Synchronous Charge Trapping Modification of a Two-Stroke Engine

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Peter A. Britanyak

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Major Professor: Karen DenBraven, Ph.D

AUTHORIZATION TO SUBMIT THESIS

This thesis of Peter A. Britanyak, submitted for the degree of Master of Science with a major in Mechanical Engineering and titled "Synchronous Charge Trapping Modification of a Two-Stroke Engine," has been reviewed in final form. Permission, as indicated by the signatures and dates given below, is now granted to submit final copies to the College of Graduate Studies for approval.

Major Professor ________________________________ Date ________
Karen DenBraven, Ph.D.

Committee
Member ________________________________ Date ________
Steven Beyerlein, Ph.D.

Committee
Member ________________________________ Date ________
David Egolf, Ph.D.

Department
Administrator ________________________________ Date ________
John Crepeau, Ph.D.

Discipline's
College Dean ________________________________ Date ________
Donald Blackketter, Ph.D.

Final Approval and Acceptance by the College of Graduate Studies

________________________________ Date _______
Nilsa Bosque-Perez, Ph.D.

Abstract

This research covers the design and testing of a modified snowmobile engine with synchronous charge trapping exhaust valves. The goal is to determine the benefits and feasibility of using this engine platform in a Clean Snowmobile Challenge (CSC) snowmobile and compare performance to the stock engine to determine if it is worth pursuing. The synchronous charge trapping engine was tuned using two methods: tuning for maximum torque and tuning for best Brake Specific Fuel Consumption (BSFC) while matching stock torque. The results from tuning for maximum torque show an increase of 32 percent in torque while only deviating up to 6 percent from the stock BSFC. When tuned to equal torque of the stock engine, fuel consumption was reduced by an average 17 percent. The results from these methods show that there is an appreciable improvement in performance that makes this technology worth developing.

Acknowledgments

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List of Abbreviations

AFR: Air-Fuel Ratio
ATDC: After Top Dead Center
ABDC: After Bottom Dead Center
BDC: Bottom Dead Center
BSFC: Brake Specific Fuel Consumption
BAT: Best Available Technology
CNC: Computer Numerical Control
CO: Carbon Monoxide
CSC: Clean Snowmobile Challenge
CVT: Continuously Variable Transmission
DI: Direct Injection
DOE: Department of Energy
EFI: Electronic Fuel Injection
EGT: Exhaust Gas Temperature
EMM: Engine Management Module
EPA: Environmental Protection Agency
EPC: Exhaust Port Closed
EPO: Exhaust Port Open
GDI: Gasoline Direct Injection
HC: Hydrocarbons
HCCI: Homogeneous Charge Compression Ignition
HP: Horse Power
HPDI: High Pressure Direct Injection
IPC: Intake Port Closed
IPO: Intake Port Open
kW: kilowatts
λ: Lambda
MAG: Magneto
NO_x: Nitrogen Oxides
NPS: National Park Service
PH: Precipitate Hardening
PSI: Pounds per Square Inch
PTO: Power Take-Off
RAVE: Rotax Adjustable Variable Exhaust
RPM: Revolutions per Minute
RTV: Room Temperature Vulcanizing
SAE: Society of Automotive Engineers
SCT: Synchronous Charge Trapping
SDI: Semi-Direct Injection
TBI: Throttle Body Injection
TDC: Top Dead Center
UHC: Unburned Hydrocarbons
UI: University of Idaho
WOT: Wide Open Throttle

1.0 Introduction

Snowmobiles have been around since the early 1900s. Their use as a quick mode of transportation across snowy terrain makes them a necessity in some areas of the world. Through the years the initial designs which consisted of modified automobiles have been refined to the point of being highly engineered performance vehicles for recreation and competition [1]. Snowmobiles are still a necessity where tracked vehicles are the only option for travel through snow covered areas. In the last ten years there has been pressure to reduce noise and exhaust emissions from on-road and off-road vehicles, including snowmobiles. These new restrictions have forced companies to develop new technology.

The University of Idaho has competed in the Clean Snowmobile Challenge (CSC) since 2001 [2]. In 2002 and 2003, the University of Idaho won this competition using a four-stroke engine platform [3, 4]. Later on, after the development of the E-TEC direct injection system on outboard two-stroke engine, the University of Idaho assembled a very competitive snowmobile adapting this new technology which won in 2007 with a gasoline direct-injected two-stroke engine [5]. Now that E-TEC injection can be found on stock snowmobiles, the University of Idaho is looking to new challenges. Synchronous Charge Trapping (SCT) is a new idea that has the potential to have a large impact on two-stroke engines and the power sport industry. By adding exhaust valves that move in sync with the crank, the exhaust port can be lowered only on the compression portion of the cycle reducing short circuited fuel. Two companies have shown interest in this new technology and have begun development on their own designs.

1.1 Research Goals

The major goal of this research is to modify a Rotax 600cc two-stroke engine and determine whether SCT is a feasable strategy for improving performance of a two-stroke engine. SCT has the potential to sharply decrease short-circuiting in the two-stroke engine. These modifications will also take the place of the tuned pipe, decreasing overall engine size. In this thesis the design and modifications to the engine will be discussed, along with the results from dynamometer testing and the conclusions concerning feasibility of using this engine platform in a snowmobile.

1.2 Clean Snowmobile Challenge

The National Park Service (NPS) is especially concerned with the use of motorized vehicles during the winter season in Yellowstone National Park. In 2010-2011 the number of snowmobiles allowed access to the park was limited to 318 per day [6]. Snowmobiles that are allowed into the park must also use Best Available Technology (BAT) [7]. BAT is based on the exhaust and sound emissions of snowmobiles as shown in Table 1. Hydrocarbon Emissions (HC), also called Unburnt Hydrocarbons (UHC), is typically higher in two-stroke engines. UHCs are caused by incomplete combustion and tuning for engine survivability. To protect engines, fuel mixtures are typically richer at higher engine speeds to prevent high temperatures, knock, and engine seizure. At lower engine speeds, the short circuiting caused by open intake and exhaust ports also causes HC emissions. Carbon Monoxide (CO) emissions are also formed due to incomplete combustion and are also typically higher in two-stroke than four-stroke engines for the same reasons as HC emissions.

National Park Service - Best Available Technology (BAT)	**Exhaust Emissions g/(kW*hr)**		**dBA**
	Hydrocarbons	**Carbon Monoxide**	
BAT Requirements	15	120	73
Average Two-Stroke (Non-BAT)	150	400	78
Univ. of Idaho 2007 E-TEC	18	68	73

Table 1: National Park Service Snowmobile Requirements for Best Available Technology Compared to Production Two-Stroke and the University of Idaho 2007 Competition Snowmobile. [5, 7]

Peak noise emitted from snowmobiles is determined using a J-192 test, in which a sound measurement is taken from 50ft away facing perpendicular to a 150ft. run. As the rider approaches the run at 35 MPH the snowmobile is held at full throttle until the end on the 150ft. run. The maximum Sound Pressure Level (SPL) is measured in decibels (dB) and then the A-weighted filter shown in Figure 1 is applied to adjust for human hearing. This is because human hearing is more sensitive in some areas of the frequency spectrum than others.

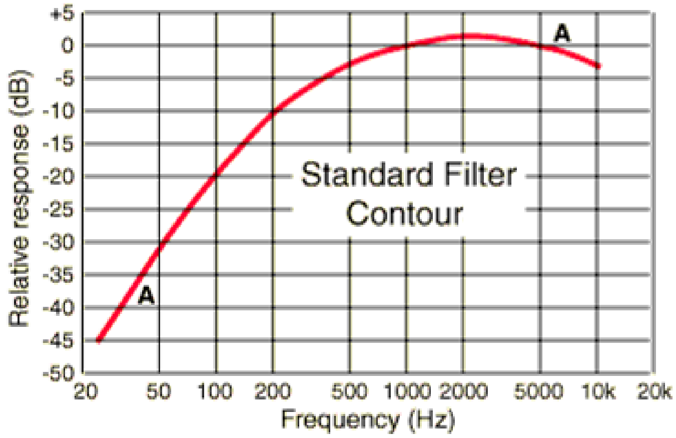


Figure 1: A-Weighted Filter [8]

As restrictions on noise and emissions tighten, snowmobiles may soon be prohibited from the park. In an effort to foster improvements in snowmobile technology, the NPS, Environmental Protection Agency (EPA), Society of Automotive Engineers (SAE), Montana Department of Environmental Quality (DEQ) and the Department of Energy (DOE) came together to create the CSC in 2000. The competition is open to Universities around the world and in 2010 had twelve competing teams [9]. The competition focuses on three areas: noise, exhaust emissions and fuel economy.

1.3 Emission Restrictions

The exhaust emissions of snowmobiles are rated by the EPA and NPS using an E score, which is based on a five mode test [10].

Mode Point	Speed [% of Rated]	Torque [% of Rated]	Weighting [%]
1	100	100	12
2	85	51	27
3	75	33	25
4	65	19	31
5	Idle	N/A	5

Table 2: EPA Mode Point Table [10]

Each mode point is a given as a percent of rated speed and percent of rated torque output of the engine. The mode points are selected and weighted based on typical snowmobile use [10]. At each mode point, emissions data are taken and weighted appropriately. The E-score is determined by Equation 1. A higher E-score represents fewer emissions.

$$E = 100\left[1 - \frac{(HC + NO_x) - 15}{150}\right] + 100\left[1 - \frac{CO}{400}\right]$$

Equation 1: E-score Calculation [10]

In 2012, the EPA will require snowmobiles to meet a minimum E score of 100; NPS BAT standards will require an E-score of 170.

2.0 Two-Stroke Engine Operation

The main advantage to a two-stroke engine is its simplicity. In a basic two-stroke engine there are only three moving parts: the crank, connecting rod and piston. Other components include the carburetor, reed cage, block, cylinder, head and tuned pipe.

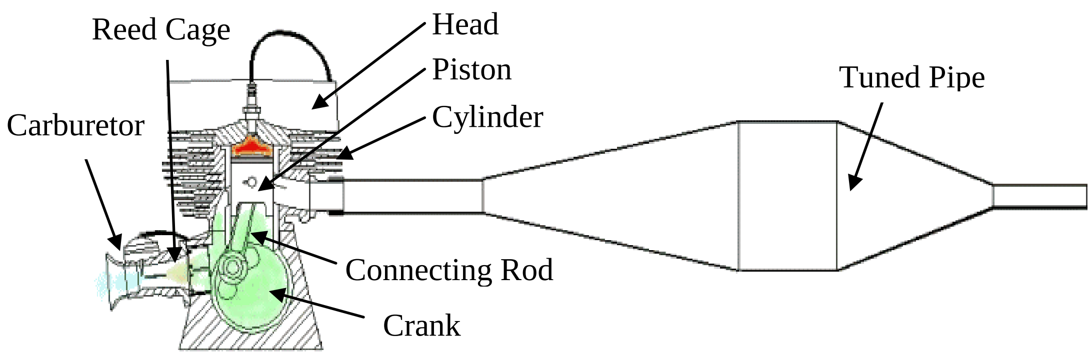


Figure 2: Two-Stroke Engine- Combustion Event [11].

The two stroke cycle begins with a combustion event with the piston near Top Dead Center (TDC), as shown in Figure 2. The increase in pressure caused by combustion will force the piston downward. As the piston travels down, the exhaust port is opened first and exhaust scavenging begins. The exhaust leaving the cylinder creates a high pressure wave that travels towards the exit of the tuned pipe, as shown in Figure 3.

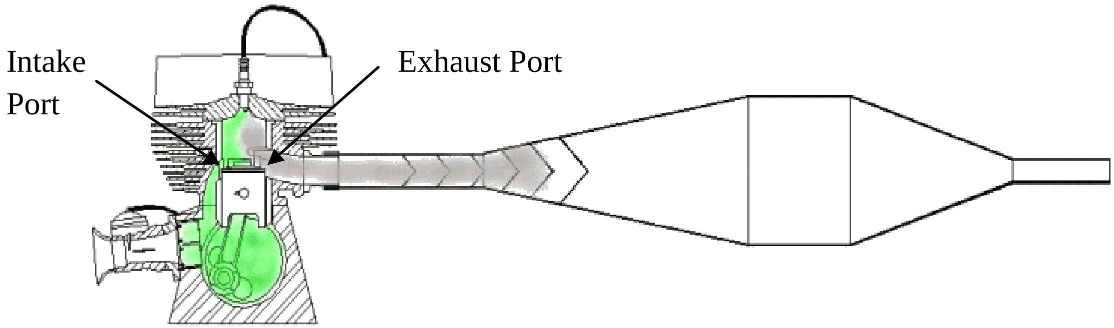


Figure 3: Two-Stroke Engine- Exhaust Pulse and Air/Fuel Mixture Intake [11].

This high pressure wave is reflected by the carefully designed diverging and converging cones in the tuned pipe. This is illustrated in Figure 4. Shortly after the exhaust port is opened, the intake ports are also uncovered. A fresh air/fuel mixture in the case is pressurized by the displacement of the piston moving downward and is forced up through the intake ports into the cylinder. At this point, both the intake and exhaust ports are open which can allow some of the entering air/fuel mixture to escape out through the exhaust. This process is called short-circuiting. Short-circuiting primarily occurs at off-tune points of the tuned pipe and is a major disadvantage to the basic two-stroke engine design.

The geometry of the tuned pipe is set up so that within a desired RPM band the high pressure exhaust wave will pull short-circuited air/fuel mixture into the pipe. Then as it is reflected, the short circuited air/fuel mixture is forced back into the cylinder, preventing short-circuiting and creating what is called a dynamic supercharging effect.

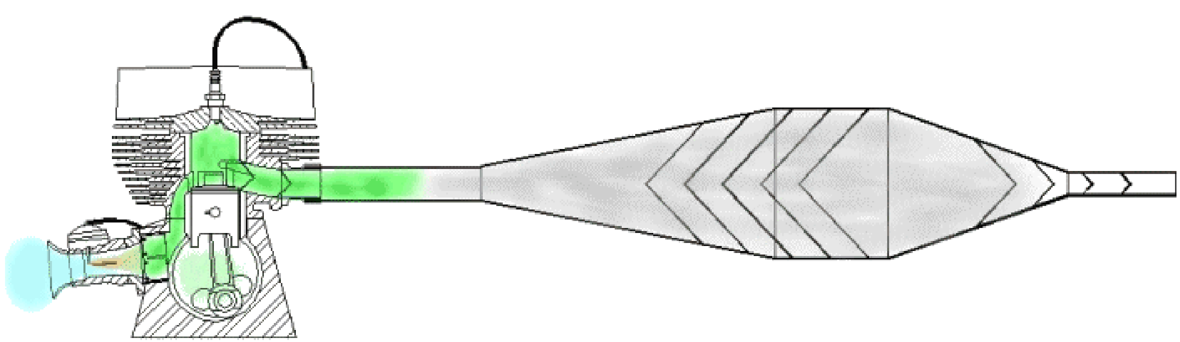


Figure 4: Two-Stroke Engine- Reflected Exhaust Pulse [11].

After Bottom Dead Center (ABDC) the piston will begin to travel upward. First the intake ports will be closed and then the exhaust port closes. As the piston rises, new air/fuel mixture is drawn into the crank case. After the piston reaches the top of the stroke, the next combustion event occurs, continuing the cycle.

2.1 Power Valves

Power valves were introduced in the production of two-stroke engines in the early 1970s. Power valves aim to improve performance at lower engine speeds where the tuned pipe is not effective. This has been done in two different ways. One option is to have a valve open a side branch chamber to the exhaust which changes the acoustic properties of the exhaust system, improving off-tune performance of the pipe. Another more common method is to raise and lower the exhaust port as a function of engine speed. This design may have valves with variable positions or multiple valves that can be successively opened or closed to vary port height. The stock Rotax 600cc two-stroke engine in Figure 5 was modified for this work had the Rotax Adjustable Variable Exhaust (RAVE) system. This system uses guillotine style slides that are able to lower the port height for lower RPM operation. Power valves are now found on many small engines used in the power-sport industry.

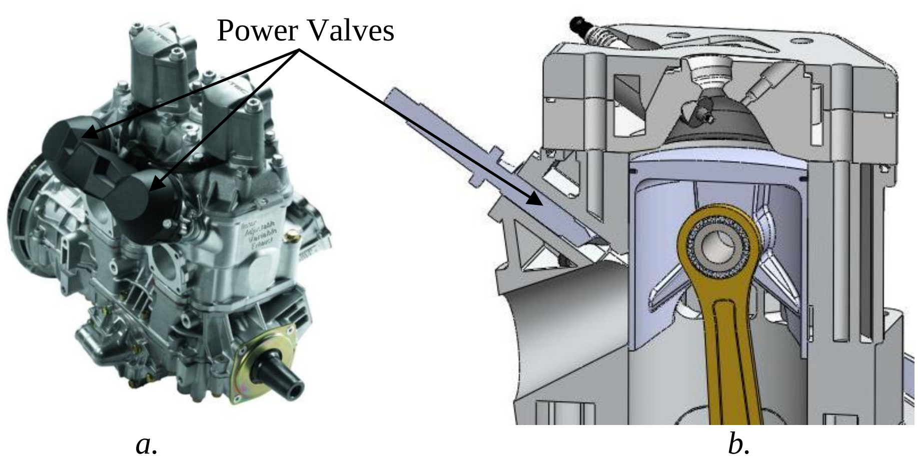


Figure 5: Stock Rotax 600cc Engine- a. Stock Rotax 600cc E-TEC b. Section View Showing Power Valve [12]

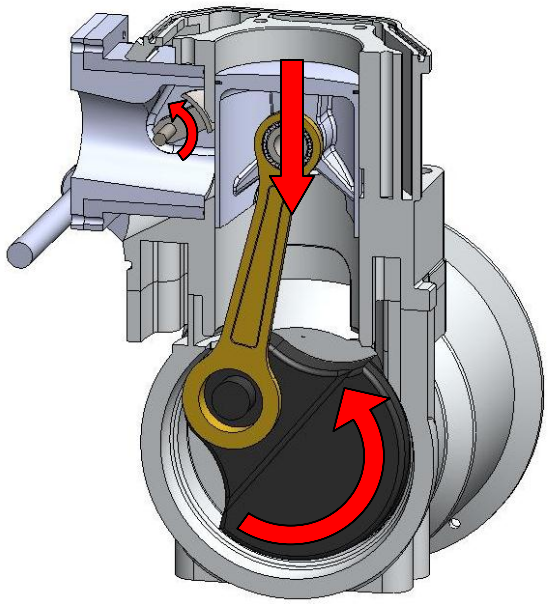
2.2 Direct Injection System

There have been many advancements in the last decade to improve the two-stroke engine that have decreased emissions and increased fuel economy. A typical carbureted snowmobile is only able to achieve an average 9 miles per gallon (MPG) [5]. The 1991 Polaris Indy 650

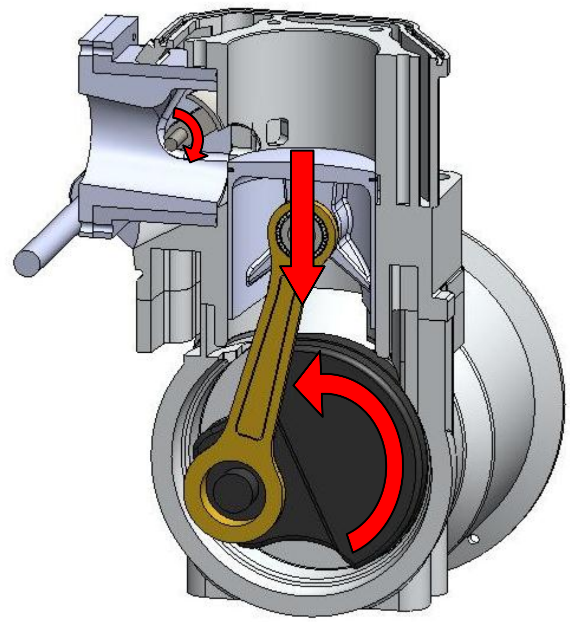
RXL EFI was the first production snowmobile with Electronic Fuel Injection (EFI), and it utilized Throttle Body Injection (TBI). EFI was a huge step in technology that was followed by Semi-Direct Injection (SDI) and recently Gasoline Direct Injection (GDI). The University of Idaho began working on applying Direct Injection (DI) to a two-stroke engine in 2003. DI injects fuel directly into the cylinder, so specially designed injectors must be used to inject fuel at a high pressure. Since the fuel is injected at a higher pressure, it takes less time to inject the required fuel, allowing for more precise injection timing control. With this technology an E-TEC system can achieve greater than 20 MPG [12]. For a more thorough explanation of DI see Justin Johnson [13] and Nate Bradbury [14]. The E-TEC DI systems can now be found on production in 600cc Skidoo snowmobiles.

3.0 Synchronous Charge Trapping

The goal of Synchronous Charge Trapping (SCT) is to keep unburnt air/fuel mixture from escaping out the exhaust port. After Top Dead Center (ATDC), the valve is synchronized so that it is fully open as the piston is at the top of the intake ports while the piston is traveling down. This allows for efficient escape of exhaust gasses. As the piston reaches Bottom Dead Center (BDC), the valve is moving to a lower position. As the piston rises After Bottom Dead Center (ABDC) the valve is in its lowest position blocking the exhaust port to prevent useable air/fuel mixture from escaping. The cycle is shown in Figure 6.



a.



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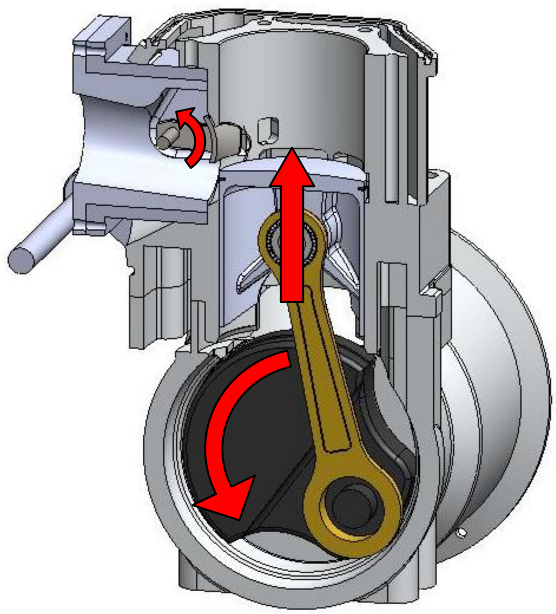

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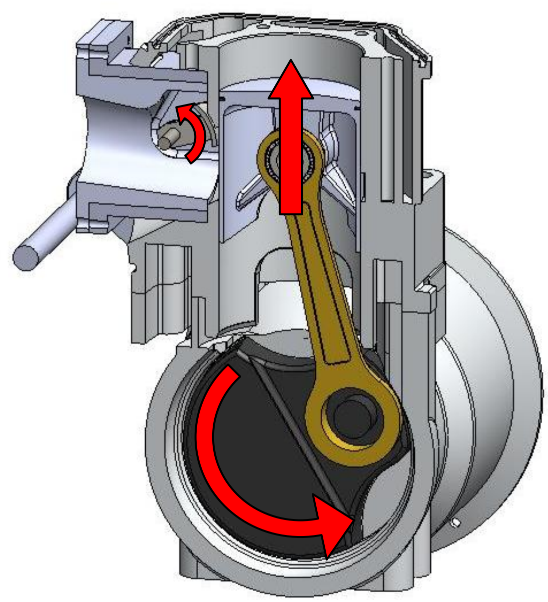


d.

Figure 6: Synchronous Charge Trapping Cycle Section View- a. After a combustion event the piston is traveling down and the valve is moving to the up position. b. As the piston opens the exhaust port the valve is in the up position. c. After BDC the piston moves to the low position blocking the exhaust port. d. As the piston moves to TDC the valve moves to the up position to begin the cycle again.

Power valves are similar to SCT valves. Both designs lower the exhaust port at lower RPMs. The advantage of power valves is their simplicity. The disadvantage is the restriction of exhaust flow. This is shown on a relative pressure vs. volume graph shown in Figure 7, comparing a standard engine cycle and an engine with SCT valves. The pressure ratio given by Equation 2, is used because the effective pressure in a two stroke engine has to take into account the force under the piston caused by the pressure in the case.

$$P_{resultant} = P_{cylinder} - P_{case}$$

Equation 2: Resultant Pressure Across Piston

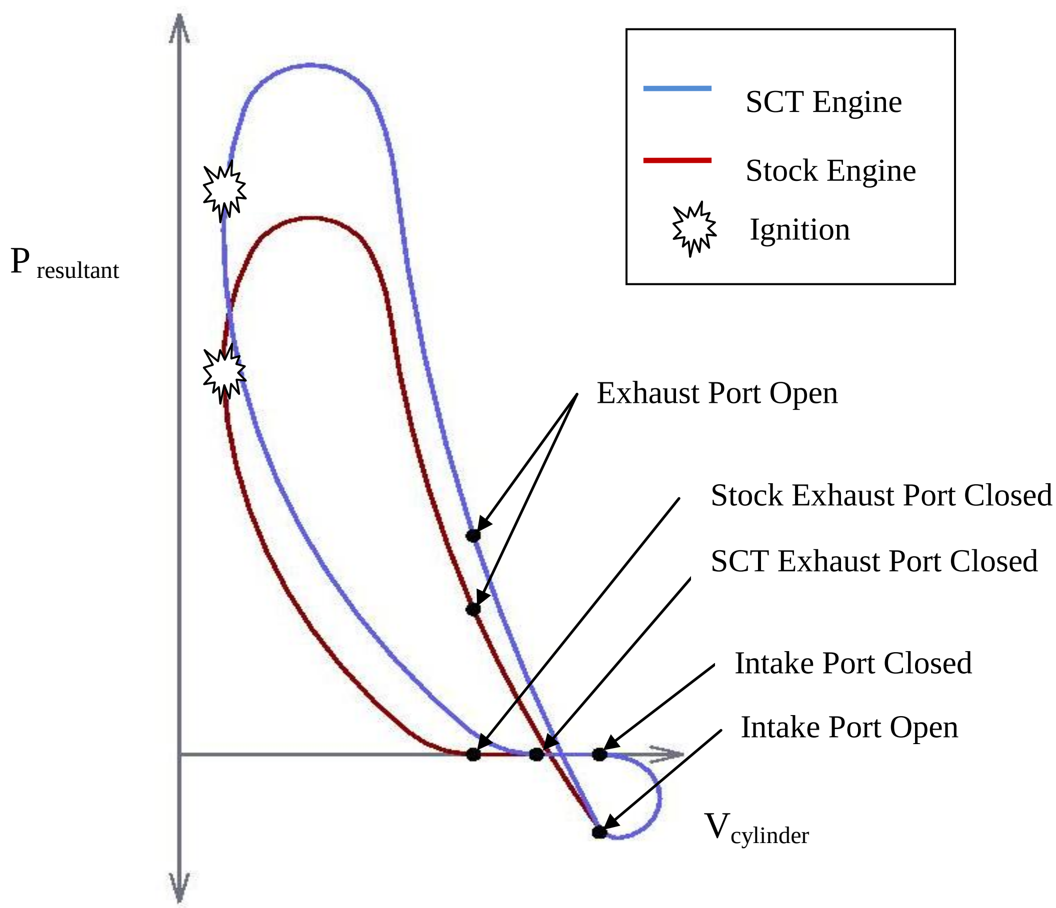


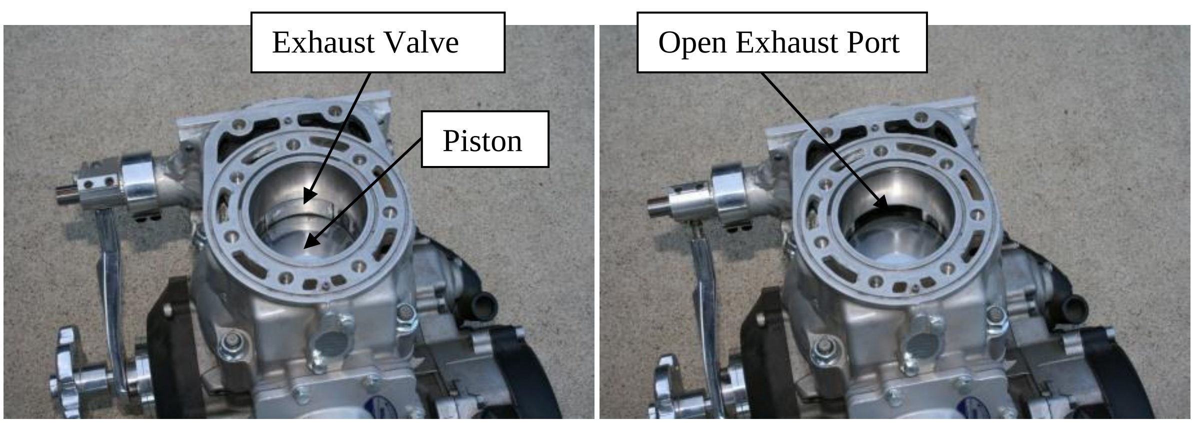
Figure 7: Theoretical Resultant Pressure vs. Volume Graph Comparing SCT to a Standard Engine at a given RPM.

On a standard two-stroke engine the exhaust port height determines the trapped volume. Blowdown begins when the exhaust port opens (EPO), and compression starts when the exhaust port closes (EPC). In a SCT engine, the exhaust port is fully open for blowdown and is lowered for the compression stroke allowing the compression stroke to begin earlier. Allthough there may be an effect on the compression ratio, the main goal of SCT is to allow efficient escape of exhaust gases and to decrease short-circuited fuel by trapping fresh air/fuel mixture.

Two-stroke tuned pipes can take up a large amount of space compared to the engine size. Not only does SCT have the potential to decrease short-circuiting, but the addition of exhaust valves also eliminates the need for a tuned pipe. Tuned pipes act as an acoustic valve, trapping useable air/fuel mixture with high pressure waves. Since mechanical exhaust valves are used the tuned pipe is no longer needed. The loss of the dynamic supercharging effect caused by the tuned pipe is made up for by the SCT valves increasing in torque througout the lower RPMs. Varying the stroke of the valves with engine speed allows the port timing to be tuned for the entire RPM band.

3.1 Boyesen SCT Design

There are currently two companies that are developing two-stroke engines with synchronous charge trapping. Boyesen Engineering has been in the power sports industry since the 1970s. Recent articles in MaxSled show their prototype designs [15]. Figure 8 Shows views of the exhaust valve in the up and down positions.



a. *b.*

Figure 8: Boyesen SCT Engine – a. Valve in Lower Position. b. Valve in Up Position. [15]

The Boyesen design uses a connecting rod driven by the crank shaft. The connecting rod is attached to a lever that actuates the exhaust valve. This system is very simple and adds only minimal complexity.

3.2 Lotus SCT Design

Lotus is also making great advances in two-stroke technology by introducing the Omnivor prototype engine. The Omnivor incorporates variable exhaust trapping, variable compression and high pressure direct injection (HPDI). The Omnivor is also able to run as a spark ignition engine and as a homogeneous charge compression ignition (HCCI) engine. The combination of these advancements allows the engine to burn a variety of fuels efficiently. What Lotus calls variable exhaust trapping is equivalent to synchronous charge trapping and is shown in Figure 9. The Lotus design has a belt driven eccentric shaft that rotates at the same speed as the crank shaft. The stroke is varied by an articulating link between the eccentric shaft and the valve link [16]. This controls the valve stroke so that it can be tuned to the engine speed for maximum performance.

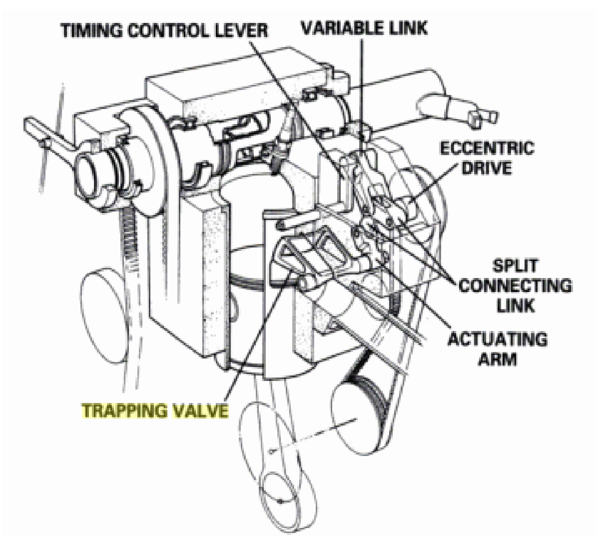


Figure 9: Lotus Omnivor Engine- Charge Trapping Valve [16, 17]

4.0 Design

There are many possibilities when considering the possible ways to design a SCT engine. Using voice coils or solenoids to electrically actuate the valves would allow for complete control and the ability to make real-time changes to the valve timing. The disadvantage of electric control is the high current demand from the actuators and the overall complexity. Since this engine may possibly be used in a snowmobile, the design had to be both adjustable and practical for use in a snowmobile chassis.

Different locations on the engine were considered for power takeoff. The mechanical oiler could be removed and replaced with electronic oiling which would allow the mechanical oiler drive to be used. The difficulty with this is that the oiler is on the intake side of the engine and the power would have to be transferred to the exhaust side of the engine. Since the oiler spins faster than the crank, gearing would have to reduce the speed back to the crank RPM. The water pump is also an option to drive the valves, as it is located on the exhaust side low between the cylinders. This would make for a shorter distance to the valve shafts. As with the oiler, the water pump turns at a higher speed and so additional gearing would need to be added to the engine.

4.1 Initial Design

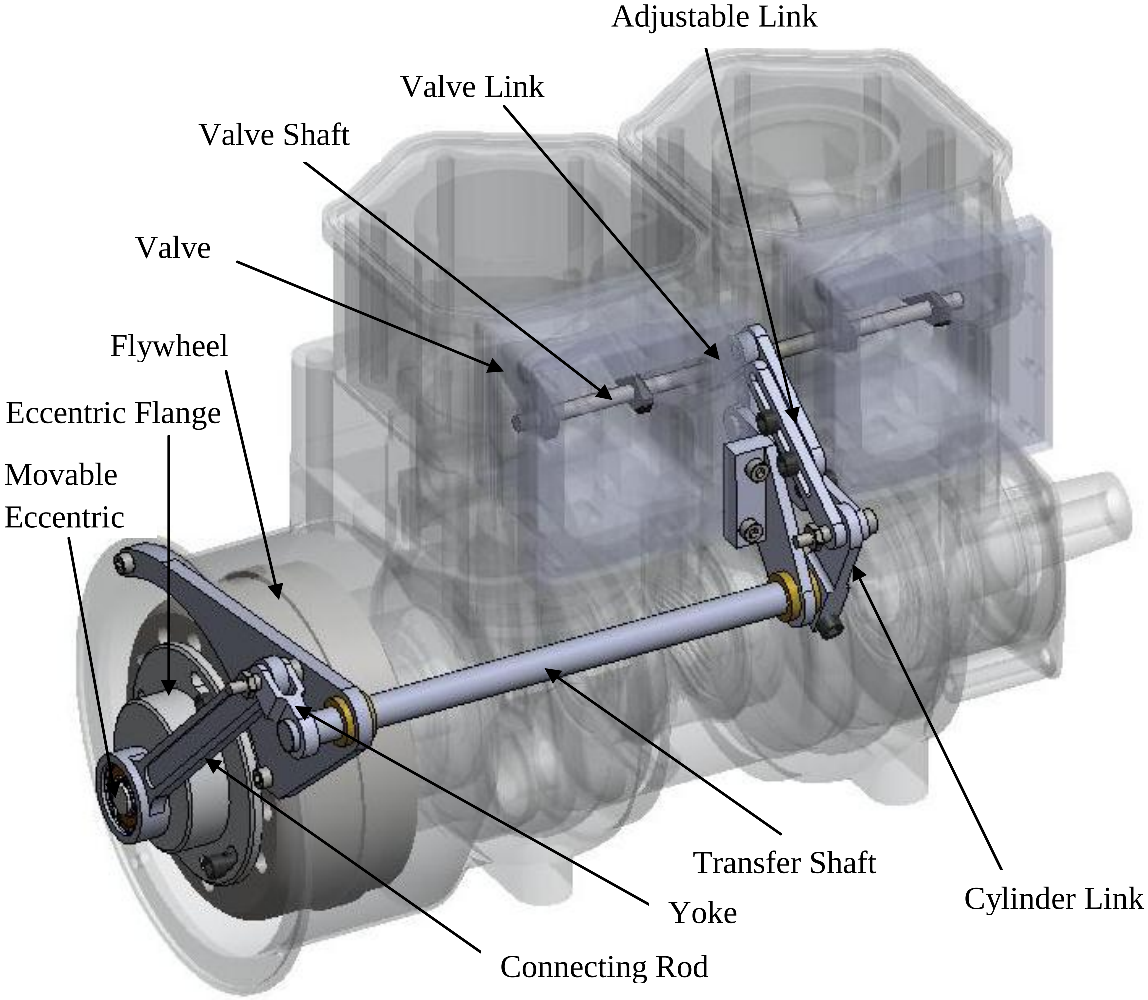


Figure 10: SCT Engine Linkage Model- Initial Design Showing Linkage

Figure 10 shows the linkage required for the exhaust valve operation. The standard components of the engine in Figure 10 are transparent. It was decided that the flywheel was the most reasonable location from which to drive the SCT linkage. On the magneto (MAG) side, the pull starter was removed. This exposed the flywheel which conveniently had a useable bolt pattern. A slotted flange was designed to attach to the flywheel. The flange supported an eccentric shaft. The slots in the flange allow the flange to be rotated, adjusting the angle with respect to the crank. The eccentric shaft was also made adjustable so that it could be moved radially in or out. A con-rod (connecting rod) connected the eccentric shaft to the yoke, which rotated the shaft transferring motion to the cylinders. The oscillating motion was transferred to a location between the cylinders on the exhaust side of the engine.

The shaft drove the cylinder link, which is connected to the two adjustable links. The adjustable links were useful in adjusting the valve position so that they moved in the same range. The valve links were attached to the valve shafts and transferred the oscillating motion into the cylinder to the exhaust valves. The MAG side link was fixed to the shaft with the cylinder link adjustable.

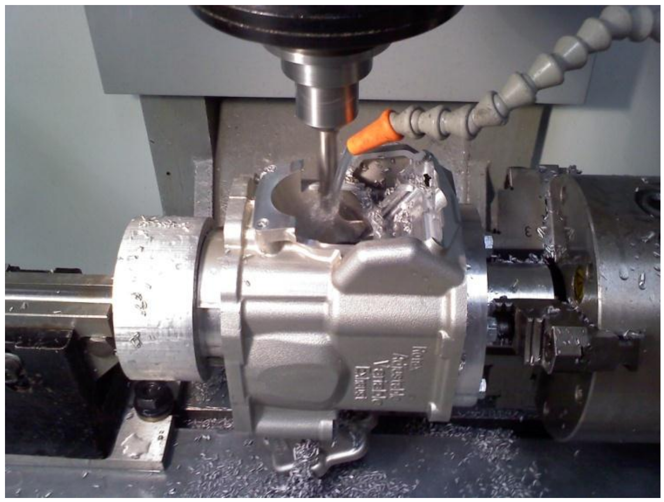


Figure 11: SCT Cylinder – Machining Process for Welded Insert.

The valve assembly needed to be removed to replace and modify valve parts. To achieve this, the exhaust side of the cylinder was machined, fill welded and re-machined to create a gasket surface. The process of machinging the exhaust side of the cylinder is shown in Figure 11.

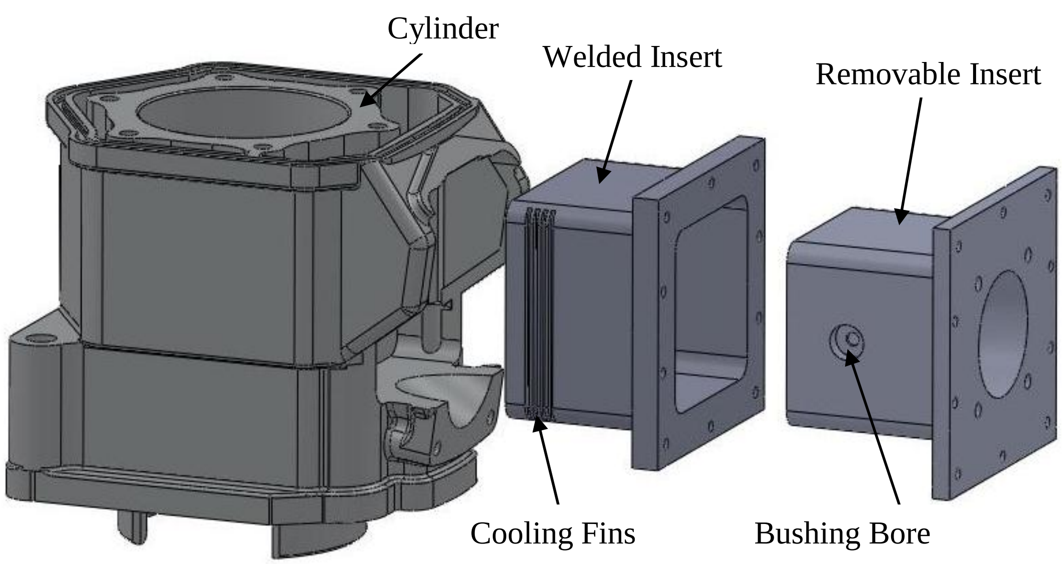


Figure 12: SCT Cylinder- Exploded View of Cylinder, Welded and Removable Insert

A permanent insert was fit into the machined gasket surface and welded into place, as shown in Figure 13. The permanent insert had a flange that allowed a removable insert to be bolted in. Because the modifications decreased the surface area that contacted the coolant, excess heat was an issue. The coolant on the exhaust side of the cylinder travels through two passages down the sides of the permanent insert. These areas were finned to try to improve heat transfer as much as possible and are shown in Figure 12 and Figure 14. The removable inserts house the high temp Graphalloy bushings that support the valve shaft. To protect the bushings from the high exhaust temps, the bore for the bushings was a blind hole with the thru hole only large enough for the valve shaft. This design greatly decreased the exposed portion of the bushing. Graphalloy bushings were chosen mainly based on availability. Graphalloy can withstand the high temperatures in the exhaust flow, but does not have the required properties to withstand the load at high RPM. The Graphite Metalizing Corp. that sells the bushing recommends that the service limit for dry applications be determined by Equation 3.

$$\frac{RPM * Total\ Shaft\ Load}{Bushing\ Length} < 46{,}000$$

Equation 3: Service Limit Calculation for Graphalloy Bushings [18]

Low oxygen environments and lubrication can increase the service limit by 7 to 10 times [18]. Exhaust is more likely to have low oxygen content. The residual two-stroke oil coats the inside of the exhaust, potentially adding lubrication to the bushings.

Figure 13: Cylinder Showing Permanent Insert, Welding and RTV Junction.

To seal the permanent insert to the cylinder, high temperature RTV sealant was applied to the sealing surface and the insert was clamped in place for welding. To prevent the RTV seal from failing, a groove was milled into the sealing face, shown in figure 14, so that the RTV could fill it and have a thickness to allow for expansion and shifting due to changes in temperature.

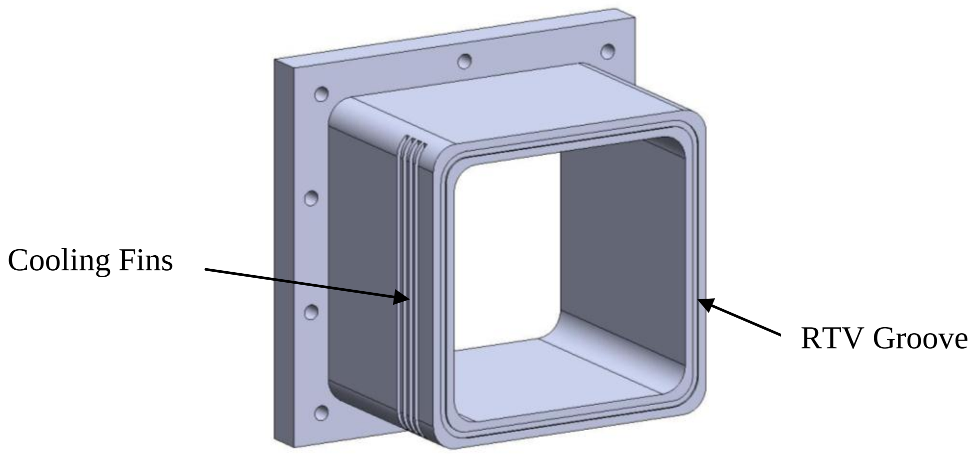


Figure 14: SCT Engine- Welded Insert View Showing Grooves

Even though the fit between the removable and permanent insert is as tight as possible, the small air gap greatly decreases heat transfer. To increase heat transfer at this boundary, a conductive paste made by AI Technology was applied to the mating surfaces of the inserts. The typical thermal conductivity of the paste is 8.6 W/(m*C) [19].

The welding of the permanent insert to the cylinder caused warpage of the top and bottom of the cylinders. The cylinders were machined so that equal amounts of material were taken from each. To compensate for the material taken from the cylinders a thicker 0.028 inch base gasket was used. A cutout also had to be machined into the cylinder for the exhaust valve. This was done using the SolidWorks model and CNC milling. A clearance of 0.040 inches was left so that the exhaust valve would not interfere with the piston or cylinder valve cutout.

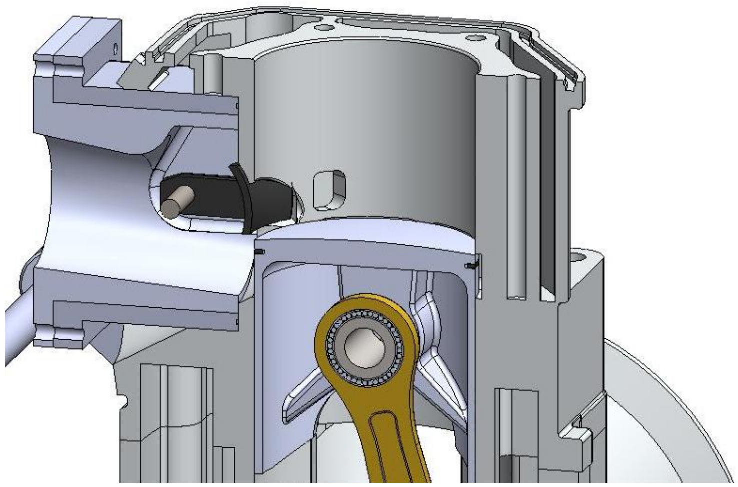


Figure 15: SCT Cylinder Assembly – Cross-Section View

To allow for the required bolt pattern, attempt was made to center the inserts and minimize warpage from welding. The exhaust tunnel was altered significantly and the outer exhaust ports were blocked. The new exhaust tunnel geometry is shown in Figure 15. The effects of these changes were not explored but need to be mentioned.

Since the tuned pipe would not be necessary with the SCT engine, a mid-pipe was fabricated to take the place of the stock pipe. The pipe was made from two and a quarter inch diameter stainless steel pipe and reduced to a two inch diameter to fit into the stock muffler. The lambda bung was placed approximately three inches from the beginning of the mid-pipe.

Figure 16: SCT Engine – Mid-Pipe

4.2 Redesign

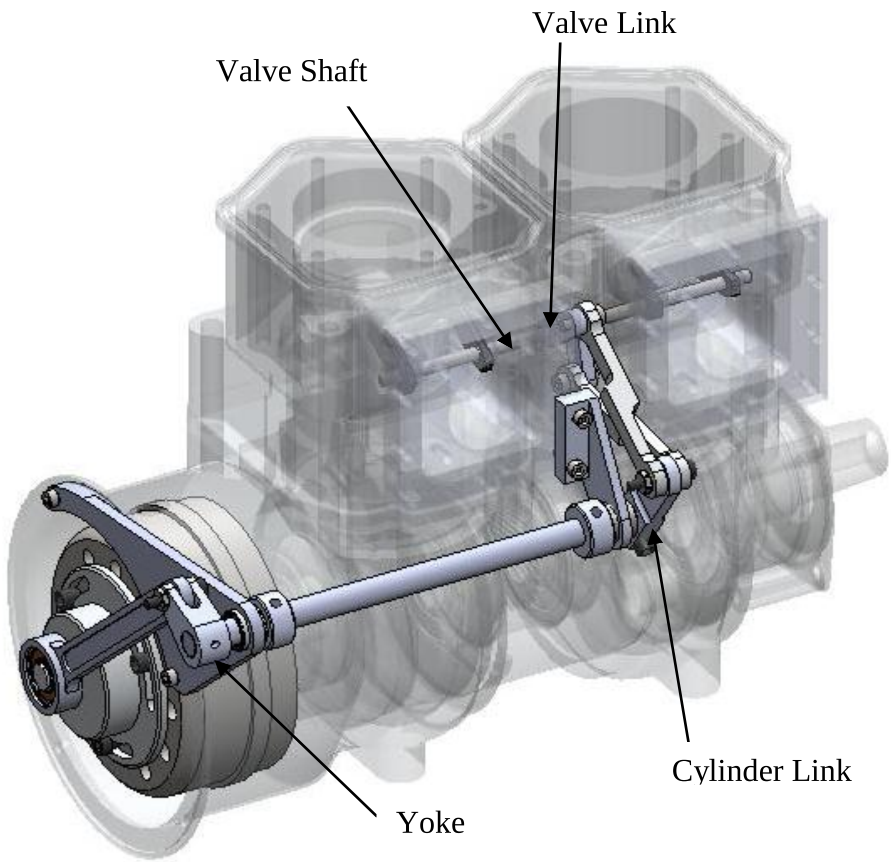


Figure 17: SCT Engine- Redesign Showing Linkage

A stroboscope was used to check for harmonic instabilities at different RPMs. There were no harmonic issues found visually, but problems occurred after short run times. It was noted that the linkage would need to be redesigned and the links would need to be made fixed instead of adjustable for reliability. The initial components helped in the learning process and gave needed experience to redesign the fixed links.

The modifications are shown in Figure 17. The yoke was thickened so it could be pinned to the long shaft. The cylinder link was also pinned to the shaft to prevent slippage. The main issue was with the valve links and valve shafts. The valve links were breaking free of the shafts. To fix this, the new valve links were made of steel with a two thousandths inch press fit. Since keying the valve link was not possible with our tooling, the link was welded to the shaft to create a permanent connection. The set screws used to fix the valves to the valve shaft also became a problem because thread locking compounds could not be used in the higher temperatures and jam nuts had no effect.

The final solution was to use #5/0 tapered pins machined from 17-4 PH stainless steel to fix the valves to the valve shaft. This required a valve leg to be removed and a new machined piece to be welded on to provide material to drill and ream into for the tapered pin. The valve with modified leg is shown in Figure 18.

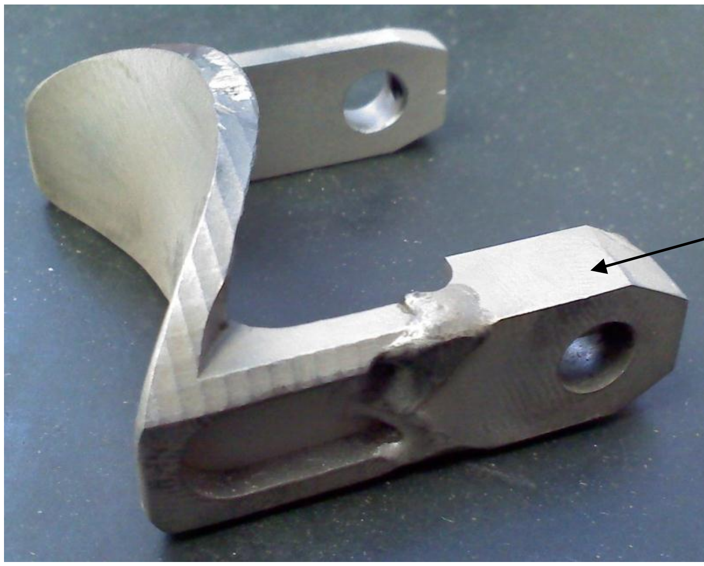


Figure 18: SCT Exhaust Valve – Modification to Valve Leg for Tapered Pin.

A Tungsten Inert Gas (TIG) setup was used to weld the valve together. Welding hardened the material to the point that it could not be machined. The valves had to be annealed in an oven before the holes could be drilled and reamed for the tapered pin. The annealing process requires the material to be heated 100 degrees Fahrenheit (F) above the critical temperature, and then slowly cooled [20]. *The Handbook of Engineering Materials* by Miner and Seastone recommends heating 440C stainless steel to between 1550-1650 degrees F for full annealing [21].

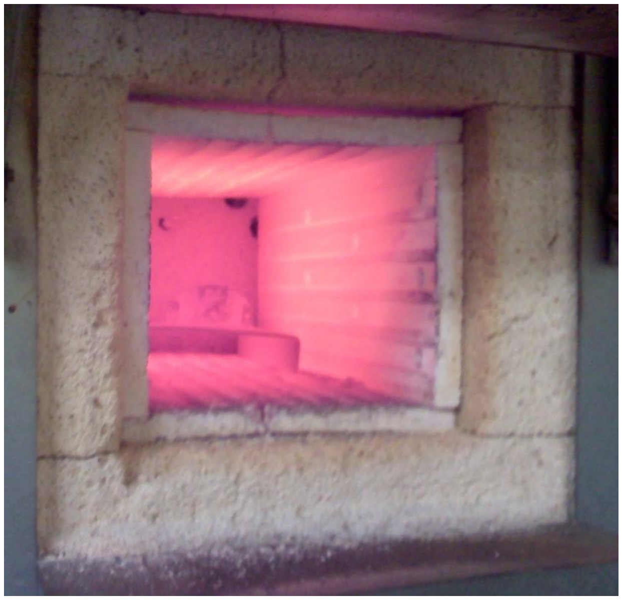


Figure 19: SCT Engine – Valve Annealing Process in Oven

The valves were heated up to 1600 degrees F and then slowly cooled by turning off the oven and opening the door approximately an inch wide, until the temperature had reached 700 degrees F. Below 700 degrees F, the valves could be removed and were left to cool in the open air. Figure 19 shows the valves being heated in the oven.

5.0 Engine Testing

The SCT modified two-stroke Rotax engine was tuned and tested at the University of Idaho. The engine was tuned using measured torque, exhaust gas temperature (EGT), lambda, head temp and in cylinder pressure traces.

5.1 Equipment

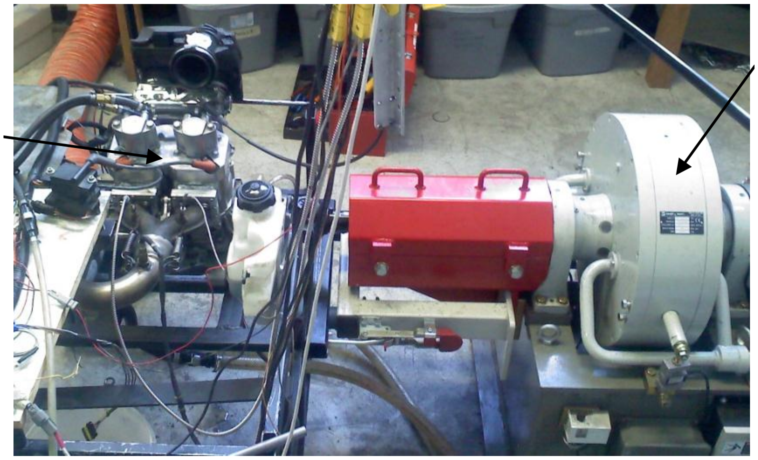


Figure 20: SCT Engine on Dynamometer

5.1.1 Dynamometer

Torque and horsepower are measured using a model FE-260-S Borgi Savri eddy current dynamometer shown in Figure 20 with a SuperFlow control/monitor system. This dynamometer is rated at a maximum power of 260 hp (191.2 kW) and speed of 12,000 rpm, which is well within the expected output of the SCT engine. The SuperFlow WinDyn Software is capable of running tests and data acquisition of connected sensors, torque and power output of the engine.

5.1.2 Oxygen Sensor

The Air/Fuel Ratio (AFR) is very useful in tuning an engine. The AFR is the mass of air divided by the mass of fuel as determined by Equation 4.

$$AFR = \frac{mass_{air}}{mass_{fuel}}$$

Equation 4: Air-Fuel Ratio

The air/fuel mixture is said to be stoichiometric if there is exactly enough oxygen to entirely burn the fuel in the mixture. The stoichiometric air/fuel mixture for gasoline is 14.7. Since the AFR changes for different fuels, lambda is typically used. Lambda is the AFR, normalized by dividing by the stoichiometric AFR and is shown in Equation 5 [22].

$$\lambda = \frac{AFR}{AFR_{Stoich.}}$$

Equation 5: Lambda Calculation

Lambda can be difficult to use with a two-stroke engine because the high UHC particulate emissions coat the sensor and the extra air that is passed through the engine give an incorrect reading [23]. The Lambda value was determined by finding rich and lean limits of combustion. Lambda was measured using an Innovate LM-1 Sensor Controller.

5.1.3 Exhaust Gas Temperature and Head Temperature

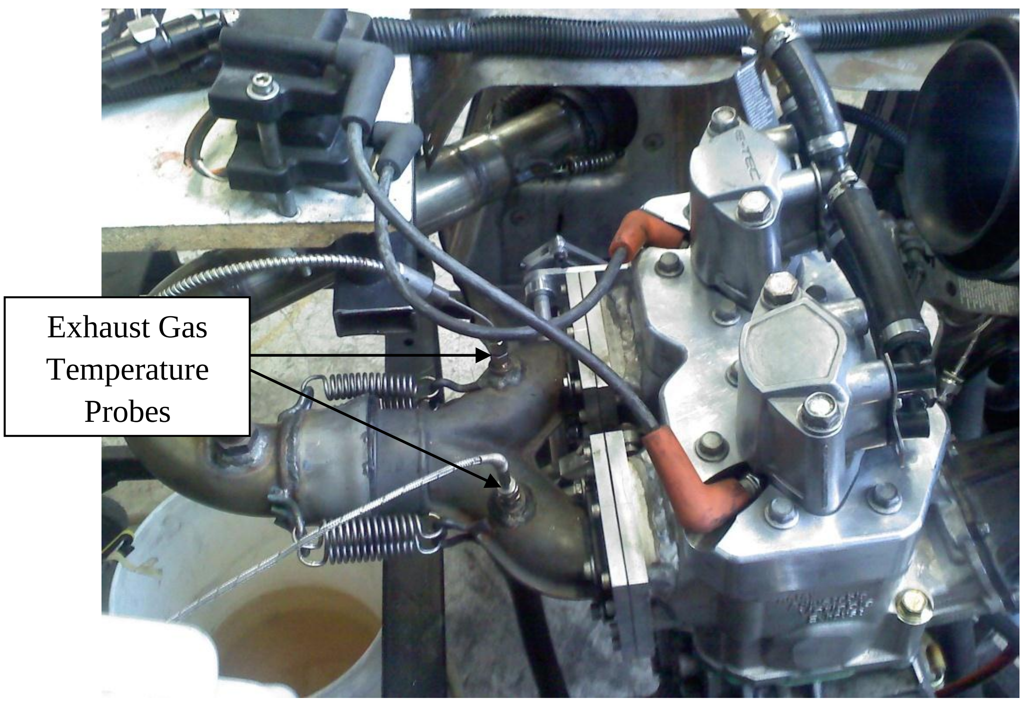


Figure 21: Y-Pipe with Exhaust Gas Temperature Probes

The exhaust gas temperature (EGT) is a very useful tool in tuning a two-stroke engine at higher temperatures to prevent engine damage. To monitor the exhaust temperature of the engine, two exhaust probes were placed in the Y-pipe to give exhaust temperatures for each cylinder. Arrows in Figure 21 point to the two EGT probes in the Y-pipe.

5.1.4 In-cylinder Pressure Trace

In-cylinder pressure was measured with a Kistler 6052C shown in Figure 22. The head was designed with pressure taps in both cylinders for installing pressure sensors. A pressure trace was only needed for one cylinder, and it was taken from the MAG side. The pressure trace gives a relative number and was not recorded. The pressure trace was solely used for detecting knock/detonation. Knock is the sound that is heard by detonation occurring in the cylinder. Detonation occurs when high in-cylinder temperatures cause fuel to auto ignite after spark initiated combustion has started. The two flame fronts collide, causing very high pressures [24]. In some cases detonation can also be caused by incorrect ignition timing. These high pressures can cause damage to the piston. When knock occurs, a spike could be seen in the pressure trace and changes could be made to the spark and fuel timing to correct the issue.

Figure 22: Kistler 6052C In-cylinder Pressure Sensor for Detecting Knock [25].

5.1.5 Fuel Cart

A 710 Max Machinery fuel measurement system was used to measure fuel consumption. The error for the system is claimed to be 0.75% or better. The fuel cart will also deliver a regulated fuel pressure up to 125 pounds per square inch (PSI) [26]. The GDI system runs at a 40 PSI. The fuel consumption is converted to a Brake Specific Fuel Consumption (BSFC). BSFC is commonly used, it is a way to relate engines of different types and sizes because the number is based on the amount of fuel used per unit of power output. The units of BSFC used was kg/(kW*hr).

5.2 Tuning Strategy

To determine if SCT is a beneficial technology to pursue the modified SCT engine was compared to that of the stock engine. The SCT engine was tuned using two different methods: matching stock torque numbers while tuning for best BSFC and tuning for maximum torque. This was done under the same operating conditions as the stock engine while reporting the resulting BSFC and torque values. Since a stock engine is tuned to be a balance between power output, fuel economy, runability and emissions it would not be a reasonable comparison to tune the SCT engine for power or fuel economy alone. By matching torque, the benefits of SCT should be seen in lower BSFC numbers indicating it is a more fuel efficient design.

6.0 Results

Unfortunately, the valves were damaged in the process of improving the design. Alternative designs for the valves are discussed in the Future Work section. Because of this issue, the engine performance data available was limited.

6.1 Tuning for BSFC While Matching Stock Torque

Results were compared with data taken by Justin Johnson [13] and are shown in Figure 23. Data was taken using the stock engine at three throttle positions: 10, 20 and 30 percent throttle. Total fuel was measured over a 30 second period for each data point and used to calculate BSFC.

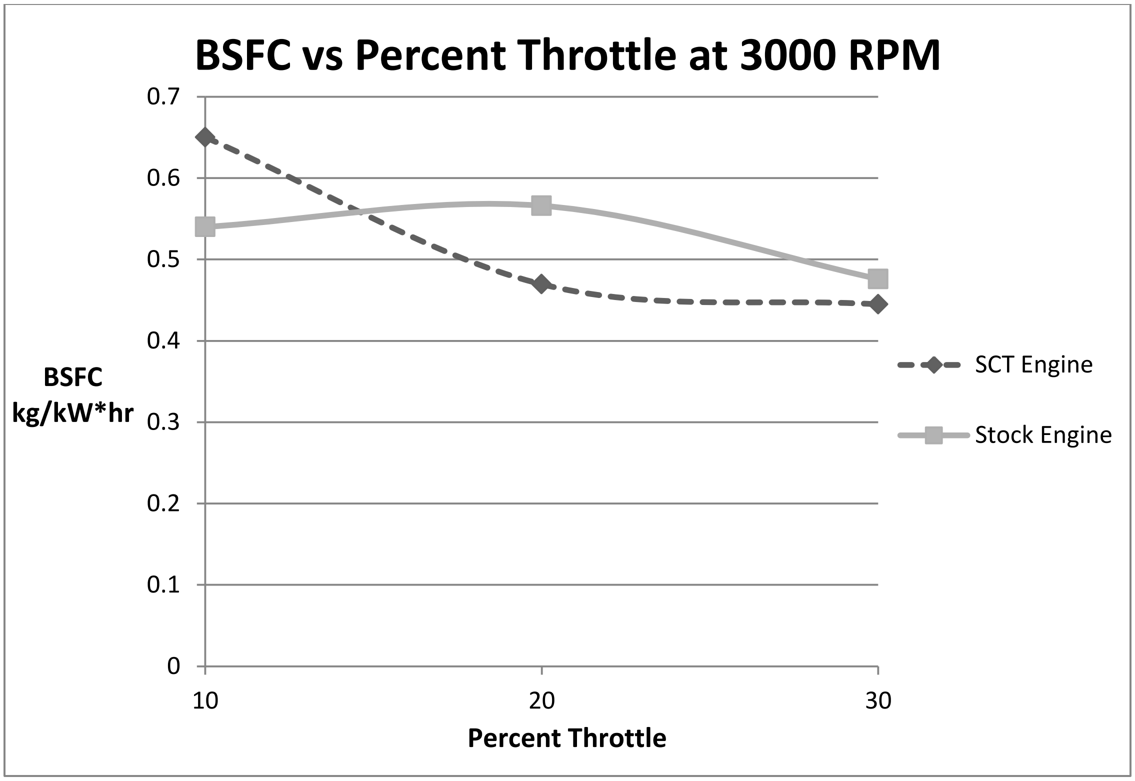


Figure 23: Graph Comparing SCT to Stock engine BSFC vs. Percent Throttle at 3000 RPM Matching Stock Torque.

The SCT engine was difficult to run at low throttle positions and engine speeds. This can be seen in the 3000 RPM comparison. The BSFC for the SCT engine is higher than the stock engine. As the throttle position and engine speed increased, the SCT engine BSFC dropped and was consistently better than the stock engine. At 3000 RPM, the SCT engine had a 17 percent increase in BSFC at 10 percent throttle. BSFC dropped by 21 percent at 20 percent throttle, and then 7 percent at 30 percent throttle.

At 3500 RPM the engine showed very good results. For 10 percent throttle the BSFC was lower by 34 percent. At 20 percent throttle the BSFC was 22 percent improved and at 30 percent throttle a decrease of 37 percent was measured.

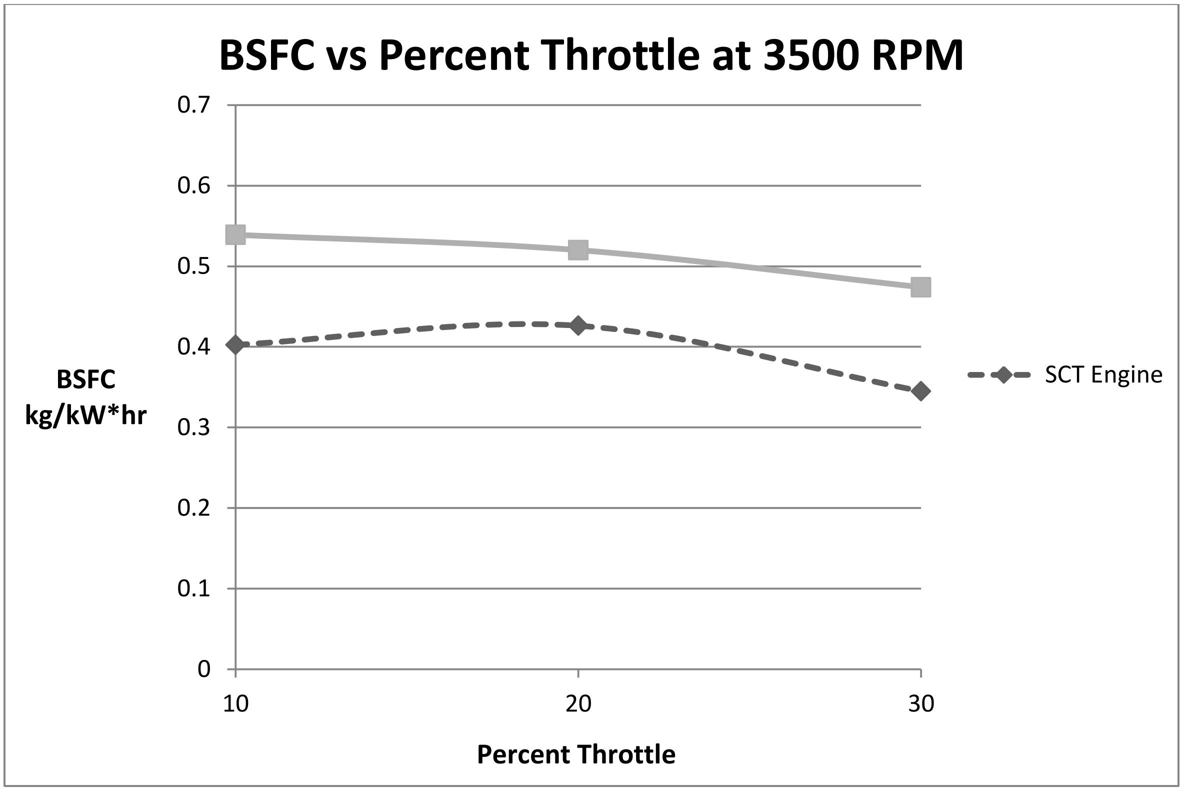


Figure 24: Graph Comparing SCT to Stock engine BSFC vs. Percent Throttle at 3500 RPM Matching Stock Torque.

6.2 Tuning for Maximum Torque

The SCT engine was also tuned for maximum torque and BSFC data was recorded and is shown in Figure 24. This time two throttle positions were used, 25 percent throttle and 50 percent throttle. Again, total fuel was measured over a 30 second period for each data point and used to calculate BSFC.

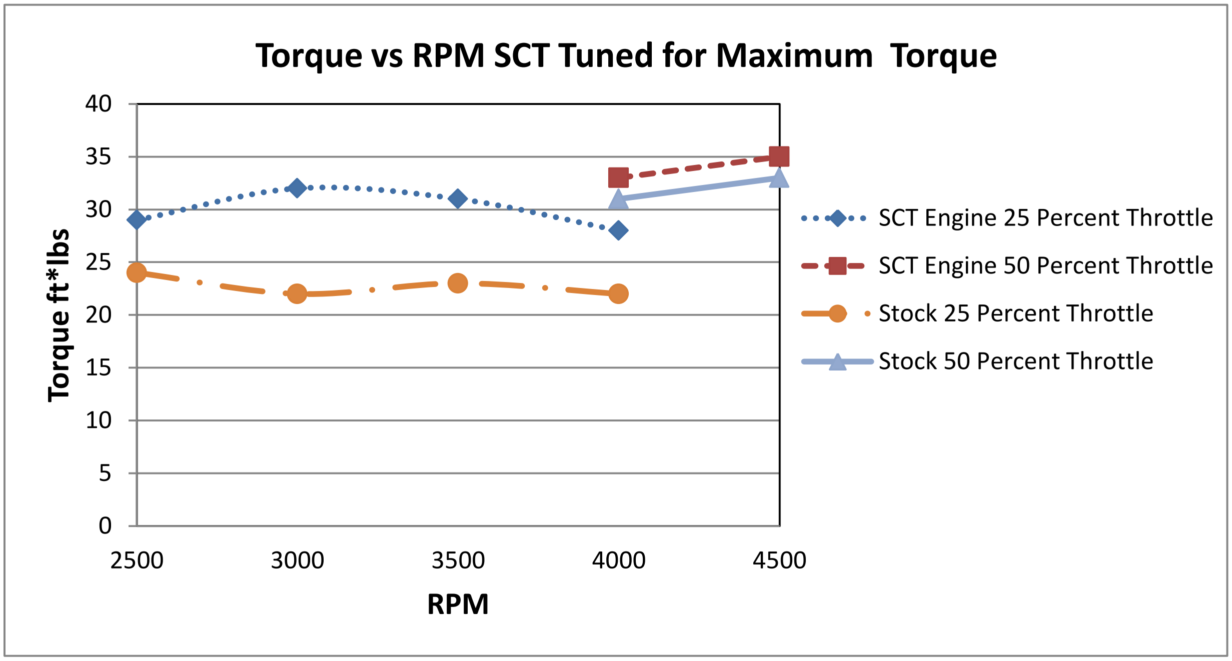


Figure 25: Graph Comparing SCT to Stock engine – Torque vs. RPM Tuning for Maximum Torque.

Upon observation of BSFC, the percent difference between the stock and SCT engine is 3 to 6 percent. The torque graph shows significant improvements at the lower RPMs. This is expected since the valve is in it lowest position, which should improve low RPM operation. In the future work section, recommendations are made for further testing necessary to explore effects of valve stroke to boost performance at higher engine speeds.

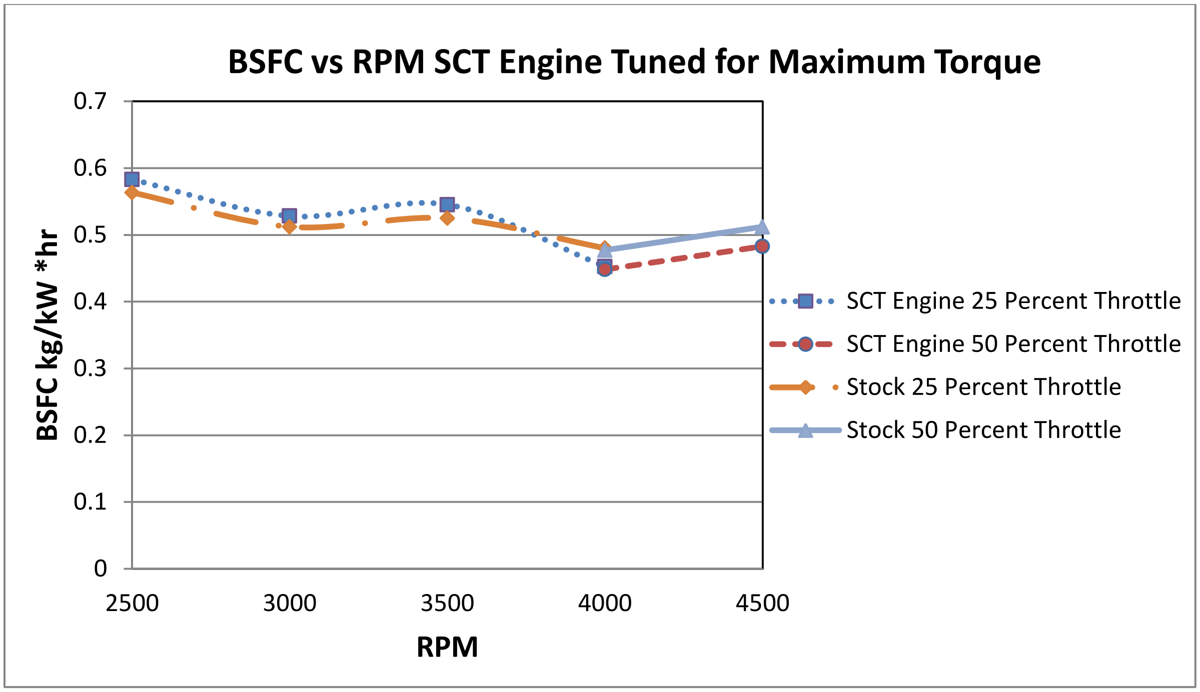


Figure 26: Fuel Used SCT vs. Stock engine – BSFC vs. RPM Tuning for Maximum Torque.

Figures 25 and 26 show that when tuning the SCT engine for torque, the fuel economy is comparable to stock engine. The difference is in the power density of the engine. When tuned for torque, the SCT engine has a higher power-to-weight ratio than the stock engine, while still matching the stock BSFC. This demonstrates that a smaller displacement SCT engine could have the same performance as a larger conventional engine. This would allow for a more compact and nimble chassis.

7.0 Conclusions

The goal of this research was to design, fabricate and test a SCT prototype engine for the purpose of determining if SCT was a system worth pursuing and developing for use in a snowmobile.

The results from tuning for best BSFC while matching stock torque shows that the SCT engine is able to perform better than the stock engine at lower engine speeds. At 20 percent throttle, the SCT engine showed significant improvement over the stock engine. As the throttle position and engine speed increased, the SCT engine BSFC dropped and consistently surpassed the stock engine. The average decrease in BSFC was 17 percent less than stock. Further testing is needed to demonstrate the performance at higher engine speeds.

The results from tuning for maximum torque also affirm the benefits of SCT. The average torque output of the engine was increased by 32 percent while differing at most 6 percent from the stock BSFC. This suggests that there is potential for performance improvements at higher engine speeds, with a shorter valve stroke. If similar gains can be achieved at higher engine speeds developing SCT engines will allow for lighter and more agile snowmobiles.

An area of concern was the life of the Graphalloy bushings supporting the valve shafts. Two bushings did have to be replaced due to damage from broken valve shafts. After testing ended, the inner dimensions of the bushings were measured and found to be unchanged. This indicated that the bushing could handle the load and RPM, possibly as a result of residual two-stroke oil in the exhaust.

8.0 Future Work

8.1 SCT Engine Recommendations

As with any new design, there are many aspects that may be improved through modifications or redesign. Future work for this project will include tuning and modifications to improve and validate the design.

8.1.1 Tuning

A major area to explore are the effects of different valve strokes. To get a base idea of the effect of the valve stroke at least three ranges would be recommended. The first range would be a full stroke from fully open to covering half of the intake port, then two thirds and one third full stroke. Power valve position is similar to SCT valve strokes. At higher engine speeds more time is needed for exhaust scavenging to occur and so a short valve stroke should improve performance at higher RPMs.

Once a reasonable amount of data is gathered defining the relation between valve stroke and engine speed, the relation between valve stroke and throttle position should be explored.

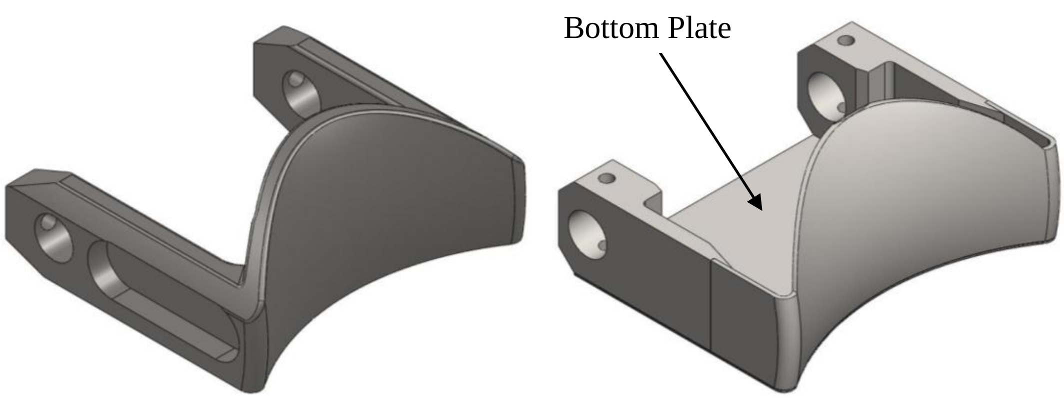
This will show if valve stroke is a function of RPM or the combination of RPM and throttle position.

8.1.2 Valves

Reducing the weight of the valves would significantly reduce the load on the linkages and Graphalloy bushings. One possible method is to make the valves from spun sheet metal. This would allow for lightweight valves that could withstand the heat of the exhaust.

The design of the valves needs to be modified to once again allow for removal in case of damage or the need to make modifications. It is possible to remove the pins locking the valve to the shaft but is not a practical design.

As the SCT design is refined, the exhaust transitions need to be improved over the current design. This would include improving the valve design to prevent turbulence from occurring behind the valve face.



a. *b.*

Figure 27: SCT Valves – a. As-Built Exhaust Valve. b. Spun Sheet Metal Exhaust Valve.

Using the process of metal spinning, the exhaust valve face could be made very thin as shown in Figure 27. A new valve was designed using this method and the calculated moment of inertia was decreased from 0.0658 lbs*in^2 to 0.0182 lbs*in^2, a reduction to 27 percent of the original valve. The new valve design could also incorporate a bottom plate to improve exhaust flows.

8.1.3 Adjustability with Engine Speed

The major advantage of SCT is the ability to tune the engine to limit short circuiting throughout the RPM band. Modifications to this design could be done in different ways. One option is to use an articulation link similar to the one found in the Lotus design. This link could be electronically controlled which would give precise control and would be helpful when tuning the engine.

Another prospect would be to use centrifugal force to move the connecting rod eccentrically to the crank. Dynamometer testing will be needed for the current design because a centrifugally controlled system would require knowing the valve stroke and engine speed relation.

8.1.4 Mid-pipe Effect on Sound

The combination of the valves and mid-pipe had an unexpected effect on sound that was noticeable in the testing laboratory. Sound data was gathered at lower engine speeds with the valves removed to determine if the mid-pipe was the main factor. Upon observation, there was no significant change in the sound level between the tuned pipe and SCT mid-pipe. At higher engine speeds, as the tuned pipe is effective the SCT mid-pipe may show a comparative decrease in sound due to its small surface area and thicker tube wall. Sound data should be taken with the valves active to determine a relative improvement.

8.2 Noise Reduction Recommendations

Even though this research is focused on the design and modifications of the engine, a main purpose of this work is to further snowmobile technology.

8.2.1 Skid Absorbers

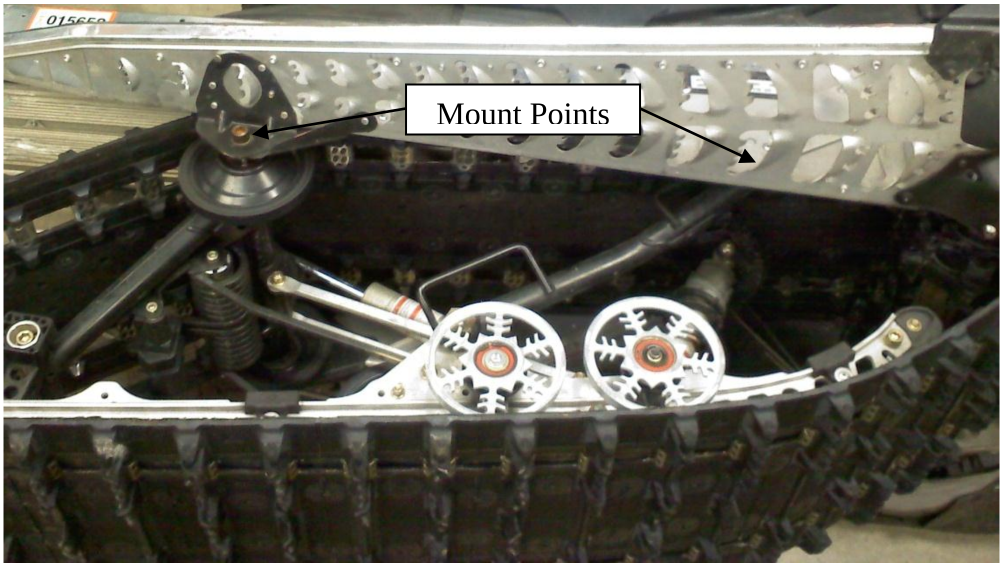


Figure 28: Skidoo MXZ Chassis- Skid to Tunnel Mounts

The skid in the stock Skidoo MXZ has metal on metal connections to the tunnel as shown in Figure 28. The tunnel is constructed from sheet aluminum and has large flat areas that can convert mechanical vibrations from the skid into sound energy. This may explain why using track skirts in the past did not significantly decrease the track noise [27, 28]. Isolating the skid from the tunnel using properly selected damping materials should be explored for its potential sound dampening benefits.

8.2.2 Laminar Flow Muffler

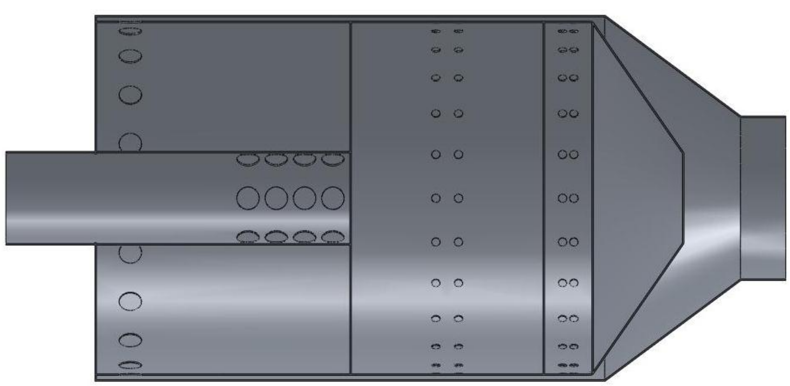


Figure 29: Laminar Flow Muffler- Model Showing Possible Design with Converging Cone for Exhaust Catalyst.

The difficulty in redesigning a muffler for a production engine is that serious work from large companies has gone into the acoustic design and so improving on the design is difficult. Gordon Blair mentions laminar flow mufflers in his book *Design and Simulation of Two-Stroke Engines* [8]. Laminar flow mufflers are designed to diffuse exhaust over a larger exit to cause laminar flow in the exiting exhaust. The advantage of a laminar flow design is that it is lightweight and can be designed to work very well within a narrow operation window.

The snowmobile Continuously Variable Transmission (CVT) shifts so that the engine at Wide Open Throttle (WOT) is at maximum RPM. Knowing the engine speed for a majority of the sound test allows the muffler to be designed for the frequency of the exhaust pulses at maximum engine speed.

UI CSC has had indeterminate results from previous laminar flow exhaust testing. One major improvement from past designs is to implement a stabbing technique mentioned by Blair [8] as a cure for turbulent noise within the muffler. This is difficult because the holes must be stabbed after the inner tube has been rolled. A punch and die set was made by members of the UI CSC team. It is currently a flat die and will flatten the rolled tube when attempting to stab the holes. A new two piece curved die will need to be made to clamp the tube in its curved shape.

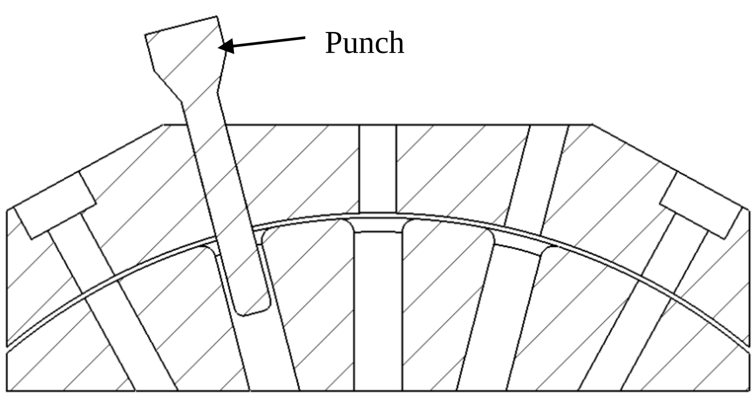


Figure 30: Laminar Flow Muffler Punch and Die Section View

Two methods have been used to model the muffler acoustics. The equations for the building blocks of the laminar flow muffler are given in Blair [8]. Another method using Helmholtz resonator equations from *Fundamentals of Acoustics* [29] was used to validate calculations.

Concerns were found with the model from Blair that makes it difficult to determine the proper way to design the muffler correctly. This is potentially from misinterpreting the equations. Both models have been well commented and are attached in Appendix F and G for further development.

Figure 29 of the muffler represents it with attached cones to reduce the exit area to fit exhaust catalyst dimensions. The effort to create laminar flow at the exit may be negated by the use of a catalyst which will have its effect on the exhaust after the laminar portion of the muffler. An option would be to move the catalyst upstream of the muffler or integrate it in the muffler to decrease radiated heat.

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Appendix A – SCT Drawing Package

ITEM#	PART NUMBER	PAGE NUMBER	QTY.
1	SCT-01-01 Eccentric Flange	39	1
2	SCT-01-02 Eccentric Moveable	40	1
3	SCT-01-03 Con-Rod	41	1
4	SCT-01-04 Yoke	42	1
5	SCT-01-05 Transfer Shaft	43	1
6	SCT-01-06 Cylinder Link	44	1
7	SCT-01-07 Link	45	2
8	SCT-02-01 valve	46	2
9	SCT-02-02 valve shaft	47	2
10	SCT-02-03 Valve Lever	48	2
11	SCT-02-04 Removable Insert	49	2
12	SCT-02-05 Welded Insert	50	2
13	SCT-02-06 Cylinder	51	2
50	SCT-02-07 valve leg	52	2
15	SCT-02-09 Tapered Pin	53	2
16	SCT-03-01 MAG Bracket	54	1
17	SCT-03-02 Cylinder Bracket	55	1
19	SCT-03-03 Mid-Pipe		1
20	Snap Ring .5 91590A122		1
21	Snap Ring 1.125 91580A221		1
22	Bushing .25x.375x.25. 6381K412		2
23	bushing .25x.5x.25 7943K22		4
24	Shoulder Screw .25x.75x10-24 90298A539		1
25	SHCS 10-24x.625 91841A011		30
26	SHCS .25-20x.5 92196A537		4
27	Nut 10-24 91841A011		2
28	Needle Bearing .5 x .6875 x .5L 5905K23		2
29	Needle bearing adapter		2
30	Stop Collar .5 8865A235		2
31	Y Pipe		1
49	Bearing .25x.375x.125 57155K355		8
50	Bearing .5x1.125x.438 2342K186		1
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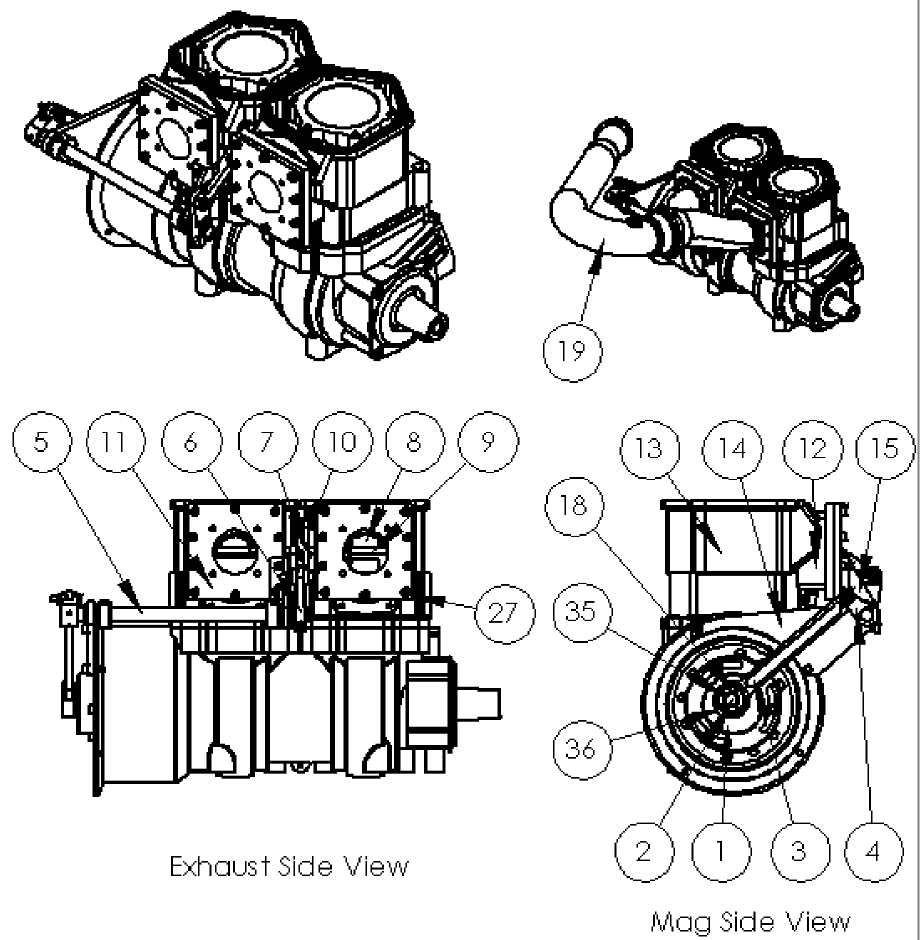

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Engine Assembly

Peter Britanyak 8/16/2010

600cc Rotax Synchronous Charge Trapping

UNIVERSITY OF IDAHO
ME DEPARTMENT

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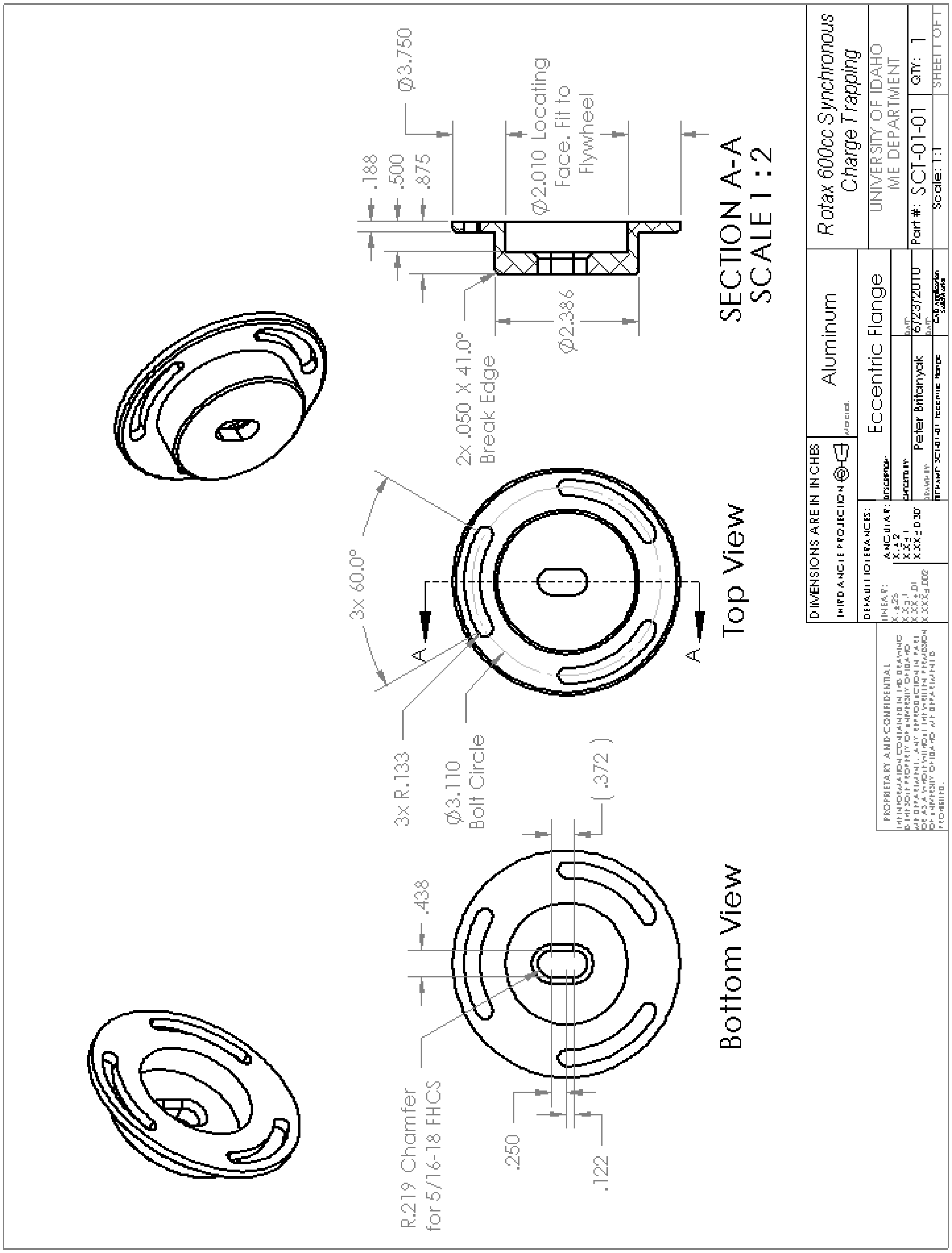
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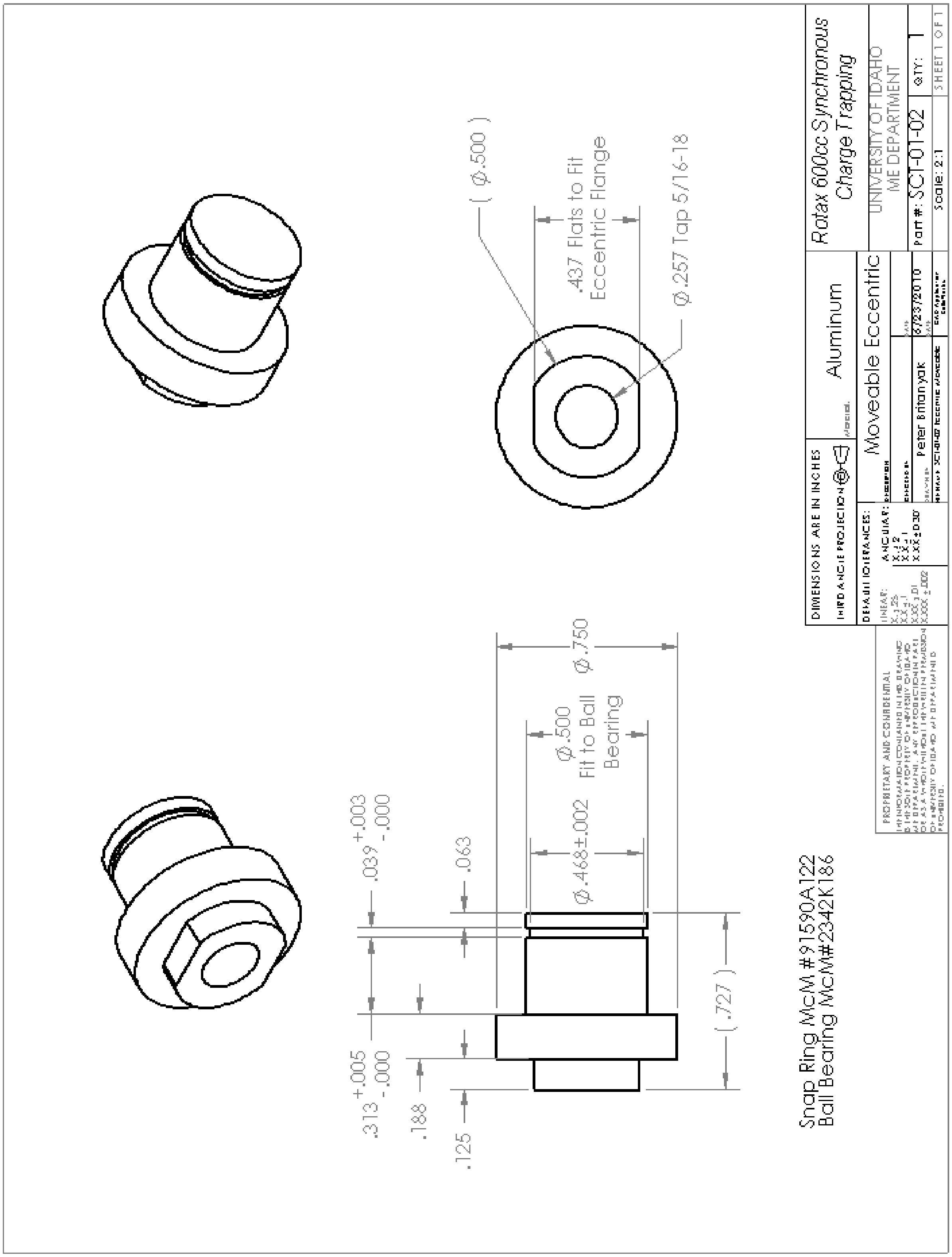
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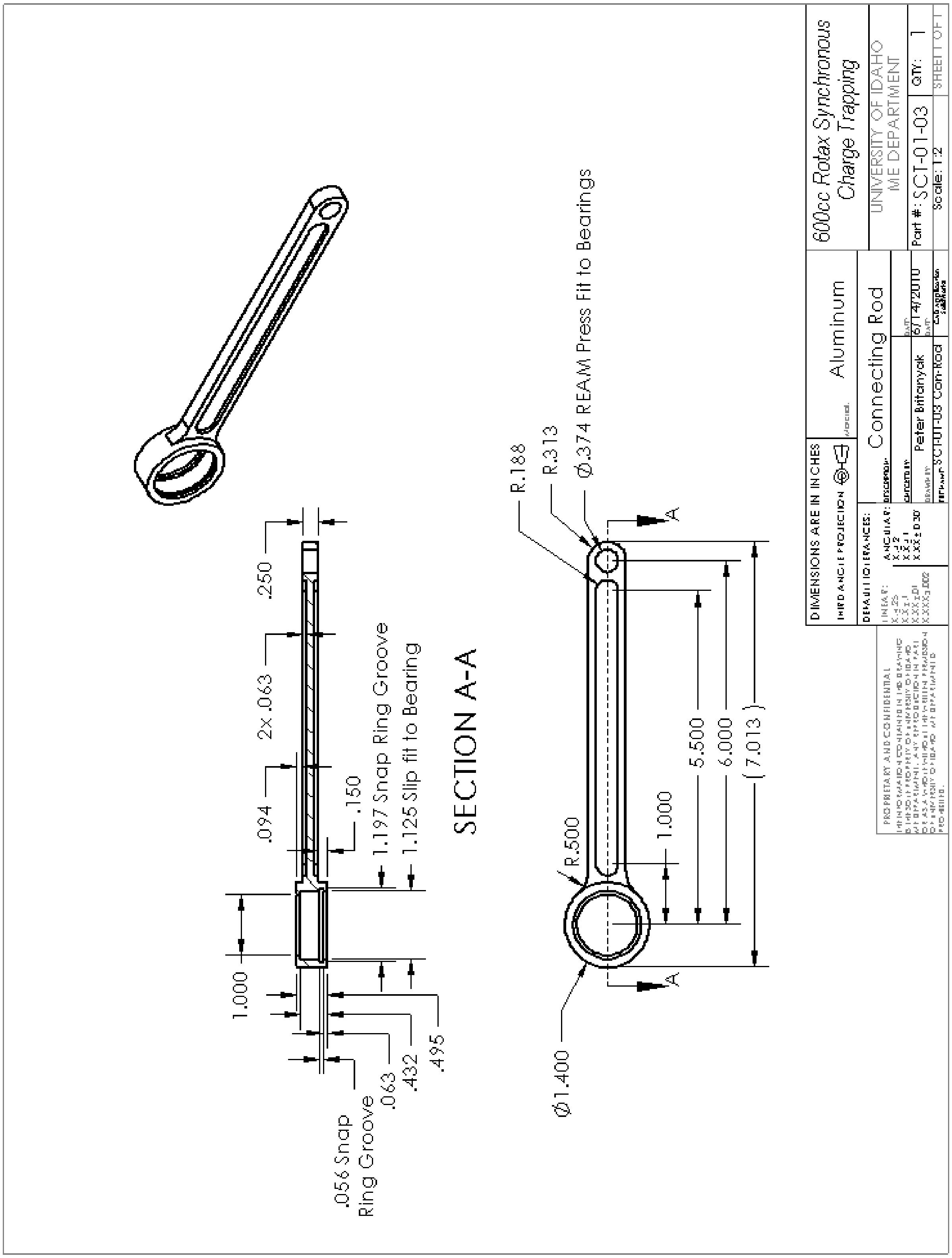
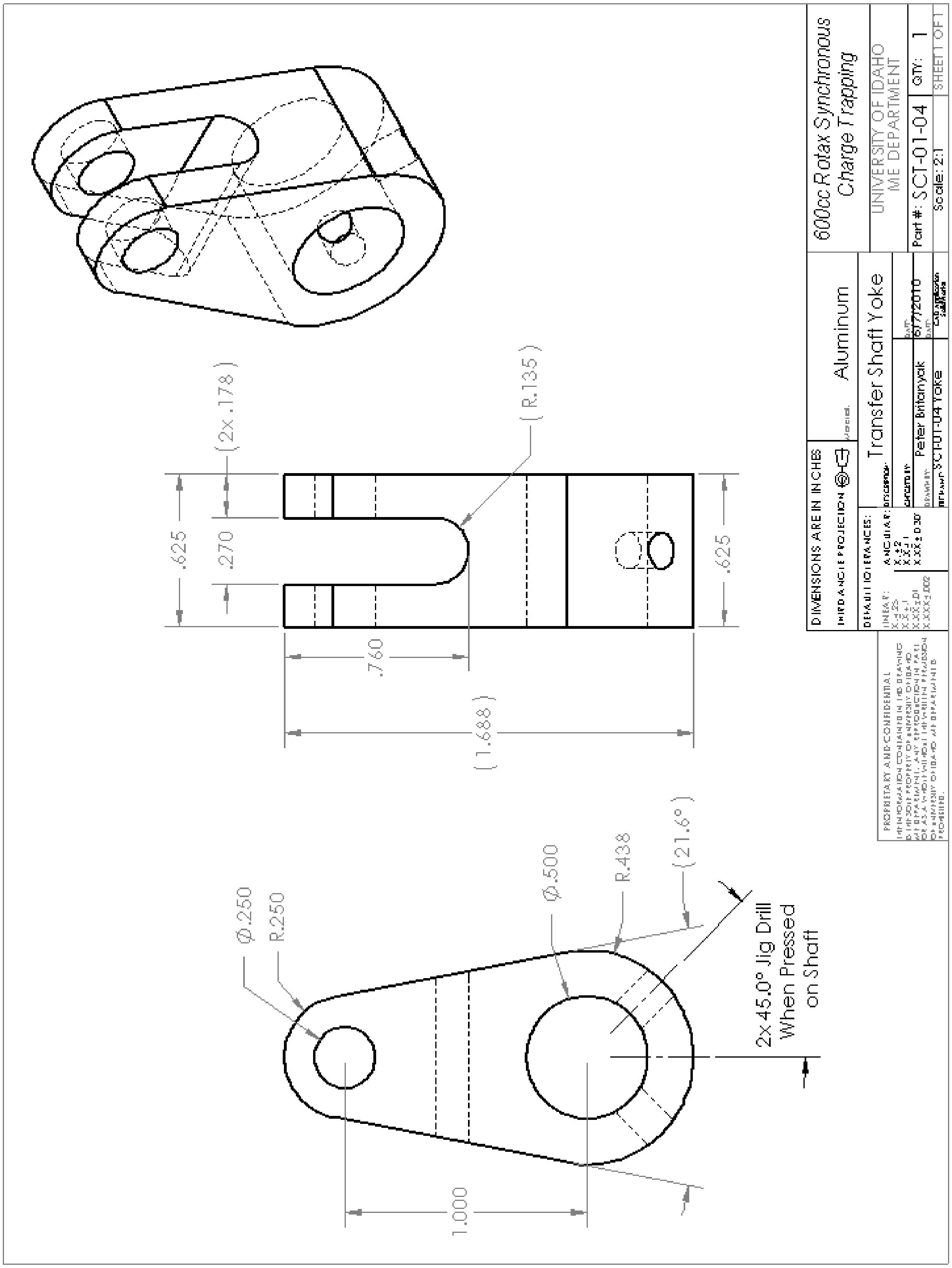
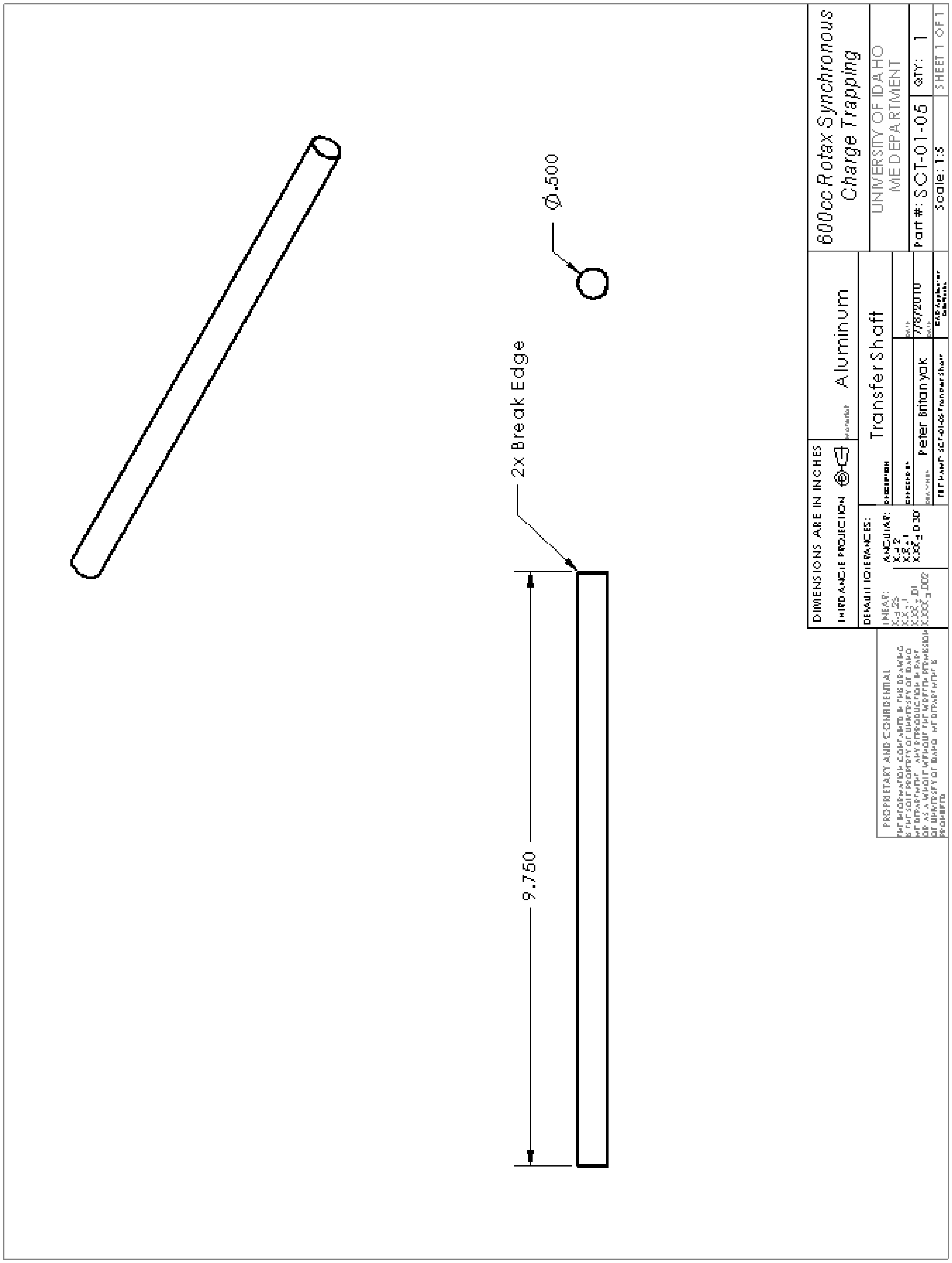
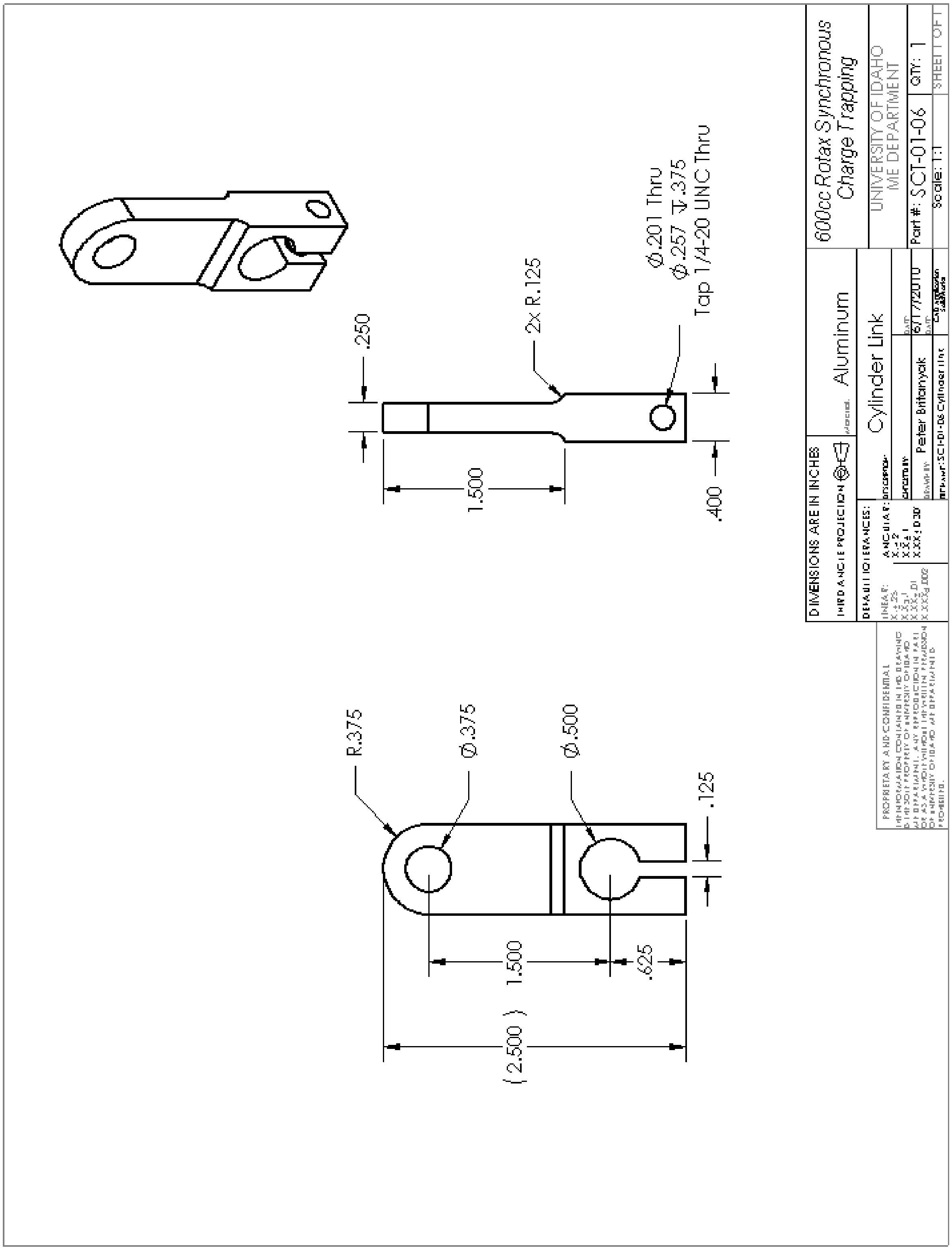
Ball Bearing McM#2342K186

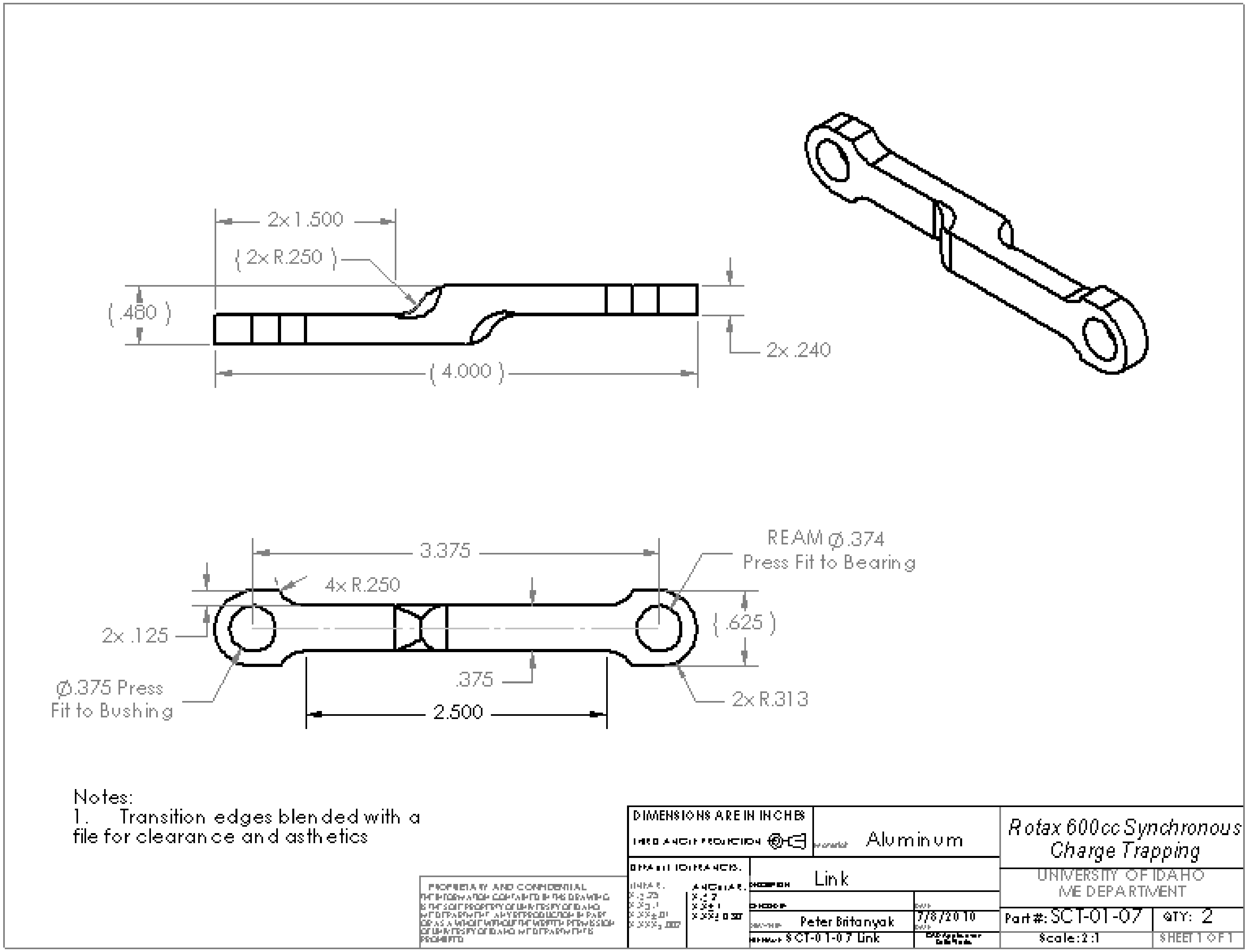

600cc Rotax Synchronous Charge Trapping
UNIVERSITY OF IDAHO
ME DEPARTMENT
Aluminum
Connecting Rod
DIMENSIONS ARE IN INCHES
Peter Britanyak
6/14/2010
Part #: SCT-01-03
QTY: 1
Scale: 1:2
SHEET 1 OF 1
SECTION A-A
.250
2x .063
.094
.150
1.197 Snap Ring Groove
1.125 Slip fit to Bearing
1.000
.056 Snap Ring Groove
.063
.432
.495
R.188
R.313
Ø.374 REAM Press Fit to Bearings
R.500
1.000
5.500
6.000
(7.013)
Ø1.400
A

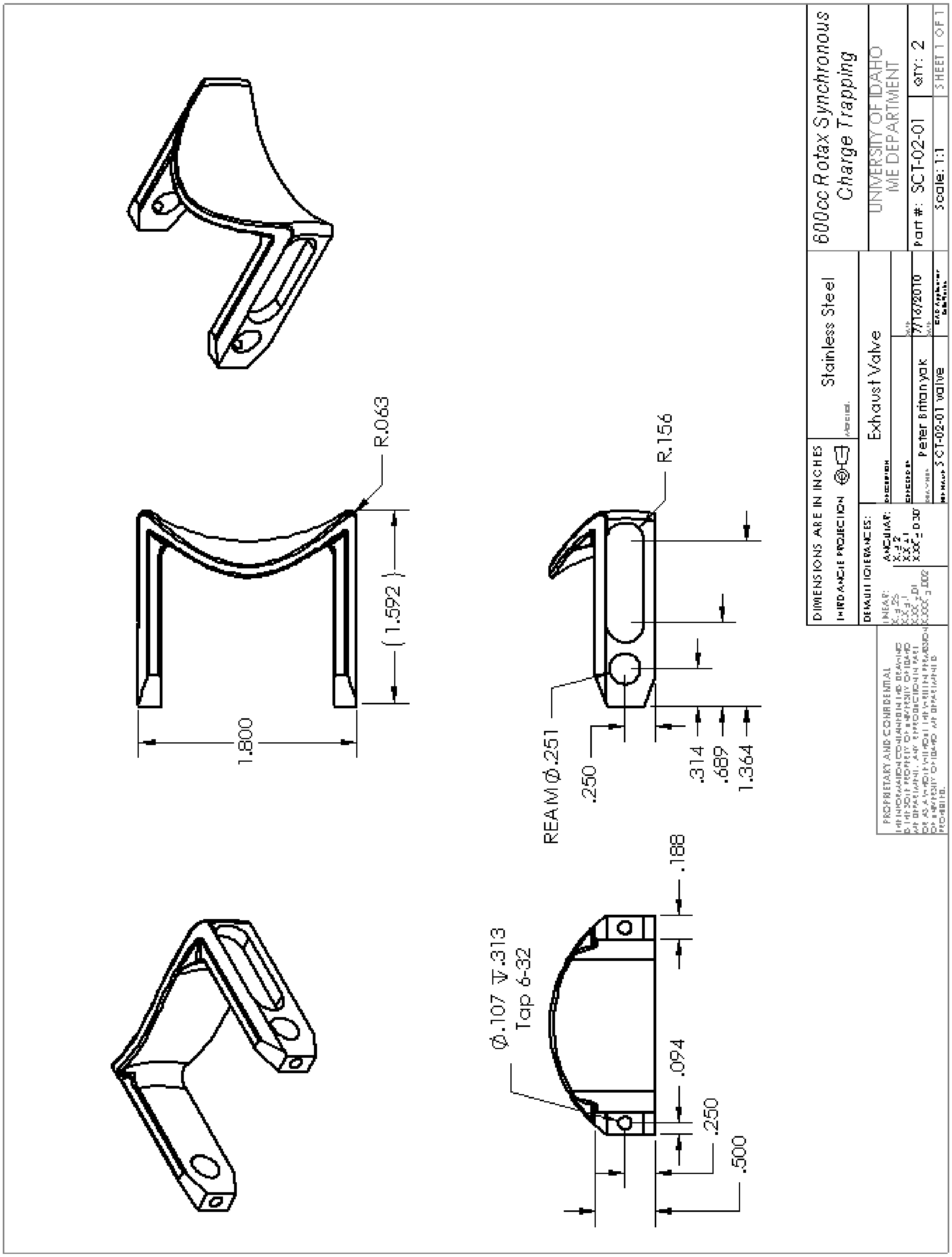

600cc Rotax Synchronous Charge Trapping
UNIVERSITY OF IDAHO ME DEPARTMENT
Part #: SCT-01-04
QTY: 1
Scale: 2:1
SHEET 1 OF 1
DIMENSIONS ARE IN INCHES
THIRD ANGLE PROJECTION
DEFAULT TOLERANCES:
Aluminum
Transfer Shaft Yoke
Peter Britanyak
6/7/2010
SCT-01-04 Yoke
PROPRIETARY AND CONFIDENTIAL
.625
.270
(2x .178)
(R.135)
.760
(1.688)
Ø.250
R.250
Ø.500
R.438
(21.6°)
2x 45.0° Jig Drill When Pressed on Shaft
1.000


2x Break Edge
9.750
Ø.500
DIMENSIONS ARE IN INCHES
Aluminum
Transfer Shaft
Peter Britanyak
7/7/2010
800cc Rotax Synchronous Charge Trapping
UNIVERSITY OF IDAHO
ME DEPARTMENT
Part #: SCT-01-05
QTY: 1
Scale: 1:5
SHEET 1 OF 1

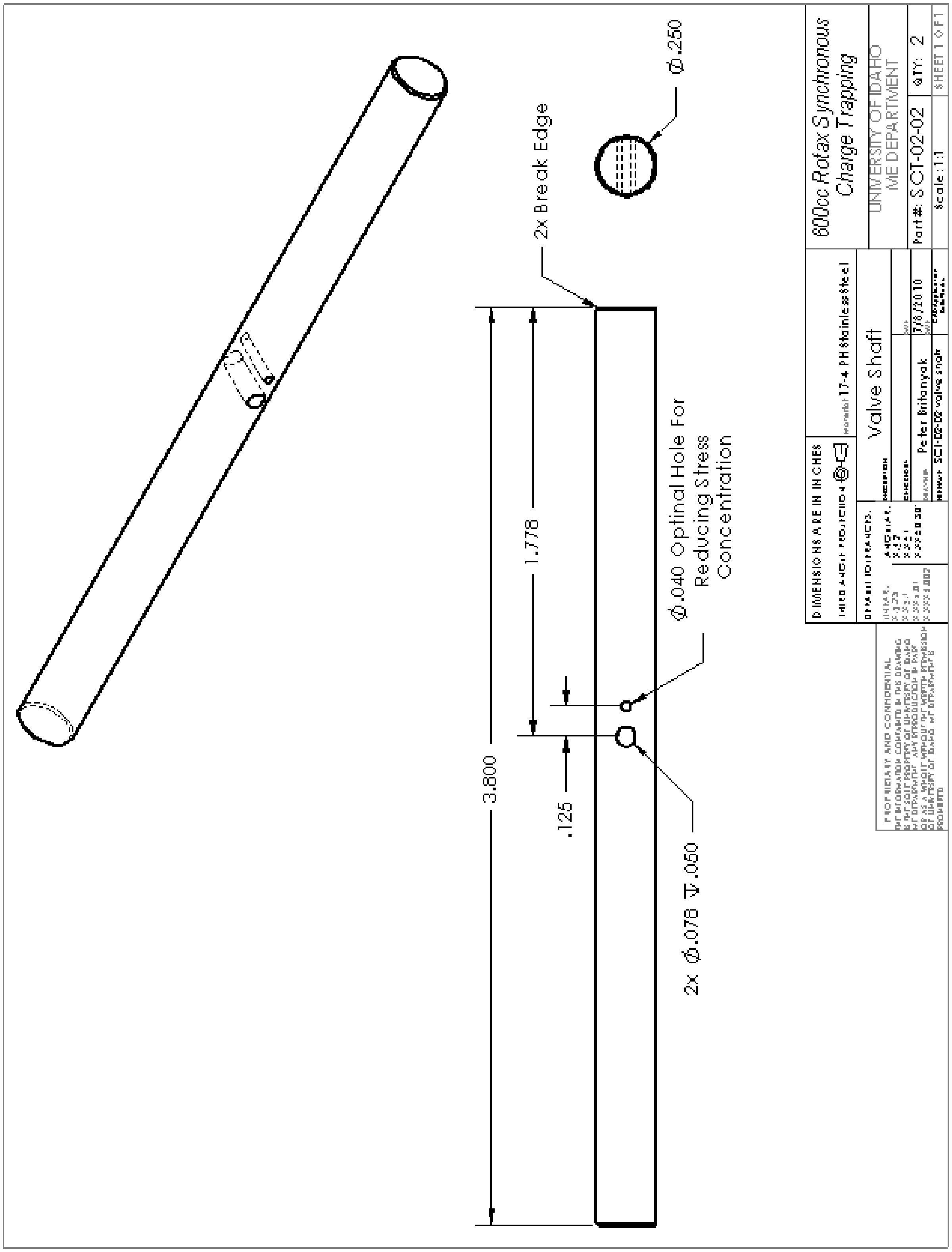


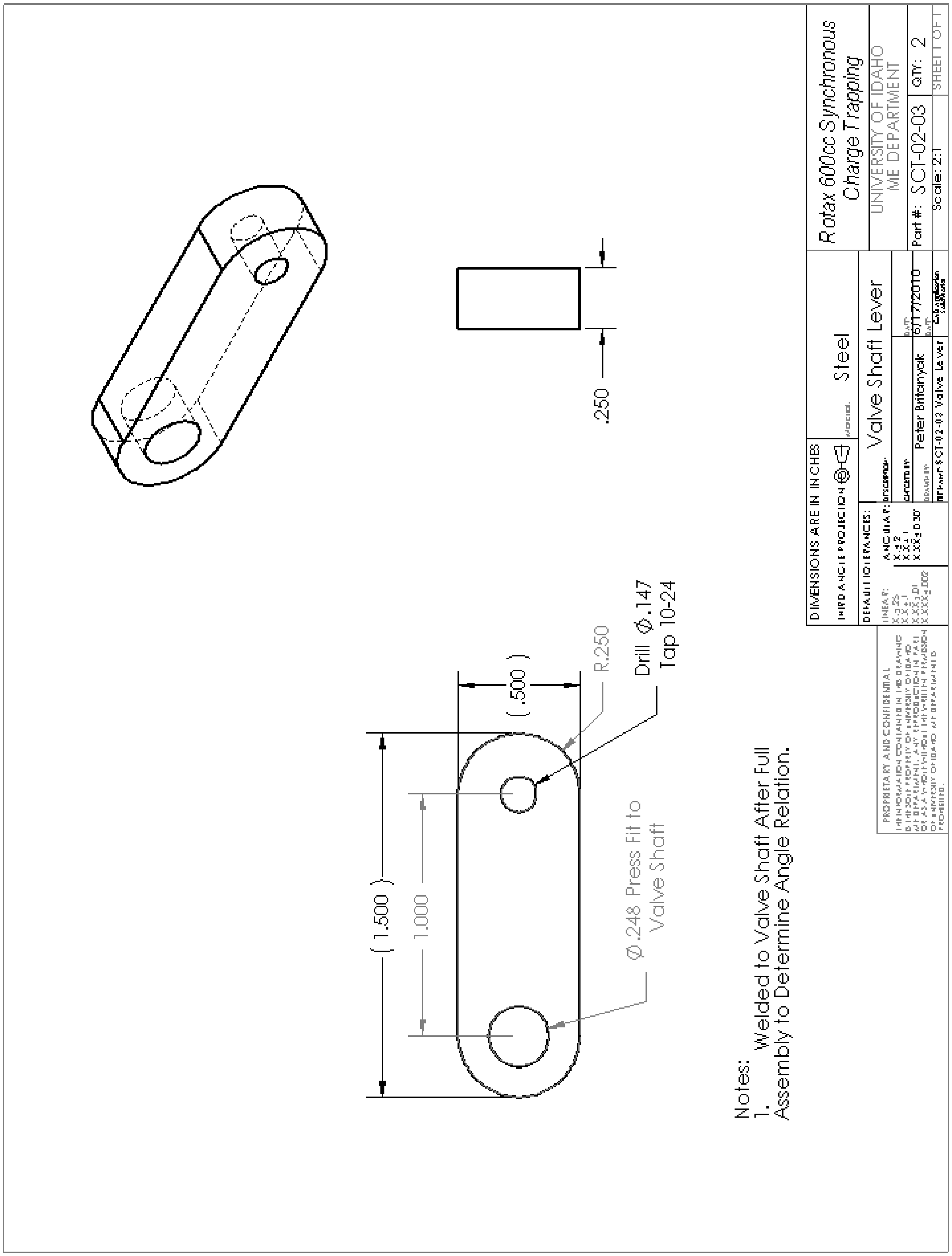
R.375
Ø.375
Ø.500
.125
1.500
.625
(2.500)
.250
2x R.125
1.500
.400
Ø.201 Thru
Ø.257 ⊽.375
Tap 1/4-20 UNC Thru
600cc Rotax Synchronous Charge Trapping
Aluminum
Cylinder Link
UNIVERSITY OF IDAHO
ME DEPARTMENT
Part #: SCT-01-06
QTY: 1
Scale: 1:1
SHEET 1 OF 1
Peter Britanyak
6/17/2010
DIMENSIONS ARE IN INCHES
DEFAULT TOLERANCES:
PROPRIETARY AND CONFIDENTIAL

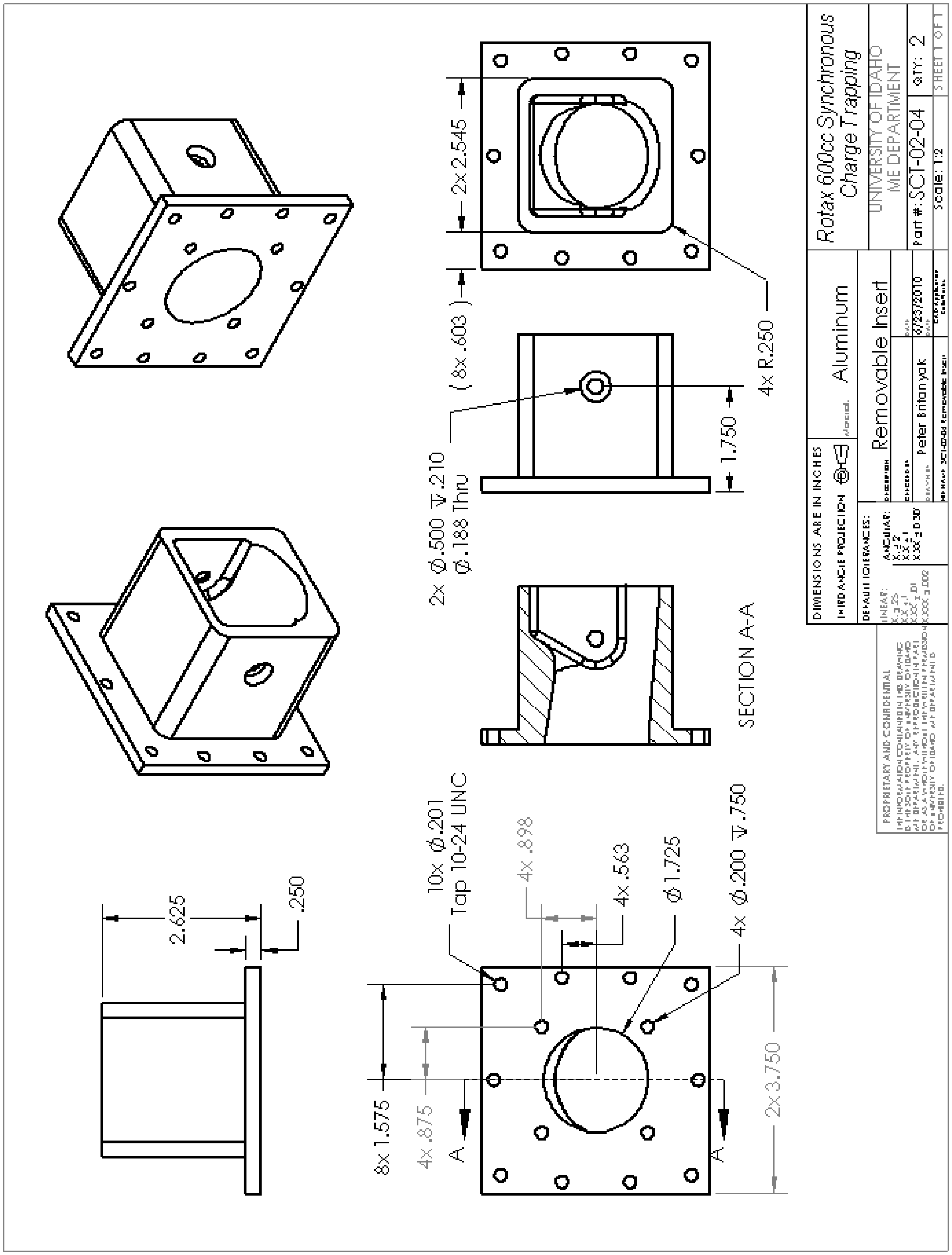

2x 1.500
(2x R.250)
(.480)
(4.000)
2x .240
3.375
4x R.250
REAM Ø.374
Press Fit to Bearing
2x .125
(.625)
Ø.375 Press
Fit to Bushing
.375
2.500
2x R.313
Notes:
1. Transition edges blended with a
file for clearance and asthetics
DIMENSIONS ARE IN INCHES
Aluminum
Link
Peter Britanyak
7/8/2010
SCT-01-07 Link
Rotax 600cc Synchronous
Charge Trapping
UNIVERSITY OF IDAHO
ME DEPARTMENT
Part #: SCT-01-07
QTY: 2
Scale: 2:1
SHEET 1 OF 1



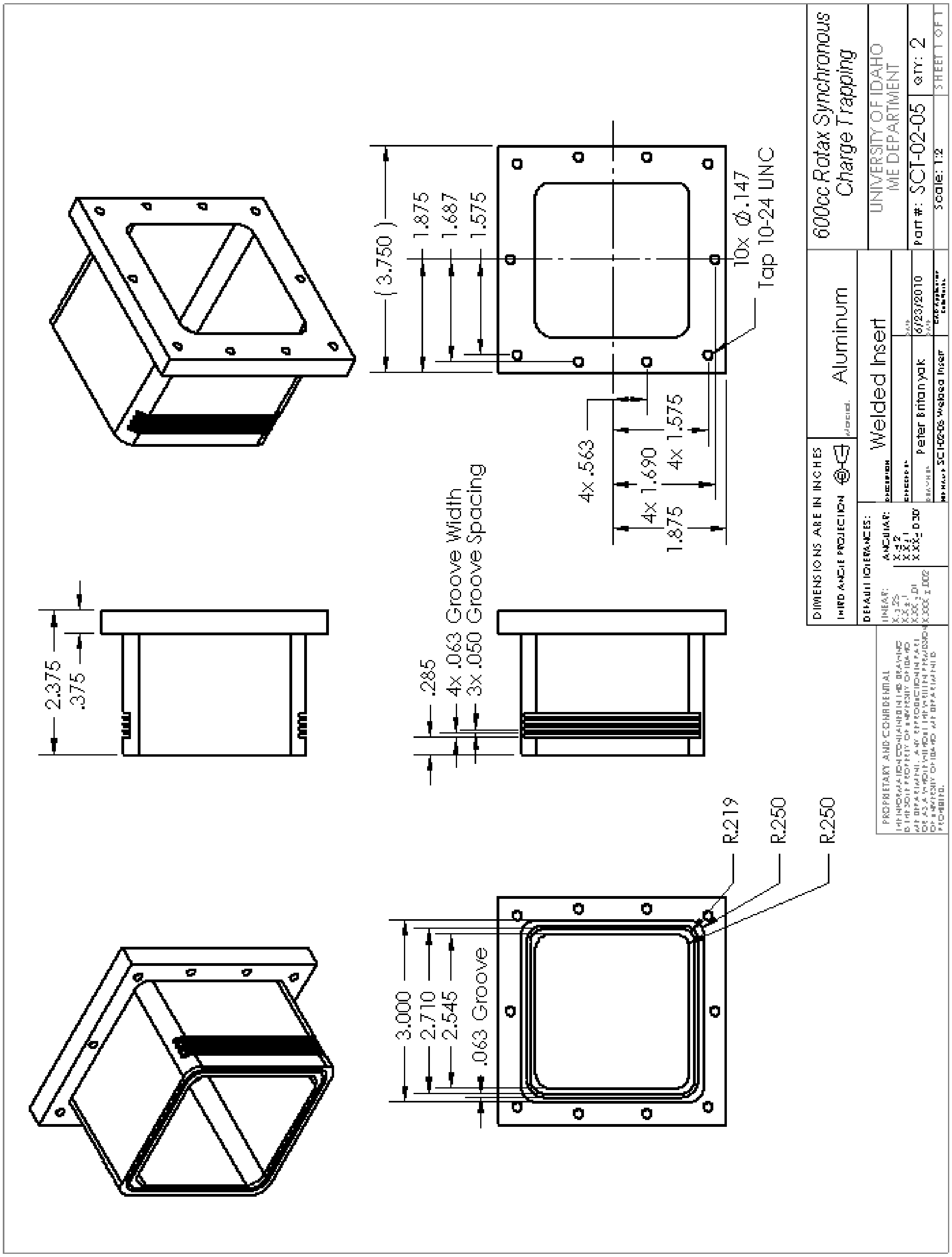
R.063
1.800
(1.592)
R.156
REAM Ø.251
.250
.314
.689
1.364
Ø.107 ▽.313
Tap 6-32
.188
.094
.250
.500
DIMENSIONS ARE IN INCHES
THIRD ANGLE PROJECTION
DEFAULT TOLERANCES:
Stainless Steel
Exhaust Valve
Peter Britanyak
7/16/2010
SCT-02-01 valve
PROPRIETARY AND CONFIDENTIAL
600cc Rotax Synchronous
Charge Trapping
UNIVERSITY OF IDAHO
ME DEPARTMENT
Part #: SCT-02-01
QTY: 2
Scale: 1:1
SHEET 1 OF 1


2x Break Edge
Ø.250
3.800
1.778
.125
Ø.040 Optinal Hole For
Reducing Stress
Concentration
2x Ø.078 ↧.050
DIMENSIONS ARE IN INCHES
Material: 17-4 PH Stainless Steel
Valve Shaft
Peter Britanyak
7/8/2010
600cc Rotax Synchronous
Charge Trapping
UNIVERSITY OF IDAHO
ME DEPARTMENT
Part #: SCT-02-02
QTY: 2
Scale: 1:1
SHEET 1 OF 1

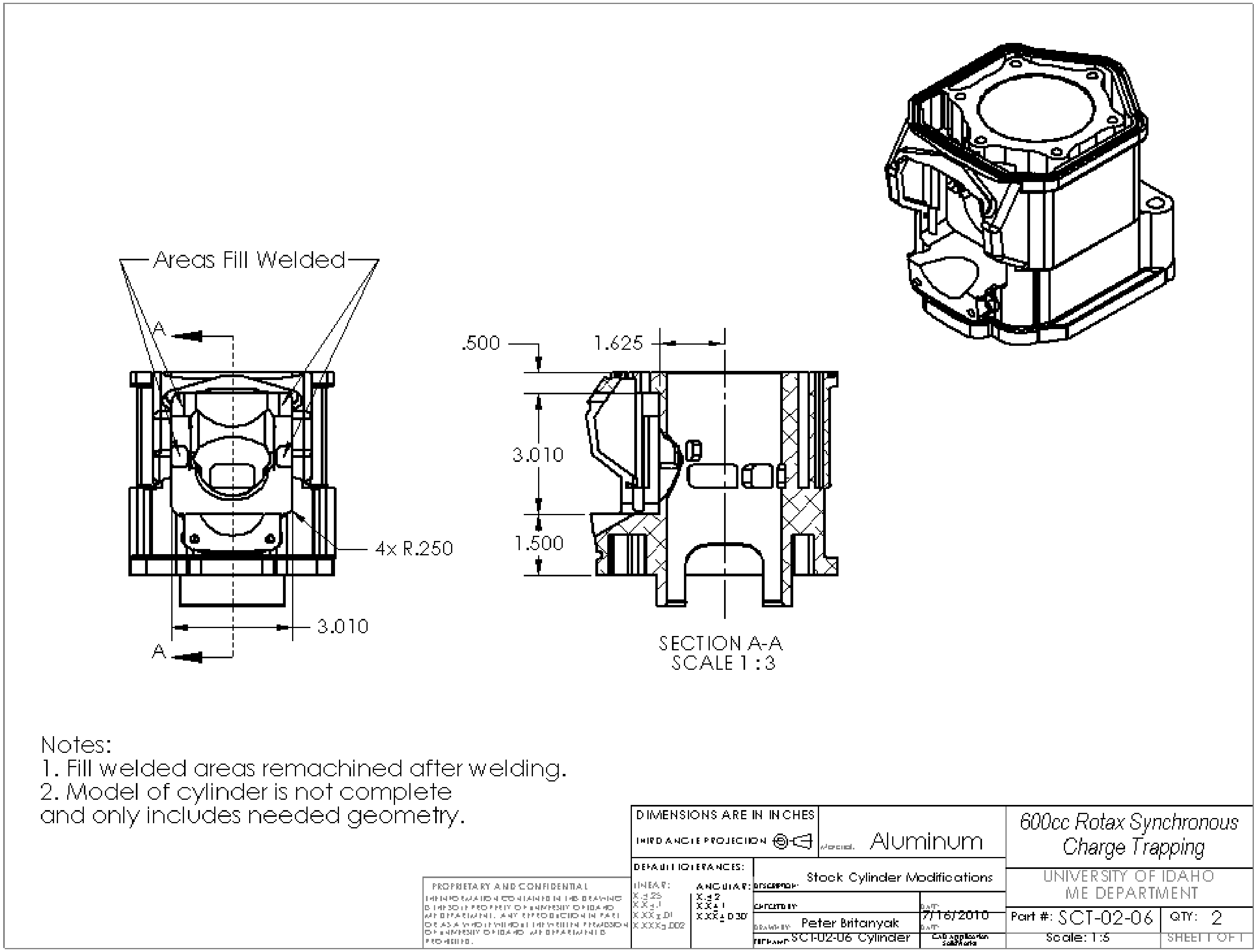

Rotax 600cc Synchronous
Charge Trapping
UNIVERSITY OF IDAHO
ME DEPARTMENT
Part #: SCT-02-03
QTY: 2
Scale: 2:1
SHEET 1 OF 1
DIMENSIONS ARE IN INCHES
Steel
Valve Shaft Lever
Peter Britanyak
6/17/2010
.250
(1.500)
1.000
(.500)
R.250
Drill Ø.147
Tap 10-24
Ø.248 Press Fit to
Valve Shaft
Notes:
1. Welded to Valve Shaft After Full
Assembly to Determine Angle Relation.



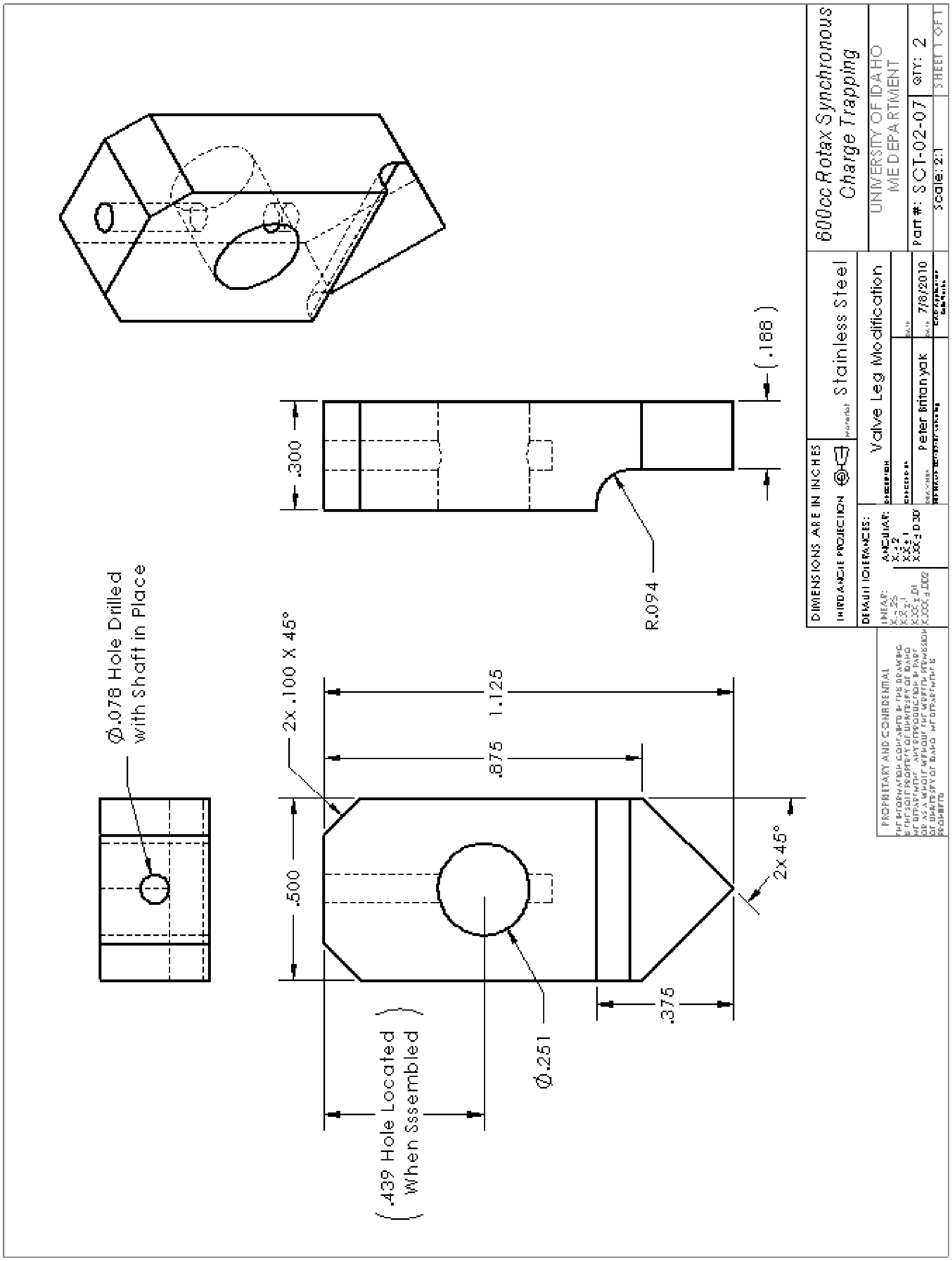
Rotax 600cc Synchronous Charge Trapping
UNIVERSITY OF IDAHO ME DEPARTMENT
Part #: SCT-02-04
QTY: 2
Scale: 1:2
SHEET 1 OF 1
DIMENSIONS ARE IN INCHES
Aluminum
Removable Insert
Peter Britanyak
6/23/2010
SECTION A-A
2x 2.545
(8x .603)
4x R.250
2x Ø.500 ⊽.210
Ø.188 Thru
1.750
2.625
.250
10x Ø.201
Tap 10-24 UNC
4x .898
4x .563
Ø1.725
4x Ø.200 ⊽.750
8x 1.575
4x .875
2x 3.750
A
A



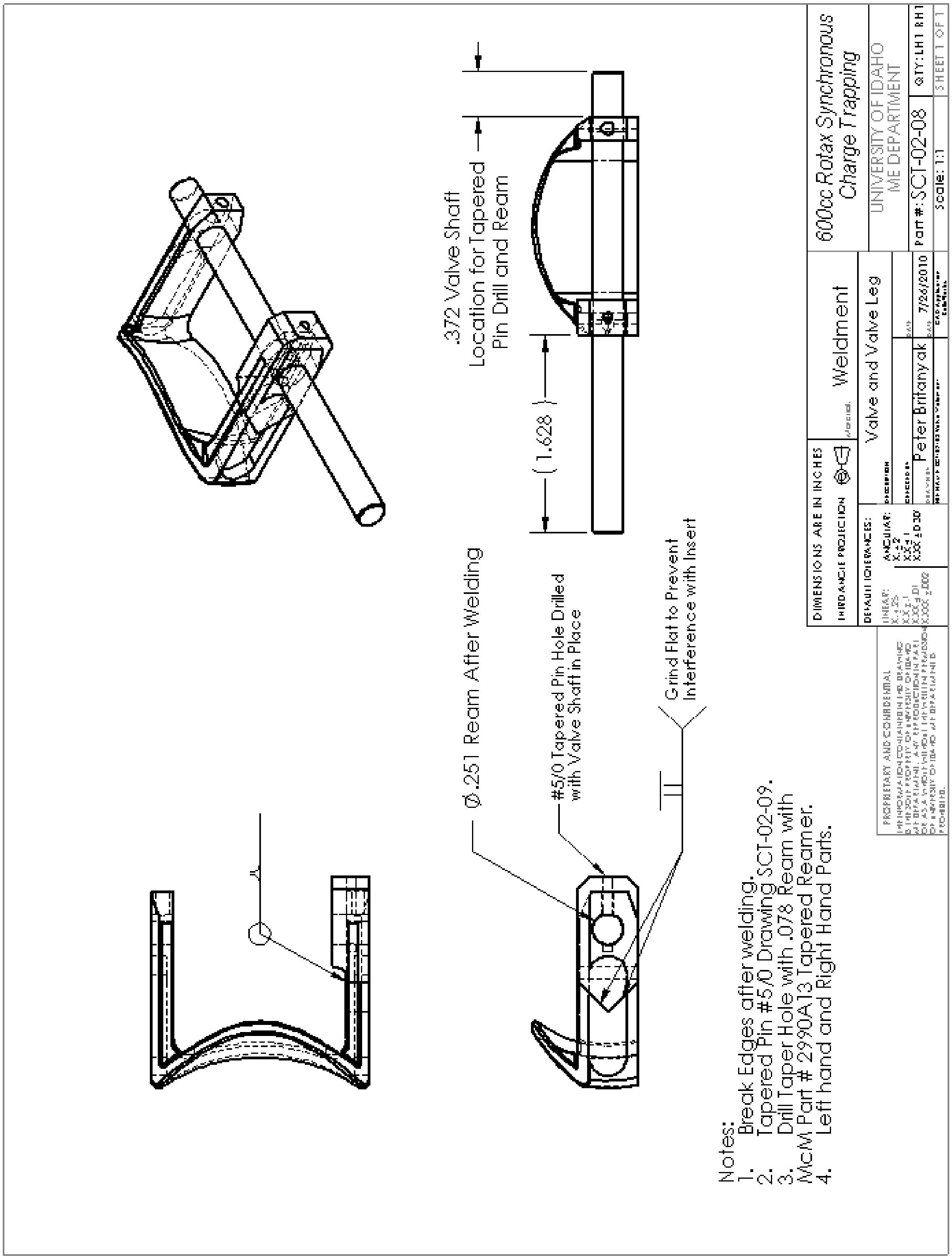
600cc Rotax Synchronous Charge Trapping
UNIVERSITY OF IDAHO ME DEPARTMENT
Part #: SCT-02-05
QTY: 2
Scale: 1:2
SHEET 1 OF 1
Aluminum
Welded Insert
Peter Britanyak
6/23/2010
DIMENSIONS ARE IN INCHES
THIRD ANGLE PROJECTION
DEFAULT TOLERANCES:
PROPRIETARY AND CONFIDENTIAL
(3.750)
1.875
1.687
1.575
10x Ø.147
Tap 10-24 UNC
4x .563
4x 1.690
4x 1.575
1.875
2.375
.375
.285
4x .063 Groove Width
3x .050 Groove Spacing
R.219
R.250
R.250
3.000
2.710
2.545
.063 Groove



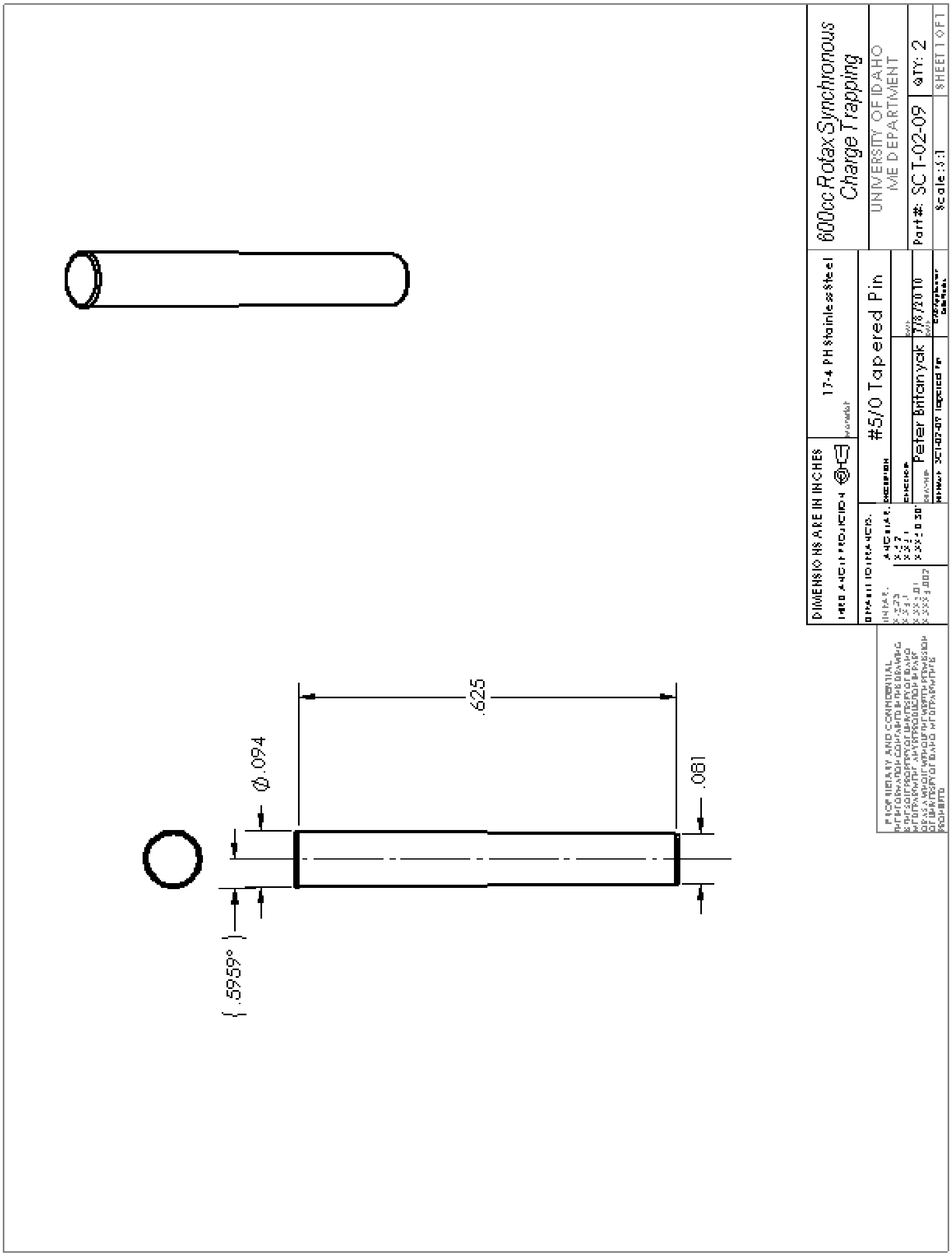
Areas Fill Welded
A
A
4x R.250
3.010
.500
1.625
3.010
1.500
SECTION A-A
SCALE 1 : 3
Notes:
1. Fill welded areas remachined after welding.
2. Model of cylinder is not complete
and only includes needed geometry.
PROPRIETARY AND CONFIDENTIAL
DIMENSIONS ARE IN INCHES
THIRD ANGLE PROJECTION
Aluminum
600cc Rotax Synchronous
Charge Trapping
DEFAULT TOLERANCES:
Stock Cylinder Modifications
UNIVERSITY OF IDAHO
ME DEPARTMENT
7/16/2010
Peter Britanyak
Part #: SCT-02-06
QTY: 2
SCT-02-06 Cylinder
Scale: 1:5
SHEET 1 OF 1

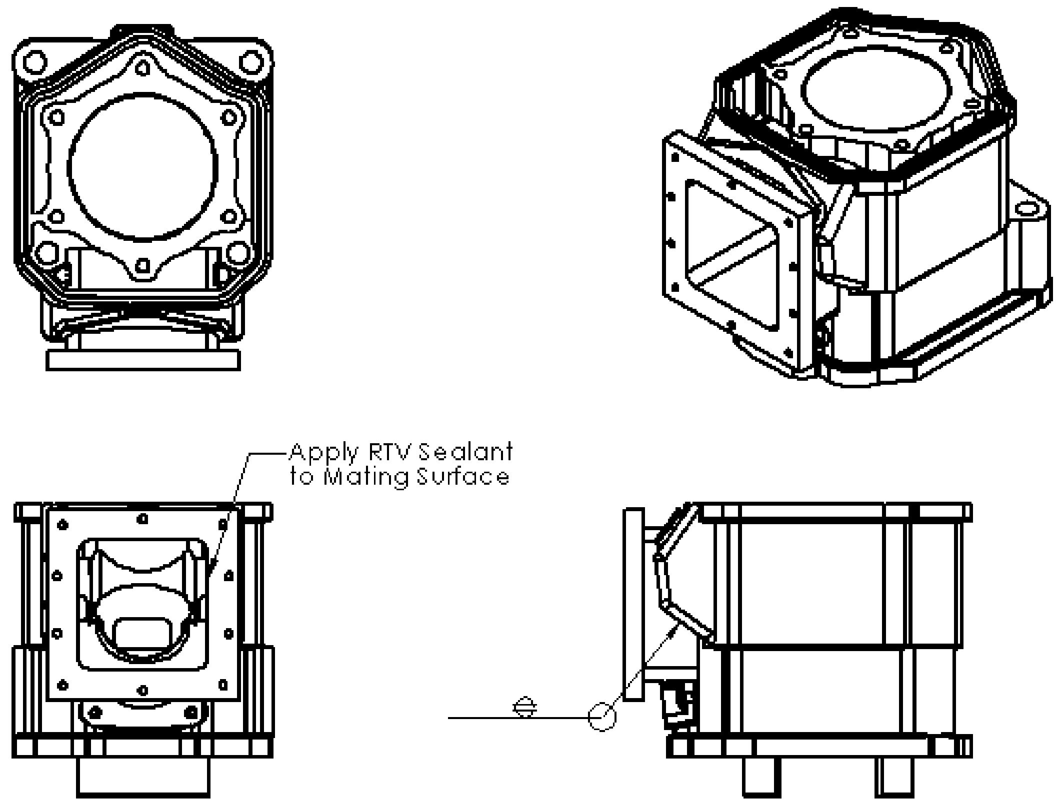


Ø.078 Hole Drilled with Shaft in Place
2x .100 X 45°
1.125
.875
.500
.300
(.188)
R.094
2x 45°
.375
Ø.251
(.439 Hole Located When Sssembled)
DIMENSIONS ARE IN INCHES
THIRD ANGLE PROJECTION
Material Stainless Steel
DEFAULT TOLERANCES:
Valve Leg Modification
Peter Britanyak
7/8/2010
PROPRIETARY AND CONFIDENTIAL
600cc Rotax Synchronous Charge Trapping
UNIVERSITY OF IDAHO ME DEPARTMENT
Part #: SCT-02-07
QTY: 2
Scale: 2:1
SHEET 1 OF 1



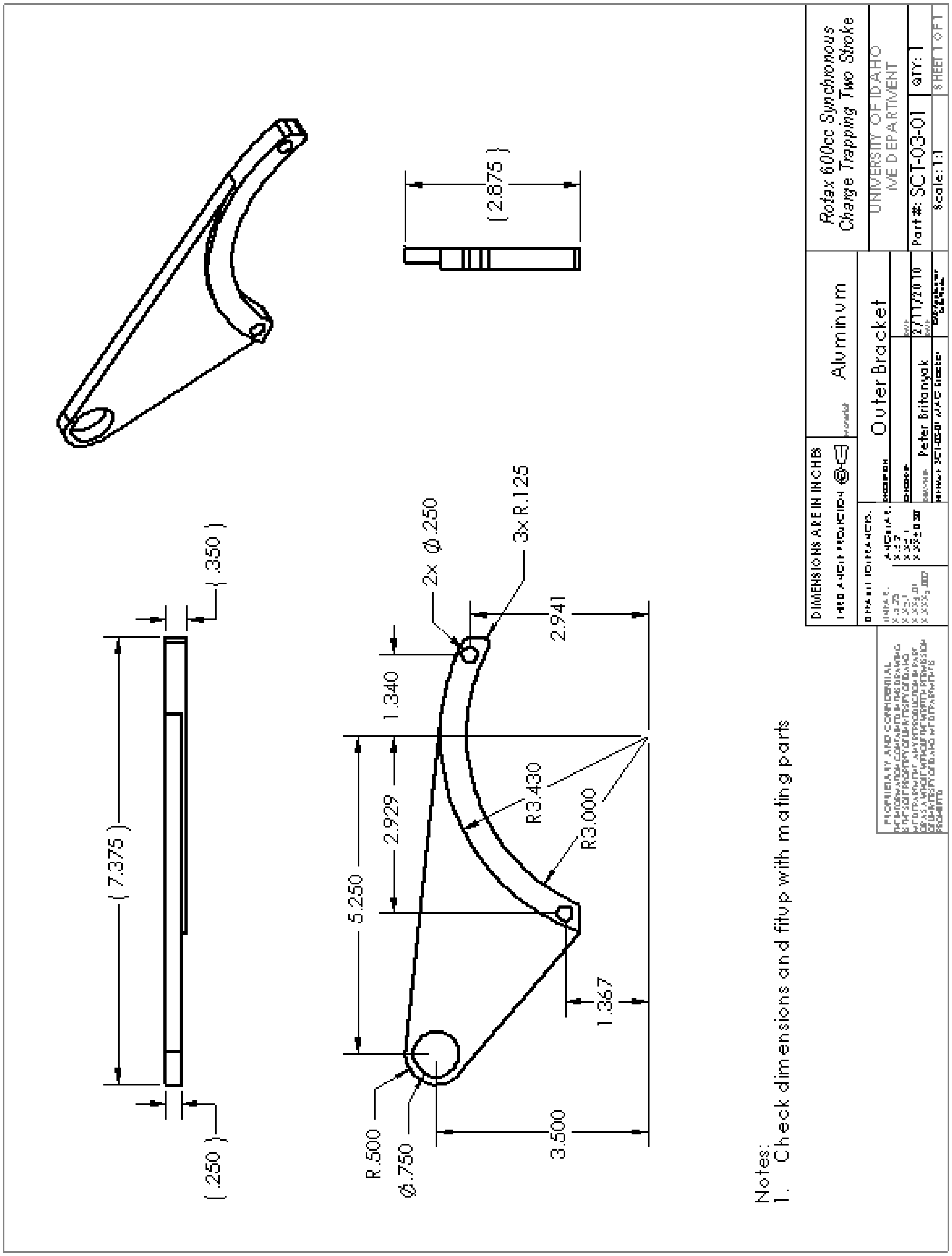
.372 Valve Shaft
Location for Tapered
Pin Drill and Ream
(1.628)
Ø.251 Ream After Welding
#5/0 Tapered Pin Hole Drilled
with Valve Shaft in Place
Grind Flat to Prevent
Interference with Insert
Notes:
1. Break Edges after welding.
2. Tapered Pin #5/0 Drawing SCT-02-09.
3. Drill Taper Hole with .078 Ream with
McM Part # 2990A13 Tapered Reamer.
4. Left hand and Right Hand Parts.
DIMENSIONS ARE IN INCHES
Weldment
Valve and Valve Leg
Peter Britanyak
7/26/2010
600cc Rotax Synchronous
Charge Trapping
UNIVERSITY OF IDAHO
ME DEPARTMENT
Part #: SCT-02-08
QTY: LH1 RH1
Scale: 1:1
SHEET 1 OF 1


.625
Ø.094
.081
(.5959°)
DIMENSIONS ARE IN INCHES
17-4 PH Stainless Steel
#5/0 Tapered Pin
Peter Britanyak
7/8/2010
600cc Rotax Synchronous Charge Trapping
UNIVERSITY OF IDAHO
ME DEPARTMENT
Part #: SCT-02-09
QTY: 2
SHEET 1 OF 1

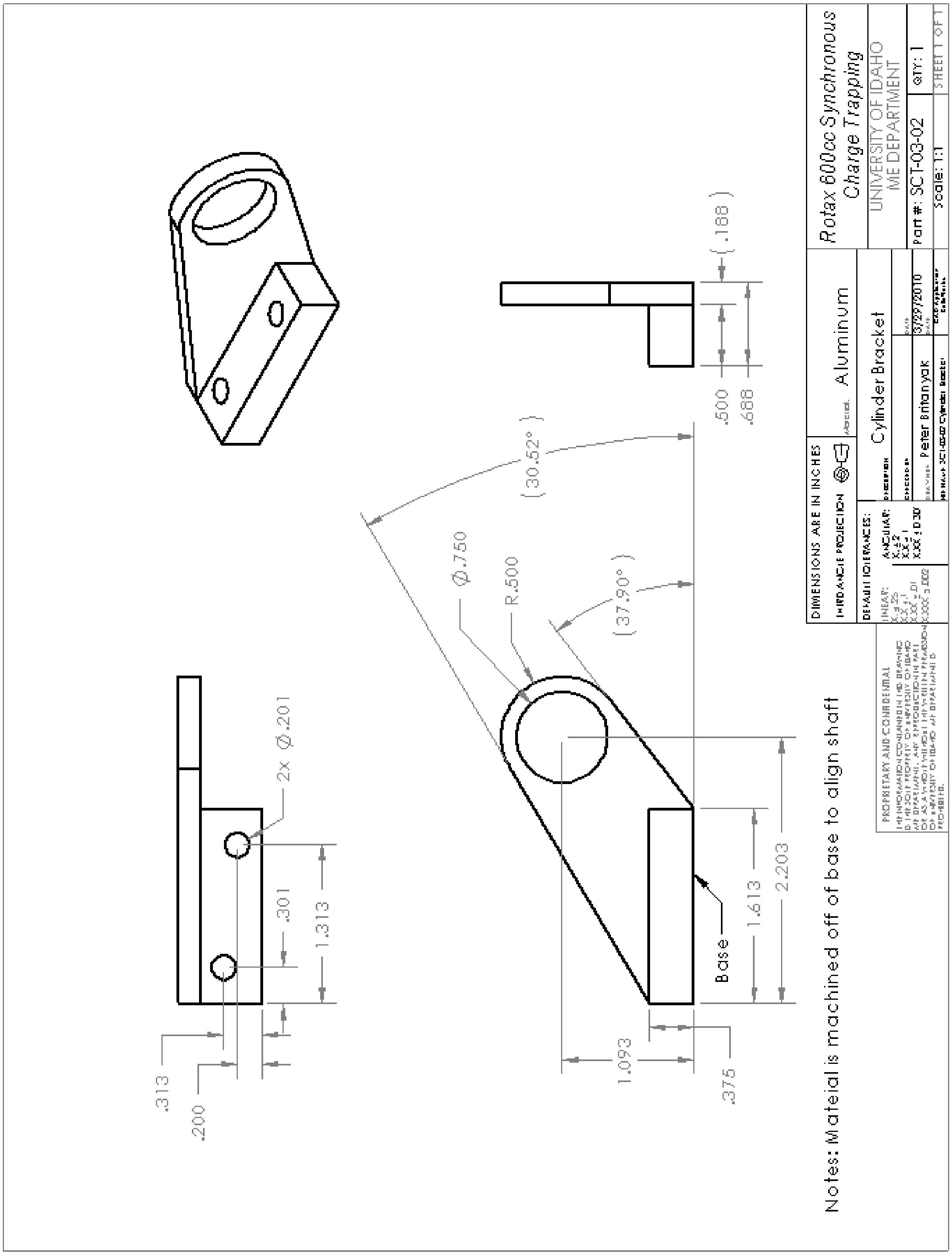


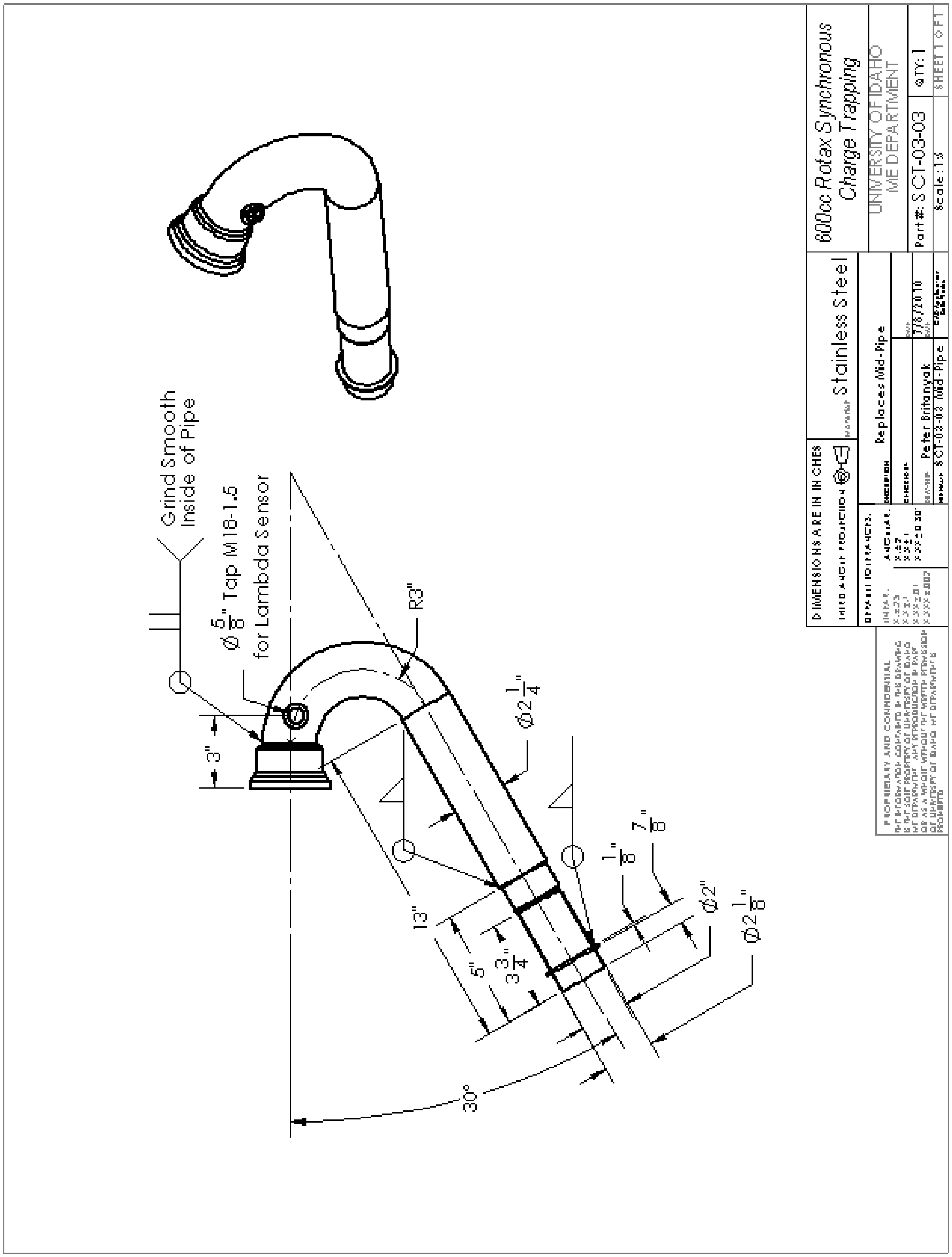
Notes:
1. Clamp Welded Insert in place for welding.
2. Remachinging will need to be done on flange face, inside surface of insert and top and bottom of jug because of warping caused by welding.
3. Base gasket thickness will need to be adjusted to compensate for material removed from top and bottom of jug.

DIMENSIONS ARE IN INCHES
THIRD ANGLE PROJECTION
MATERIAL: Weldment
600cc Rotax Synchronous Charge Trapping
DEFAULT TOLERANCES:
LINEAR: X ±.25, X.X ±.1, X.XX ±.01, X.XXX ±.002
ANGULAR: X ±2, X.X ±1, X.XX ±0 30'
DESCRIPTION: Cylinder and Insert Weldment
UNIVERSITY OF IDAHO
ME DEPARTMENT
CHECKED:
DRAWN: Peter Britanyak
DATE: 7/26/2010
Part #: SCT-02-09
QTY: 2
Scale: 1:4
SHEET 1 OF 1

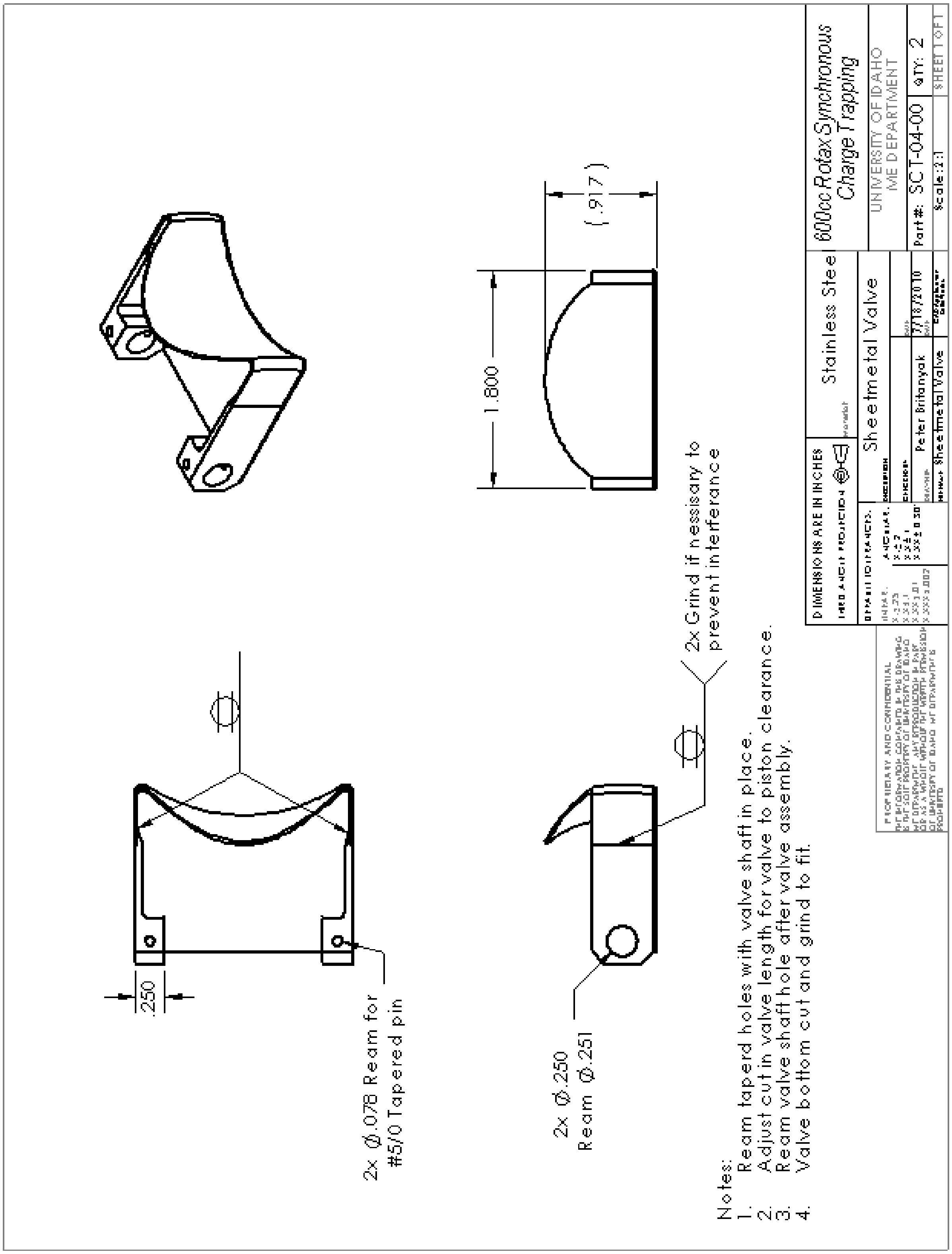


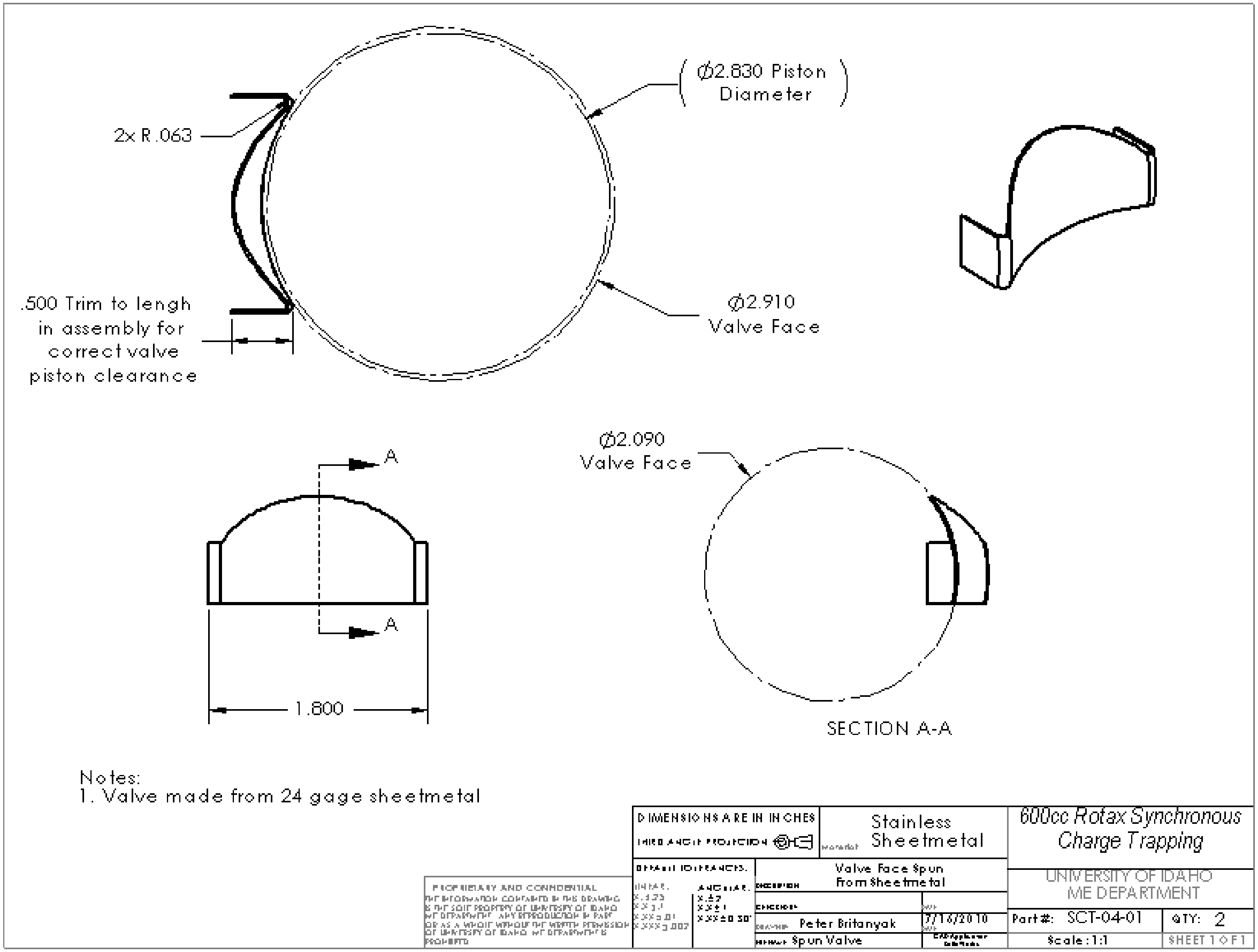
Rotax 600cc Synchronous Charge Trapping Two Stroke
UNIVERSITY OF IDAHO ME DEPARTMENT
Part #: SCT-03-01
QTY: 1
Scale: 1:1
SHEET 1 OF 1
Aluminum
Outer Bracket
Peter Britanyak
2/11/2010
DIMENSIONS ARE IN INCHES
PROPRIETARY AND CONFIDENTIAL
(2.875)
(.350)
(7.375)
(.250)
2x ∅.250
3x R.125
2.941
1.340
2.929
5.250
R3.430
R3.000
1.367
R.500
∅.750
3.500
Notes:
1. Check dimensions and fitup with mating parts


.313
.200
.301
1.313
2x Ø.201
1.093
.375
Base
1.613
2.203
Ø.750
R.500
(37.90°)
(30.52°)
.500
.688
(.188)
Notes: Mateial is machined off of base to align shaft
DIMENSIONS ARE IN INCHES
Aluminum
Cylinder Bracket
Peter Britanyak
3/29/2010
Rotax 600cc Synchronous
Charge Trapping
UNIVERSITY OF IDAHO
ME DEPARTMENT
Part #: SCT-03-02
QTY: 1
Scale: 1:1
SHEET 1 OF 1

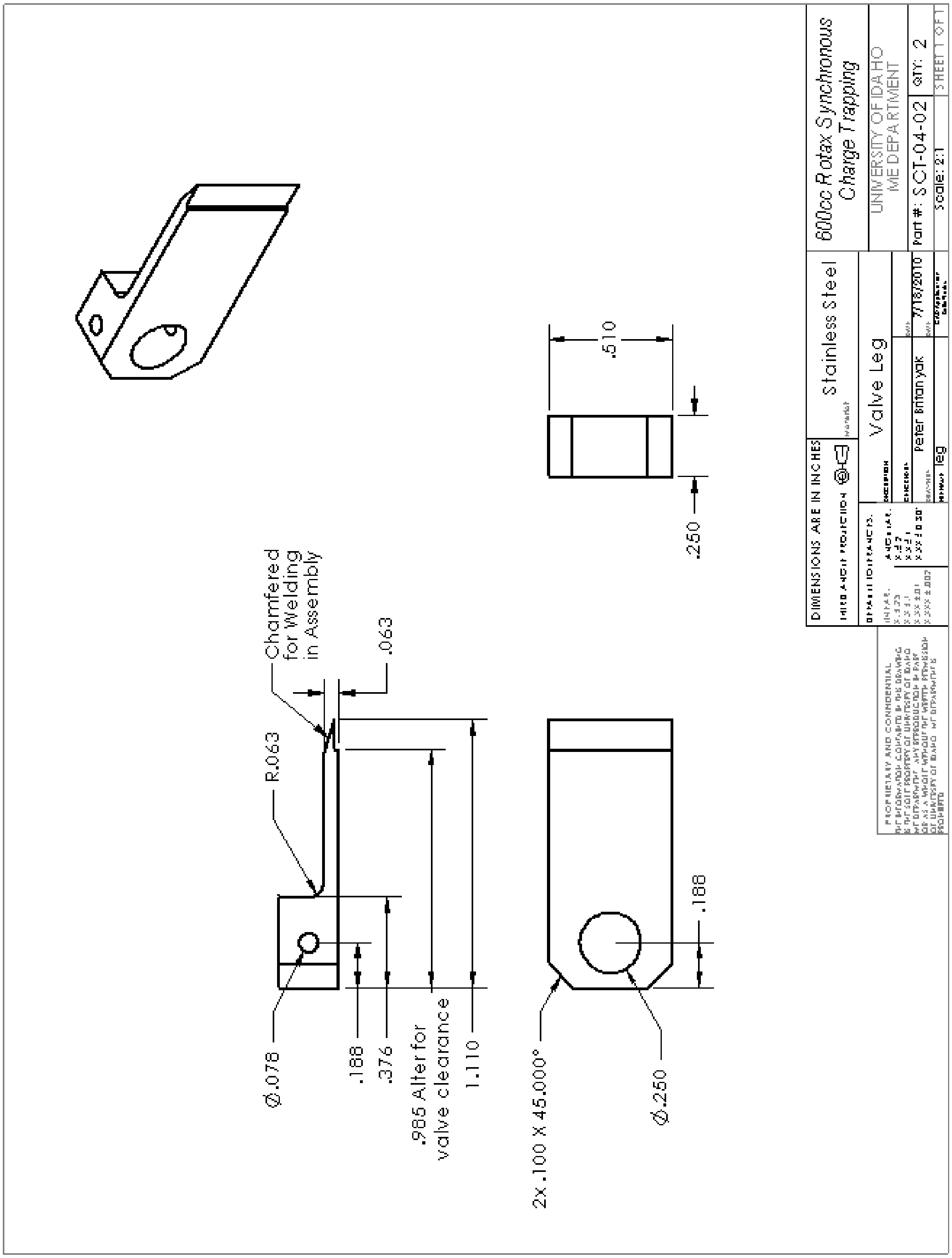

Grind Smooth
Inside of Pipe
Ø 5/8" Tap M18-1.5
for Lambda Sensor
R3"
Ø2 1/4"
3"
13"
5"
3 3/4"
1/8"
7/8"
Ø2"
Ø2 1/8"
30°
DIMENSIONS ARE IN INCHES
Stainless Steel
Replaces Mid-Pipe
Peter Britanyak
7/8/2010
600cc Rotax Synchronous
Charge Trapping
UNIVERSITY OF IDAHO
ME DEPARTMENT
Part #: SCT-03-03
QTY: 1
SHEET 1 OF 1

Appendix B – Valve Redesign Drawings





Ø2.830 Piston Diameter
2x R.063
.500 Trim to lengh in assembly for correct valve piston clearance
Ø2.910 Valve Face
Ø2.090 Valve Face
A
A
1.800
SECTION A-A
Notes:
1. Valve made from 24 gage sheetmetal
DIMENSIONS ARE IN INCHES
THIRD ANGLE PROJECTION
Stainless Sheetmetal
600cc Rotax Synchronous Charge Trapping
Valve Face Spun From Sheetmetal
UNIVERSITY OF IDAHO ME DEPARTMENT
Peter Britanyak
7/16/2010
Part #: SCT-04-01
QTY: 2
Spun Valve
Scale: 1:1
SHEET 1 OF 1



Ø.078
.188
.376
.985 Alter for valve clearance
1.110
R.063
Chamfered for Welding in Assembly
.063
2x .100 X 45.000°
Ø.250
.188
.250
.510
DIMENSIONS ARE IN INCHES
Stainless Steel
Valve Leg
Peter Britanyak
7/18/2010
leg
800cc Rotax Synchronous Charge Trapping
UNIVERSITY OF IDAHO ME DEPARTMENT
Part #: SCT-04-02
QTY: 2
Scale: 2:1
SHEET 1 OF 1
PROPRIETARY AND CONFIDENTIAL

Appendix C – Valve Shaft Endurance Calculations

TK Solver REPORT

Valve Shaft Calculations

Rules Sheet

Rules

;Stress Concentration Shigley [19] pg.217

$$ke = \frac{1}{q \cdot (kt - 1)}$$

;Size Factor

$$kb = \left[\frac{r}{.3}\right]^{-.1133}$$

;Marin Equation

$$Se = Su \cdot .5 \cdot ka \cdot kb \cdot kc \cdot kd \cdot ke$$

;Shear Stress

$$J = \frac{\pi \cdot (2 \cdot r)^4}{32}$$

$$\tau = \frac{T \cdot r}{J}$$

Variables Sheet

Input	Name	Output	Unit	Comment
.9	q			
2.6	kt			
	kb	1.104277		Size Factor
.125	r			[in] Shaft Radius
	Se	24021.222007		Endurace Strength
160000	Su			[psi] Ultimate Strength for 17-4 PH Stainless
.78	ka			Surface Factor from Stevens pg.81
.577	kc			Load Factor
.87	kd			Temp Factor
	ke	0.694444		Misc. - Stress Concentration Factor
	τ	7170.885116		Shear Stress
	J	0.000383		
22	T			[in*lb] Torque

Appendix D – Tapered Pin Endurance Calculations

TK Solver REPORT

Valve Pin Calculations

Rules Sheet

Rules

;Stress Concentration Shigley [19]

$$ke = \frac{1}{1 + q \cdot (kt - 1)}$$

;Size Factor

$$kb = \left[\frac{dia\_pin}{.3}\right]^{-.1133}$$

;Marin Equation

$$Se = .5 \cdot ka \cdot kb \cdot kc \cdot kd \cdot Su$$

;Sum the Moments

$$M\_valve \cdot g\_force \cdot r\_CG = F \cdot r\_shaft$$

;Shear Stress

$$\tau = \frac{F}{2 \cdot \left[\frac{\pi \cdot dia\_pin^2}{4}\right]}$$

Variables Sheet

Input	Name	Output	Unit	Comment
.1122	M_valve			[lbm] Mass of the Valve
68	g_force			Calculated G Force
.5	r_CG			[in] Distance from Valve CG to Pivot
	F	30.5184		[lb] Shearing Force
.125	r_shaft			[in] Valve Shaft Radius
	τ	2508.860648		[psi] Shear Stress on Pin
.088	dia_pin			[in] Average Diameter Of Tapered Pin
1	ke			Misc. Factor
	q			
	kt			
	kb	1.149074		Size Factor
	Se	10343.043779		[psi] Endurance Strength
.78	ka			Surface Finish
.577	kc			Load Factor
.5	kd			Temp Factor
160000	Su			[psi] Ultimate Strength of 17-4 PH Stainless Steel

Appendix E – Valve Force Calculations

TK Solver REPORT

Valve Force Calculations

Rules Sheet

Rules

$$rev\_min \cdot \left[\frac{1}{60}\right] = rev\_sec$$

$$travel \cdot .5 \cdot rev\_sec = ave\_vel$$

$$ave\_vel \cdot 2 \cdot 2 = max\_vel$$

$$max\_accel = \frac{max\_vel - 0}{\frac{12 \cdot .5 \cdot travel}{max\_vel}}$$

$$\frac{max\_accel}{32.174} = g\_force$$

$$g\_force \cdot (mass) = load$$

Variables Sheet

Input	Name	Output	Unit	Comment
.1122	mass			[lbm] measured in solidworks
8000	rev_min			[rev/min] RPM of crank
	rev_sec	133.333333		[rev/sec]
.56	travel			[in] distance valve CG moves per half revolution of the crank
	ave_vel	37.333333		[in/s] average velocity to travel valve distance
	max_vel	149.333333		[in/s]
	max_accel	6637.037037		[in/s^2]
	g_force	206.285729		
	load	23.145259		[lbf] weight of valve due to acceleration

Appendix F – Laminar Flow Muffler Calculations - Blair Equations

TK Solver REPORT

Laminar Flow Muffler Math Model Using Blair Equations [10]

Rules Sheet

Rules

$$\rho = \frac{P}{R \cdot T}$$

$$Reout = \frac{\rho \cdot cout \cdot Dout}{\mu}$$

$$Rein = \frac{\rho \cdot cin \cdot Din}{\mu}$$

$$xg = \frac{Db - \left[Db^2 - Din^2\right]^{\left[\frac{1}{2}\right]}}{2}$$

$$Dout = 2 \cdot xg$$

$$Ain = \frac{pi \cdot Din^2}{4}$$

$$Aout = \frac{pi}{4} \cdot \left[Db^2 - (Db - 2 \cdot xg)^2\right]$$

$$Aout = Ain \cdot restriction$$

$$Length = L1 + L2 + L3$$

;Volume 1

$$L1 = \frac{a0}{12 \cdot f1}$$

$$V1 = \left[\frac{\pi \cdot Di^2}{4}\right] \cdot L1 - \left[\frac{\pi \cdot Din^2}{4}\right] \cdot L1$$

$$A1 = \frac{pi \cdot Dh1^2}{4} \cdot Nh1$$

;Volume 2

$$V2 = \frac{\pi \cdot L2 \cdot Di^2}{4}$$

$$Kh2 = \frac{Nh2 \cdot \left[\frac{pi \cdot Dh2^2}{4}\right]}{xt + .8 \cdot \left[\frac{\pi \cdot Dh2^2}{4}\right]}$$

Rules

$$f2 = \frac{a0}{2 \cdot \pi} \cdot \left[\frac{Kh2}{V2}\right]^{.5}$$

$$A2 = \frac{pi \cdot Dh2^2}{4} \cdot Nh2$$

;Volume 3

$$V3 = \frac{\pi \cdot L3 \cdot Di^2}{4}$$

$$Kh3 = \frac{\frac{Nh3 \cdot pi \cdot Dh3^2}{4}}{xt + \frac{.8 \cdot pi \cdot Dh3^2}{4}}$$

$$f3 = \frac{a0}{2 \cdot pi} \cdot \left[\frac{Kh3}{V3}\right]^{.5}$$

$$A3 = \frac{pi \cdot Dh3^2}{4} \cdot Nh3$$

;Conversions

$$Lin = \frac{Length}{.0254}$$

$$Di = Db - 2 \cdot xg$$

$$Db = Db\_in \cdot .0254$$

$$Din = Din\_in \cdot .0254$$

$$xg = xg\_in \cdot .0254$$

$$Lin1 = \frac{L1}{.0254}$$

$$Lin2 = \frac{L2}{.0254}$$

$$Lin3 = \frac{L3}{.0254}$$

Variables Sheet

Input	Name	Output	Unit	Comment
	ρ	0.541911		Density of Exhaust- Kg/m^3
45	cin			Exhaust Velocity- m/s
140000	P			Exhaust Pressure- Pa
287.05	R			Exhaust Gas Constant- J/(Kg*K)
900	T			Exhaust Temp- K
3.14	pi			

Input	Name	Output	Unit	Comment
10	Db_in			Silencer Body I.D.- in
2	Din_in			Inlet Pipe Diameter- in
	Ain	0.002026		Inlet Flow Area- m^2
	Aout	0.002026		Oulet Flow Area- m^2
1	restriction			Ratio of inlet area to oulet area
	Dout	0.005132		Hydraulic Diameter
0.000040	μ			Dynamic Viscosity- N*s/m
40	cout			Velocity of Exiting Exhaust
0.000813	xt			Perforated Wall Thickness
	Di	0.248868		Inner Silencer Dia. (Helmholtz Dia.)- m
	Reout	2778.806397		Reynolds Number for Outlet
	Rein	30945.765955		Reynolds Number for Inlet
	xg_in	0.101021		Annular Gap Dimension- in
	V1	0.006722		1st Helmholtz Volume
	L1	0.144206		1st Helmholtz Length
	V2	0.003274		2nd Helmholtz Volume
	L2	0.067302		2nd Helmholtz Length
	V3	0.001229		3rd Helmholtz Volume
	L3	0.025264		3rd Helmholtz Length
	Length	0.236771		Length of Exhaust Pipe
20	Nh1			Number of Holes in Volume 1
.0127	Dh1			Dia. of Holes in Volume 1
	Kh2	0.172642		Conductivity of Holes in Volume 2
20	Nh2			Number of Holes in Volume 2
.003	Dh2			Dia. of Holes in Volume 2
	Kh3	0.258964		Conductivity of Holes in Volume 3
30	Nh3			Number of Holes in Volume 3
.003	Dh3			Dia. of Holes in Volume 3
256	f1			Natural Freq. of Volume 1
512	f2			Natural Freq. of Volume 2
1024	f3			Natural Freq. of Volume 3
443	a0			Acoustical Wave Propagation Velocity- m/s
	Lin	9.321709		Total Length in Inches
	Lin1	5.677391		
	Lin2	2.649677		
	Lin3	0.994641		
	A1	0.002532		
	A2	0.000141		
	A3	0.000212		
	Db	.254		
	Din	.0508		
255	f			

Input	Name	Output	Unit	Comment
	xg	0.002566		

Appendix G – Laminar Flow Muffler Calculations - Helmholtz Approximation

TK Solver REPORT

Laminar Flow Muffler Math Model Using Helmholtz Approximations [28]

Rules Sheet

Rules

$$\rho = \frac{P}{R \cdot T}$$

$$Reout = \frac{\rho \cdot cout \cdot Dout}{\mu}$$

$$Rein = \frac{\rho \cdot cin \cdot Din}{\mu}$$

$$xg = \frac{Db - \left[Db^2 - Din^2\right]^{\left[\frac{1}{2}\right]}}{2}$$

$$Dout = 2 \cdot xg$$

$$Ain = \frac{pi \cdot Din^2}{4}$$

$$Aout = \frac{pi}{4} \cdot \left[Db^2 - (Db - 2 \cdot xg)^2\right]$$

$$Aout = Ain \cdot restriction$$

$$Length = L1 + L2 + L3$$

;Volume 1

$$L1 = \frac{a0}{12 \cdot f1}$$

$$V1 = \left[\frac{\pi \cdot Di^2}{4}\right] \cdot L1 - \left[\frac{\pi \cdot Din^2}{4}\right] \cdot L1$$

$$A1 = \frac{pi \cdot Dh1^2}{4} \cdot Nh1$$

;Volume 2

$$V2 = \frac{\pi \cdot L2 \cdot Di^2}{4}$$

$$f2 = \frac{a0}{2 \cdot \pi} \cdot \left[\frac{A2}{V2 \cdot 1.7 \cdot \left[\frac{A2}{\pi}\right]^{.5}}\right]^{.5}$$

Rules

$$A2 = \frac{pi \cdot Dh2^2}{4} \cdot Nh2$$

;Volume 3

$$V3 = \frac{\pi \cdot L3 \cdot Di^2}{4}$$

$$f3 = \frac{a0}{2 \cdot \pi} \cdot \left[\frac{A3}{V3 \cdot 1.7 \cdot \left[\frac{A3}{\pi} \right]^{.5}} \right]^{.5}$$

$$A3 = \frac{pi \cdot Dh3^2}{4} \cdot Nh3$$

;Conversions

$$Lin = \frac{Length}{.0254}$$

$$Di = Db - 2 \cdot xg$$

$$Db = Db\_in \cdot .0254$$

$$Din = Din\_in \cdot .0254$$

$$xg = xg\_in \cdot .0254$$

$$Lin1 = \frac{L1}{.0254}$$

$$Lin2 = \frac{L2}{.0254}$$

$$Lin3 = \frac{L3}{.0254}$$

Variables Sheet

Input	Name	Output	Unit	Comment
	ρ	0.541911		Density of Exhaust- Kg/m^3
45	cin			Exhaust Velocity- m/s
140000	P			Exhaust Pressure- Pa
287.05	R			Exhaust Gas Constant- J/(Kg*K)
900	T			Exhaust Temp- K
3.14	pi			
6	Db_in			Silencer Body I.D.- in
2	Din_in			Inlet Pipe Diameter- in
	Ain	0.002026		Inlet Flow Area- m^2
	Aout	0.002026		Oulet Flow Area- m^2
1	restriction			Ratio of inlet area to oulet area
	Dout	0.008716		Hydraulic Diameter
0.000040	μ			Dynamic Viscosity- N*s/m

Input	Name	Output	Unit	Comment
40	cout			Velocity of Exiting Exhaust
0.000813	xt			Perforated Wall Thickness
	Di	0.143684		Inner Silencer Dia. (Helmholtz Dia.)- m
	Reout	4719.514704		Reynolds Number for Outlet
	Rein	30945.765955		Reynolds Number for Inlet
	xg_in	0.171573		Annular Gap Dimension- in
	V1	0.002046		1st Helmholtz Volume
	L1	0.144206		1st Helmholtz Length
	V2	0.000651		2nd Helmholtz Volume
	L2	0.040168		2nd Helmholtz Length
	V3	0.000163		3rd Helmholtz Volume
	L3	0.010042		3rd Helmholtz Length
	Length	0.194415		Length of Exhaust Pipe
30	Nh1			Number of Holes in Volume 1
.0127	Dh1			Dia. of Holes in Volume 1
60	Nh2			Number of Holes in Volume 2
.0048	Dh2			Dia. of Holes in Volume 2
60	Nh3			Number of Holes in Volume 3
.0048	Dh3			Dia. of Holes in Volume 3
256	f1			Natural Freq. of Volume 1
512	f2			Natural Freq. of Volume 2
1024	f3			Natural Freq. of Volume 3
443	a0			Acoustical Wave Propagation Velocity- m/s
	Lin1	5.677391		
	Lin2	1.581410		
	Lin3	0.395353		
	A1	0.003798		
	A2	0.001085		
	A3	0.001085		
	Db	.1524		
	Din	.0508		
255	f			
	xg	0.004358		
	Lin	7.654153		Total Length in Inches