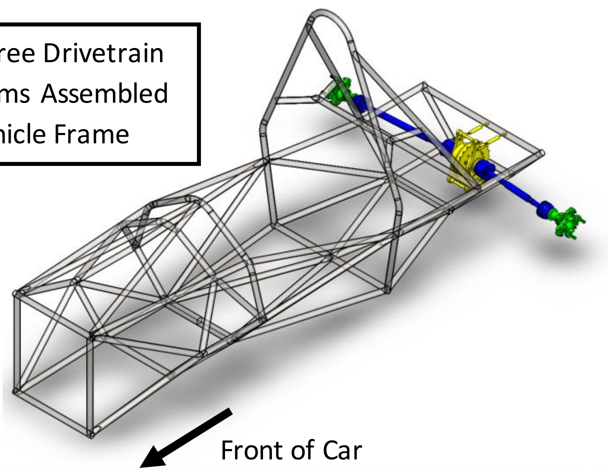
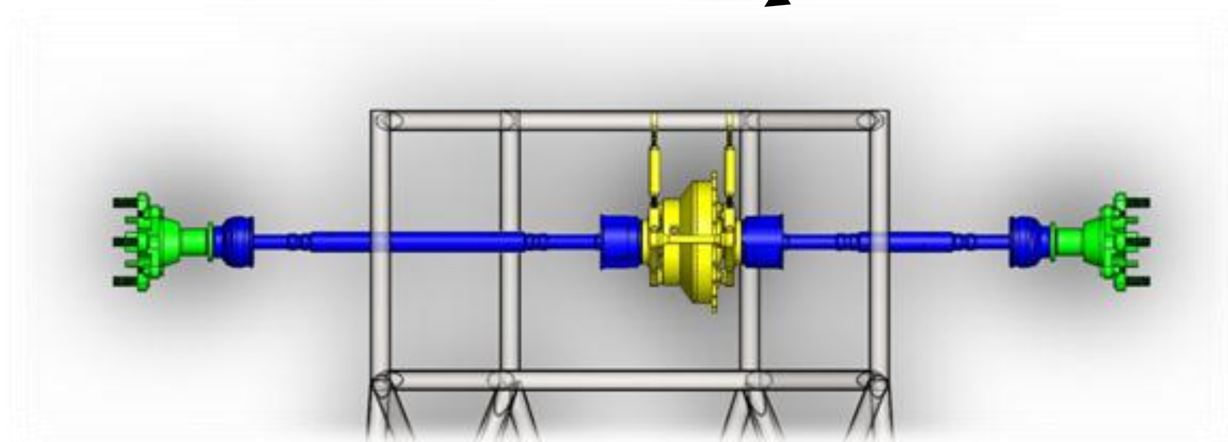
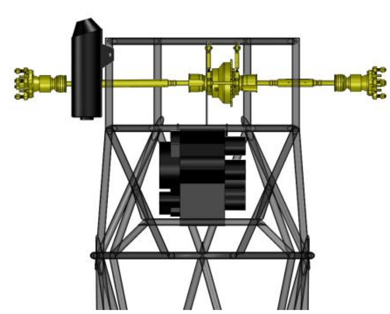


Fig 1) Three Drivetrain Subsystems Assembled in Vehicle Frame







It is of the utmost importance that these three systems be as lightweight as possible in order to maintain the vehicles maximum power to weight ratio, as strong/durable as possible to prevent failure, and as stiff/rigid as possible to transfer the engine's torque to the wheels with minimum energy losses. The reminder of the *Project Aspect 1: Drivetrain* section is organized in these three subsystems moving from the interior or the vehicle out—differential, axles, hubs.

Differential:

Housing and Upright Design

Differential Selection

Before the housing and uprights were designed a limited slip differential was chosen. To replace the torsen differential in the old car, a clutch type differential was chosen. The clutch type's smoother operation leads to better handling overall so it is an upgrade from the current system. The differential was chosen from a Honda TRX ATV because the ATV has a comparable power to weight ratio to the SAE car.

Constraints

-Must be of limited slip type
-Cost must fit within $1000 project budget

Metrics

Based off improving upon last year's design

Cost: Differential cost less than $600
Weight*: Total weight less than 7lbs
Size (between uprights): Less than 10 inches
Durability (number of seals for housing case): Less than 4 seals

**Weight includes components of the differential and housing assembly, excluded are the uprights and sprocket which are comparable weight in both systems.*

Comparison

	Old	New	Savings
Cost ($)	628*	534*	94 (14%)
Weight (lbs)	7.75	6.10	1.65 (21%)
Size (inches)	10.25	4.96	5.29 (51%)
Durability	Leaks	TBD	---

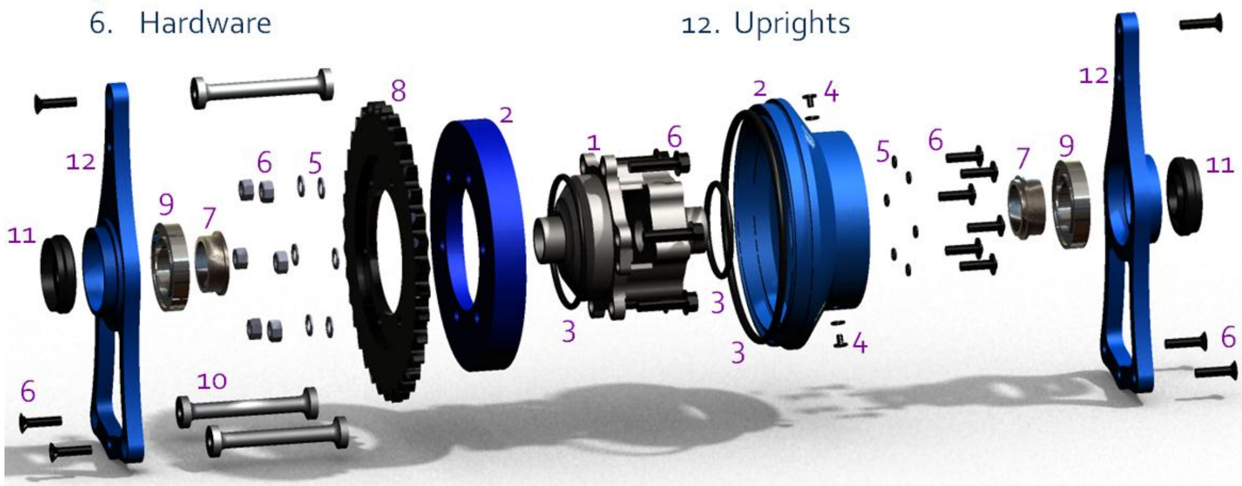
**Cost of material was normalized using McMaster Carr to find prices of old material vs. new material. Actual material for new differential was purchased locally with significant savings. Hardware is assumed to be of similar cost for both differentials so it is neglected from the cost comparisons.*

Final Design Considerations

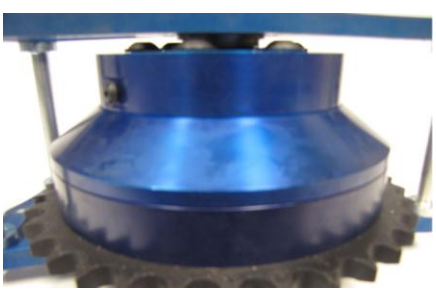
Geometric Constraints

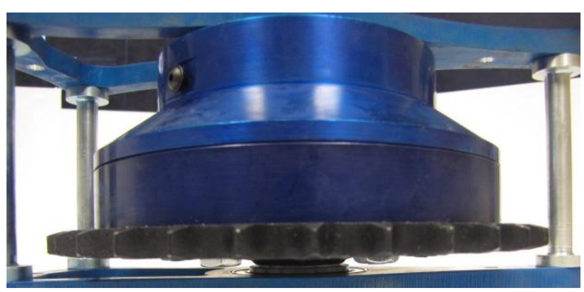
One of the biggest concerns when building the differential housing is the use of stock mounting locations and geometry. The exploded view below shows how the components line up with each other such that the actual Honda differential did not have to be modified.

1. Honda TRX Limited Slip Differential
2. Differential Housing
3. Housing O-ring Seals
4. Drain Bolt and Seal
5. Hardware Seals
6. Hardware
7. Press Fit Adapter
8. Sprocket
9. Bearing
10. Upright Connections
11. Axle Seals
12. Uprights



The biggest downfall to the geometric constraints is the inability to easily disassemble the differential housing. As previously stated the space between the uprights has to be so small because the axle stub shafts are very short and have to be snapped into the differential while being seated against the axle seals. As seen in the cross sectional view of the differential, there is no room for further axle seals to be added between the upright bearing and the carrier. Since axle seals could not be added in this space, a simple press fit is used to seal between the bearing and carrier. An adapter piece fits the 29mm outer diameter of the carrier while also press fitting into the 30 mm inner diameter of the bearing. Below are pictures of the actual assembly showing how close the clearances are (both sides being about 0.15 inches).





Modification to Design

Machine Shop Advisement

A great asset to the successful build of the differential assembly is greatly due to the help of the mechanic in charge of the mechanical engineering machine shop, Mr. Beard. While working though each drawing to help best come up with a method of properly manufacturing each part, Mr. Beard advised several changes to the design of the differential housing. Most of these changes were done to either make assembly or manufacturing easier, but in the end simplified the part leading to a more elegant design overall.

The first change suggested by Mr. Beard was the decreased radius of the taper on the upright connections. The radius decreases stress concentrations to some degree, but is mostly aesthetic with little validation to use the specified radius. In the machine shop was a bit that could be used on the lathe that had a generic radius built in. This tool still provides a radius and would be much more consistent than attempting to fillet the part by hand so it was used instead of what was specified in the drawing.

The sprocket side of the housing had features that could be revised to ease machining with little to no loss in the mechanics of the housing. Initially there was a lip on this half of the housing on both the inside and the outside of the interface with the non-sprocket side. After some discussion it was decided that solely the outer lip could locate the non-sprocket side of the housing. With this realization the inner lip neither improved the ability to assemble the housing nor did it provide any benefit to sealing. Also since the lip would be difficult to machine due to tool size restrictions in the shop it was decided that the removal of the inner lip would only decrease weight of the overall assembly.

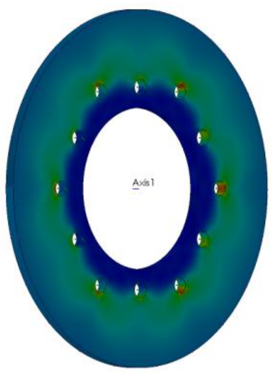
The next thing up for discussion was the fitment of the sprocket and non-sprocket sides of the housing. Since a face sealed o-ring is the sealing feature at the interface of both halves of the housing, having the two aluminum pieces meet face to face at the lip interface could be detrimental. If the aluminum faces met it would restrict the compression of the o-ring which could go unnoticed after assembly. Since this lack of compression would result in failure to hold a seal at the interface it was decided to make the housing with a small gap between the two housing halves. An added benefit to this decision is the removal of the small groove which was initially a feature for a screw driver to pry the casing apart. With the small gap between halves of the housing, this feature was removed from the part.

Lastly a final feature was removed from the sprocket side of the differential housing. The initial plan included aluminum tabs that protruded through the sprocket. The idea was to provide further support to the sprocket, such that the bolts would not have to support as much of the load in the case of the friction between the sprocket and housing was not enough to prevent slippage. Although some analysis proved that the bolts would have significantly more strength than necessary to support the possible impulse torque, the feature remained on the drawing to be machined. After some discussion this feature proved to be very difficult to machine and it was suggested that dowels be pressed fit into the face of the sprocket half of the housing rather than a direct protrusion. Seeing as dowels would have little to no reasonable effect in the case of loading, this feature was removed from the part.

Final Testing and Validation

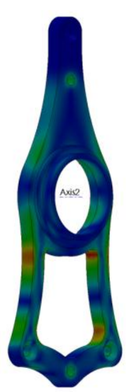
Although final testing that can be done before the differential is actually being run in the car is less than optimal, there is little physical validation that can be done until the car is up and running. Two things that are possible to check at this stage include upright integrity and low pressure sealing.

First a simplified sprocket was created to represent the case of improper tightening of the nuts. In this case the bolts and tabs coming through the sprocket are under a load equal to the reaction due to an impulse. Each bolt in this simulation experiences 1,478.1psi which is extremely insignificant considering the bolts are rated at a minimum of 150,000 psi. The actual load will be withstood by the frictional force between the bolt and nut that sandwich the assembly. Due to this sandwich effect the assembly acts as if it was one piece from the applied toque through to the carrier. In this way we were able to effectively apply the torque directly from the sprocket to the carrier.



1,478.1 (psi)	
1,231.9	
1,108.8	
862.6	
739.5	
493.4	
370.3	
247.2	
0.1	

Next the upright was tested for the case of a bearing seizing at the same time as an impulse occurred. This is, like the previous case, another highly unlikely scenario to prove the strength of the uprights. The aluminum with a yield strength of 35,000psi can easily withstand this worst case scenario with a safety factor greater than three.



17,700.4 (psi)	
14,225.4	
13,275.3	
10,325.3	
8,850.2	
7,375.2	
5,900.2	
2,950.1	
0.1	

Next a simple spin test was performed on the differential assembly. With a theoretical volume of 375ml, 100ml of gear oil was poured into the housing. The housing was then sealed back up and spun 50 times. Spinning the differential helped ensure complete coverage of the gear oil along the internal surface of the housing. No leaks were found as a result of this test. A picture of the setup can be seen below.

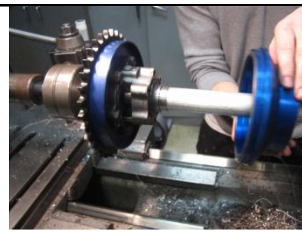
As previously stated, true driven tests are the only way to really prove the integrity of the housing. This is addressed in future suggestions on how to test and deal with any issues that may come up.

Disassembly

Due to the geometric constraints mentioned earlier the differential is not easily disassembled. Fortunately as with most differential assemblies like the one in this application, there are snap rings that make inserting the axles into the differential relatively permanent anyway. If however the differential must be disassembled this outline of guidelines will help decrease risk of injury or excessive damage to the assembly.

First it is important to note the locations at which the assembly can be pulled apart. For a visual of these areas, look at the press fits labeled 1c and 2c in the differential image showing at sealing features. Note that only the non-sprocket side must be removed for access to the carrier internals. A tool known as a puller can be used to safely spate the press fit parts from their otherwise permanent locations. To use the puller a simple extension should be machined such that when the jaws pull on the upright the center of the puller applies force to the carrier or adapter.

Removal of any housing bolts or nuts other than the upright bolts and drain plug will result in an expansion of the differential unless counteracted with some clamping force. This clamping for can be effectively established using lathe and a small adapter that fits into the end of the differential past the bearing.

		
Step 1: Removal of hardware, after clamping the housing in the lathe	Step 2: Removal of non-sprocket side of housing	Step 3: Insertion of maintenance bolts and unclamping

After clamping the differential in the lathe disassembly becomes pretty straight forward. In step three it is important not to forget to re-insert the maintenance bolts (refer to differential assembly step 1 for clarification.

Differential the Path Forward

Having manufactured the differential housing with little time to spare along with the car being a couple months away from completion, final testing will be done in the future. Seeing as the differential is basically one piece once the axles are snapped into the carrier, failed tests will have to have reasonable solutions that do not require disassembly of the system. The testable metric of the differential is not leaking. This is a yes or no metric. There are several places that could cause a failure to pass this metric. Although this housing was obviously designed to prevent leakage from all of these areas, by realizing them as areas of possible failure the amount of engineering that went into this system can be validated.

Since leaking is something that can occur randomly compared to the failure of other systems, it is best tested under real conditions. With the strong possibility of a long period of testing time before competition, there should be enough run time on the differential by competition to prove its success. Of all the areas of competition that the differential could fail, the endurance section would be the area of the greatest possibility of failure. Since the differential will be under high stress for a run time of 45 minutes, any small leakage could grow into a serious problem by the end of the race. By running tests on the car under similar conditions we will be able to test for leakage in the differential.

Careful consideration went into every possible location where oil could leak from. In theory all of these predetermined problem areas should be properly sealed, however, failure of any of these seals cannot be ignored. Below is a section view showing each of the design features that will prevent leakage. They are broken up into three categories where category “a” designates seal interfaces, “b” designates o-ring interfaces, and “c” shows press fit interfaces.

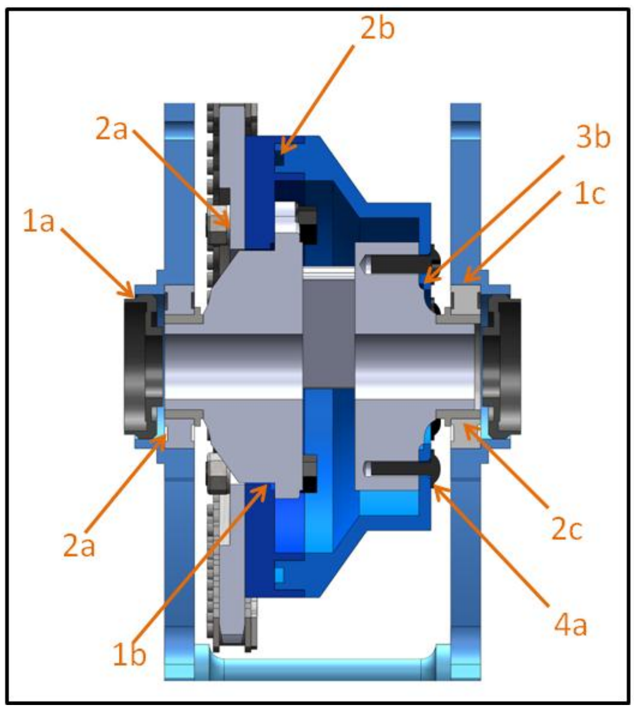
-1a: Axle seal/Upright Interface
-2a: Pressure sealing washer/Nut/Sprocket
-3a: Inner and outer bearing seals
-1b: Sprocket side housing/carrier o-ring interface
-2b: Sprocket side/ non-sprocket side housing o-ring interface
-3b: Non-sprocket side housing/carrier o-ring interface
-1c: Bearing inner diameter/adapter press fits or outer diameter/upright press fits
-2c: Adapter/carrier or adapter/ bearing press fits (same as bearing/adapter press fit mentioned in 1c)

Knowing these areas for failure gives some basis for how to repair these possible leak points. Since the differential will be used in future years its good for the creators to give the users feedback on how they could fix these potential problems.

1a: The axle seal was purchased with a package of components that can be purchased on eBay for a Honda TRX differential. If there is a leak along the outer edge of the seal than gasket sealant can be applied to the upright/axle seal interface. If the leak is coming from the axle/gasket interface, then a new seal should be purchased. If there is an urgent need for replacement, there are two seals off of the purchased Honda TRX differential that can be used as a short term substitute.

2a: The pressure sealing washer was used as a precaution to prevent any oil from escaping through the clearance holes for the M8 bolts that hold the sprocket and sprocket side housing to the carrier. There are several spares of these. If switching the washers is needed, it is essential that at least two of the opposing M8 be left tightened at any one moment. If all nuts are released, the carrier will expand.

3a: This is a location that is partially covered and sees much less oil contact than most of the other interfaces. If leakage occurs here the bearing seal may be damaged. Over filling the carrier could increase the chance of leakage in this area since it would increase the amount of contact the bearing seals have with the oil.



1b: Out of all of the o-ring seals, this one has the greatest chance of failure. The o-ring groove was cut such that the compression is slightly less than the optimal 25%. Replacing the o-ring completely would require disassembly of the housing, but there is a simple fix to this issue. Application of gasket sealant in the groove between the Honda TRX carrier and the sprocket would provide a quick and simple fix to this issue.

2b: Although this o-ring showed significant compression during assembly and was designed for the optimal compression, it is unlikely that a leak would start at the interface of the two housing sides. If however a leak does start in this area, a groove was left between the two housing that gives clearance for

gasket sealant to be used. Fill the gap with sealant all the way around the differential to fix the leak.
3b: This is the same type of seal as 1b, but the compression was designed and machined to give the optimal 25% compression. In theory there should never be an issue with this seal, but gasket sealant can be applied in the same way as it was in 1b.
1c: If a leak occurs between the press fit of the bearing than gasket sealant should be applied first and re-tested. If the problem persists then disassembly may be required. If the leak occurs between the bearing and upright a new upright should be made to correct for the lose press fit. If the leakage is between the bearing and the adapter, then the adapter should be replaced with careful consideration to the tolerance of the press fit.
2c: Similarly to the bearing press fit, this issue may require disassembly of the housing in order to fix. Re-machining of the adapter would be necessary to correct for the lose press fit that's causing leakage.

Axles

Changes in Scope

Because the axle outer diameter is so small, we have disregarded the spring pin as a viable option. Cutting a hole in the axle would introduce too much of a stress concentration to be effective. Further, time and budget constraints have also contributed to our dismissal of this option.

Brief Description of Final Axle Design

The design of the axle extension has changed since phase three. After fabricating a sample axle extension with specifications from the previous ohase, we received overwhelming feedback about the high weight of the axle extension. In designing for reliability as the most important metric, the original design of the axle extensions had a factor of safety in place that guaranteed the ale to hold up to the stresses that it would be facing. This resulted in a larger outer diameter than necessary, which increased the weight. Significant weight was shaved off both the short and long axle extensions after the importance of low weight was increased. Now, both axle extensions have an outer diameter of 7/8”. For detailed assembly instructions and design package, see Appendix

Metrics

Clean Interface

Target Value: Yes

Actual Value: Yes

Description: Axles fit inside the differential and are the appropriate length to create a vehicle track of 48”, and fit within the constraints of the FSAE suspension team's specifications.

Cost

Target Value: Team cost less than $1000

Actual Value: $244.94 dollars (Under budget)

Description: Our final project cost came in under budget, with the axle's only expenditure being the test material, and two ATV axles.

Weight

Target Value: 6.9 lbs per axle (weight of previous custom axle)

Actual Value: 6.3 lbs short axle, 6.9 lbs long axle

Description: To minimize the weight, deflection had to be sacrificed. This decision was driven by feedback from our sponsor and advisor. However, the two axles weigh a combined 0.6 lbs less than the custom-made axles from last year.

Deflection at 20 ft-lbs

Target Value: 0.68 Degrees (Deflection of Custom Axles)

Actual Value: 0.61 Degrees in short axle, 0.82 degrees in long axle

Description: At normal driving conditions, the engine will produce a maximum of 20 ft-lbs of torque in each axle. To most efficiently transfer power to the wheels, we want to minimize deflection. Our initial goal was to match the deflections of the short and long axle, but based on feedback from our sponsor and advisor, the increase in weight was not worth matching deflections so precisely. With less than half of a degree of deflection difference between the two axles, it is insignificant to increase the amount of material to match the deflections compared to the weight savings gained by using the shorter diameter extensions.

Allowable Static Torque

Target Value: 165 ft-lbs

Actual Value: 335 ft-lbs

Description: Test axles were made with similar material and tested in torsion, and the allowable torque before failure was found to be approximately 335 ft-lbs before the weld points gave out and the axle extension slipped relative to the ATV Axles

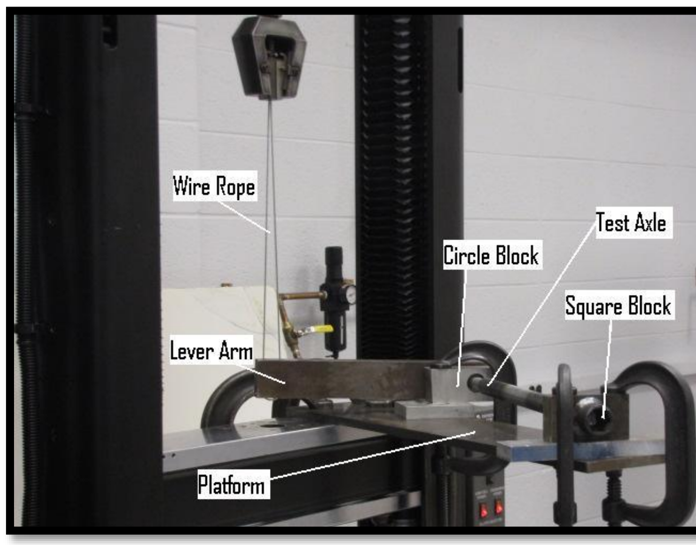
Validation

Experiment

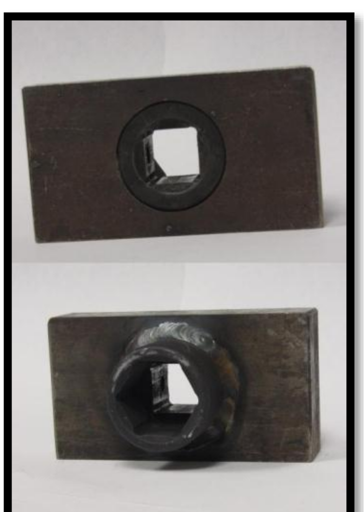
In order to experiment on the design of the new axles, we will have to test a similar axle assembly of pieces in torsion. Due to the budget constraints, the test materials will be made of mild steel, while the ATV Axles are high carbon, hardened steel, and the tube extensions are heat treated 4130 Chromoly Steel. These test results will hold true for the final design.

This experiment will focus on quantifying the amount of torque that can be applied through the axle without making the axle extension slip, but after the experiment, we will be able to determine the weld strength, failure modes, maximum allowable torque through the shaft, and accuracy of testing. Five test setups will be created and tested, each testing a specific design parameter of the axle extension.

1. Short extension, press fit only
2. Short extension, press fit and spot welded (x2)
3. Long extension (same OD as short extension) press fit and spot welded
4. Long extension (different OD as short extension to match torsional deflection) press fit and spot welded.

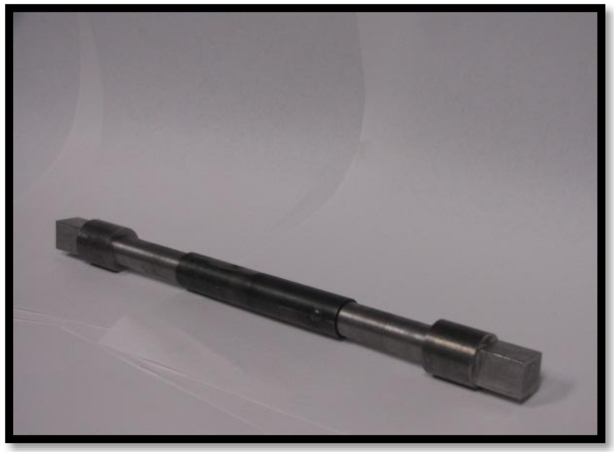


The test apparatus, with one of the test specimens being tested, can be shown in picture right. The setup requires a 1/2" steel plate to be fixed to the Instron machine using C-clamps, which creates a platform for the test apparatus. The test apparatus consists of a "Square Block" and "Lever Arm" which fix the test axle in place, and a circle block, which lets the axle spin through it. When the Instron machine moves down, tension is applied to the wire rope, which applies a torque (at the distance of the lever arm, which is 12") on the test axle specimen.



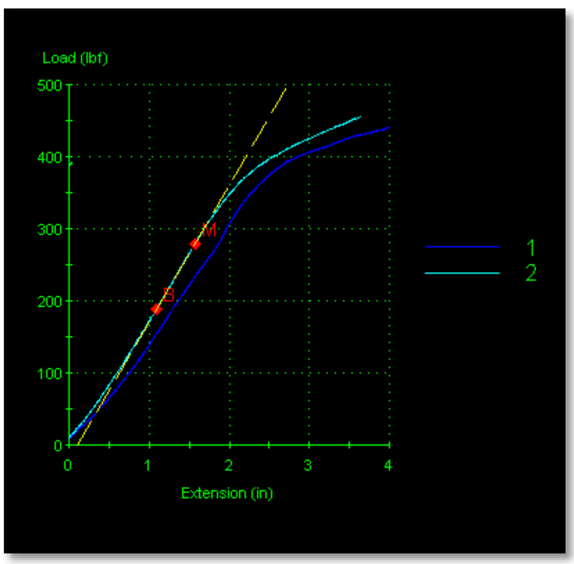
The picture to the leftshows the front and back views of the "Square Block", which is a piece of 2" x 4" steel with a ¾" Drive Socket press fit and welded into it. The test axle specimen fits securely in the square and the Square Block is fixed to the testing apparatus using C-Clamps. To make sure the components are fixed, squares were cut into each side of the ATV axle extension to fix them to the testing apparatus.

Each test extension will have two shafts press fit into both sides, which represent the ATV axle. We will use thermal expansion in order to make sure the axle shafts will slide smoothly into the extension during the press fit. In order to guarantee the 0.0013" press fit that is necessary, the extension will be heated to no more than 727 F (to conserve the heat treatment and ensure that the eutectic temperature is not reached), and the axle shafts will be cooled in a freezer to approximately 50 F, and the components will slip easily in to each other. This process will be repeated with the final axle components. Figure right shows one of the specimens that was tested.



Results

The results of the experiment showed the allowable torque through the axles to be much higher than what we were aiming for. Unfortunately, the weld strength was not able to be calculated, because of complications with one of the test specimens. However, the other information that was needed is calculated, and the axles are able to be validated for use in the design.



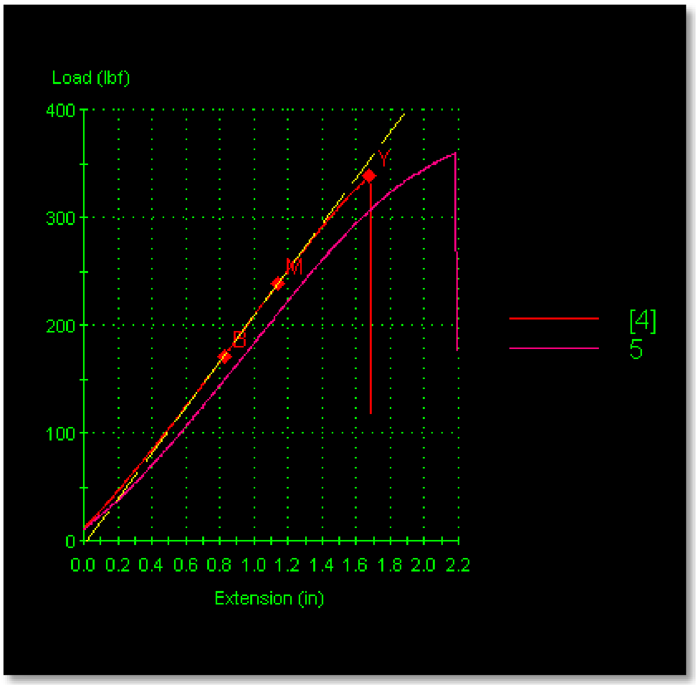
The graph to the left shows the readout of two trials of the long axle extensions. Trial 1 shows the load vs. extension of the Instron machine of the 7/8" Outer diameter tube, and Trial 2 shows the same of the 1" outer diameter tube. The relationship is consistent with what would be theoretically calculated; the larger diameter tube has higher loads with the same amount of deflection. The shape of these graphs resembles an "S" because the Instron machine pulls straight up, and the lever arm is constantly changing at an angle. Also, a large amount of deflection was noticed in the platform, which may have changed the shape of the graph. Ignoring this deflection, and based on the final angles of the lever arm and the final loads applied, the maximum torque that the axles were tested to was:

7/8" OD Axle – 458 lbs* 0.942 ft = 431 ft-lbs

1" OD Axle – 446 lbs * 0.95 ft = 423 ft-lbs

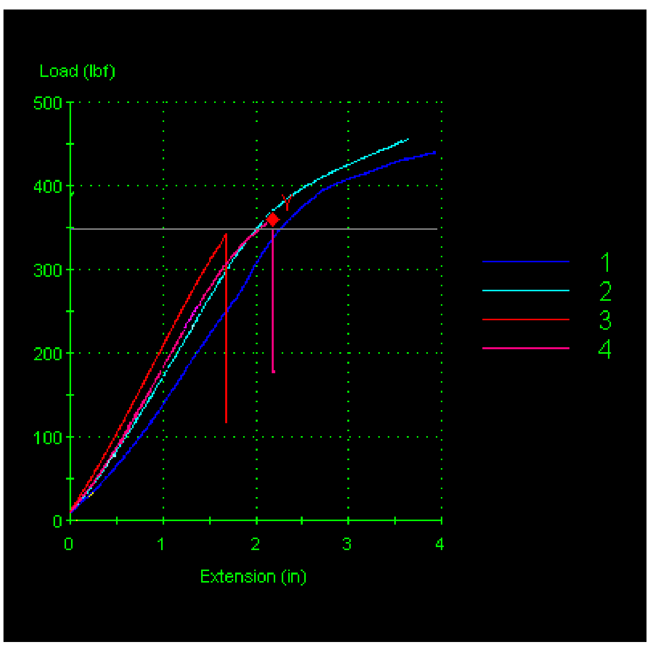
The tests were stopped at these points due to the fact that an unsafe amount of deflection was seen in the platform. Therefore, we cannot draw too many conclusions from this data about the actual performance of the axle.

The results for the short axle extension were more conclusive than the previous set, and actually showed a failure point. The readouts from the Instron can be seen in the image to the side. Two identical short axle specimens were tested in the same fashion, without noticing a deflection in the platform. The graphs of the load vs. extension show a much more linear relationship, which correlates with less deflection and error in the system. The failure points of both axle specimens can be clearly seen in the figure, which happened to be the weld points breaking, and the axle slipping inside the extension. Based on the final angles of the lever arm and the final loads applied, the maximum torque that the short axles could hold before slipping was:



Trial 4 – 328.39 lbs * 0.991 ft = 325.45 ft-lbs

Trial 5 – 349.56 lbs * 0.983 ft = 343.61 ft-lbs



We can attribute the differences in torque between the inaccuracies in the press fit; achieving a 0.0013" press fit exactly is nearly impossible given the accuracy of the machines in the machine shop. The most accurate machining increments we have available to us is 0.0005," therefore the press fits can range from anywhere between 0.0013" – 0.0018." Each interference was calculated before actually performing the press fit, and it can be shown that trial 5, which had a press fit of 0.0015", where the press fit in trial 4 had a press fit of 0.0013".

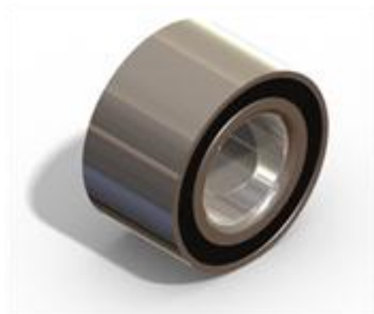
Averaging the two experimental values, we can find the maximum allowable torque, before failure at the weld points, is **335 ± 10 ft-lbs.** Based on the linearity of the long axle specimens before this value, it is strongly hypothesized that the long axle extensions can hold at least this same value of allowable torque. To further demonstrate this hypothesis, we can look at both graphs overlapped. Before the gray line (average point of failure for the short axle specimen) both trials 1 and 2 (long axle specimen trials) exhibit more linear behavior and deflection in the platform was not noticeable. Because the same press fit is used in the short and long axle, we have come to the conclusion that the press fit is just as strong in both cases, and the failure point remains constant in both short and long axle.

Hubs:

Brief Overview of Design process (Phases 1-3):

The rear wheel hubs are the outermost component of the drivetrain. The hubs serve four primary purposes…

1. The hub's lug bolts hold the wheels to the vehicle.
2. The hub, which is press fit into a wheel bearing, allows the wheels to rotate with respect to the "fixed" suspension upright.
3. The hub's internal splines mesh with those on the axle's stub shaft, allowing the engines power/torque to be transferred to the wheels to propel the car.
4. The hubs hold the brake rotor so that braking force can be transferred to the wheels to slow the car.



Bearing Selection:

64mm Taylor Racing Products Wheel Bearing were selected for use in the hub assemblies because they are slightly cheaper and considerable more compact then the Porsche 911 wheel bearings which were also being considered. These were the same bearings used on the 09-10 car and they weigh 74017lb each. They also each cost only $39. Since the bearings are the dividing line between our project and suspension's project we each paid for one.

With the bearing selected, the suspension team was able to move forward designing the uprights and bearing retainer that enclose it (Fig 20). For details on this design, see the Suspension team's reports.

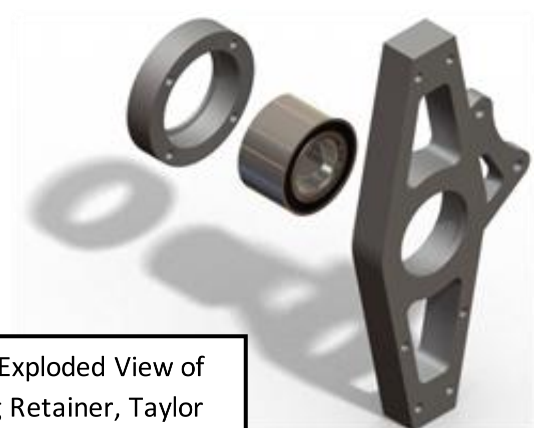


Fig 20) Exploded View of Bearing Retainer, Taylor Racing Bearing, and Upright

Internal Geometric Constraints

Dimensions in this section are not exact as indicated by the ~ symbol. Exact values can be found in the machinists drawings located in the appendix.

The primary influence on the hubs design were the necessary dimensional constraints over the hub's span. In other words, the hubs' exact role in bringing together the axles, suspension components, and wheels had to be determined. First, a close look had to be taken at the axle's stub shaft and how the whole internal assembly would fit together.

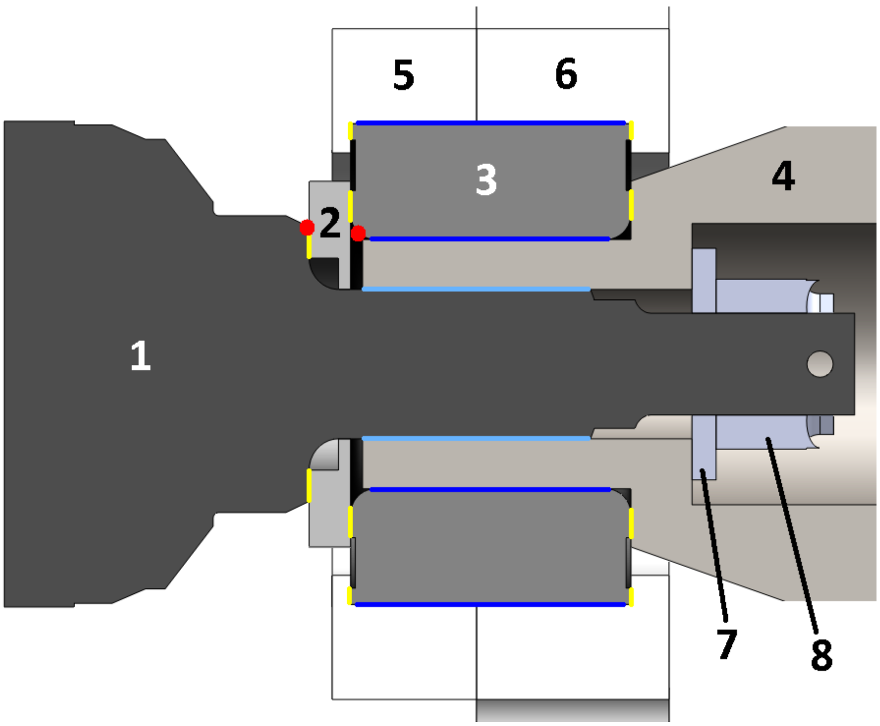


Fig 21) Cross Section of Internal Hub Assembly

In the above figure (Fig 21): yellow represents contact surfaces, light blue represents splined interfaces, and dark blue represents press fits.

1. Axle Stub Shaft (Spindle)
2. Spacer
3. Taylor 64mm Wheel Bearing
4. Hub (only partially shown)
5. Bearing Retainer (from suspension)
6. Upright (from suspension)
7. Washer
8. Castle nut

There are several key things to note in this assembly. First of all, the bearing is the only thing in contact with, and the bearing exterior is effectively one with, suspension's upright and bearing retainer. Next, notice the hub is press fit into the bearing interior to allow for rotation and interfaces with the stub shaft via splining. This allows to allow torque to be transferred from the axle to the hub. Finally, note that spacer (2) allows the bearing interior to be effectively sandwiched between the stub shaft and hub when the castle nut (8) is installed. This eliminates any chance of force transmitted down the axle overcoming the friction from the hub's press fit and sliding everything axially out of the bearing. Without the spacer (2), the radius on the bearing would awkwardly contact the point at the bottom of the small chamfer on the stub shaft (red dots on figure).

External Geometric Constraints

Dimensions in this section are not exact as indicated by the ~ symbol and machinists drawings in the appendix should be examined for exact values.

Once clear of the wheel bearing/upright assembly there were still numerous dimensions that the hub MUST conform to (Fig 22). Several of the most critical ones can be seen in the figure below...

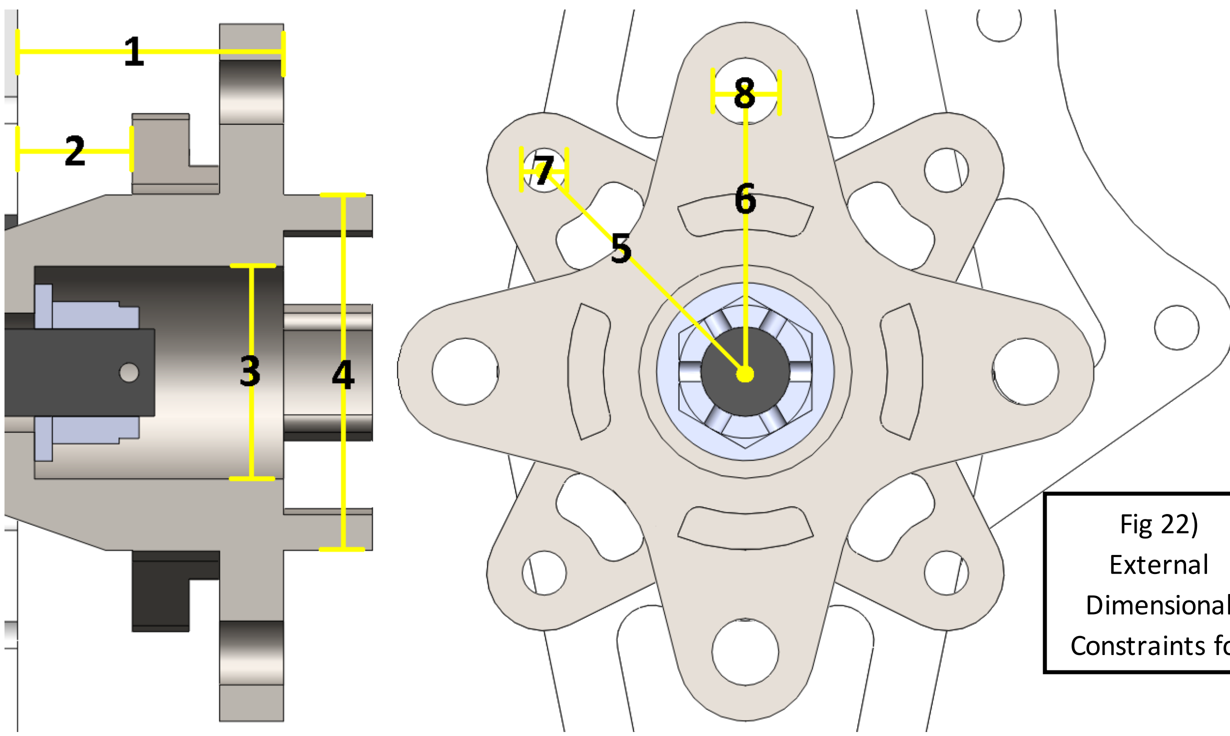
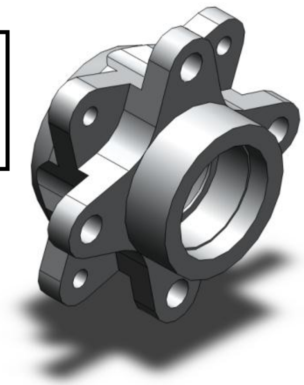


Fig 22) External Dimensional Constraints for

1. The distance from the wheel mount plane to the upright must be such that the wheel has clearance internally for the brake rotor and caliper (~1.87in)
2. The distance from the brake rotor mount surface to the upright is critical when considering the brake caliper, which is mounted to the upright, reaching over the rotor (~.81in)
3. A diameter of ~1.5in must be here to insure a socket can be inserted to tighten the castle nut
4. A diameter of ~2.5in must be here to fit into the hole in the wheel to ease mounting and provide brief wheel attachment in the event of lug bolt failure

5. Provided radius of brake rotor bolt pattern (~2in)
6. Provided radius of wheel lug pattern (~1.97in)
7. Provided diameter of brake rotor bolt (~.31in)
8. Provided diameter of lug bolt (~.47in)

Fig 23) Aluminum Front Hub from 09-10 Car



General Concept:

The original concept for the hubs is the double 4 point star design utilized in the 09-10 aluminum front hubs (Fig 23). Dr. Timmins also provided our team with a Porsche 911 wheel hub as a possible model for our design. The 911 hub does not have a brake rotor mounting surface the way that ours needs to and has a complete disc instead of separate feet which extend to support the lug bolts. While this solid disc idea is great for torsional rigidity it presents a huge machining issue since there must be a ~.5in gap and then four feet or another disc for the brake rotor mount. Discussion with Steve Beard in the ME student shop revealed that the shop is not equipped to machine such a cravis and not removing that material would make the hub far too heavy. Using the disc idea for the wheel mounting and having offset pedestals for rotor mounting was also considered but under heavy braking the shear in these offsets would easily exceed the yield strength. Therefore, the double 4 point star concept was settled upon.

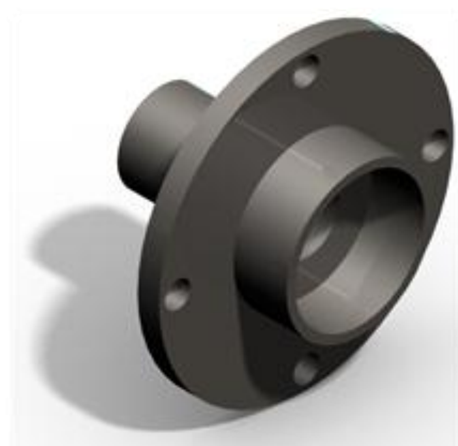


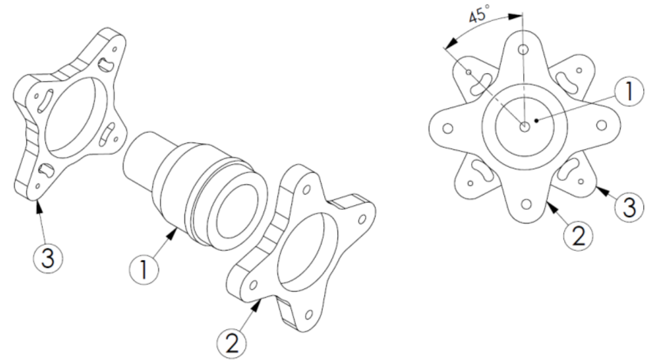
Fig 24) Steel Taylor Racing Wheel Hub (No Brake Rotor Mounting Surface)

In terms of material it was decided that steel should be used due to the strength and hardness requirements of the splines. Again Steve Beard had concerns about machining such a part out of steel. As long as the splines must be steel the only other concept that was developed was machining the majority of the hub out of aluminum, to be almost identical to the 09-10 front hub, and pressing and bolting it to a steel insert with splines. This idea was rejected due to the intricacies of press fitting the two different types of metal together. This idea would also end up saving very little weight, if any, by the time the design was beefed up to be just as strong as steel made from aluminum. The only benefit would have been a considerable ease of machining but it was determined that a guaranteed quality end produce should take priority over manufacturing ease.

In the end, Mr. Martelli, from Martelli's metal fabrication helped us come up with the idea to make the hubs out of three pieces of steel; a center core and 2 separately machined mounting stars. The core is 4130 steel and the mount shapes are cut from A-36 domestic plate.

Final Design Generation Process and Validation

Keeping in mind the geometric constraints, the double 4 point star design, as seen in the 09-10 front hub, and the 09-10 rear hub from Taylor Racing (Fig 24), which can be used to observe the perfect, professionally designed, interface with the Taylor 64mm wheel bearing an initial design was established. Hand solid mechanics calculations and many iterations of *SolidWorks* FEA simulation where then used to optimize key dimensions and validate the final design.



SolidWorks FEA Simulation:

This section highlights just five of the key scenarios examined. The images in this section are primarily of an early design iteration and do not reflect the final design.

Resting (Fig 25): The resting simulation was set up to examine the worst case scenario of the car sitting at rest. This was done by fixing the surfaces that contact the wheel bearing and applying a vertical load equal to the entire weight of the car (500lb) distributed over the 4 lug bolt holes. This obviously acts like the whole car is resting on one rear wheel for worse case.

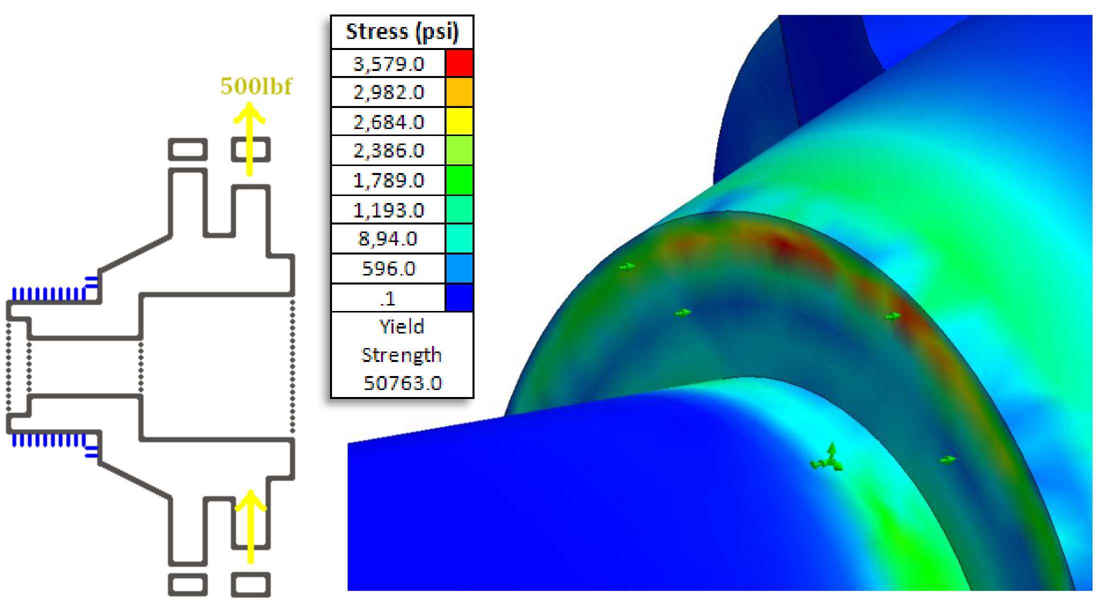


Fig 25) Resting Scenario Simulation with stress concentration snap shot

Accelerating (Fig 26): For the acceleration simulation the lug holes were fixed in space to act like the wheel is stationary with respect to or "glued" to the ground. Though the engine produces only 58ft-lbs the internal hub area where the splines would be was loaded with the maximum shock torque value of 165ft-lbs (1980lb-in). *See derivation in the Axle section*

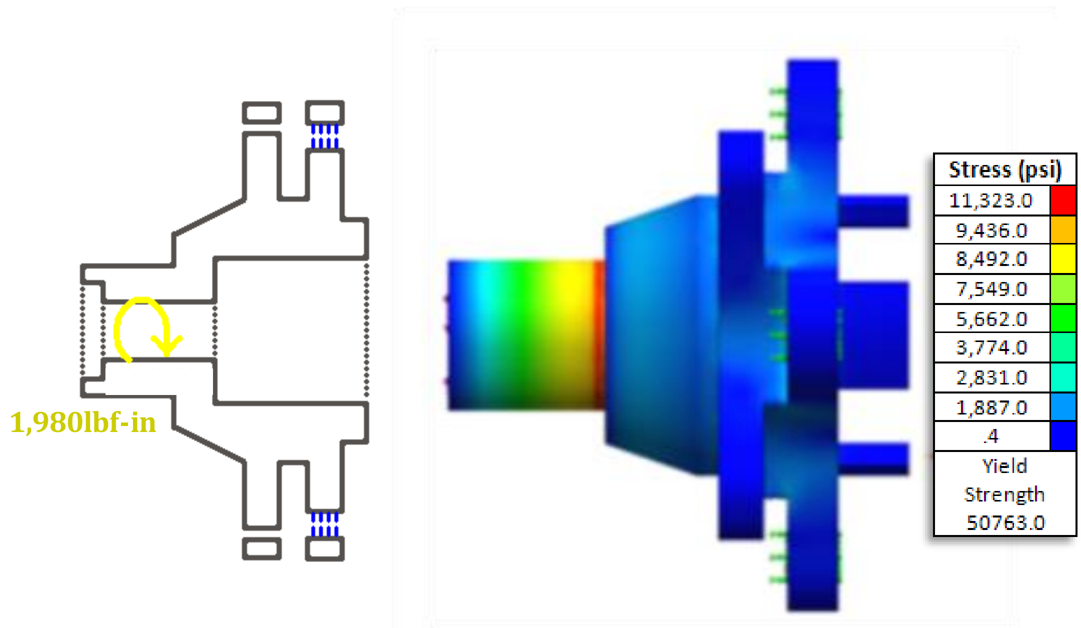
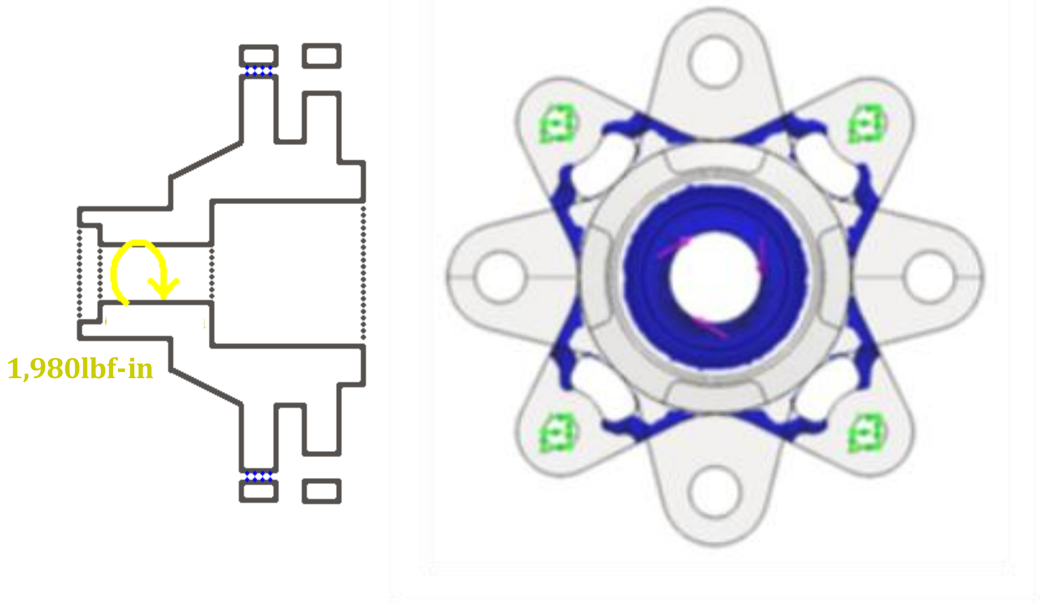


Fig 26) Accelerating Scenario Simulation with stress plot

Braking Type 1 (Fig 27): The first braking simulation that was completed was done very similarly to the acceleration case. This time instead of the lug holes being held fixed the brake rotor mount holes were fixed and the same dynamic torque load of 165ft-lbs (1980lb-in) was placed on the hub where the splines should be. This effectively creates the situation where the brake caliper has the rotor locked and the clutch is dropped.

Fig 27) Braking Type 1 Scenario Simulation with 53% design insight plot

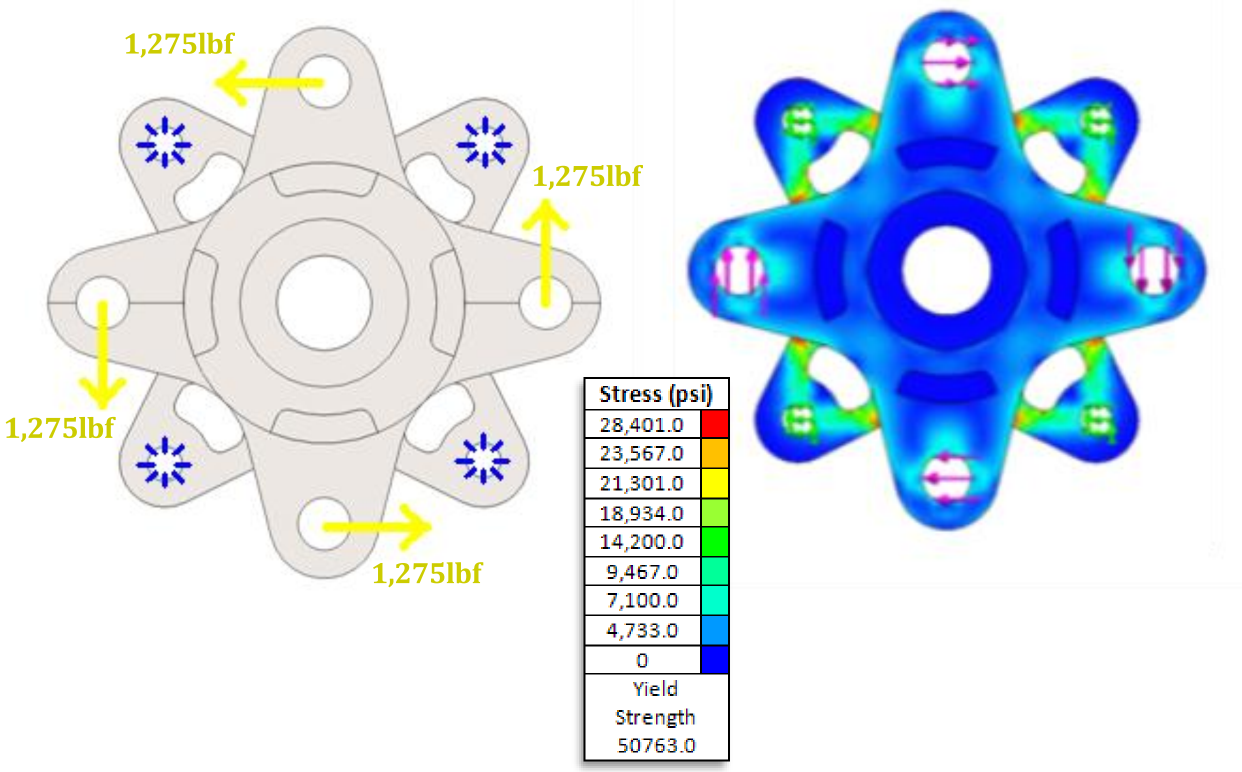


Braking Type 2 (Fig 28): The second type of brake testing, and the more realistic, is where the brake rotor is again locked but torque is being applied from the tire's friction on the road. This worst case frictional torque was calculated by using the whole weight of the car on one wheel and an exaggerated frictional coefficient of 2.

$$\frac{N * \mu_k * r_{Wheel}}{r_{LugBolts} * Number\ of\ Lug\ Bolts} = Force\ Load\ on\ each\ Lug\ Hole$$

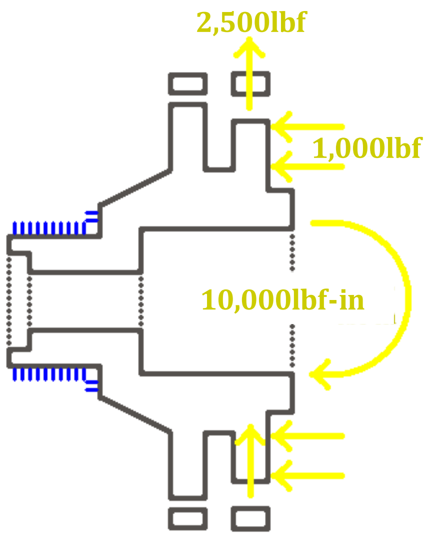
$$\frac{500lbf * 2 * 10in}{1.96in * 4} = 1275lbf$$

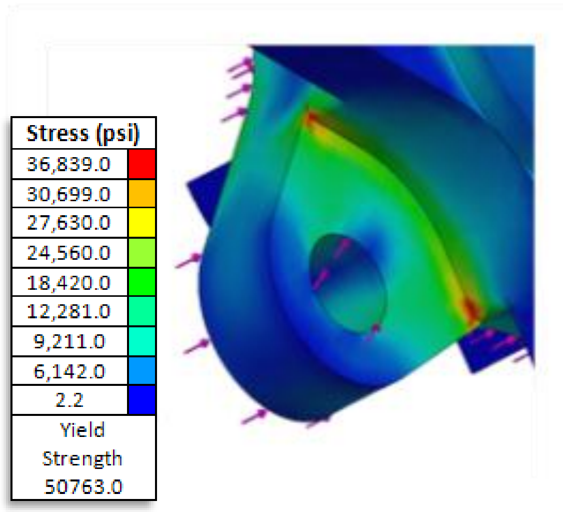
Fig 28) Braking Type 2 Scenario Simulation with stress plot



Cornering (Fig 29): The final critical simulation conducted was that of extreme cornering. As indicated in the figure below the hub bearing contact surfaces were once again fixed. Due to weight shift to the outside tires as well as dynamic shock loading values provided by suspension a vertical force of 5g's, five times the weight off the car, 2500lb was used. In addition, forces were added to simulate a 2g corner with the tire not slipping on the ground. These included a 1000lb (500lb * 2) force inward on the hub and a 10,000in-lb (1000lb force acting at radius 10in) torque on the hub acting around the tire center. All these loads combine the make a very worst case situation.

Fig 29) Cornering Scenario Simulation with stress concentration snap shot





It is not necessary to examine the hub/tire on the inside of a corner since friction would be very low from the wheel lifting and the potential for failure is primarily in the lug bolts.

Eventually a final design emerged which was optimized for weight and factor of safety in the five critical loading scenarios described above.

Also note that an aluminum wheel guide was designed to bolt to the front of the hub to ease wheel mounting.

See appendix for machinists drawings of finalized hub.

Final Design and Part:

All hand strength calculations and SolidWorks simulations were done assuming AISI 4130 steel. It is important to note that the part will be even stronger after RCV's heat treatment. Images of the final hub are shown in Fig 30.

Both hubs were successfully manufactured due to a tremendously generous donation of time and material by Mr. Martelli. The core was machined on the lathe by a professional machinist and the two mounts were water jet cut out of plate. The assembly was then pressed together and welded by a professional welder using appropriate heat treatable weld filler.

Shown below are the factors of safety and maximum deflection generated by the five critical simulation cases on the final design. Some of the studies were also run on the steel Taylor hubs and front aluminum hubs for comparison.

	Factor of Safety			Maximum Deflection (in)		
	Taylor Racing	Aluminum Front	New Steel Rear	Taylor Racing	Aluminum Front	New Steel Rear
Resting	14.99	---	19.44	0.000128	---	0.0002269
Acceleration	3.58	---	5.83	0.000544	---	0.0003871
Braking Type 1	---	---	7.37	---	---	0.0003772
Braking Type 2	---	0.26	2.78	---	0.00306	0.001916
Cornering	0.98	---	2.64	0.00691	---	0.004217

Weight (lb)	2.04	0.94	3.17

Key Observations:

Weight: The final design has a weight of 3.17lb each. Though this is heavier than both the Taylor and aluminum hub it is actually quite low. It is important to remember that the Taylor hub doesn't have a rotor mount plane and the hub being designed must extend outward over 1.5in farther. Since extending a solid cylinder out 1.5in from the current Taylor wheel mount surface would create a part that weighs ~8.63lb it is clear that the 3.17lb design is fairly minimal. It is not worth comparing weight to that of the aluminum hub since it is obviously a lighter material and since it is a front hub the wheel bearing fits inside of it greatly reducing internal material.

Comparison with Taylor Hub: The Taylor hub was tested in the 3 simulations that didn't involve braking since they do not have a brake rotor mount. As indicated by the table the designed hub had a larger factor of safety then the Taylor hub for all 3 tests.

Comparison with Aluminum Hub: Since the aluminum hub does have both a brake rotor and wheel mount plane but lacks the mount shaft that presses into the wheel bearing the only test run on it was Braking Type 2. The new steel hub far out preformed the aluminum one in this scenario by a factor of safety of 2.5.

Greatly Exaggerated Scenarios: The fact that the Taylor Hub has a factor of safety less than one for cornering and the Aluminum hub has a factor of safety far less than one for Braking 2 justifies that the scenarios used for simulation are extremely work case. The designed steel hub will hold up perfectly considering that it has a factor of safety above one for both of these worst scenarios and considering that both the Taylor and aluminum hub have held up perfectly on the 09-10 car.

Fatigue:

Full fatigue analysis was conducted on the two scenarios with the lowest factors of safety, braking type 2 and cornering, to determine the lifespan of the part under these loadings. The simulations were set up to run the 2 cases in a fully reversed mode to portray hub rotation until part failure based on SolidWorks S-N Curves derived from Young's modulus.

- If the car was driven in the cornering scenario for 28hrs it would fail from fatigue
- If the car was driven in the braking type 2 scenario for 33hrs it would fail from fatigue

These results are fine since the car is only driven for around 30 hours over the course of the year/season and little if any of that time will be spent in these very extreme loading situations.

Successful Cost Minimization:

Only $130 for the pair after splining and heat treating due to Mr. Martelli's generous donations

Current Status:

The hubs are the only component of our project which did not end up properly installed on the final car. This is due to the fact that the broaching and heat treating procedure has at least a 2 week lead time and Dr. Timmins advised us that it would be best to not send the hubs away but instead keep them to show at the presentation.

Finish Machining:

Once the hubs were received from Martelli's there was a bit of machining that had to be done to finish the pieces.

1. The wheel mount surface had to be flattened on the lathe
2. Material had to be taken off the brake rotor mount surface to ensure proper dimensioning between the wheel and brake mount planes
3. The lug bolt holes had to be boarded out for a tight permanent press fit
4. The brake rotor bolt holes had to be drilled larger and tapped
5. The holes to bolt the aluminum wheel guide to had to be drilled, tapped, and counter board

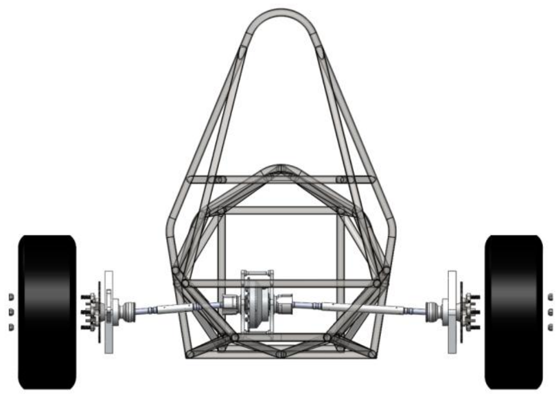
Path Forward:

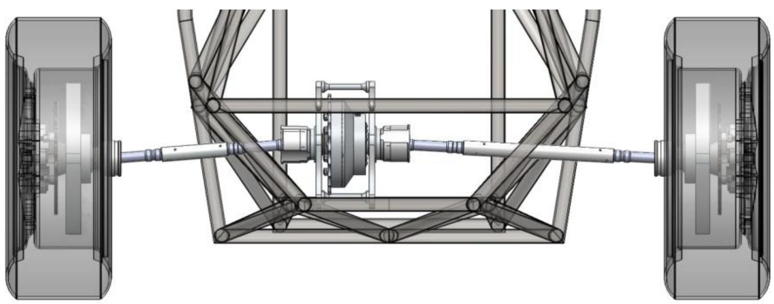
With senior design drawing to a close the hubs will promptly be finished. A good bit of the finish machining described above has already been done but there is a little to finish and that will be the first order of business. Next, the hubs will promptly be sent away for broaching and heat treating. When they return, around the beginning of winter session, all threads will be re-tapped, as slight warping may have occurred from heat treating. The lug bolts and wheel bearing can then be pressed in place and the unit can be assembled in the uprights from suspension. Once installed on the car the hubs performance and integrity can be observed.

Complete Drivetrain:

Following are some images of all 3 drivetrain subsystems assembled together and placed in chassis' steel frame (Fig 32).

Some of these images include uprights, bearing retainers, rims, tires, and brake rotors which FSAE Powertrain is not responsible for. Also note that due to time constraints these depictions are SolidWorks screen shots and are thus not rendered to photograph quality with SW PhotoWorks.





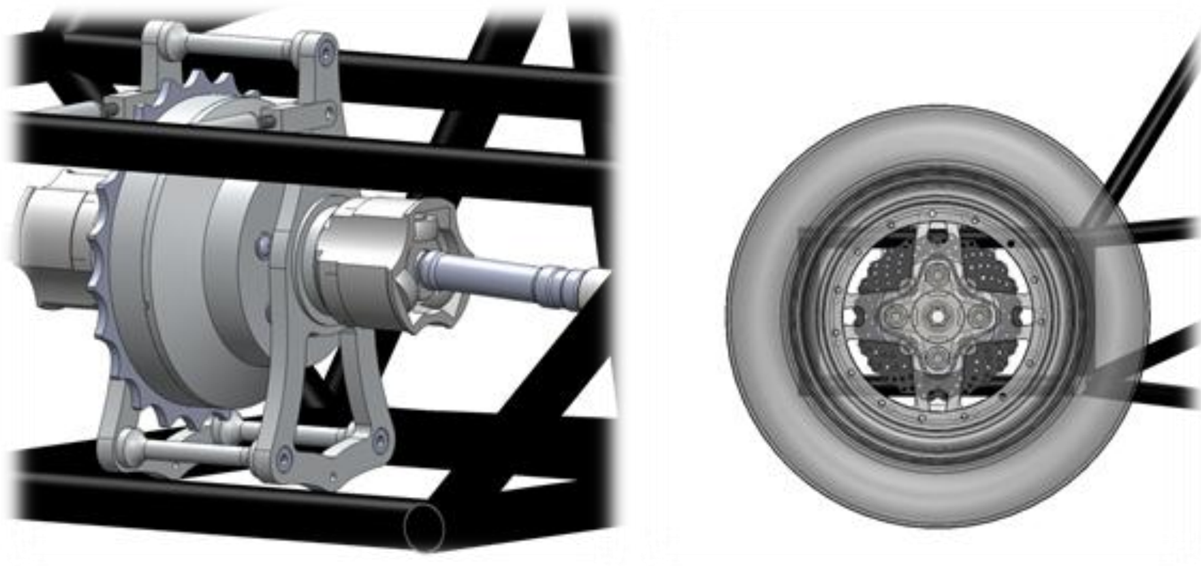
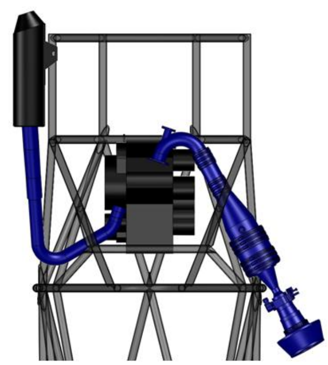


Fig 32) Several views of the complete drive train assembly.

Project Aspect 2: Air System



The second aspect of our project is the air system. The air system is what allows the engine to breath and can itself be broken down into 2 projects; air intake and exhaust. Both are covered in detail in the following section.

Because a large amount of our team's time so far has been used for final drive design some basic information about both systems that may have been omitted in previous reports must be covered.

Air Intake:

Phase 4 Scope Changes:

The original plan was to create an adjustable air intake, run a set of experiments with the car on the dyno to judge performance of different arrangements, and create detailed plans for the final intake to be manufactured over winter session. Our team did successfully do a dyno day with the old air system to gather baseline data from the car and fired up the car with the new air system to check for structural integrity. Unfortunately though, due to the highly compromised status of the club's 09-10 car and the high amount of business at Injection Connection the big experimental dyno day was not completed. Therefore the deliverable for this aspect of our project became the prototype itself to be used as a tuning tool for years to come. A path forward for the testing and final air intake manufacturing to be done early in winter session is also provided.

Overview of Phases 1-3:

All the air that enters the engine is delivered via the air intake. An effectively designed and optimized air intake can improve the volumetric efficiency of any engine by up to ~10% over a missing or poorly designed intake which can actually be highly detrimental to performance.

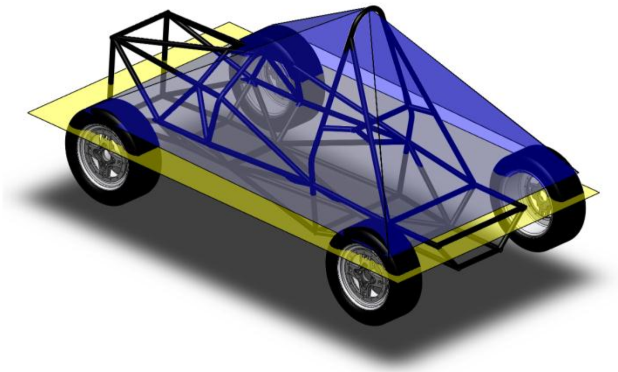
Constraints:

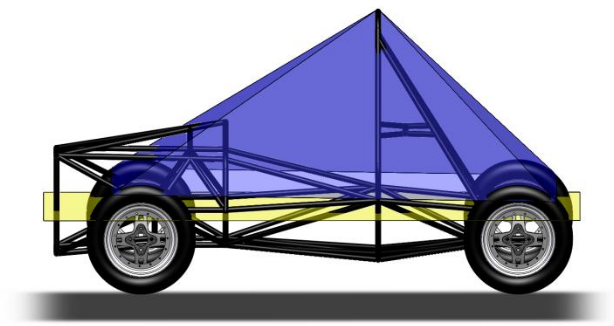
For engine to function…

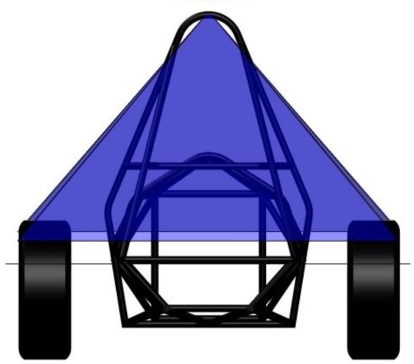
- Must have an air filtration component of some kind to clean the inlet air and prevent fouling of engine components
- Must have a throttle body to control the amount of air able to flow into the engine
- Must have fuel injector port from where the fuel injector (FI) can spray fuel
- Must have a short "runner" tube where the fuel injector port is located some distance from the engine valves to allow the fuel to mix with the flowing air
- Must have a small "Fuel Pressure Regulator Local FI Ambient Reference Point" for a hose from the pressure regulator to attach to
- Must have a throttle position sensor and mass flow sensor to feed data to the on board ECU unit (only one of these at a time, used along with rpm readings, is necessary but we must provide ports for both)
- Must mount securely to the engine head with gasket to prevent air leakage

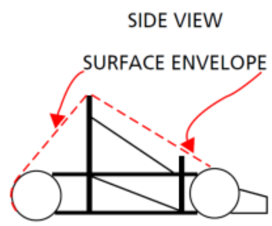
Additional from FSAE Competition Rulebook (Section 8B)…

- Must have a intake support of some kind that prevents structural stress in the intake
- Must have a ridged 20mm circular restrictor between the throttle body and engine
- The restrictor must be easily accessible for the competition judges to measure
- All intake components must fit within the pyramid envelope created by connecting the top of the main roll bar to the outside tangent edges of the wheels (Blue in Fig)
- All intake components less than 350m from the ground must be shieled "from side or rear impact collisions by structure built to Rule B.3.24 *or B.3.31* as applicable" (Below yellow in figure)





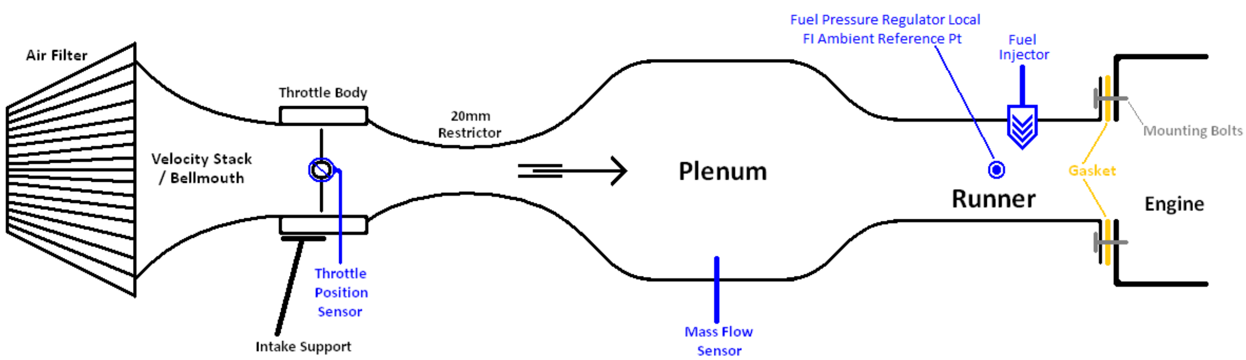


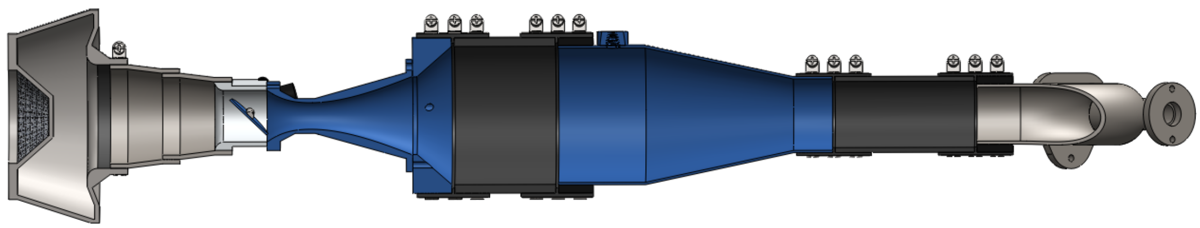


Additional from Mr. Shwarz…

- Must have a velocity stack/bellmouth that smoothly transitions from the air filter to the throttle body

In addition to all the necessary components touched on above, an air plenum will also be incorporated into the system mainly to improve engine responsiveness. A schematic of the basic air intake layout is below along with a cross sectional *Solidworks* rendering of the final prototype followed by details regarding each component of the system…





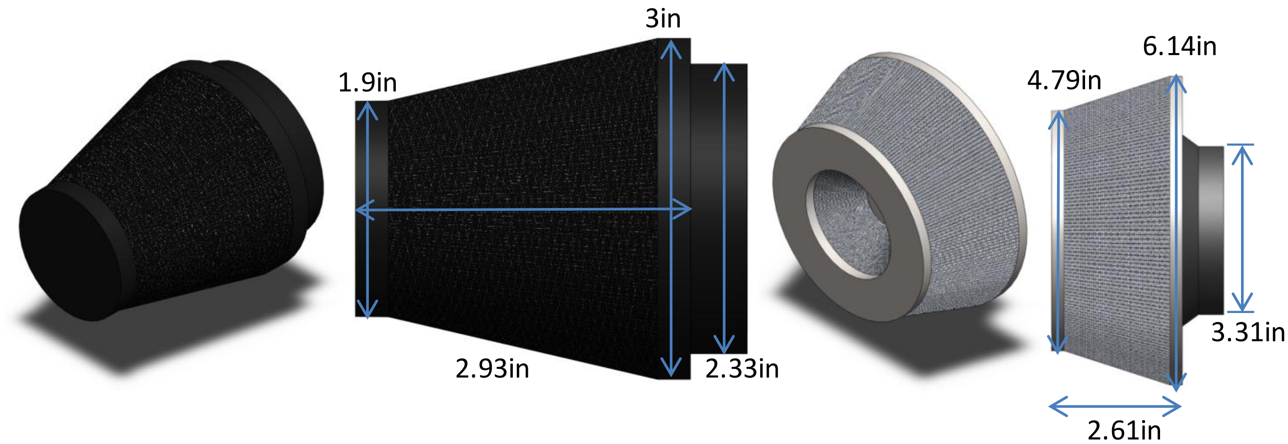
Basic Component Order Validation:

The first true design consideration for the air intake was what order the intake components should be lined up in. The first thing that was done was to determine how the system should begin. Two concepts for this were researched in detail; 1) Have an air inlet and snorkel tube feeding air into an air box which would contain a rectangular filtration component like in most full sized automobiles or 2) simple have the system begin with a conical or cylindrical air filter sticking out in the ambient air like it does on the 09-10 car. Option 2 was selected primarily due to weight, packaging, and increased system complexity concerns of option 1. The addition of complexity largely is because the flow and resonant frequency of an additional pipe (the snorkel) and chamber (the air box) would have to be examined. Utilizing a sheet filter in an air box is also a poor use of space considering that if a sheet filter were simply rolled up, it would become a conical/cylindrical filter and be far more compact.

Once it was clear that the air system would simple begin with a conical or cylindrical air filter like on the 09-10 car the next consideration was the order of the major components that are on the clean side of the air filter. The main three are the required 20mm restricted, the air plenum, and the throttle body. While the throttle body would ideally be as close to the engine block as possible, the SAE rulebook mandates that the restrictor be between the throttle and engine. Again, every order that the three could be arranged in was investigated and a decision was made. The decided upon order of throttle body, restrictor, and plenum is shown in the above schematic. It is the best option because it allows the storage of air in the plenum for use in the event of vigorous acceleration before additional external air has a chance to pass through the fully open throttle body and restrictor. This allows for a very responsive vehicle. There can be short moments during acceleration that the engine feels unrestricted as is “eats” the air from the plenum which does not have to pass through the restrictor as it has previously done. If the restrictor was downstream of the plenum this would not be the case and the plenum would effectively be useless.

Filtration Components (Air Filter): The purpose of the air filter is to extract dirt, dust, and other foreign particles from the air before it flows into the engine. This air cleaning prevents foiling of critical engine components including the valves and allows for better long term performance and reduced maintenance.

Selection Validation: An air filter had to be selected for use on the new car. Since maximum ease of air travel through the filter is desired Mr. Schwarz asserted that bigger is better up to the point where packaging becomes an issue. In terms of shape, both the conical and cylindrical types of filters can be mounted via a simple hose clamp so neither makes of breaks the selection process. Due to our team’s extremely tight budget the first options that were examined were the collection of air filters in the SAE stock room. The filter that was selected is examined below…

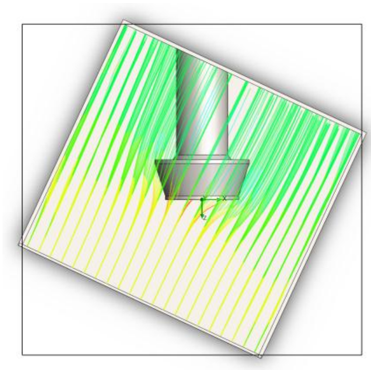


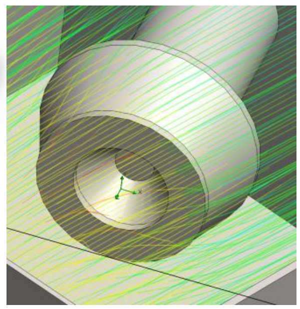
	Weight (lb)	Filtration Material	Filtration Material Surface Area (SA) (in^2)
Old Filter	.3125	Oil Gauze	17.69
New Filter	.9375	Dry Gauze	43.24

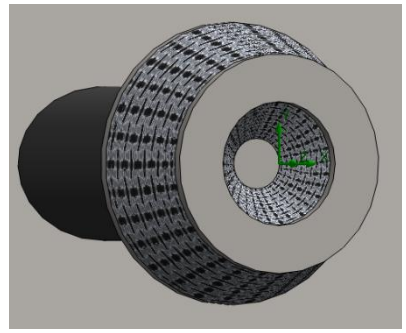
Benefits: The air intake to be used was a lucky find in the SAE stock room. It is a dry gauze filter which is good because it has a long life span and can be easily cleaned. It also does not have to be periodically oiled like the current filter which obviously reduces car maintenance. The new filter has nearly 2.5 times more filtration area then the filter from the 09-10 car greatly increasing the ease of air flow and reducing the likelihood of filter fouling. The inverted conical section in the center of the filter is also a positive feature as it further increases surface area and thus adds in the effective use of space. The filter is also free as it was found in the SAE room.

Drawbacks: The only drawbacks of the new filter are weight and size increase. These issues are obviously inherent to using a larger filter and the increases are not large enough to warrant concern. As seen in the above table weight only increases.625lb and the for size, the largest diameter doubles while filter depth actually decreases slightly. This doubling of diameter does not present any packaging concerns onboard the vehicle.

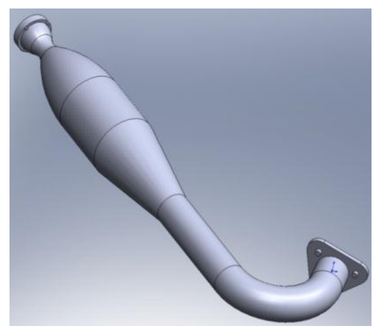
Orientation Validation: The filter selected for use was very easy to orient properly to intake air. While a cylindrical filter would work relatively poorly if it's "top" surface without filtration material were perpendicular to the air flow passing the car our filter with its external and internal inverted conical sections can be situated in almost any way. Below are some snap shots of a basic CFD Simulation showing the filter oriented at 25° to the flow of passing air as if it were mounted on the end of the air intake of the 09-10 car. This is pertinent because the new air intake will have a similar arrangement on the new car and thus will likely place the filter at a similar angle to the cars motion. This simulation was done with air traveling 35mph (the cars ~ average speed). Note: The whole internal inverted conical section as well as the portion of the external cone visible in the view from what would be the front of the car (the right image) are areas where the dynamic pressure of the air will create a minimal ram air effect adding flow into the intake. All other areas will also work but will rely solely on the creation of negative pressure inside the filter to draw air in.





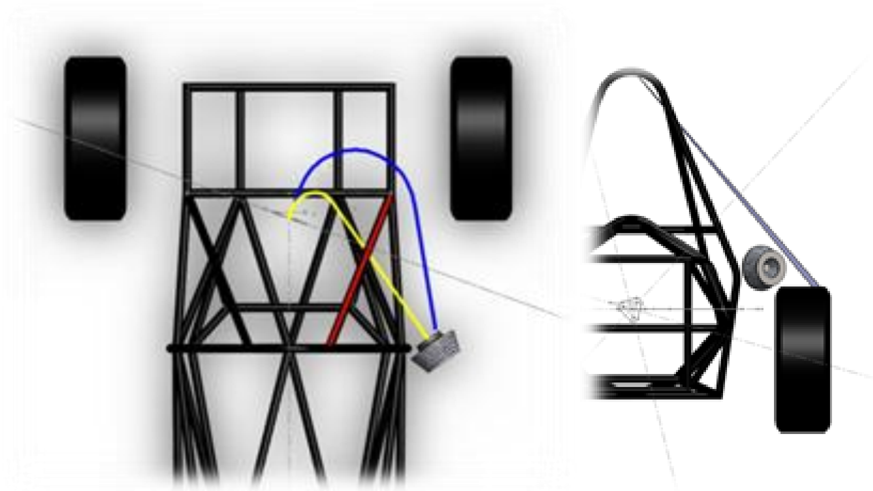


Positioning (General Intake Path) Validation: It was critical that the air filter be positioned on the vehicle so it receives a constant stream of cool, undisturbed, ambient air regardless of the vehicle's speed and direction of travel. It was also important to remember that the air filter, which is the farthest out component of the intake, must be within the pyramid envelope. Since the engine is a single cylinder the intake must therefore be a single somewhat lengthy component similar to what is on the 09-10 car. Thus the only place that the filter can go without exiting the pyramid is just outside the frame to the side of the driver's seat. There would be more positioning options if the engine had multi-cylinders and the intake with a plenum was situated close to the engine head with short runners leading to each cylinder.



Below are some images of the path (shown in yellow) that the intake will take leading from the head air inlet, that is positioned where it will actually be when the engine is mounted in the new frame, to the air filter. This intake path is very similar to the one on the 09-10 car.

Note: The blue path is an old design iteration which was eliminated but it may be called on for use if the dyno experimentation calls for a very long runner.



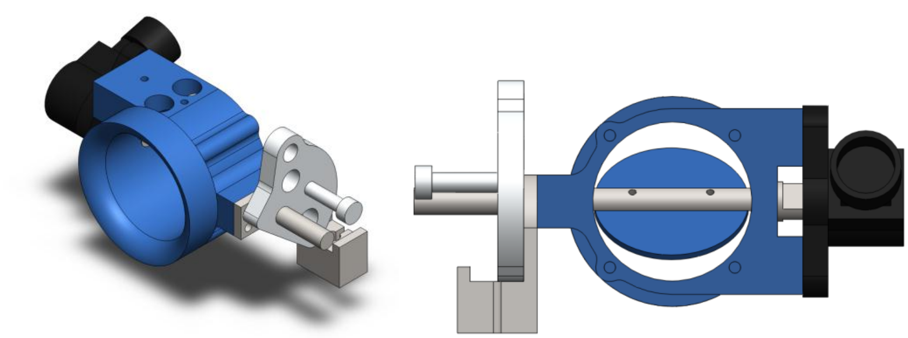
This air filter position is also ideal because just below it there will be sidepods mounted to the vehicle (not yet shown in chassis' design). Air will travel smoothly along the top of the sidepod straight into the filter.

Note: As per our customers request the interior lip of the air filter was rounded to ease the flow of air into the rest of the intake.

Throttle Body Validation: The purpose of the air intake incorporated throttle body is to control the amount of air which is allowed to flow to the engine.

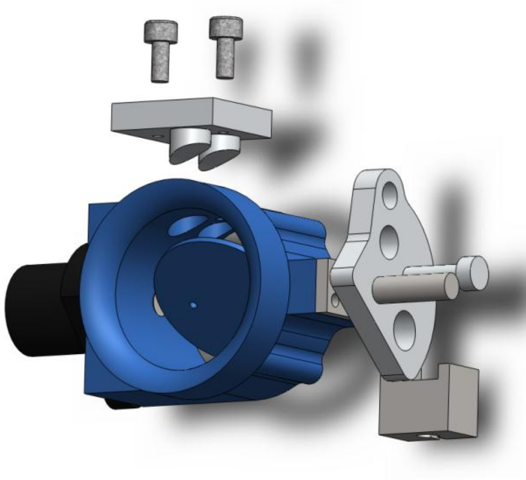
For details about how the throttle body is actuated based on the drivers input on the accelerator see the Driver Controls report.

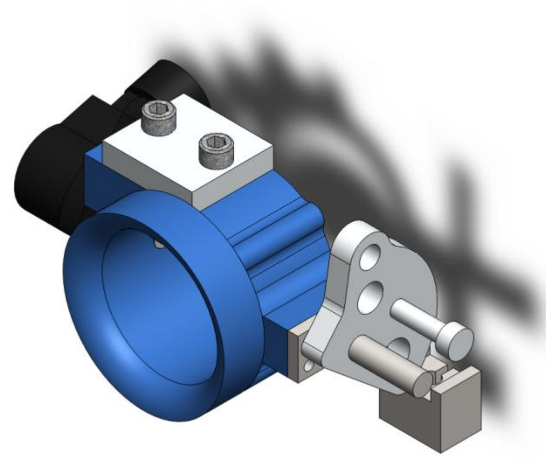
Again, due to our tight budget we will be utilizing a throttle body from the SAE stock shelves (pictured below) in the 2010-11 car.



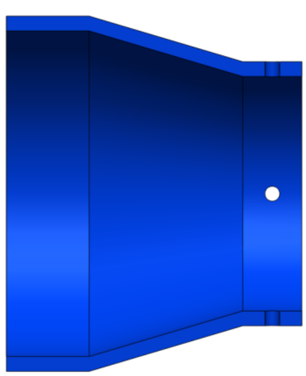
In the above images the anodized blue aluminum is the main part of the throttle body, the black plastic is the throttle body position sensor, and the non-anodized metal components are what links to a cable provided by driver controls to actuate the throttle. This throttle body was custom machined in house for use on the SAE car several years ago and is of relatively high quality and durable design. It is also pretty light weight, only .5lb with sensor.

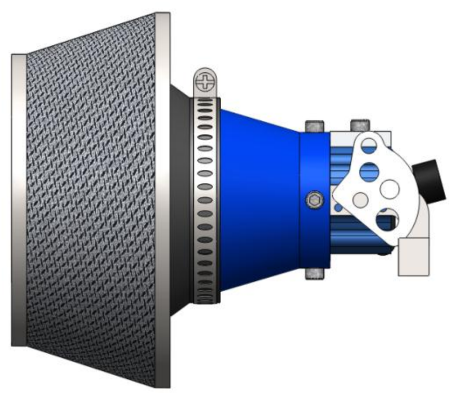
Note: The two holes on the top of the blue body are for a vacuum sensor which can be installed and hooked up to the ECU if desired in the future. For now these holes will be plugged with a custom part as shown below…





Velocity Stack/Bellmouth Validation: Per our customer's request a velocity stack had to be designed to smoothly transition from the new air filter to the throttle body. The velocity stack, also referred to as an intake bellmouth or funnel, starts with a large area at the filter and smoothly works its way down to the diameter of the throttle body. This reduction of area accelerates the air, thus dropping its pressure and easing its journey through the restrictor. This added speed through the throttle body and restrictor will also help to build more pressure in the plenum to further increase engine responsiveness. The designed velocity stack is shown below…





Due to budget and time limitations the velocity stack ended up being 2 steel pipe reductions placed together to transition from the 2in throttle body to the 3in filter.

Restrictor: The next item that was considered was the 20mm restrictor required by the SAE rules. This section that attaches to the throttle body, tapers down to the 20mm diameter, and expands back out into the plenum is the first of several that make up the main body of the intake.

Intake Support: There will be a short structural member attached to the frame via bolt and that supports the restrictor as it is the thinnest area of the intake can be secured with a simple hose clamp. This support will greatly reduce if not eliminate structural stress of the intake which is not designed to be a force carrying component.

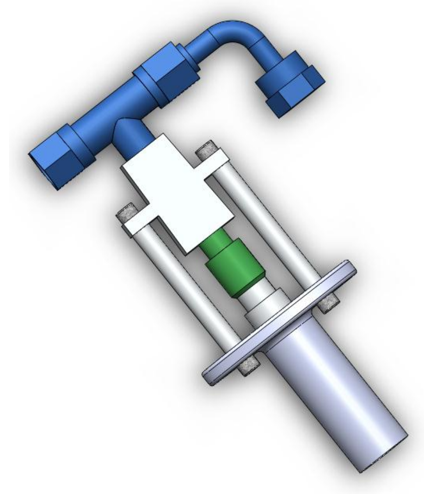
Prototype Concept Validation: Once past the restrictor there are only 2 main components of the intake left; the air plenum and runner. These also happen to be the biggest components and have by far have the biggest effect on engine performance. Some relatively common practices in performance air systems are the use of baffles to change the effective length of the runner and moving internals to change the effective volume of the plenum at different rpms. Such devices and design features are considered outside the scope of our project this year but may very well be perused in future years. The interior of all components will be as smooth as possible to reduce frictional surface drag and ease air flow.

For this year, the goal is to design an intake with a plenum volume and runner length that provide good engine power and responsiveness over the whole range of rpms. We will not be seeking peaks where power is amazing in a small rpm range and terrible everywhere else. It has been determined that the 09-10 car can put out ~39hp and ~52.88 ft-lbs of torque. Using one of the excel spreadsheets from Dr. Timmins Powertrain Theory class it was determined that the 10% increase in volumetric efficiency, that should correspond to a perfectly optimized intake, would yield gains of 3.9hp and 5.28ft-lbs.

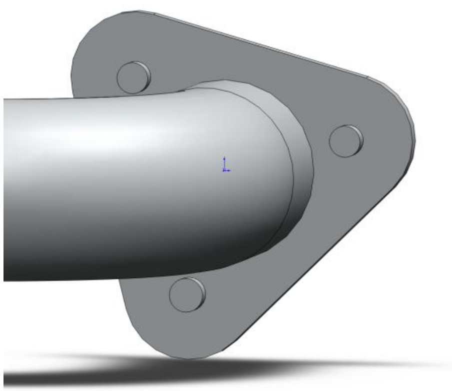
Without any more specific data it is very hard to complete any resonant frequency or acoustic wave calculations which would normally be done for intake design. This lead to the decision that the best way to get the proper diameters and lengths (volumes) of the plenum and runner would be experimentally. Both of these components have been designed to be expandable over a certain range so the car can be set up on a dyno and ever combination can be tried. For each arrangement the engine will be given a basic tune and data will be pulled at 30, 50, 70, and 100% open throttle as to not miss any disturbing peaks in performance. 3D plots can then be generated with runner length vs. plenum length vs. horsepower or torque. Once the basic sizes are determined both *SolidWorksFlowWorks 2010* and *Fluent* will be used to perfect the final intake design before it is presented as a deliverable to the SAE club for manufacturing. With actual data collected proper calculations will also be able to be done to confirm theoretical performance.

Expandable Plenum Validation: The plenum on the final prototype is adjustable from 2.5*511cc to 5*511 cc via a set of PVC inserts. The 09-10 plenum was 2.5 times the displacement and all research indicated that the plenum should be larger than that. Mr. Schwarz recommended the volume be ~3 times the displacement.

Straight Expandable Runner/Primary Length Validation: The runner, or primary length, is the tube that connects the plenum to the engine. In general, a longer runner should be used to maximize low-speed torque and a shorter one should be used to maximize (or at least prevent from decreasing) high speed torque. The adjustability on the new prototype spans from 11.3 to 17.3in. The 09-10 runner was 13.3in long.



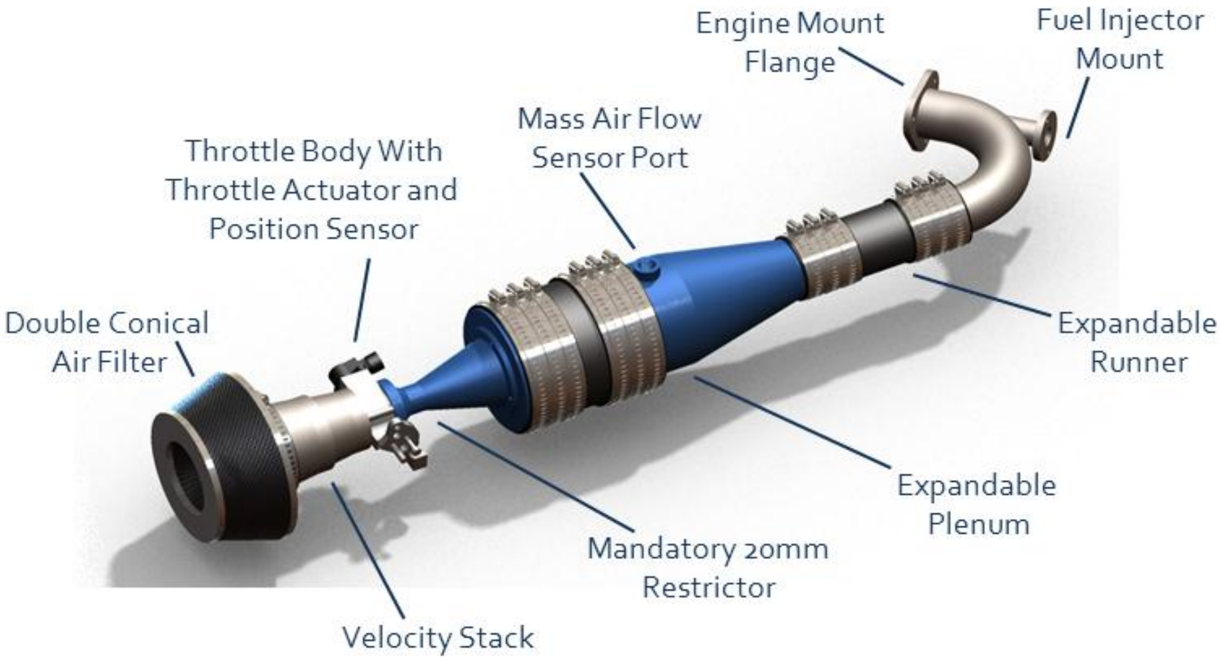
Runner/Primary Length Curve: When making this bend, there are two critical considerations that seem to have been disregarded on the 09-10 car. The first is that the bend should allow the intake to get away from the hot engine as quickly as possible, instead of wrapping around it. The second is that the bend should be as gradual as possible as to not restrict flow. Internally, the different sections of the intake need to be connected smoothly. The engines fuel injector (shown right) sprays fuel into the runner tube. Located just before the fuel injector, a small hose must run from the intake to the fuel pressure regulator. This provides the regulator with the "ambient" pressure local to where the fuel is injected so it can compare fuel pressure to that instead of the true ambient pressure outside of the car. It is also the location of the fuel pressure regulator local fuel injector ambient reference point and the fuel injector itself.



Engine Mounting: Again, very similar to the 09-10 car arrangement three bolts will be used to hold the intake firmly to the engine head. A rubber gasket will also be utilized to prevent air and fuel leakage and hamper the transfer of heat and vibration from the engine into the intake.

Completed Prototype:

The prototype was successfully manufactured. Due to time and resource constraints it was largely made form donated materials and components found in the SAE shop. *See appendix for machinist's drawings of custom components.*



Path Forward: The intake's path forward into winter session is pretty clear. First the 09-10 car must be tuned up and the experimental dyno day must take place. The data from that can be examined with special focus on…

1. Power Curve - power (kW) vs. rpm for a particular engine load
2. Torque Curve - torque (Nm) vs. rpm for a particular engine load
3. Brake-Specific Fuel Consumption (bsfc) - the mass of fuel consumed in g/kWh at a particular engine speed and load

This should yield an approximate ideal plenum volume and runner length with can be used for further optimization. These experimentally found values must be used along with CFD software and hand calculation to design and optimize the final air intake that will be built for the 2010-11 car. A basic CFD run for the 09-10 intake can be see below…

Average Engine Air Draw = Avg. 7500 rpm/2 for double cam rotation/4 for only want intake stage out of 4 step cycle = 937 breathes/min/60s = 51.61 breathes/sec

511cc intake 15.61 times/sec = 8120 cc/s*1.2kg/m^3 ➔

Average Mass Flow into the Combustion Chamber = .009744 kg/s

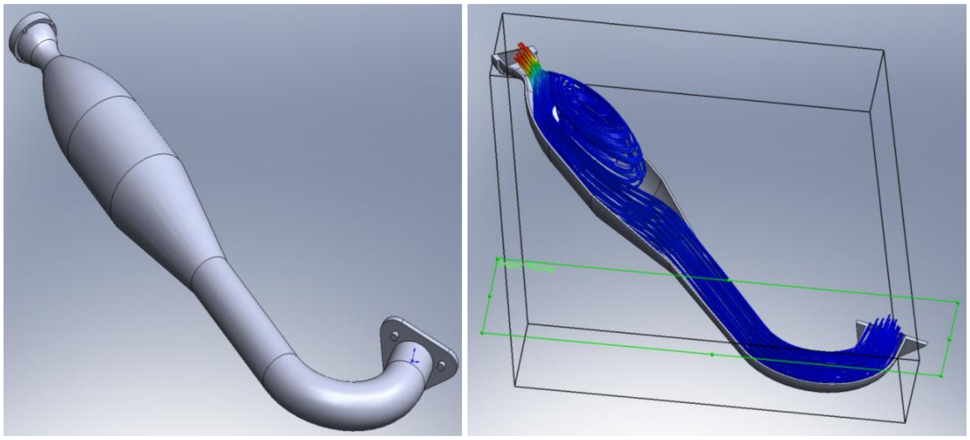


Figure 15: SolidWorks CFD simulation of existing intake (from 09-10 UDSAE car).

The final intake, to be manufactured over winter session, will have the same PVC and aluminum construction as the 09-10 intake. Once it is assembled it can be installed on the car and its performance can be observed.

Exhaust

Change of Scope

As previously mentioned, the scope of the air system has changed due to the unavailability of a dynamometer and complications to the test vehicle. Therefore, it will be impossible to physically test for horsepower performance gains in the exhaust before the end of the semester. Because the air plenum volume is subject to change, as well as the overall engine tune, a final design for the exhaust is not possible. However, an expandable exhaust is manufactured, which can function completely as an exhaust system for the new car. Further, we will show in the coming section that the weight and noise requirements have been validated, and a balance between performance and noise can be made once a final air intake volume is set, and the engine is tuned.

Brief Description of Exhaust

The test exhaust system was made using a single header pipe that connects the combustion chamber to the muffler. The largest factor taken into account with the exhaust system was the cost; all available free tubing was used during the design process, which included one stainless steel J-bend tube. This tube was cut and welded into a shape that would use less bends than the previous design (to improve flow characteristics) while still producing less than 110 db of sound, per the FSAE rules. Outer Diameter and thickness were set at 1 5/8" and 16-gauge, respectively; because of the free material we were given. The main parameter was changed in our design is a length of straight stainless steel tubing running to the muffler. The overall length of the exhaust runs from 27" TO 35". For assembly instructions for the exhaust, see Appendix.

The test exhaust is made up of two pieces and is tightly connected by a stainless steel band clamp. Of these two pieces and the band clamp, there are has eight major components;

Piece One

1. Collar – Machined with mild steel to interface with the combustion chamber and the header. It is press fit and TIG welded into the beginning of the exhaust header.
2. Flange – A simple steel flange to hold the exhaust system securely to the engine. This was taken from the old exhaust system.
3. Header – This 1-5/8” OD tube runs from the collar to the transition. The length of the header will determine the power and noise characteristics of the exhaust system.

Piece Two

4. Transition –A simple 1/16” wall thickness, 2” to 1-3/4” Transition. The 1-3/4 side of the transition slides over the 1-5/8” header, to raise or lower the total length of the exhaust system. The 2” side is welded to a 3-1/2” length, 2” OD, stainless steel tube, which is welded to the muffler (this 2” tube was out into the system to match the previous exhaust design). The transition is made of mild steel and was one of our two purchases made in this system, which cost $4.50.
5. O2 Sensor Mount – This is an 18 mm pipe fitting which houses the O2 sensor. It is located on the wall of the 2” tube, 15 degrees offset from the centerline. The O2 sensor mount was created by purchasing a spark plug housing (whose threads fit the O2 sensor), and modifying in order to create accurate measurements of the Air: Fuel Ratio. (Not on car)
6. Muffler – The muffler was taken off the previous car. It is a FMF Q4 muffler, which was highly recommended from members of the FSAE club to be placed on the new car. The muffler is welded to the 2” tube and includes a flange that is connected to the chassis. It is the largest, most noise cancelling muffler that is freely available for Team Powertrain to use.
7. Muffler Mounting System – This mounting system takes advantage of the previous muffler mount and ads a piece to smoothly facilitate lengthening of the exhaust system.

Metrics

The primary focus of the metrics of the exhaust prototype has shifted away from performance, and towards exhaust noise. Because our dyno day at Injection connection was cancelled, there is no way to validate performance gains in horsepower and torque until after the semester is over.

A single noise trial was made at a short setting of the exhaust system, where the exhaust was set to a main tube length of 30”. At 10,500 RPM, the engine made 120 dB, fast weighting. While the reading would indicate that we have failed the noise test, many outside factors may have negatively influenced its outcome;

a. The test was conducted in the outside parking lot with large buildings around it, so sound may have bounced back and influenced the readings. Traveling to a proper location to test for noise with the car could not be achieved in the past few days, once the exhaust was completed.
b. The muffler has not yet been repacked, which would definitely lower the sound reading. New muffler gauze is available to us in from the FSAE club and once it is repacked, noise level will go down.
c. The final air intake and final engine tune have not yet been finalized. Therefore, any noise targets that are made at this point will not stay the same after those systems are finalized.

More importantly, the reading was taken can show that the exhaust system was fully functional and in the general range of decibel values that would pass the noise test. Further testing in the winter term will conclude the validation of this noise metric.

Validation

Envelope

The FSAE rules govern the furthest positions outside the car that the exhaust can protrude. The exhaust cannot be more than 17.7 inches behind the centerline of the rear axle, and cannot be be higher than 23.6 inches above the ground.

At the furthest extension, the axles are at a height of 21.5" above the ground, and they are 16" behind the centerline of the axle. So, the exhaust system satisfies the requirements for enveloping for competition.

Noise

The exhaust prototype can be used to test for noise and find a theoretical relationship between primary exhaust tube length and engine noise. To ensure the exhaust prototype will pass the noise test, the new exhaust prototype was modeled after the previous exhaust system, which passed the noise test. Many attributes were carried over to the prototype design to mimic the previous system, which was successful. Although constrained by costs, all of the materials that were used in the manufacture of the new exhaust prototype were free of charge, and provided to our group by the FSAE club at Delaware. The muffler, initial length of the primary tube, and copper insulating wrap were all taken from the old design and implemented in the new exhaust prototype. Accounting for any new flow gains from the intake and new engine tune, the exhaust prototype can extend up to seven inches longer than the previous system, to ensure that the noise requirement will be met.

Path Forward

In order to perform the proper noise test, the muffler gauze first needs to repacked, which will lower sound readings. Then an O2 sensor bung needs to be welded into the end of the main tube, in order to properly measure the air:fuel ratio during dyno testing. Finally, a stainless steel band clamp needs to be purchased to create an airtight seal between the two sliding exhaust pieces.

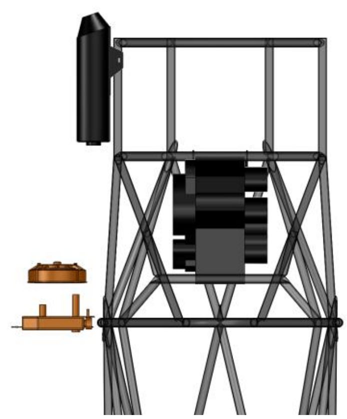
After these items are completed, there will be preliminary dyno testing done in the first week of winter session, where an air intake volume and engine tune will be finalized. For this initial dyno testing, the furthest length of exhaust will be used, and baseline performance data will be collected for this length of exhaust main tube.

After the dyno day, the car will be taken to an open area, and iterations of noise tests will be done, decreasing the length of the main tube after each trial, until the noise readings hit 110 dB. Once this location is set, a second dyno day will prove any performance increases associated with the change in the main tube.

The new exhaust prototype uses stainless steel tubes and one temporary mild steel transition. The stainless steel will hold not rust and be reliable for the competition, and once a final primary length is set, the galvanized steel transition will be replaced with a stainless steel one.

Project Aspect 3: Cooling

Description



The cooling system of the car is responsible for preventing engine seize and overheating. The system involves a radiator, fan, and coolant. Our team has successfully selected the components to be used in the system and proved that their performance on the car will surpass our metrics.

Constraints

The following constraints had to be taken into account when designing the cooling system for the 2011 FSAE car:

1. Must only use water through radiator as coolant
2. Hot coolant must be shielded or insulated from direct contact with the driver
3. Radiator pull fan must not drain battery when running with all other vehicle electronics
4. Radiator must be oriented perpendicular to the air flow
5. System must have sealed ducting
6. Assembly must fit within chassis design

Final Product

The final cooling assembly is focused around 2 main components: radiator and fan. The radiator is from a 2004-05 Yamaha YFZ450 ATV. The fan is a 7”-diameter pull fan from Derale Cooling Products. 1” OD rubber heater hose connect the inlet and outlet radiator ports to their respective ports on the engine. The pressure-relief valve on the radiator is not being utilized—the coolant passes through a pressure-relief valve attached to the engine. The radiator is currently bolted to the right side of the chassis via the stock radiator tabs and chassis tabs.

Actual photo of fan on chassis

ares

Actual photo of radiator on chassis

res

Inlet and outlet hoses into engine from radiator

Co
mp
arison to 2009-10 Cooling System

	Old System	New System	Improvement
Radiator Core Length	7.1 in	7.4 in	+ 0.3 in
Radiator Core Width	9.2 in	11 in	+ 1.8 in
# of tubes	21	18	---
Fan Amperage Draw	unknown	4.8 A	---
Inlet Radiator Temperature	138 F (332 K)	126 F (325 K)	- 12 degrees
Outlet Radiator Temperature	115 F (319 K)	97 F (309 K)	- 18 degrees

The new radiator features an increased core surface area—that is, the area of tubes that actually comes in contact with the moving air is increased (24.6% larger). The new system also not only lowers both the inlet and outlet temperatures but produces a large difference between the 2 temperatures. This results in more heat being dissipated.

Validation of Metrics

	Metric Target Value	Old System	New System	Savings
Cost	< $15	unknown	$0	meets metric
Total Weight	< 5 lbs	4.75 lbs	4.25 lbs	0.5 lbs (12%)

Volume Space	< 2,496 cubic inches (8"x13"x24")	500 cubic inches (10"x10"x5")	1,465 cubic inches (7.4"x11"x18")	meets metric
Amperage Draw	< 5 A	unknown	4.8 A	meets metric
Heat Dissipated	> 17,000 W	16,861 W	20,753 W	3,892 W (23%)

As shown in the preceding table, the new cooling system meets all of its metrics: cost, weight, volume, amperage draw, and heat dissipated. It also weighs less and dissipates more heat than the old system. The assembly will fit within the space specified by SAE Team Chassis and the electric fan draws an acceptable amperage according to SAE Team Driver Controls.

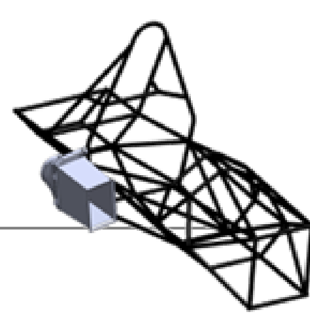
Final Cost Analysis

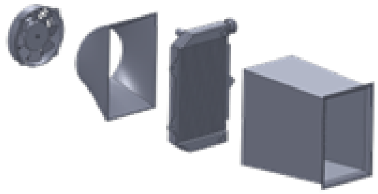
As shown in the table below, the final cooling system fell well within the budget allotted to it. In fact, development of the assembly incurred no financial cost to the project. All components were either donated or already available in the SAE stock room.

Item	Cost	Notes
(1) 2004-05 Yamaha YFZ450 radiator	$0.00	Available from UD SAE
(1) 7" Tornado	$0.00	Available from UD SAE
(6 feet) rubber heater hose	$0.00	Donated by Michael Honeychuck
(4) 1" hose clamps	$0.00	Donated by Michael Honeychuck
TOTAL	**$0.00**	**Within $15 budget**

Path Forward

The only major manufacturing left to be completed for this aspect of the project is radiator ducting. Our team has design drawings and dimensions that we recommend using when developing the sealed ducts (attached in Appendix). This addition of ducting is what will make the electric fan a permanent component of the cooling assembly. Currently only the radiator is bolted to the chassis—there are no mounting tabs for the fan. Additionally, because of the distance between the radiator and fan, the fan is only effective with some type of sealed ducting connecting it to the radiator. Manufacturing will take place over Winter Session 2011. The entire cooling assembly will be bolted to the right side of the chassis as shown in the figure below.





Project Schedule:

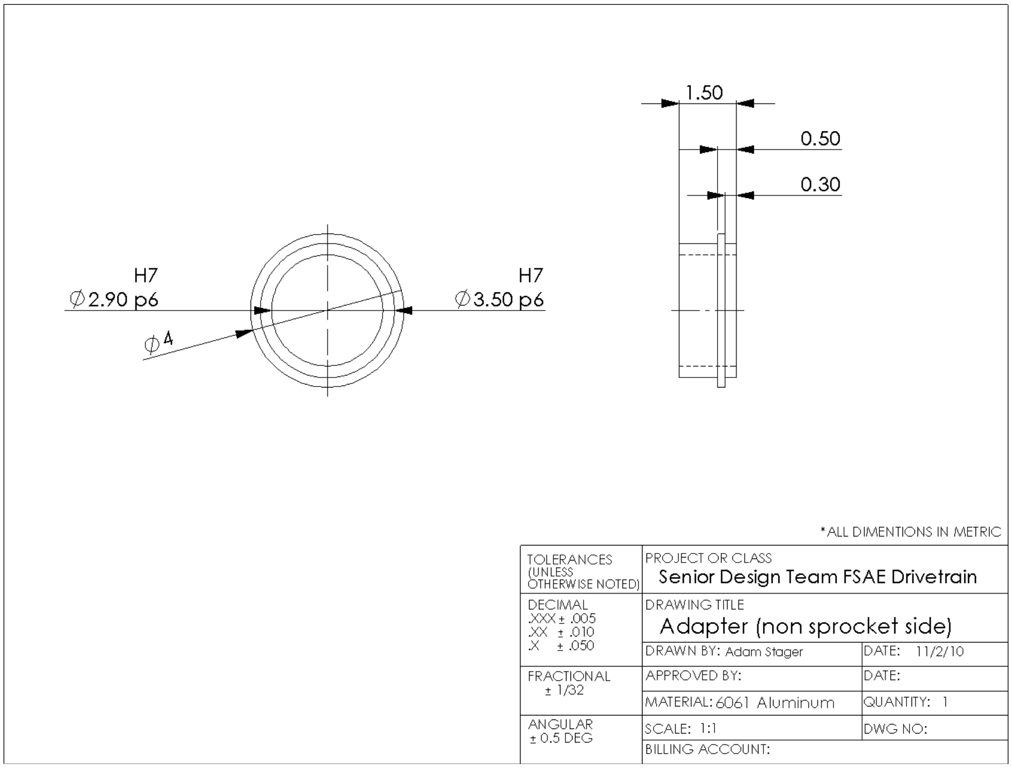
For oversized project schedule, please visit the FSAE Powertrain Google Docs Folder

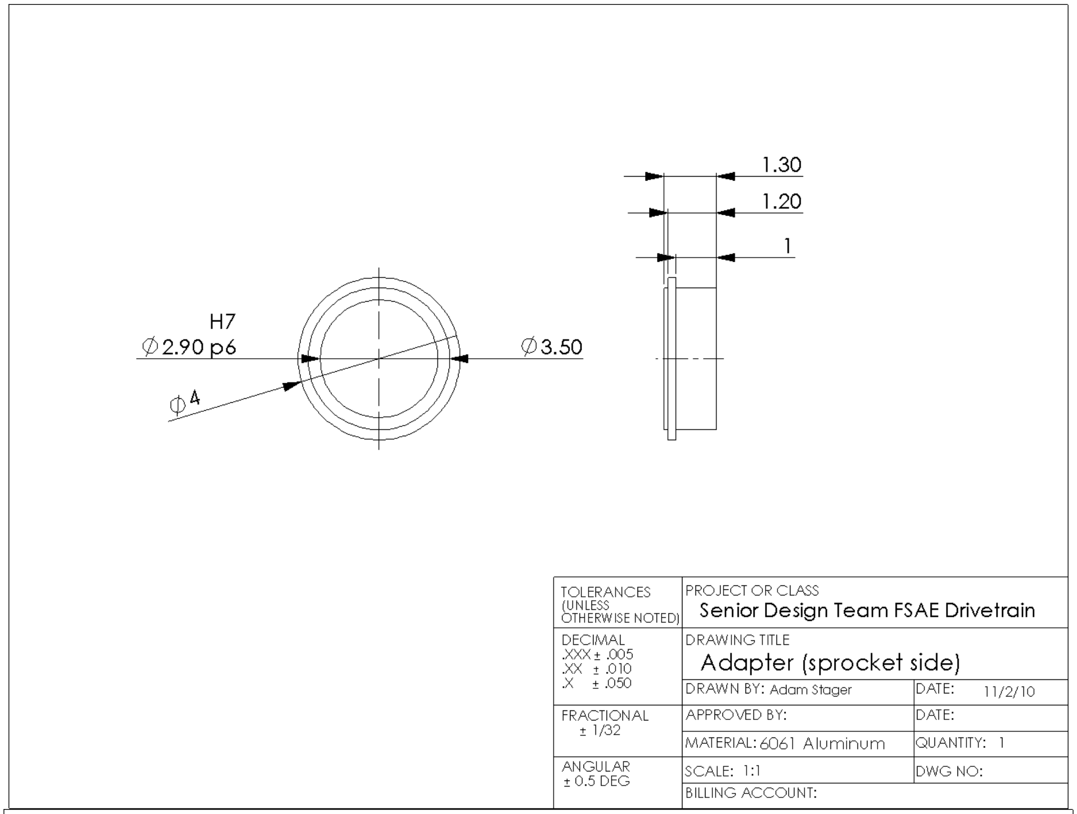
Budget:

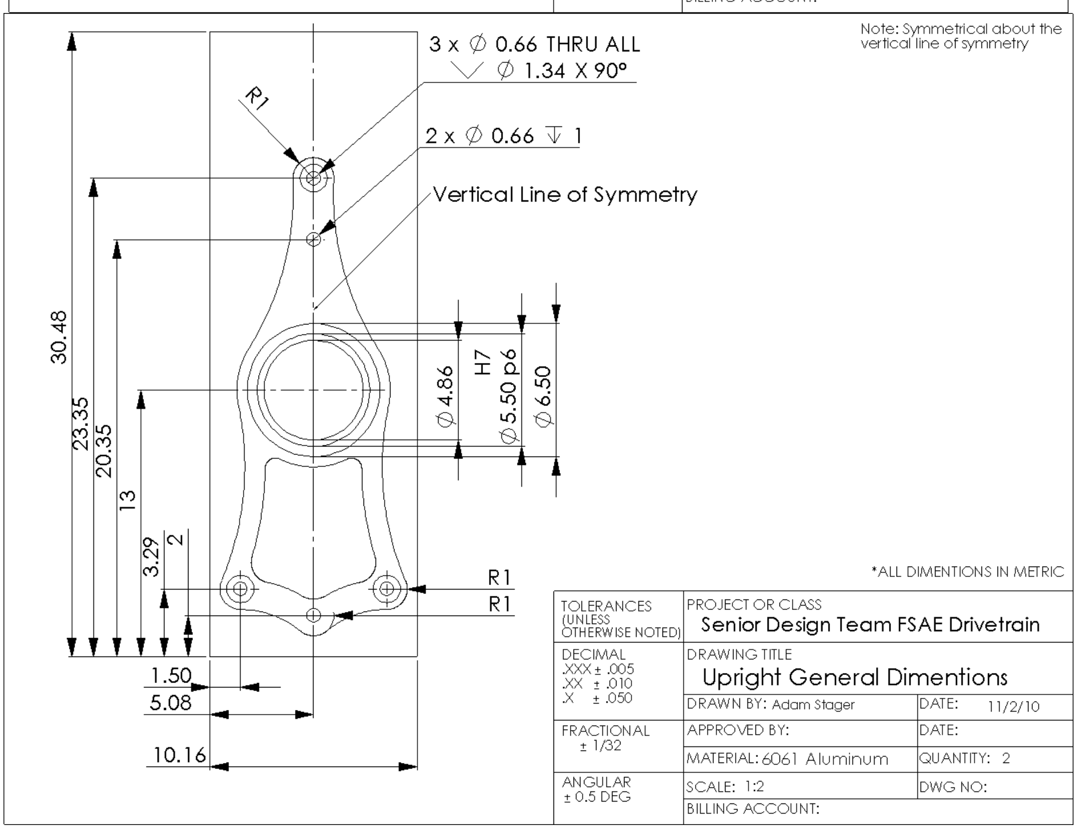
For oversized project budget, please visit the FSAE Powertrain Google Docs Folder or Sakai log book

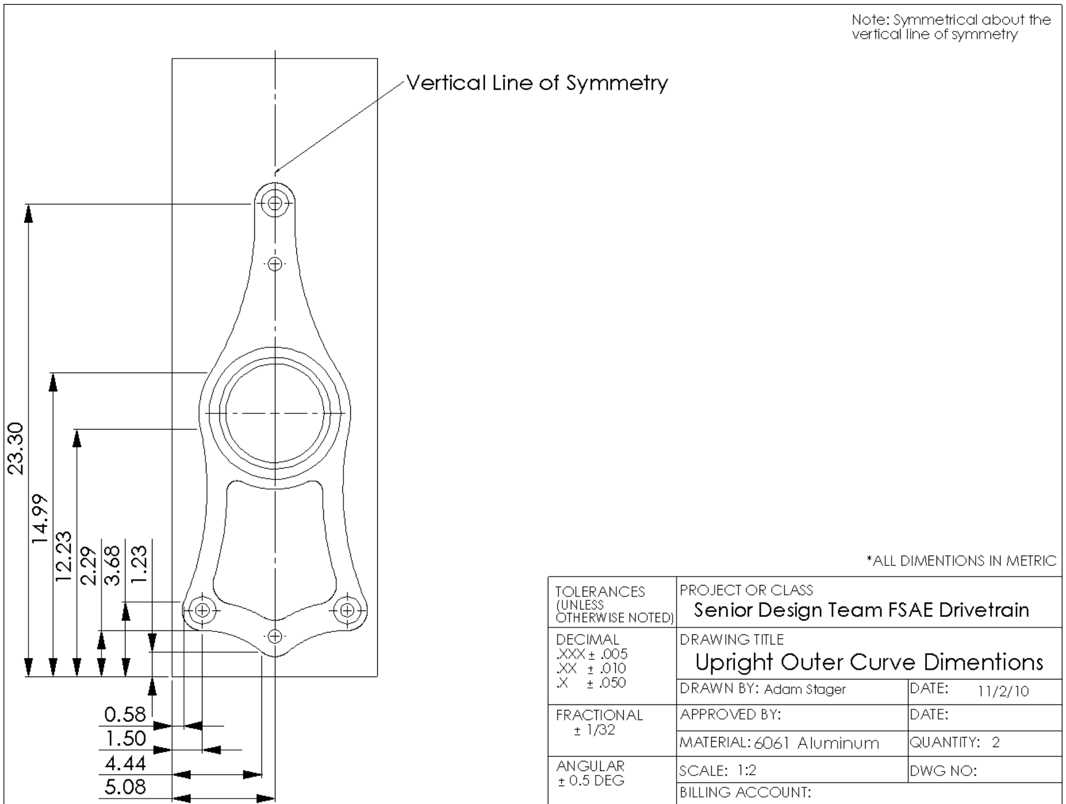
Note: For actual manufacturing full page machinists drawings will be printed out. The smaller images of the drawings in the following appendices are primarily meant to help reinforce which parts will be manufactured in house and how that will be accomplished.

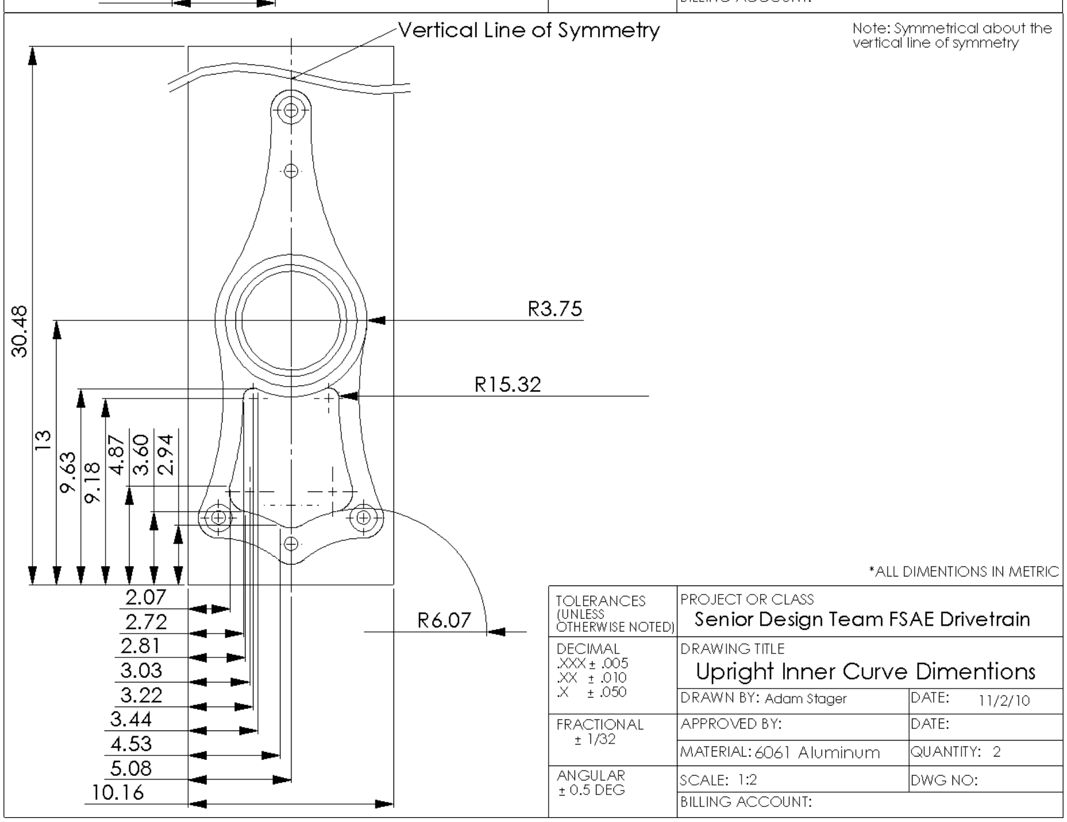
Appendix A (Differential Machinists Drawings):

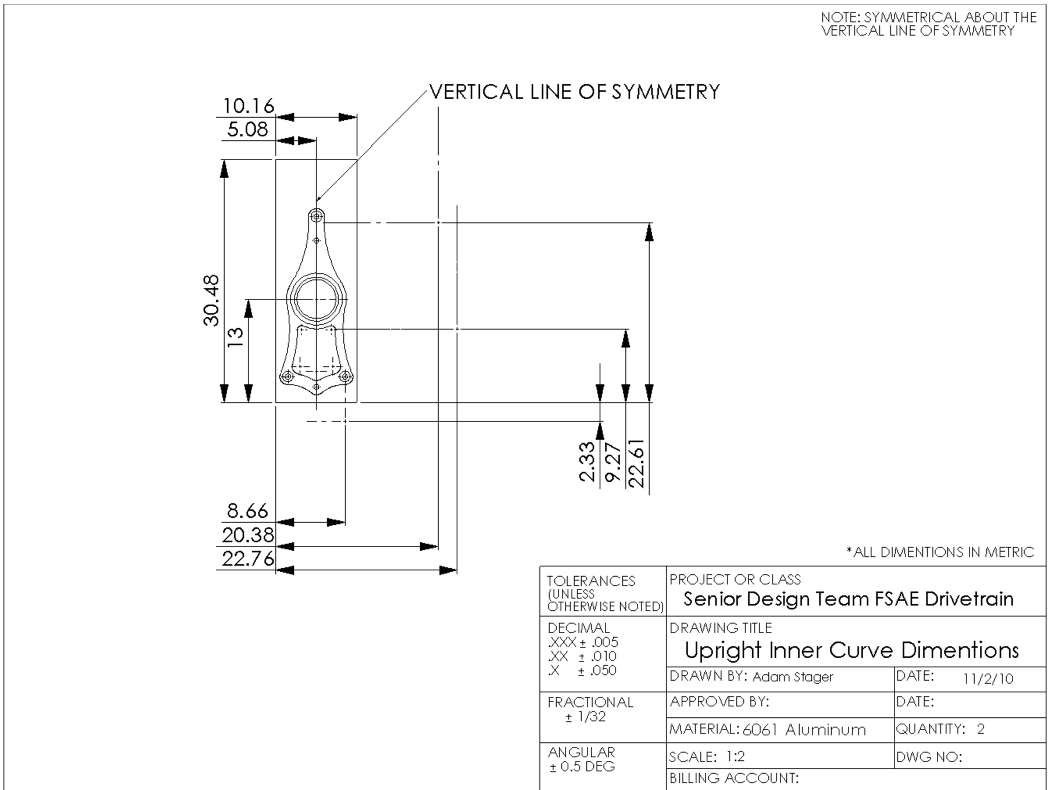


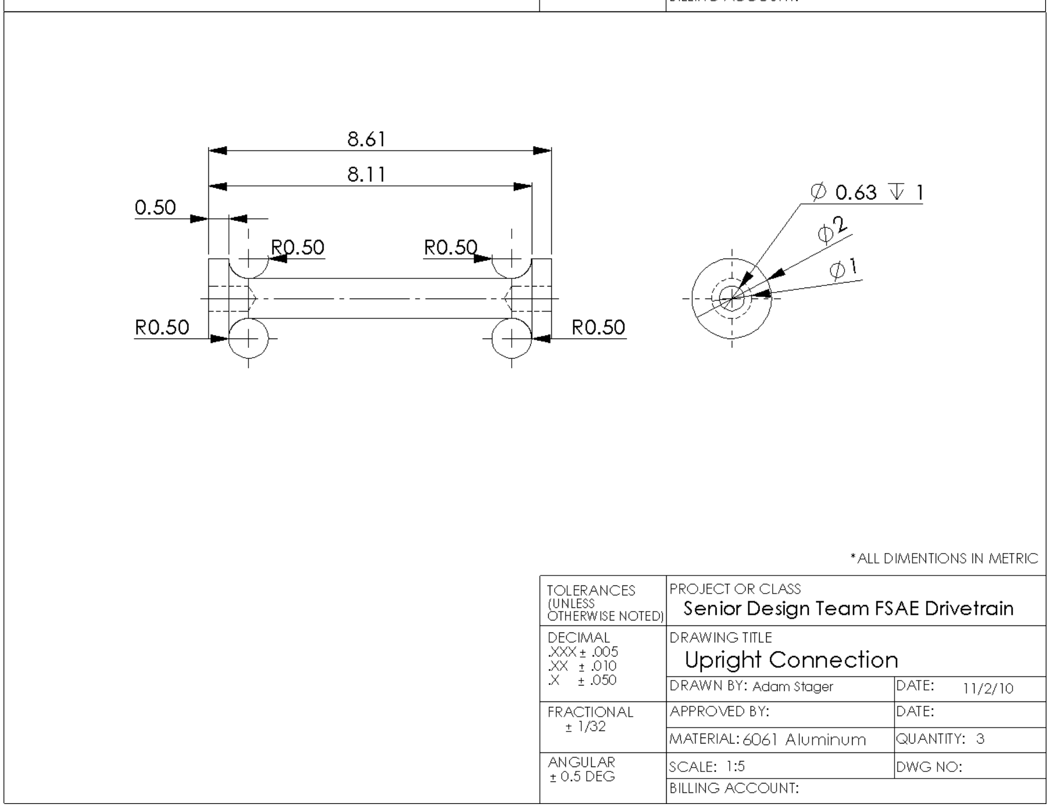

1.30
1.20
1
H7
∅2.90 p6
∅3.50
∅4
TOLERANCES (UNLESS OTHERWISE NOTED)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS
Senior Design Team FSAE Drivetrain
DRAWING TITLE
Adapter (sprocket side)
DRAWN BY: Adam Stager
DATE: 11/2/10
APPROVED BY:
DATE:
MATERIAL: 6061 Aluminum
QUANTITY: 1
SCALE: 1:1
DWG NO:
BILLING ACCOUNT:

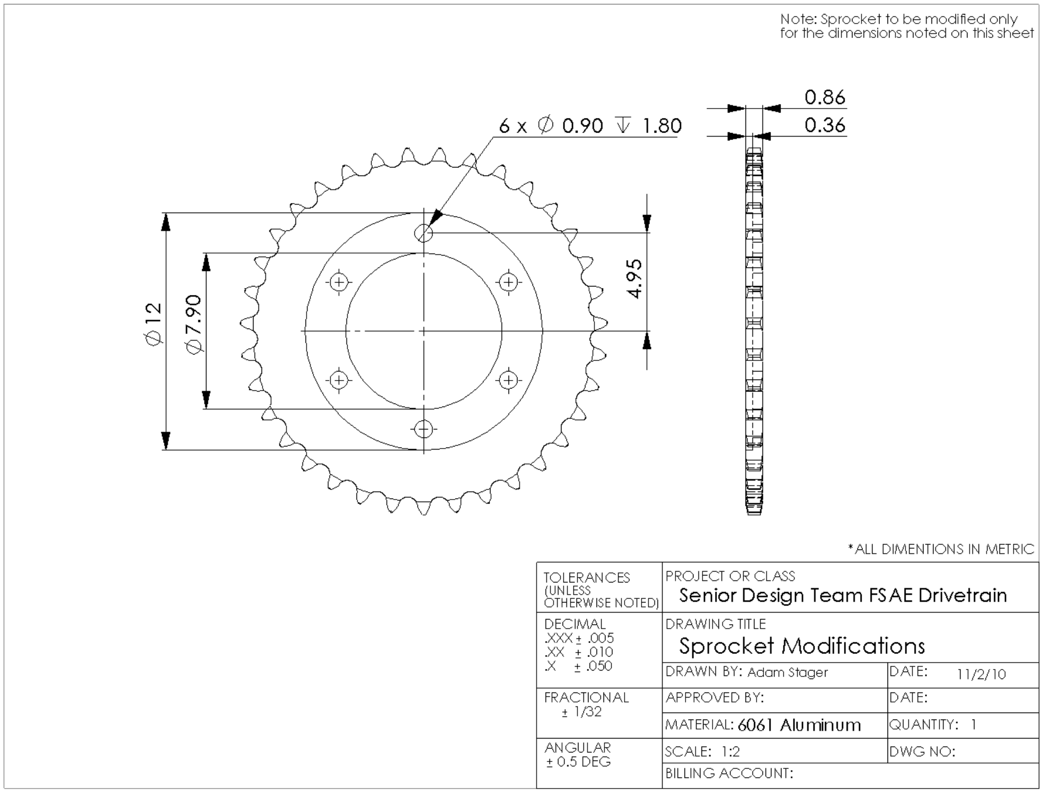

3 x ∅ 0.66 THRU ALL
∅ 1.34 X 90°
R1
2 x ∅ 0.66 ↧ 1
Vertical Line of Symmetry
Note: Symmetrical about the vertical line of symmetry
30.48
23.35
20.35
13
3.29
2
∅4.86
H7
∅5.50 p6
∅6.50
R1
R1
1.50
5.08
10.16
*ALL DIMENTIONS IN METRIC
TOLERANCES (UNLESS OTHERWISE NOTED)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS
Senior Design Team FSAE Drivetrain
DRAWING TITLE
Upright General Dimentions
DRAWN BY: Adam Stager
DATE: 11/2/10
APPROVED BY:
DATE:
MATERIAL: 6061 Aluminum
QUANTITY: 2
SCALE: 1:2
DWG NO:
BILLING ACCOUNT:

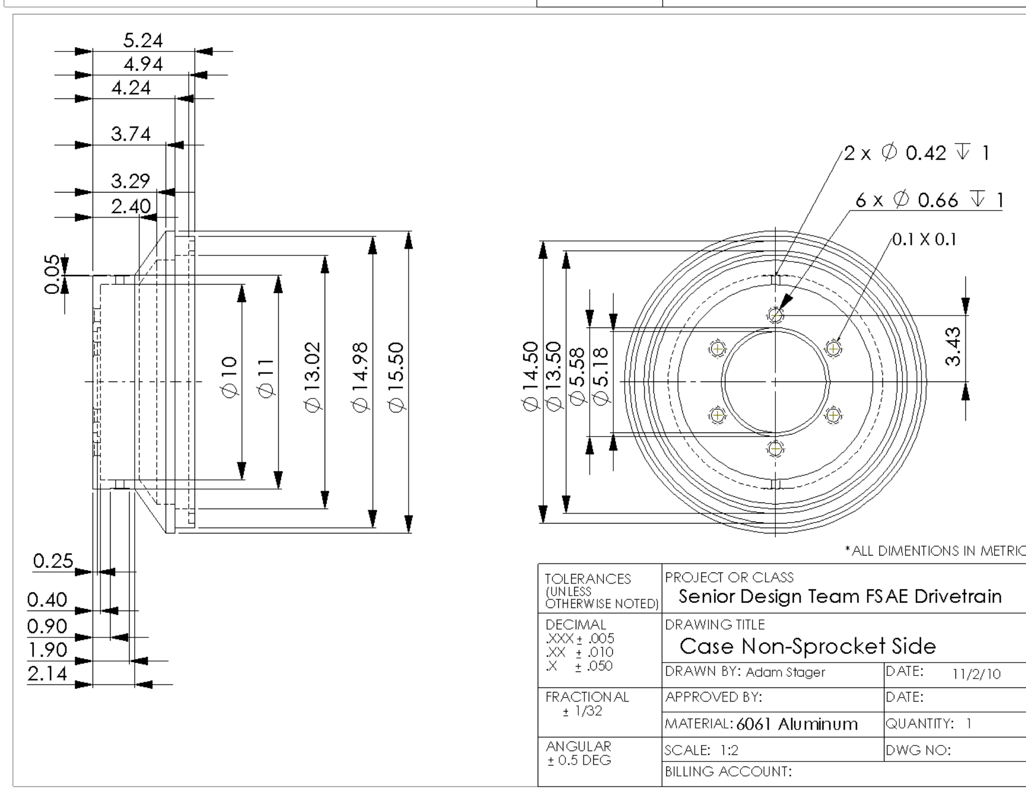

Note: Symmetrical about the vertical line of symmetry
Vertical Line of Symmetry
23.30
14.99
12.23
2.29
3.68
1.23
0.58
1.50
4.44
5.08
*ALL DIMENTIONS IN METRIC
TOLERANCES (UNLESS OTHERWISE NOTED)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS
Senior Design Team FSAE Drivetrain
DRAWING TITLE
Upright Outer Curve Dimentions
DRAWN BY: Adam Stager
DATE: 11/2/10
APPROVED BY:
DATE:
MATERIAL: 6061 Aluminum
QUANTITY: 2
SCALE: 1:2
DWG NO:
BILLING ACCOUNT:

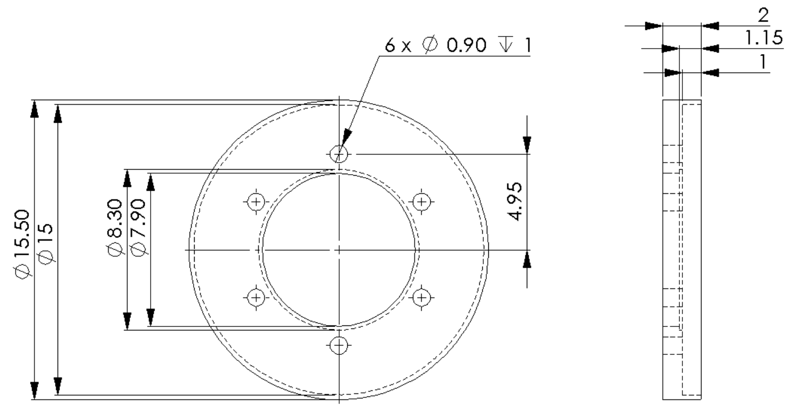

Vertical Line of Symmetry
Note: Symmetrical about the vertical line of symmetry
30.48
13
9.63
9.18
4.87
3.60
2.94
R3.75
R15.32
R6.07
2.07
2.72
2.81
3.03
3.22
3.44
4.53
5.08
10.16
*ALL DIMENTIONS IN METRIC
TOLERANCES (UNLESS OTHERWISE NOTED)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS
Senior Design Team FSAE Drivetrain
DRAWING TITLE
Upright Inner Curve Dimentions
DRAWN BY: Adam Stager
DATE: 11/2/10
APPROVED BY:
DATE:
MATERIAL: 6061 Aluminum
QUANTITY: 2
SCALE: 1:2
DWG NO:
BILLING ACCOUNT:


NOTE: SYMMETRICAL ABOUT THE VERTICAL LINE OF SYMMETRY
VERTICAL LINE OF SYMMETRY
10.16
5.08
30.48
13
2.33
9.27
22.61
8.66
20.38
22.76
*ALL DIMENTIONS IN METRIC
TOLERANCES (UNLESS OTHERWISE NOTED)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS
Senior Design Team FSAE Drivetrain
DRAWING TITLE
Upright Inner Curve Dimentions
DRAWN BY: Adam Stager
DATE: 11/2/10
APPROVED BY:
DATE:
MATERIAL: 6061 Aluminum
QUANTITY: 2
SCALE: 1:2
DWG NO:
BILLING ACCOUNT:


8.61
8.11
0.50
R0.50
R0.50
R0.50
R0.50
∅ 0.63 ↧ 1
∅2
∅1
*ALL DIMENTIONS IN METRIC
TOLERANCES (UNLESS OTHERWISE NOTED)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS
Senior Design Team FSAE Drivetrain
DRAWING TITLE
Upright Connection
DRAWN BY: Adam Stager
DATE: 11/2/10
APPROVED BY:
DATE:
MATERIAL: 6061 Aluminum
QUANTITY: 3
SCALE: 1:5
DWG NO:
BILLING ACCOUNT:

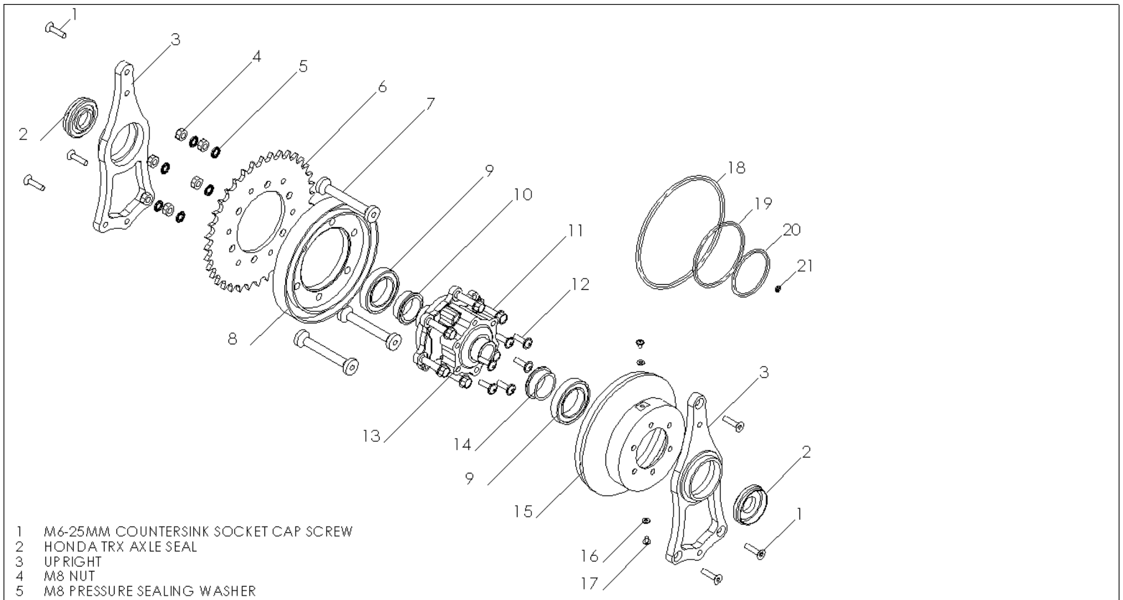

Note: Sprocket to be modified only for the dimensions noted on this sheet
6 x ∅ 0.90 ↧ 1.80
0.86
0.36
∅12
∅7.90
4.95
*ALL DIMENTIONS IN METRIC
TOLERANCES (UNLESS OTHERWISE NOTED)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS
Senior Design Team FSAE Drivetrain
DRAWING TITLE
Sprocket Modifications
DRAWN BY: Adam Stager
DATE: 11/2/10
APPROVED BY:
DATE:
MATERIAL: 6061 Aluminum
QUANTITY: 1
SCALE: 1:2
DWG NO:
BILLING ACCOUNT:


5.24
4.94
4.24
3.74
3.29
2.40
0.05
∅10
∅11
∅13.02
∅14.98
∅15.50
0.25
0.40
0.90
1.90
2.14
2 x ∅ 0.42 ↧ 1
6 x ∅ 0.66 ↧ 1
0.1 X 0.1
∅14.50
∅13.50
∅5.58
∅5.18
3.43
*ALL DIMENTIONS IN METRIC
TOLERANCES (UNLESS OTHERWISE NOTED)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS
Senior Design Team FSAE Drivetrain
DRAWING TITLE
Case Non-Sprocket Side
DRAWN BY: Adam Stager
DATE: 11/2/10
APPROVED BY:
DATE:
MATERIAL: 6061 Aluminum
QUANTITY: 1
SCALE: 1:2
DWG NO:
BILLING ACCOUNT:



*ALL DIMENTIONS IN METRIC

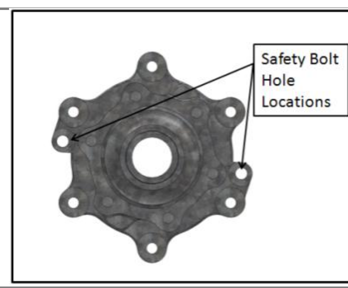
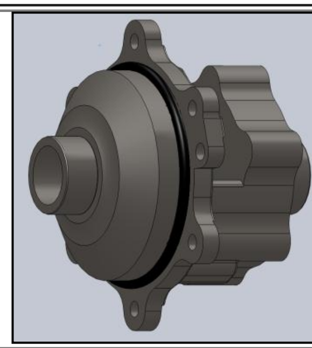
TOLERANCES (UNLESS OTHERWISE NOTED)	PROJECT OR CLASS Senior Design Team FSAE Drivetrain	
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050	DRAWING TITLE Case Sprocket Side	
	DRAWN BY: Adam Stager	DATE: 11/2/10
FRACTIONAL ± 1/32	APPROVED BY:	DATE:
	MATERIAL: 6061 Aluminum	QUANTITY: 1
ANGULAR ± 0.5 DEG	SCALE: 1:2	DWG NO:
	BILLING ACCOUNT:	

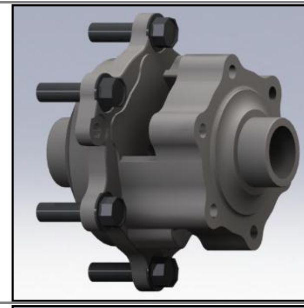
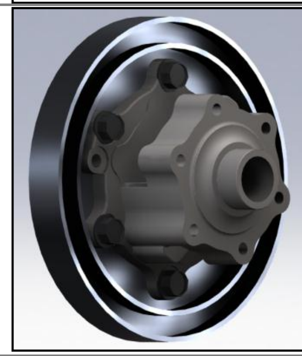
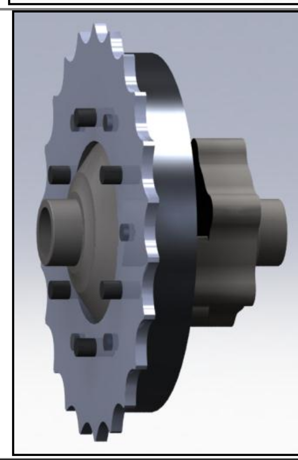
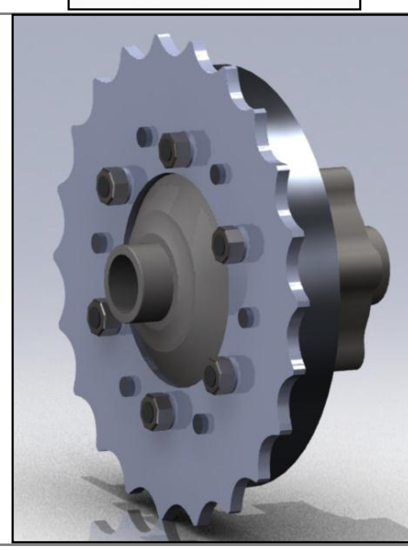


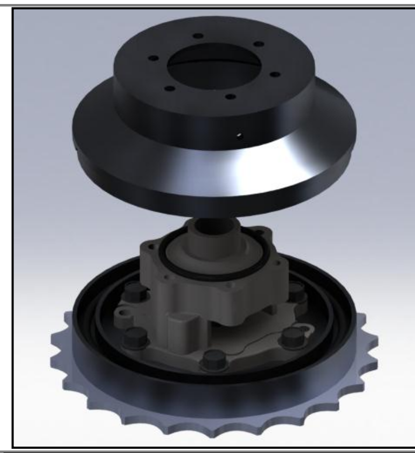
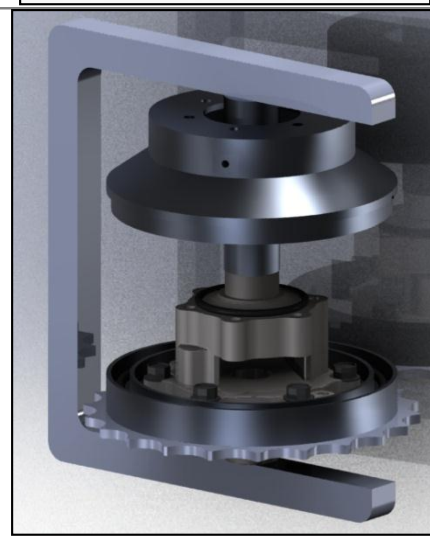
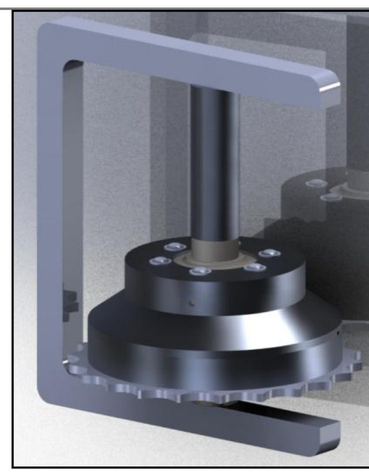
1 M6-25MM COUNTERSINK SOCKET CAP SCREW
2 HONDA TRX AXLE SEAL
3 UPRIGHT
4 M8 NUT
5 M8 PRESSURE SEALING WASHER
6 MODIFIED 8IN DIA-35 TOOTH SPROCKET
7 UPRIGHT CONNECTOR
8 DIFFERENTIAL HOUSING (SPROCKET SIDE)
9 35MM ID-55MMOD BALL BEARING
10 ADJUSTMENT SHIM (SPROCKET SIDE)
11 M8-30MM HEX BOLT WITH FLANGE
12 M6-20MM ALLEN SOCKET CAP SCREW
13 HONDA TRX DIFFERENTIAL CARRIER
14 ADJUSTMENT SHIM (NON-SPROCKET SIDE)
15 DIFFERENTIAL HOUSING (NON-SPROCKET SIDE)
16 M4 DRAIN WASHER
17 M4-6MM SOCKET CAP SCREW
18 135MM ID 145MM OD O-RING
19 78MM ID 82MM OD O-RING
20 55MM ID 59MM OD O-RING
21 5MM ID 7MM OD O-RING

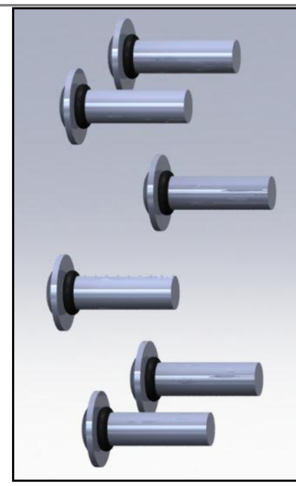
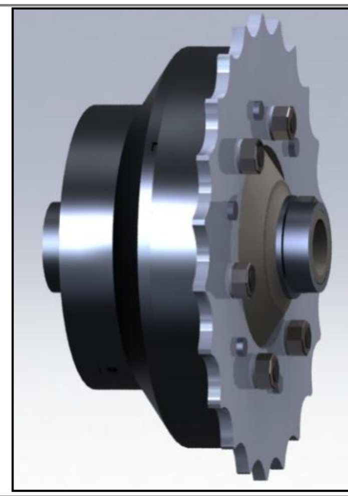
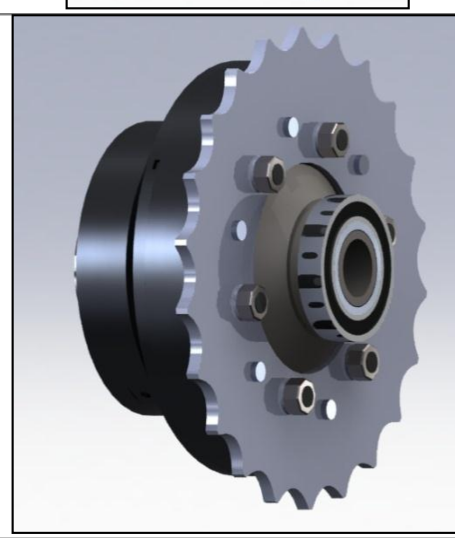
TOLERANCES (UNLESS OTHERWISE NOTED)	PROJECT OR CLASS Senior Design Team FSAE Drivetrain	
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050	DRAWING TITLE DIFFERENTIAL ASSEMBLY	
	DRAWN BY: Adam Stager	DATE: 11/2/10
FRACTIONAL ± 1/32	APPROVED BY:	DATE:
	MATERIAL:	QUANTITY: 1
ANGULAR ± 0.5 DEG	SCALE: 1:10	DWG NO:
	BILLING ACCOUNT:	

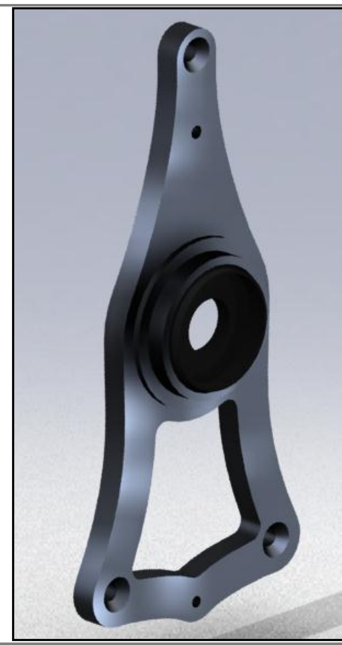
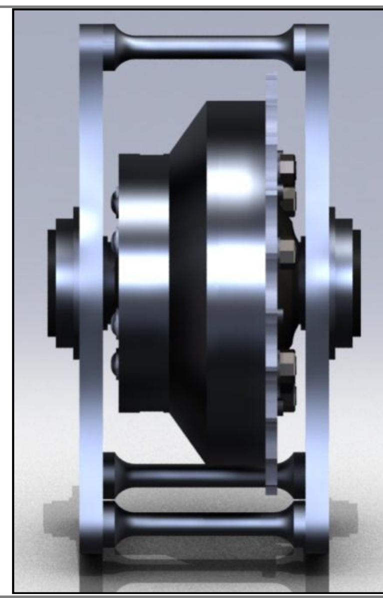
Appendix B (Differential Assembly Guide):

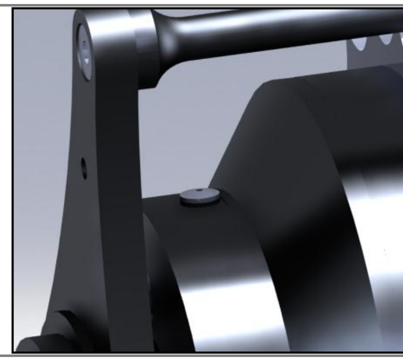
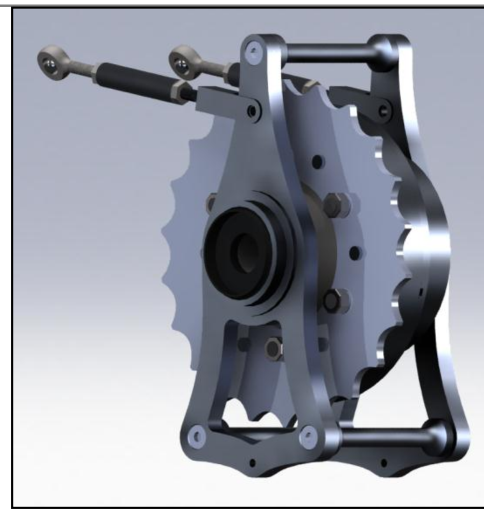
Step 1: Application of safety bolts which hold together the Honda TRX differential carrier (shown in black [1]) while the ring gear is removed	
Step 2: Apply ID78mm OD82mmO ring to the inside groove of the sprocket-side casing	

Step 3: Insert 6xBolt M8- 30mm into the Honda TRX differential carrier with the threads pointing outward as shown in [1]	
Step 4: Slide the sprocket-side case over the ends of the 6xBolt M8- 30mm	
Step 5: Slide the sprocket over the 6xBolt M8- 30mm	
Step 6: Slide washers and then nuts onto the 6xBolt M8- 30mm bolts	

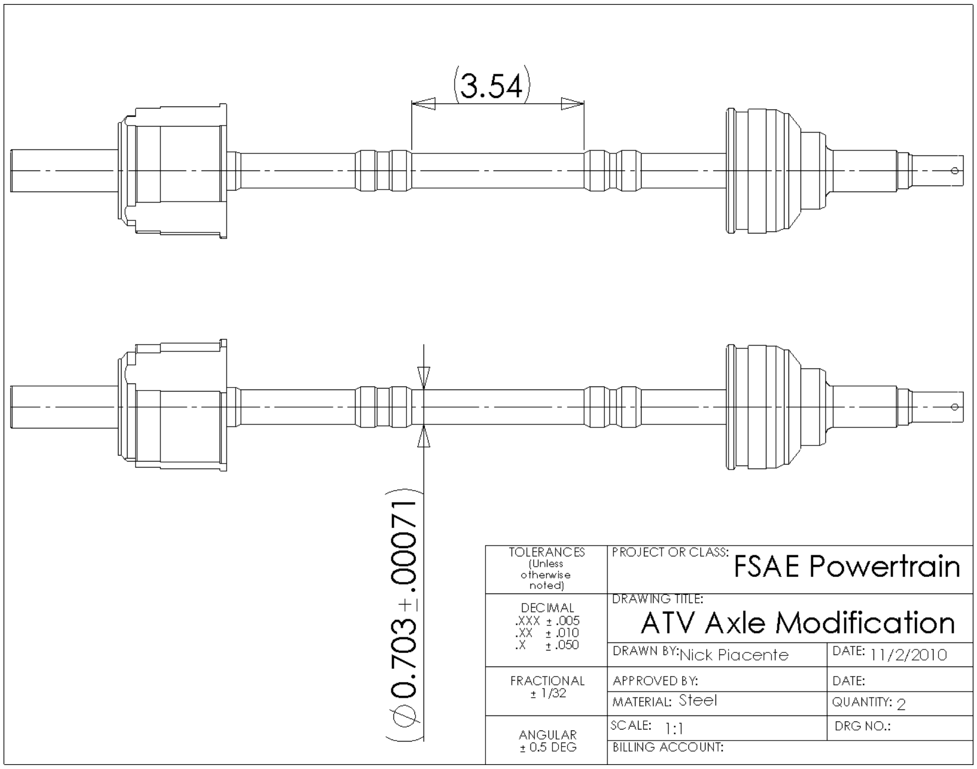
Step 7: Insert O ring ID55mm OD59mm and O ring ID135mmOD145mm into their appropriate grooves	
Step 8: Clamp the assembly using a clamp extension with the case between the clamp and pipe extension (the lathe can be used to clamp the differential)	
Step 9: Remove the bolts holding together the stock (non-sprocket side of the stock carrier)	
Step 10: Align the bolt mounting locations while sliding the case interface, making sure not to twist the case once the O rings make contact (bolts shown inserted in picture)	

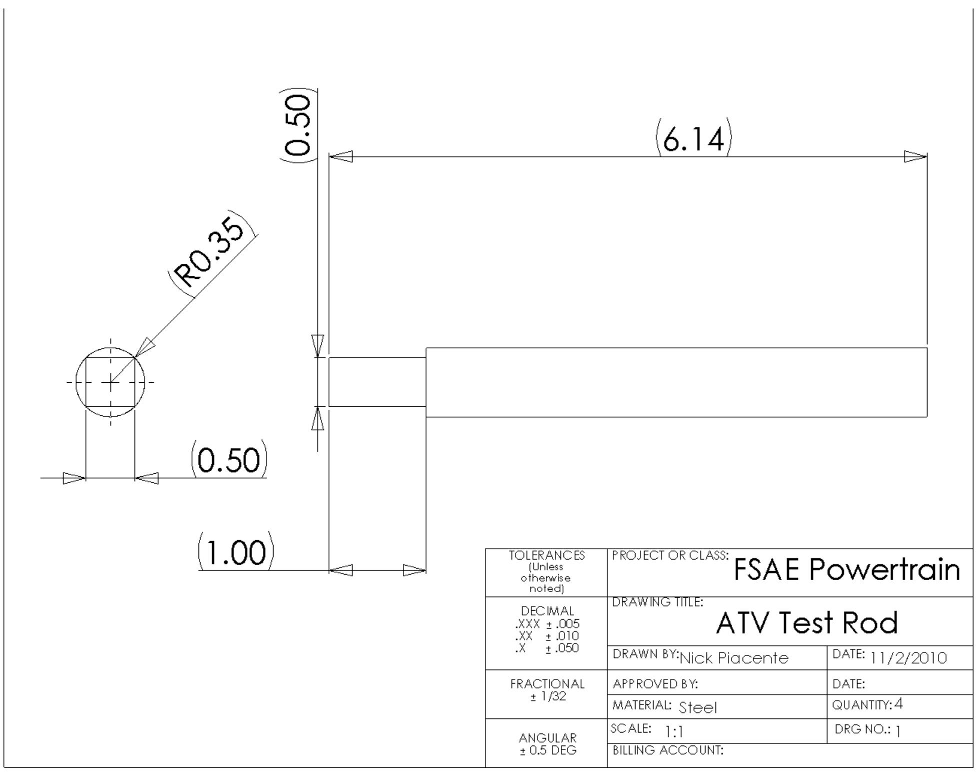
Step 11: Insert and tighten the non-sprocket side 6xBolt M6- 20mm with O ring ID5mm OD7mm lubedwith oil and pushed over the threads up to the bolt head as shown in [3]	
Step 12: Remove the clamp and clamp extension	
Step 13: Press fit the mounting adapters over each side of the carrier shown in black [1]	
Step 14: Press fit the bearing over the mounting adapters as shown in [4]	

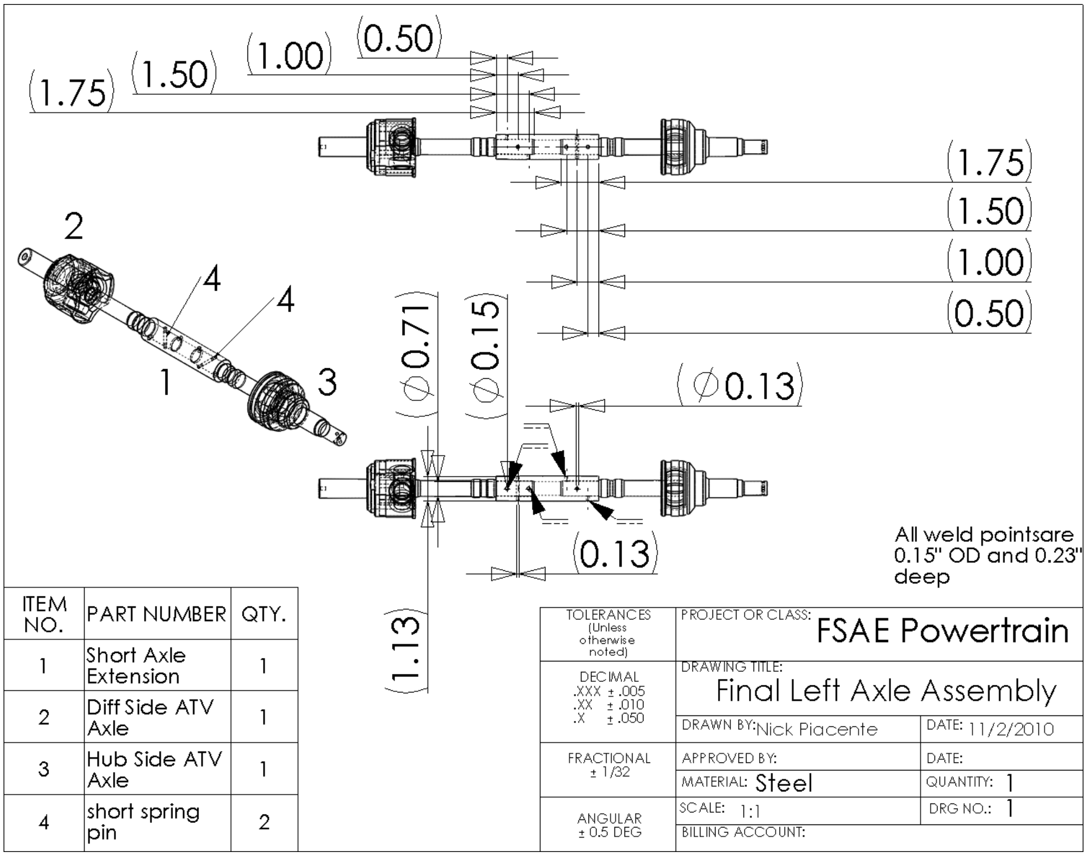
Step 15: Insert the axle seals into the smaller diameter side of each upright	
Step 16: Sandwich the uprights against the bearings	
Step 17: Slide chain over the sprocket	
Step 18: Align the upright links and 3xBolt M6-25mm as shown by [5]	
Step 19: Tighten each bolt little by little moving in a circular pattern from bolt to bolt until the bearings are fully pressed into the uprights (there should be no gap between the uprights and the upright links	

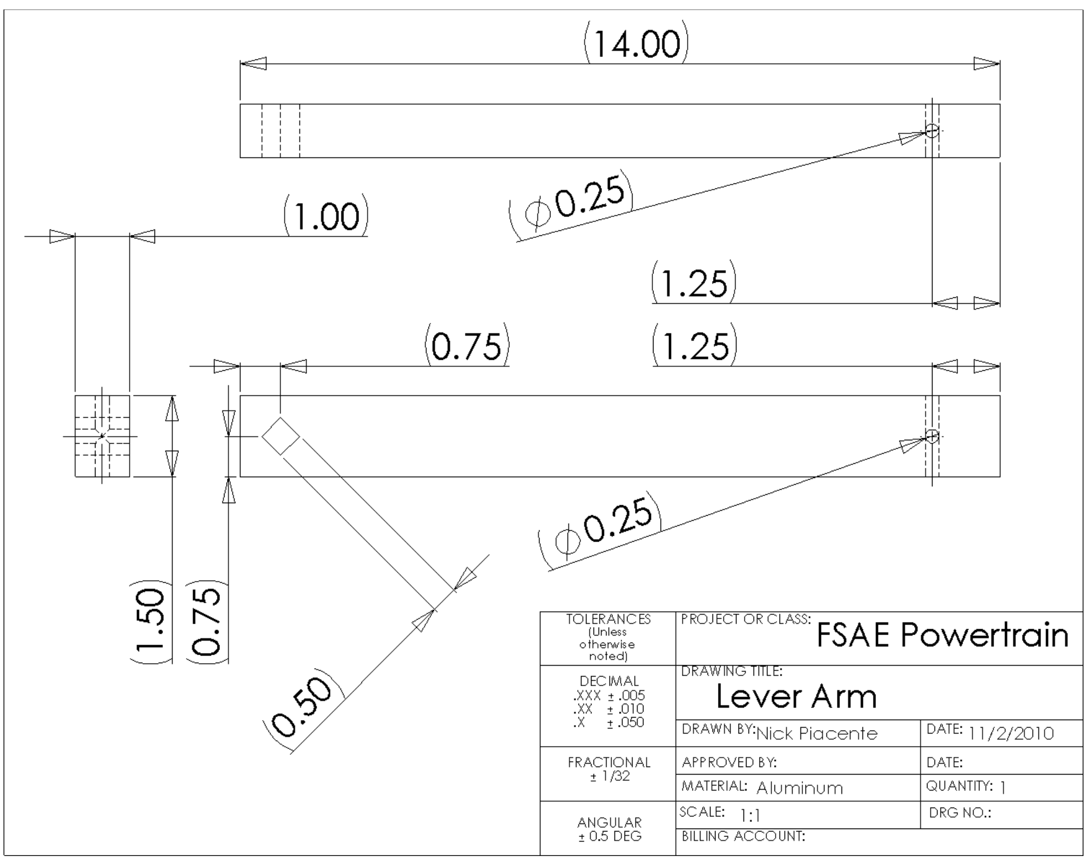
Step 20: Insert the drain bolt and seal on the bottom side of the case	
Step 21: Pour oil into the casing until it is half way full	
Step 22: Seal the case with the top drain bolt and seal	
Step 23: Interface the bottom pivot with chassis	
Step 24: Attach chain tensioner (shown in green horizontally off upright) and tighten until no slack is present in chain as shown by [6]	

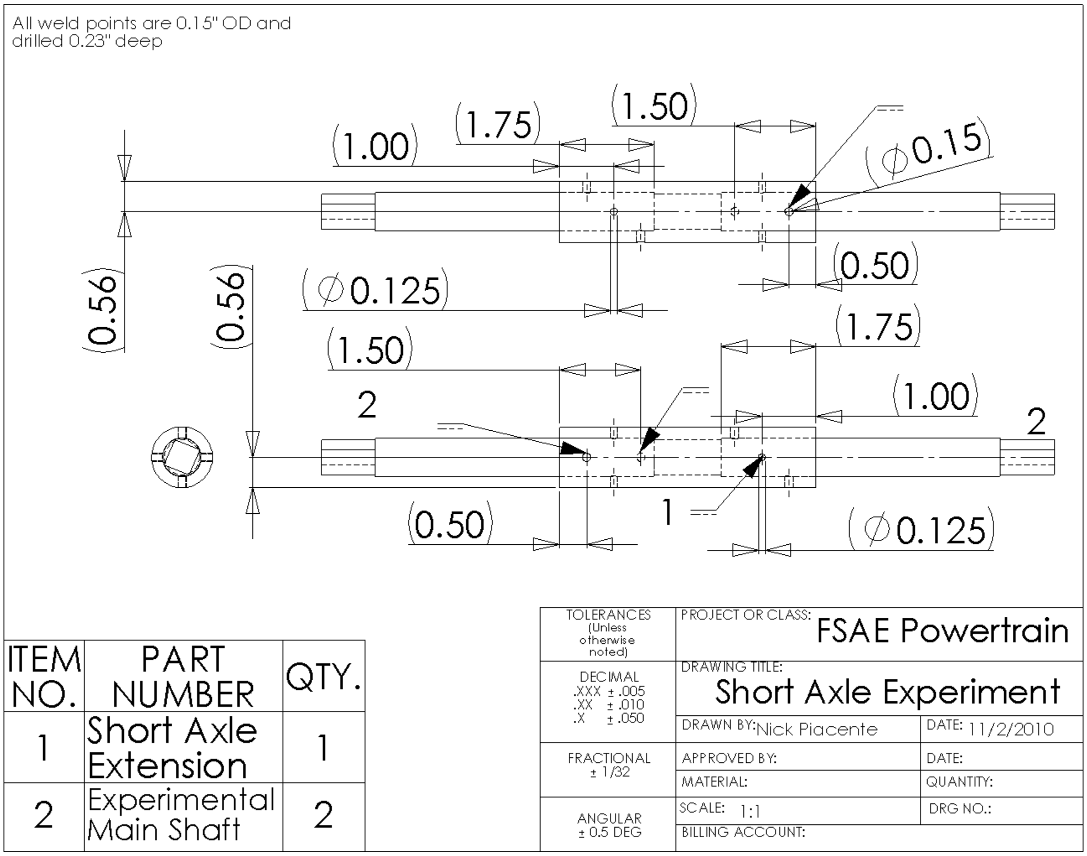
Appendix C (Axle Machinists Drawings):

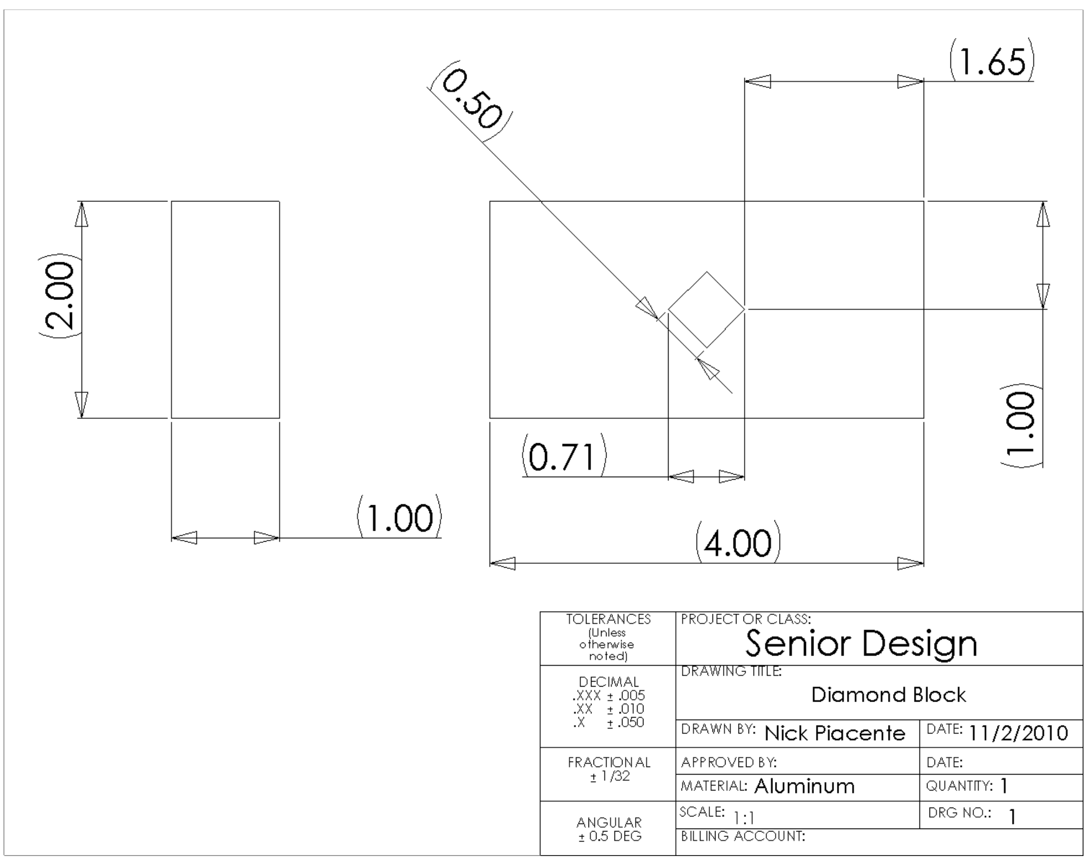




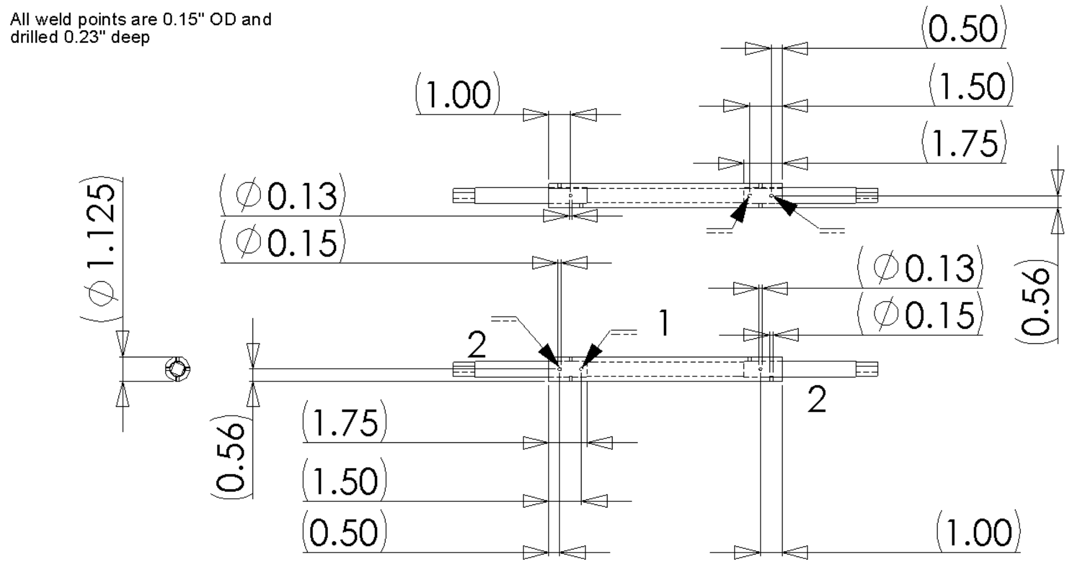

(1.75) (1.50) (1.00) (0.50)
(1.75)
(1.50)
(1.00)
(0.50)
2
4
4
1
3
(⌀0.71)
(⌀0.15)
(⌀0.13)
(0.13)
(1.13)
All weld pointsare 0.15" OD and 0.23" deep
ITEM NO. | PART NUMBER | QTY.
1 | Short Axle Extension | 1
2 | Diff Side ATV Axle | 1
3 | Hub Side ATV Axle | 1
4 | short spring pin | 2
TOLERANCES (Unless otherwise noted)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS: FSAE Powertrain
DRAWING TITLE: Final Left Axle Assembly
DRAWN BY: Nick Piacente
DATE: 11/2/2010
APPROVED BY:
DATE:
MATERIAL: Steel
QUANTITY: 1
SCALE: 1:1
DRG NO.: 1
BILLING ACCOUNT:


(14.00)
(⌀0.25)
(1.00)
(1.25)
(0.75)
(1.25)
(⌀0.25)
(1.50)
(0.75)
(0.50)
TOLERANCES (Unless otherwise noted)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS: FSAE Powertrain
DRAWING TITLE: Lever Arm
DRAWN BY: Nick Piacente
DATE: 11/2/2010
APPROVED BY:
DATE:
MATERIAL: Aluminum
QUANTITY: 1
SCALE: 1:1
DRG NO.:
BILLING ACCOUNT:


All weld points are 0.15" OD and drilled 0.23" deep
(1.00)
(1.75)
(1.50)
(⌀0.15)
(0.50)
(0.56)
(0.56)
(⌀0.125)
(1.50)
(1.75)
(1.00)
2
2
1
(0.50)
(⌀0.125)
ITEM NO.
PART NUMBER
QTY.
1
Short Axle Extension
1
2
Experimental Main Shaft
2
TOLERANCES (Unless otherwise noted)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS: FSAE Powertrain
DRAWING TITLE: Short Axle Experiment
DRAWN BY: Nick Piacente
DATE: 11/2/2010
APPROVED BY:
DATE:
MATERIAL:
QUANTITY:
SCALE: 1:1
DRG NO.:
BILLING ACCOUNT:

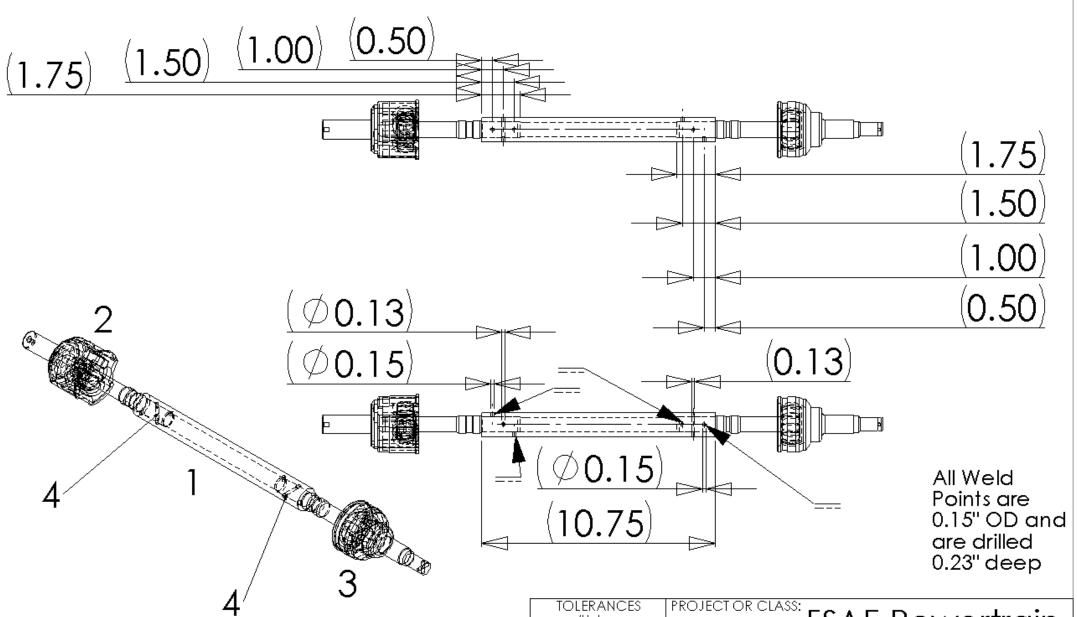

(0.50)
(1.65)
(2.00)
(0.71)
(1.00)
(1.00)
(4.00)
TOLERANCES (Unless otherwise noted)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS: Senior Design
DRAWING TITLE: Diamond Block
DRAWN BY: Nick Piacente
DATE: 11/2/2010
APPROVED BY:
DATE:
MATERIAL: Aluminum
QUANTITY: 1
SCALE: 1:1
DRG NO.: 1
BILLING ACCOUNT:

All weld points are 0.15" OD and drilled 0.23" deep



ITEM NO.	PART NUMBER	QTY.
1	Long Axle Extension	1
2	Experimental Main Shaft	2

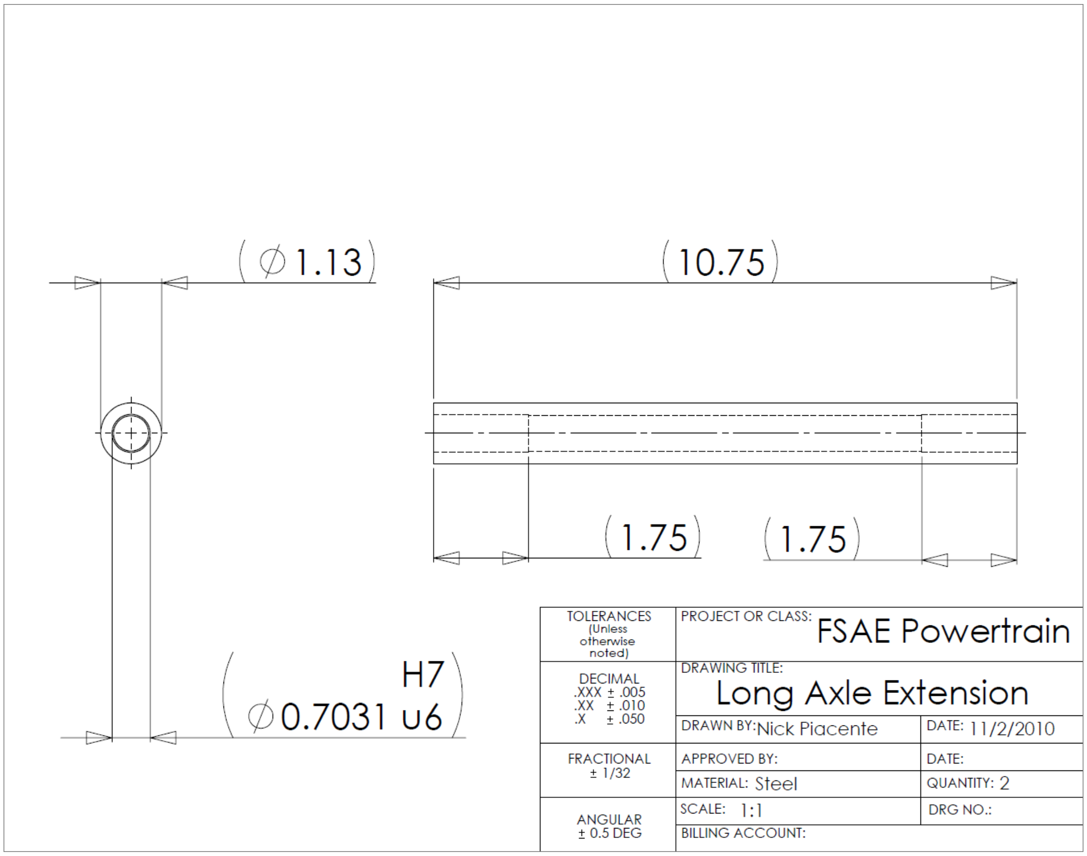
TOLERANCES (Unless otherwise noted)	PROJECT OR CLASS: FSAE Powertrain	
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050	DRAWING TITLE: Long Axle Experiment	
	DRAWN BY: Nick Piacente	DATE: 11/2/2010
FRACTIONAL ± 1/32	APPROVED BY:	DATE:
	MATERIAL: Steel	QUANTITY: 1
ANGULAR ± 0.5 DEG	SCALE: 1:1	DRG NO.:
	BILLING ACCOUNT:	



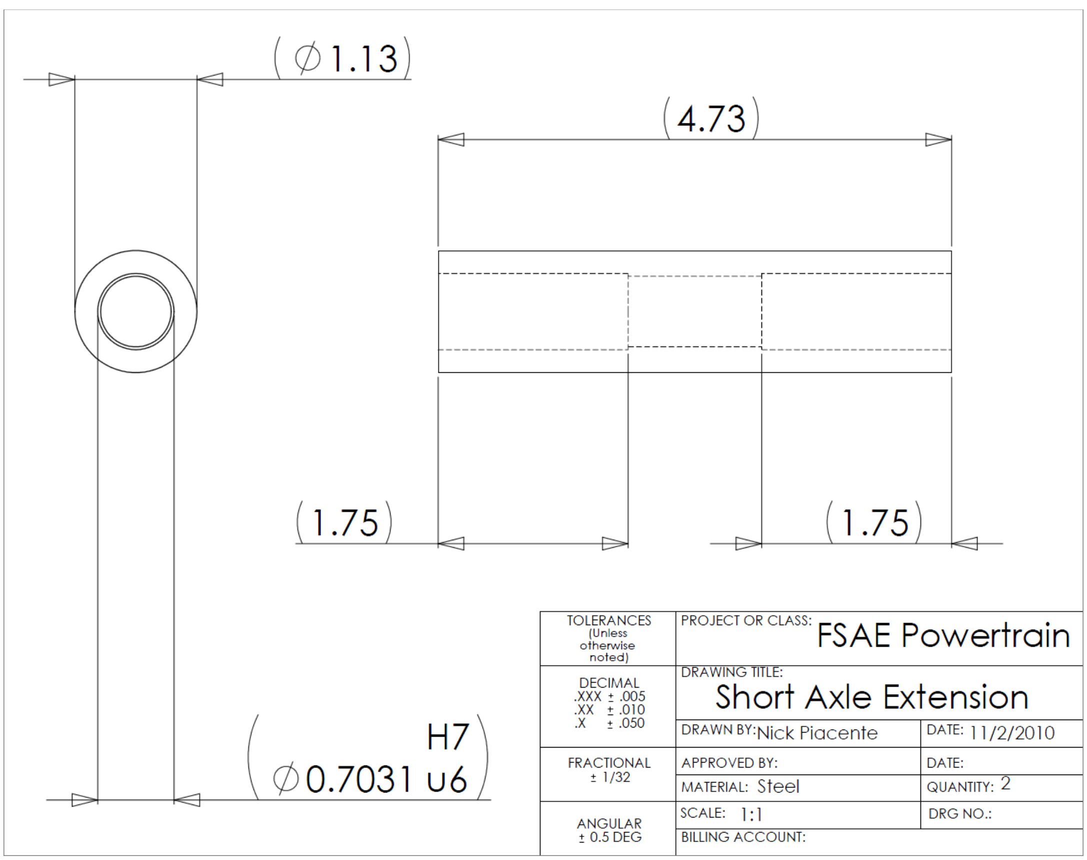
All Weld Points are 0.15" OD and are drilled 0.23" deep

ITEM NO.	PART NUMBER	QTY.
1	new long extension	1
2	Diff Side ATV Axle	1
3	Hub Side ATV Axle	1
4	Spring Pin	2

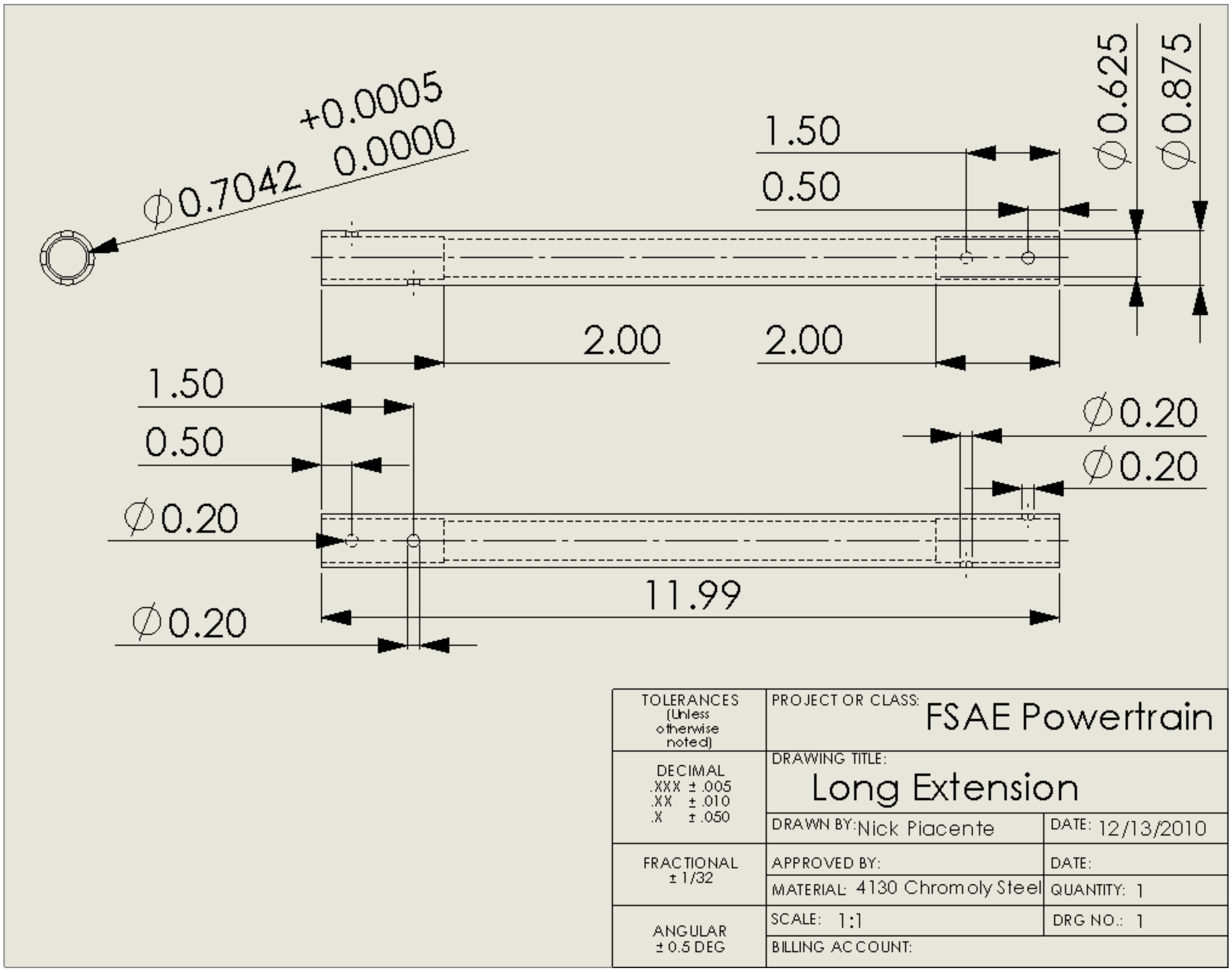
TOLERANCES (Unless otherwise noted)	PROJECT OR CLASS: FSAE Powertrain	
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050	DRAWING TITLE: Final Right Axle	
	DRAWN BY: Nick Piacente	DATE: 11/2/2010
FRACTIONAL ± 1/32	APPROVED BY:	DATE:
	MATERIAL: Steel	QUANTITY: 1
ANGULAR ± 0.5 DEG	SCALE: 1:1	DRG NO.: 1
	BILLING ACCOUNT:	



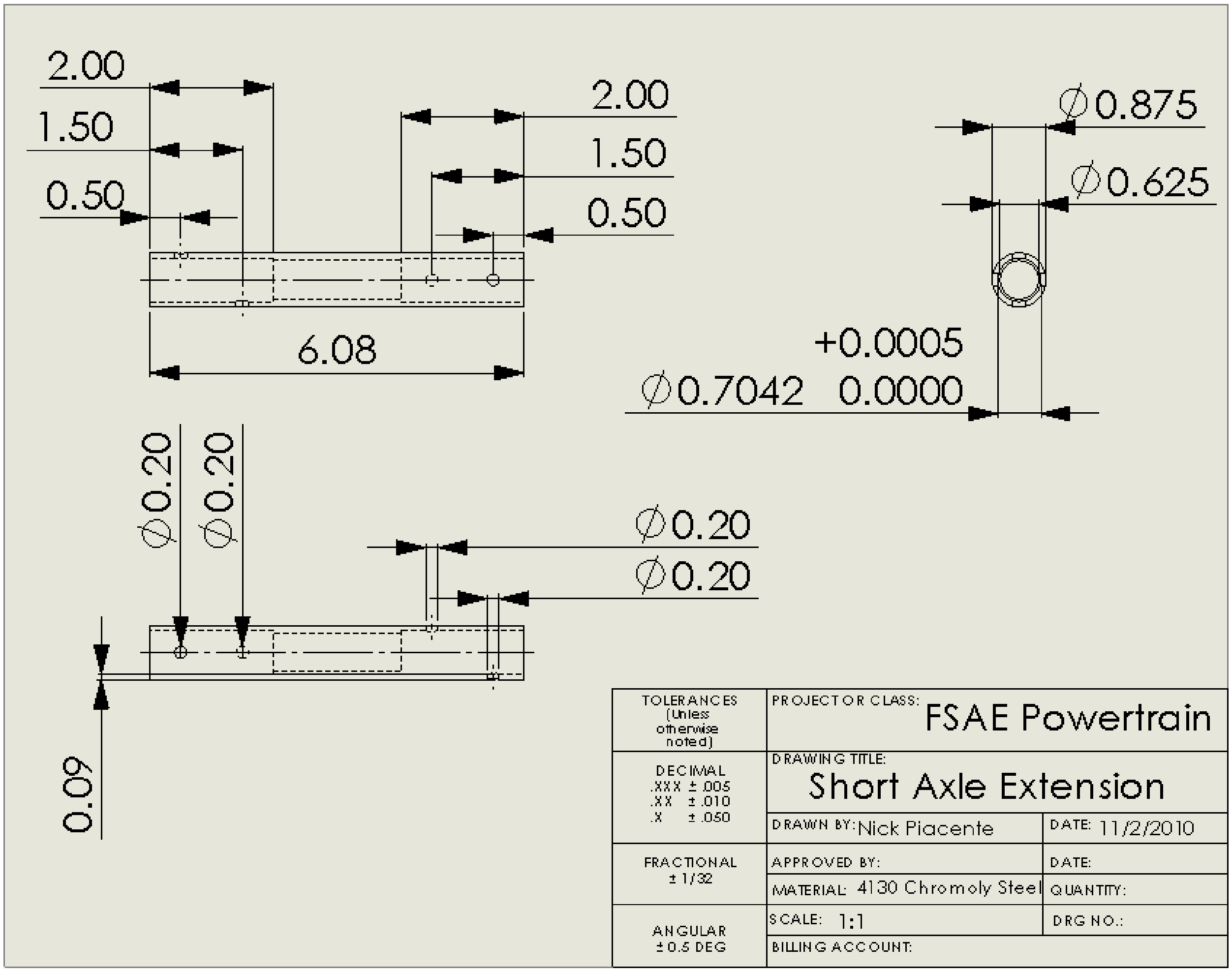
Ø1.13
10.75
1.75
1.75
H7
Ø0.7031 u6
TOLERANCES (Unless otherwise noted)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS: FSAE Powertrain
DRAWING TITLE: Long Axle Extension
DRAWN BY: Nick Piacente
DATE: 11/2/2010
APPROVED BY:
DATE:
MATERIAL: Steel
QUANTITY: 2
SCALE: 1:1
DRG NO.:
BILLING ACCOUNT:



Ø1.13
4.73
1.75
1.75
H7
Ø0.7031 u6
TOLERANCES (Unless otherwise noted)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS: FSAE Powertrain
DRAWING TITLE: Short Axle Extension
DRAWN BY: Nick Piacente
DATE: 11/2/2010
APPROVED BY:
DATE:
MATERIAL: Steel
QUANTITY: 2
SCALE: 1:1
DRG NO.:
BILLING ACCOUNT:



+0.0005
∅0.7042 0.0000
1.50
0.50
∅0.625
∅0.875
2.00
2.00
1.50
0.50
∅0.20
∅0.20
∅0.20
11.99
∅0.20
TOLERANCES (Unless otherwise noted)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS: FSAE Powertrain
DRAWING TITLE: Long Extension
DRAWN BY: Nick Piacente
DATE: 12/13/2010
APPROVED BY:
DATE:
MATERIAL: 4130 Chromoly Steel
QUANTITY: 1
SCALE: 1:1
DRG NO.: 1
BILLING ACCOUNT:



Appendix D (Final Axle Fabrication):

1. Cut 4130 chromoly tubing in the horizontal band saw. The two lengths necessary are 6-1/4" and 12-1/4", respectively.
2. Face the ends and cut to length on the lathe, using the carbide cutting tools at around 500 RPM. The small extention should be 6.08 $\pm$0.01" and the long extention should be 11.99 ± 0.01".
3. Use the boring bar to bore the inner diameter (2" deep) to between 0.7037" – 0.7042" to guarantee at least a 0.0013" press fit. Repeat this step for both sides of each axle extension.
4. Chamfer the extensions using a 45 degree cutting tool. Move only 0.1" into each extension on both sides. The extensions are now ready.
5. Mark the center of the Honda TRX axle. (This can be done by measuring 1.75" from the end of the flare in the ATV axle.) Cut the Honda TRX axle down the center with a reciprocating saw.
6. Grind down the ends of the ATV axle to make sure there are no burrs or rough edges.
7. Cool the ATV axles in the freezer until they reach approximately 40 degrees.
8. Clamp one of the extensions in a vice and start heating it with an acetalyne torch. Use a heat gun to make sure you are around 500 F before attempting the press fit. Make sure the extension does not reach above 727 F (eutectic point) which would result in a phase change of the solid.

9. Once 500 F is reached, have a partner grab the ATV axle from the freezer. Carefully but swiftly drive the ATV axle into the extension. If the temperature gradient between the two metals is what it should be, the ATV axle should slide in the extension easily.
10. Let the extension cool and repeat steps 8 and 9 for the other side of the extension. Repeat for the second extension.
11. Once the axles are press fit together, drill 2 holes (1/2” from the end, and 1-1/2” from the end, 180 degrees apart) through the extension, and touching the ATV axle. Repeat for the other side of the axle, and for the second axle assembly.

TIG Weld over each hole drilled and let sit. The axles are now complete!

Appendix E (Wheel Hop Torque Calculation):

Angular Momentum (L) of the spinning axle-wheel system can be given by:

$$L = I\omega = \tau \Delta t$$

Where I is the moment of inertia of the wheel and axle (calculated numerically), ω is the angular velocity value described earlier, Δt is the time period where the constant, frictional torque is applied, and τ is the frictional torque applied to the axle, (which the axle needs to withstand).

To find Δt, start by defining the frictional torque τ as

$$\tau = I\alpha = I\frac{d\omega}{dt} = -F_k\delta = -\mu_k mg\delta$$

Where δ is the radius of the tire. Rearranging and integrating:

$$\int d\omega = \int -\frac{\mu_k mg\delta}{I}dt$$

$$\omega(t) = -\frac{\mu_k mg\delta t}{I} + \omega_o$$

Because $\omega(t_f) = 0$,

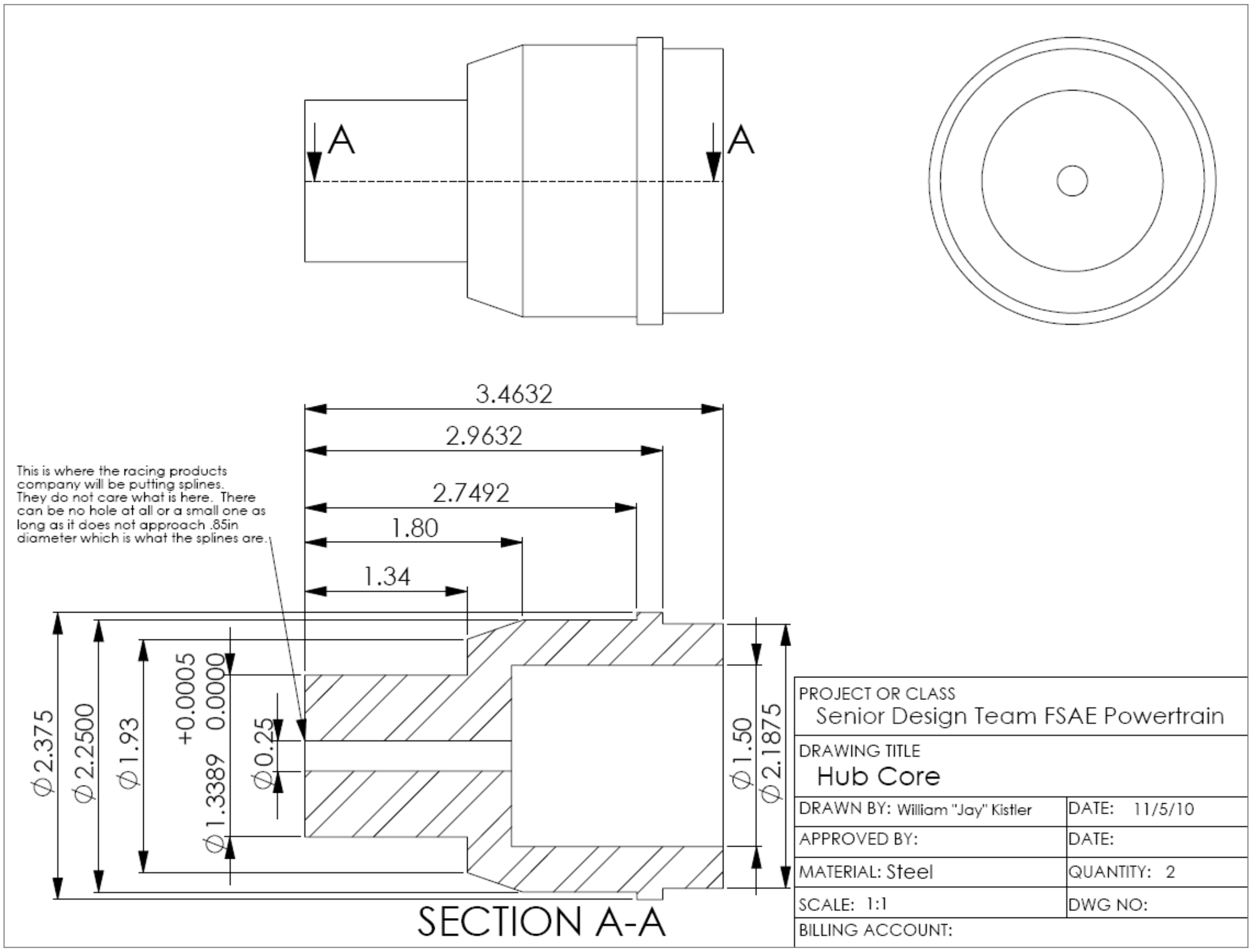
$$t_f = \frac{\omega_0 I}{\mu_k mg\delta t} = .015s$$

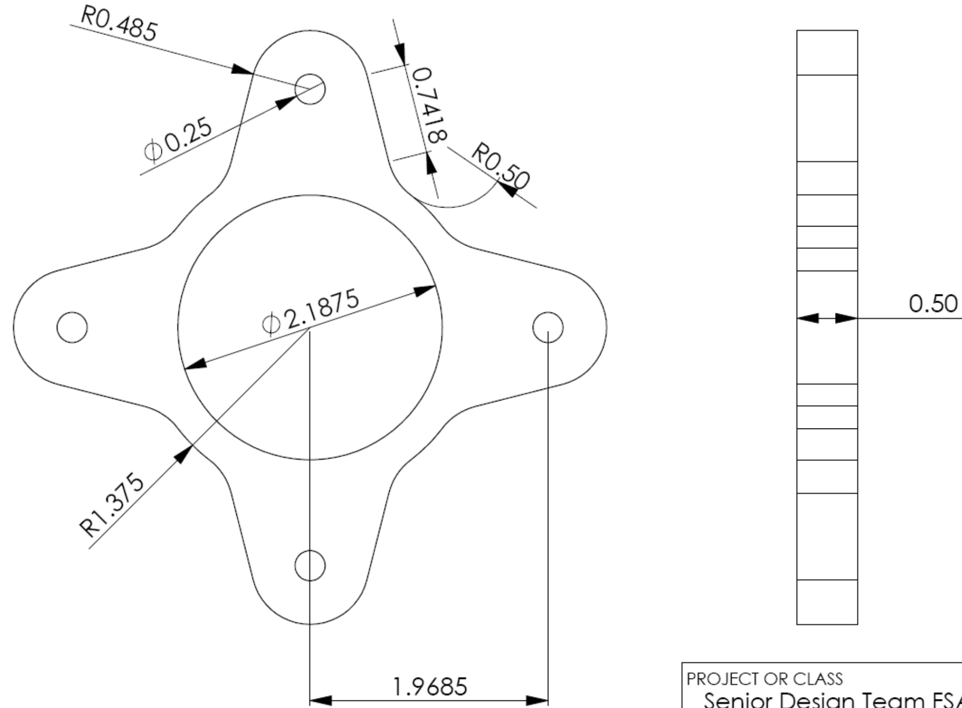
Rearranging the first equation,

$$\tau = \frac{I\omega}{\Delta t} = 165\ foot\ pounds$$

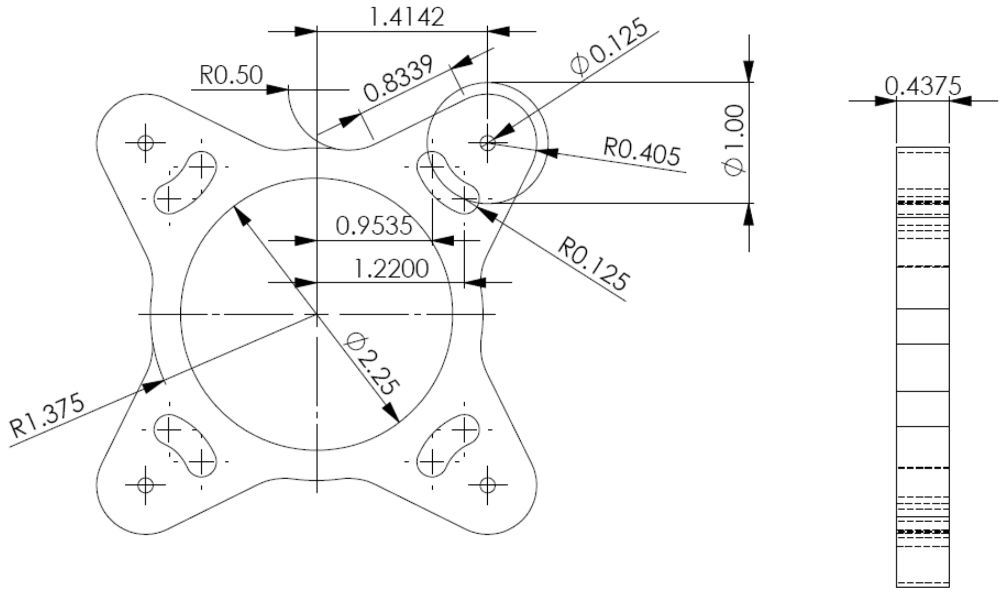
This analysis is only valid assuming the joining methods (described later) will sufficiently hold the axle in one piece (i.e. no relative motion between the ATV Axle and the axle extension)

Appendix F (Hub Machinists Drawings):

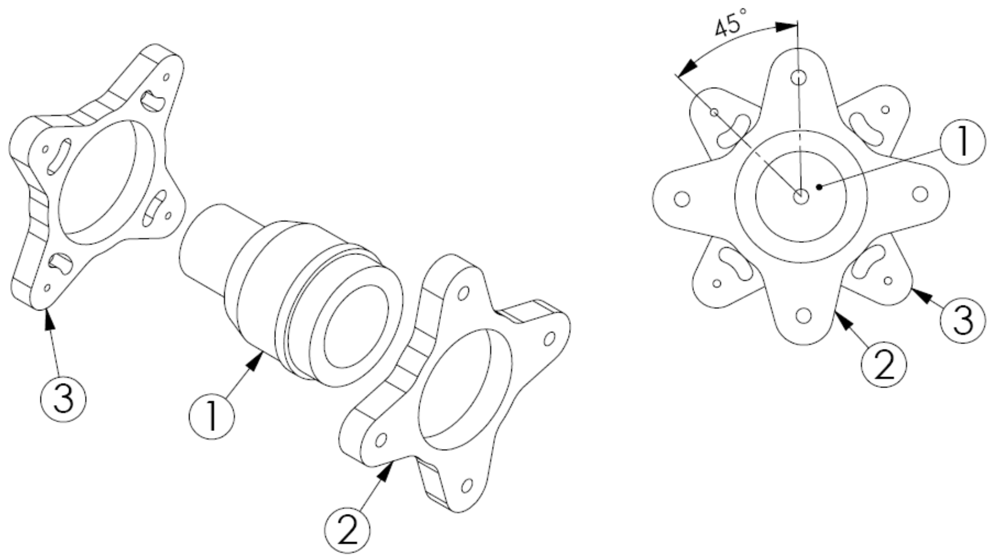




PROJECT OR CLASS Senior Design Team FSAE Powertrain	
DRAWING TITLE Hub Wheel Star	
DRAWN BY: William "Jay" Kistler	DATE: 11/5/10
APPROVED BY:	DATE:
MATERIAL: Steel	QUANTITY: 2
SCALE: 1:1	DWG NO:
BILLING ACCOUNT:	

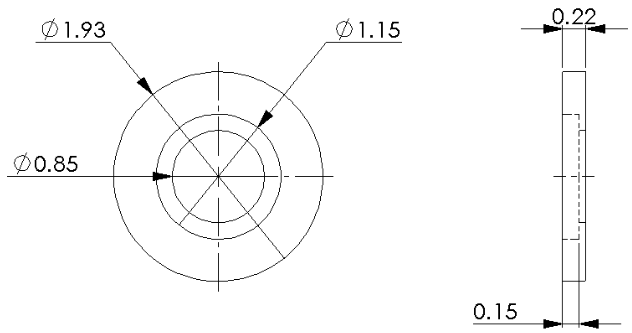


PROJECT OR CLASS Senior Design Team FSAE Powertrain	
DRAWING TITLE Hub Brake Rotor Star	
DRAWN BY: William "Jay" Kistler	DATE: 11/5/10
APPROVED BY:	DATE:
MATERIAL: Steel	QUANTITY: 2
SCALE: 1:1	DWG NO:
BILLING ACCOUNT:	

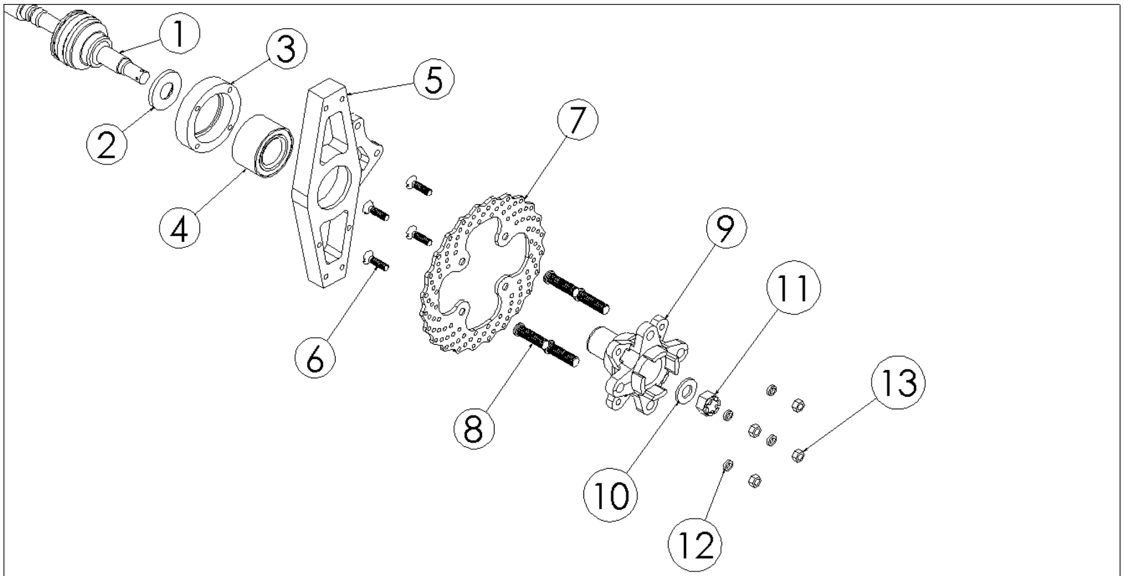


1. Hub Core
2. Hub Wheel Star
3. Hub Brake Rotor Star

PROJECT OR CLASS Senior Design Team FSAE Powertrain	
DRAWING TITLE Hub Assembly	
DRAWN BY: William "Jay" Kistler	DATE: 11/5/10
APPROVED BY:	DATE:
MATERIAL: Steel	QUANTITY: 2
SCALE: 1:2	DWG NO:
BILLING ACCOUNT:	



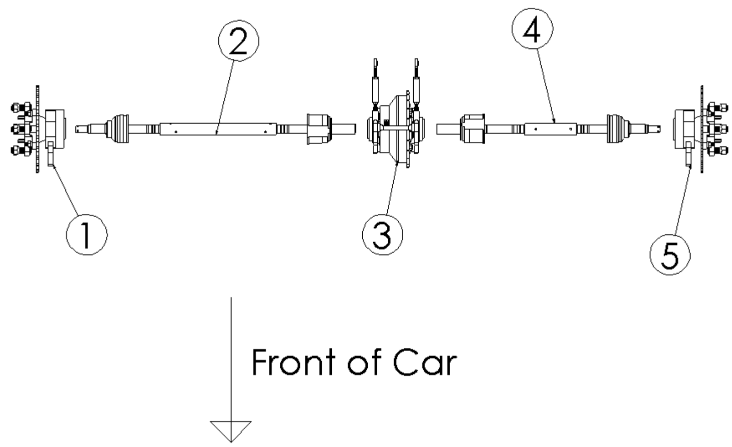
TOLERANCES (UNLESS OTHERWISE NOTED)	PROJECT OR CLASS Senior Design Team FSAE Drivetrain	
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050	DRAWING TITLE Hub Spacer	
	DRAWN BY: William "Jay" Kistler	DATE: 11/5/10
FRACTIONAL ± 1/32	APPROVED BY:	DATE:
	MATERIAL: 1020 Steel	QUANTITY: 1
ANGULAR ± 0.5 DEG	SCALE: 1:1	DWG NO: 7 of 8
	BILLING ACCOUNT:	



1. Stub Shaft (Spindle) From Axle
2. Hub Spacer
3. Bearing Retainer (From Suspension)
4. 64mm Taylor Racing Wheel Bearing
5. Upright (From Suspension)
6. 4 x Brake Rotor Bolts
7. Brake Rotor
8. 4 x Lug Bolts
9. Hub
10. Washer
11. Castle / Lock Nut
12. Brake Rotor Washers
13. Brake Rotor Nuts

TOLERANCES (UNLESS OTHERWISE NOTED)	PROJECT OR CLASS: Senior Design Team FSAE Drivetrain	
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050	DRAWING TITLE: Hub Assembly	
	DRAWN BY: William "Jay" Kistler	DATE: 11/2/10
FRACTIONAL ± 1/32	APPROVED BY:	DATE:
	MATERIAL:	QUANTITY:
ANGULAR ± 0.5 DEG	SCALE: 1:10	DWG NO: 8 of 8
	BILLING ACCOUNT:	

Appendix G (Drivetrain Assembly Drawings):

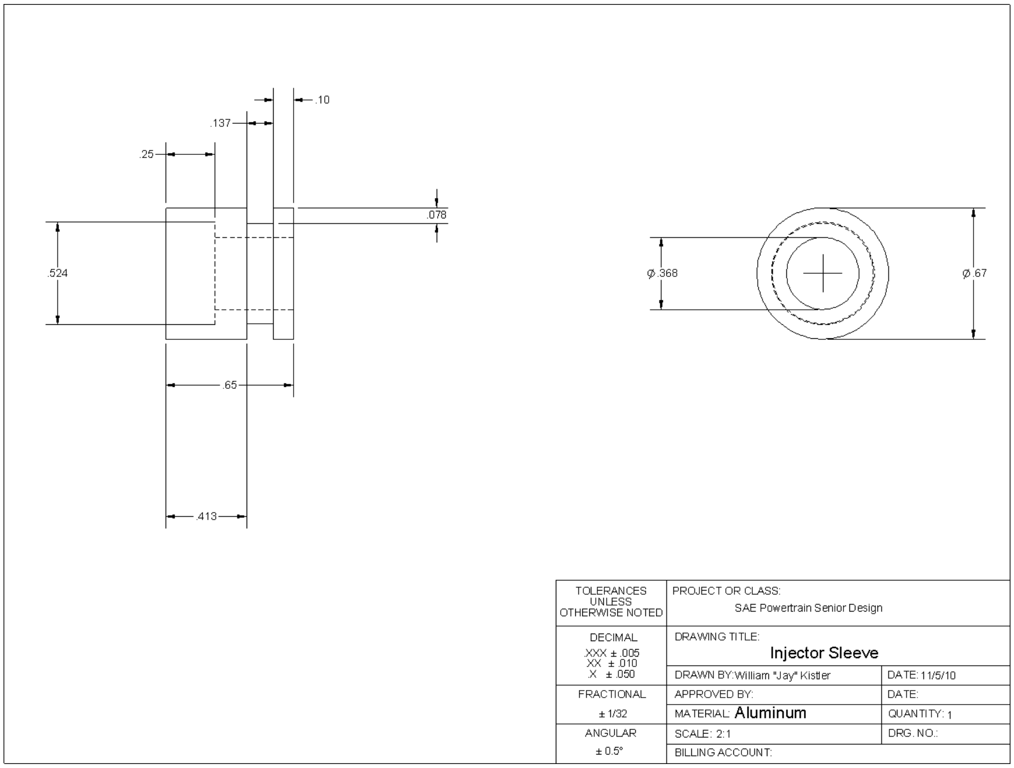


1. Right Hub Assembly
2. Final Right Axle Assembly
3. Differential Assembly
4. Final Left Axle Assembly
5. Left Hub Assembly

* Only difference between hub assemblies is upright

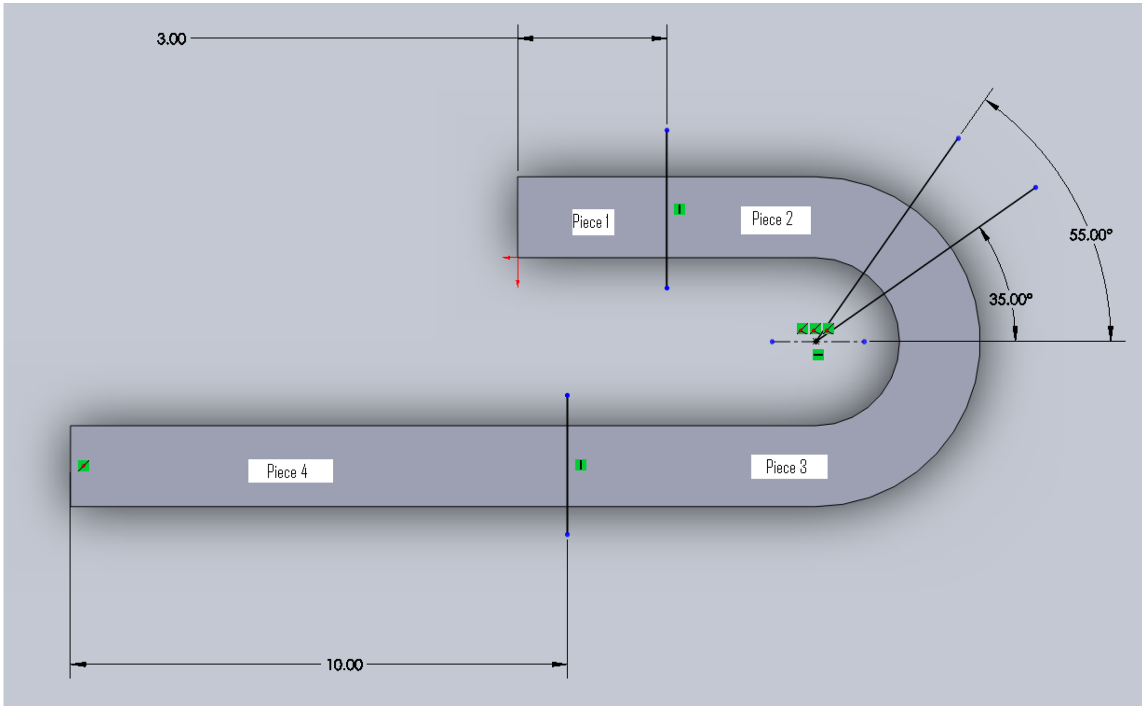
TOLERANCES (UNLESS OTHERWISE NOTED)	PROJECT OR CLASS Senior Design Team FSAE Drivetrain	
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050	DRAWING TITLE Drivetrain Assembly	
	DRAWN BY: William "Jay" Kistler	DATE: 11/2/10
FRACTIONAL ± 1/32	APPROVED BY:	DATE:
	MATERIAL:	QUANTITY:
ANGULAR ± 0.5 DEG	SCALE: 1:10	DWG NO:
	BILLING ACCOUNT:	

Appendix H (Air Intake Machinists Drawings):



Appendix I (Final Exhaust Prototype Fabrication):

1. Starting with the stainless steel J-tube, make the following cuts with a horizontal bandsaw, while cleaning up the edges on the belt sander:

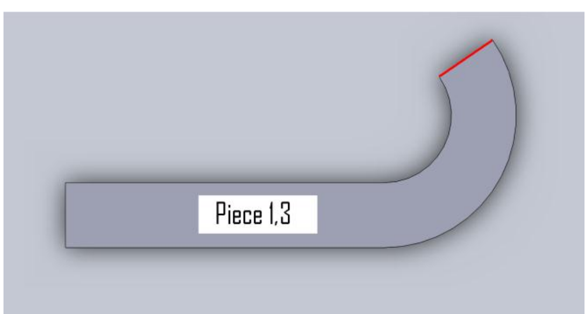


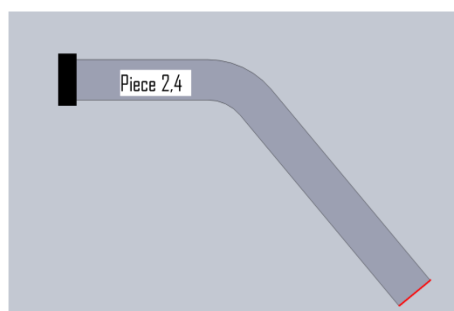
2. Line the straight side of Piece 1 with the straight side of Piece 3 and weld around the perimeter of the tube.

3. Line one of the straight parts of Piece 4 with the curved side of piece 2 and weld the two together.

4. Weld the collar to the straight side of piece 2.

Note: If done correctly, the following pieces should be created:



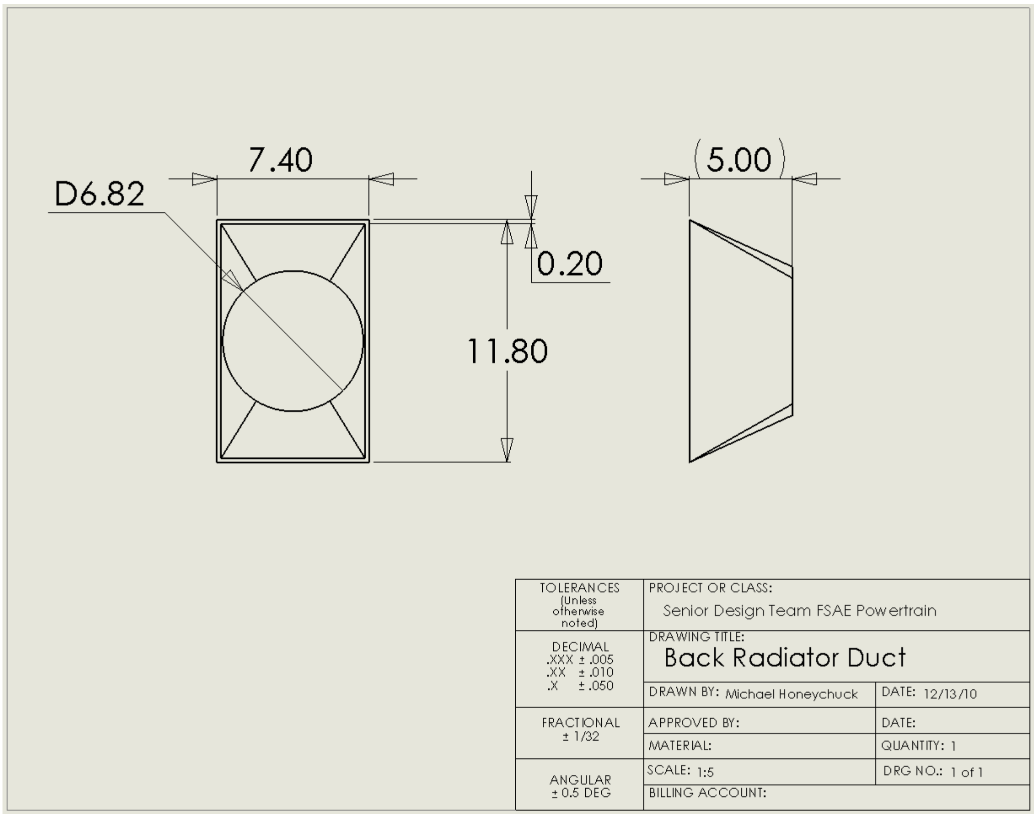


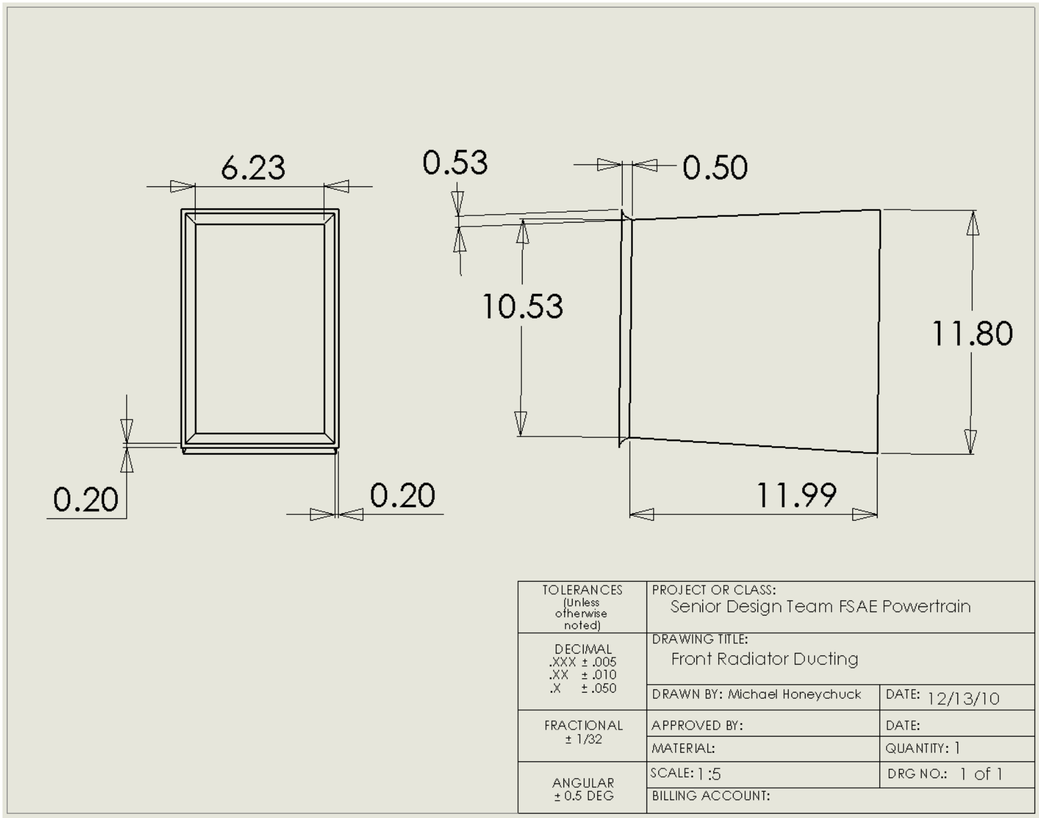
5. Line up the straight end of piece 2,4 with the curved piece of piece 1,3 (this is where the red lines would line up). Rotate piece 1,3 down (CCW) approximately 20 degrees, and weld the pieces together.
6. Wrap piece 2,4 with copper insulating header wrap, using stainless steel band clamps to secure the ends.

7. Attach this assembly to the engine using the exhaust flange, orienting the tube outlet to be parallel with the ground.
8. Take a 2” OD (3” length), 2” to 1-3/4” transition, and 1-3/4” OD (3” length) tubes, and weld them together in that order.
9. Weld the other 2” OD end to the FMF Q4 muffler.
10. Wrap from the edge of the muffler down to the 1-3/4” tube with copper insulating header wrap. Use stainless steel band clamps to secure the ends.
11. Slide the 1-3/4” OD end over the assembly attached to the car, and clamp down to the desired main tube length!

Note: If done correctly, the exhaust should look like this:


Appendix J (Radiator Duct Designs):




6.23
0.53
0.50
10.53
11.80
0.20
0.20
11.99
TOLERANCES (Unless otherwise noted)
DECIMAL .XXX ± .005 .XX ± .010 .X ± .050
FRACTIONAL ± 1/32
ANGULAR ± 0.5 DEG
PROJECT OR CLASS: Senior Design Team FSAE Powertrain
DRAWING TITLE: Front Radiator Ducting
DRAWN BY: Michael Honeychuck
DATE: 12/13/10
APPROVED BY:
DATE:
MATERIAL:
QUANTITY: 1
SCALE: 1:5
DRG NO.: 1 of 1
BILLING ACCOUNT: