Development and Calibration of the Internal Combustion Engine used in the University of Idaho Hybrid Vehicle Powertrain

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Authorization to Submit Thesis

This thesis of Rory Lilley, submitted for the degree of Master of Science with a Major in Mechanical Engineering and titled "Development and Calibration of the Internal Combustion Engine used in the University of Idaho Hybrid Vehicle Powertrain," has been review in final form. Permission, as indicated by the signatures and dates below, is now granted to submit final copies to the College of Graduate Studies for approval.

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Abstract

Since 2009 the University of Idaho has developed a custom 250 cc engine for its Formula Hybrid vehicle program. This platform won the International SAE Formula Hybrid competition in 2014. In part this resulted from extensive hardware modifications, particularly the lubrication system, interface between stock components, and the manufacturing of a custom crank case. Equally important was extensive engine calibration and engine controller setup using a Motec M800. This thesis documents the rationale behind the various engine hardware modifications and outlines a simple calibration technique for spark timing and fuel injection timing as well as duration. Detailed information is given about Motec engine parameters for replicating stable, high performance operation of the custom 250cc engine across a broad range of load conditions. Future Formula Hybrid competition teams should find this to be a valuable source document and training document for powertrain development.

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I also acknowledge the work done by the previous generations of the FHSAE team's whose dedication and perseverance led to the development of the one of the best student built four stroke vehicles the University of Idaho has built. This vehicle which has defined my time and education at the University of Idaho would not have been possible with a few past members of the team. Sam Wos who was always willing to listen and lead, Chris Eacker who was also willing to help conceptualize new system and ideas, and Dr. Mostafa Asfoor for being there to solve so many problems and providing a new vision for the future development of the vehicle.

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Chapter 1: Introduction

A custom SI engine developed at the University of Idaho has been the center piece of the Formula Hybrid vehicle development program. The engine consists of parts from several different models and years of Yamaha 250cc motorcycle engines. Along with several different sets of donated parts from different platforms, the engine also features several upgrades to the basic functionality of the system. These upgrades consist of an aftermarket clutch, a wet oil sump conversion, a custom electronic fuel injection system and a custom final drive and several more. The most significant deviation from a stock system is the use of a billet engine case with a unique layout for packaging. To fully understand the functionality of the current engine platform it is important to understand the rationale behind modifications to the case, clutch, and lubrication system.

1.1 Previous Engine Development (2010-2013)

Chapter 2 explains what engine some of the different components came from and why they were chosen and what benefit they give the performance of the vehicle. Discussed is the use of several major aftermarket components and the purpose they serve along with any modifications performed to the components. The discussion also includes previous modifications to the oiling system as well as the overall performance of the vehicle previously achieved.

1.2: Current Engine Modifications (2013-2015)

Given the state of the performance of the engine along with the alarming frequency of malfunctions several modification were done to the engine to improve reliability and performance. Chapter 3 goes into detail the modifications that have been made to the engine over the past two years. These modifications include modifications to the crankshaft support to limit the amount of axial play in the crankshaft as well as the oil delivery system. This chapter also discusses several modifications made to the custom engine to improve systems or points of failure that have caused catastrophic and systematic failures of the engine costing thousands of dollars over the past several years of development.

One such modification was the extensive rerouting of oil to subsystems used in the engine such as the transmission, clutch and differential. The support structure of the differential input shaft was also modified to address several issues apparent in the previous design. This chapter also goes into detail about several other components and their designs and performance over the past two years.

1.3: Dynamometer Setup and Test Engine Layout

To fully realize the potential of the custom engine used in the FHSAE project a dynamometer is used to calibrate and test the engine. The setup and test engine layout is discussed in Chapter 4 where the modifications to the coupling system and test stand are examined. Chapter 4 also goes into the setup and input parameters of the dynamometer used at the University of Idaho's Small Engine Research Facility.

The test engine used with the dynamometer is a stock 2005 YZ250F engine that features only the smallest modifications to allow for the EFI conversion of the engine. A stock engine offers several advantageous features that allow for sustained testing duration and limit the possible failures that can naturally occur with custom components. The stock engine is also used for its ease of maintenance and easily attainable components should a failure occur.

1.4: Engine Calibration

The calibration of the engine is one of the more important aspects of an engine's functionality. Chapter 5 goes over the importance of calibration as in introduction and an introduction into the use and operation of the Motec ECU used on the vehicle. The chapter also covers the basic calibration technique used on the engine with a discussion on the important concerns while conducting basic calibration of the engine.

Chapter 5 also goes into the performance characteristics that the vehicle is capable of currently. A comparison is also drawn between what the vehicle was previously able to achieve in terms of several performance characteristics including acceleration, fuel economy, and time based events. A short discussion also looks at the performance of the engine with comparison to the engine model created at the University of Idaho.

1.5: Future Work

Chapter 6 discusses the conclusions regarding the work presented, looking at the success of the modifications done to the engine. The chapter also discusses the work done with the calibration of the engine and compares the experimental data collected from the engine with a 2 Zone Heat Release Model developed at the University of Idaho. Chapter 6 also makes recommendations regarding future development of existing systems and new calibration techniques to improve the performance of the internal combustion engine.

Chapter 2: Engine Development Prior to 2013

To gain a better understanding of the engine significant time was devoted to the research of the developments that have gone into the engine. This includes discovering where different engine components came from and why they were chosen for the engine. The billet case which is the single greatest custom manufactured component was also looked at to determine the source of some of the issues with the engine. Other important features were also researched such as the EFI conversion done to the engine. The hybrid architecture of the engine is also discussed, primarily about the change of the architecture that took place a year after the engine was fabricated.

2.1 Different Engine Donors

Given the nature of the engine in question it is plain to see that it is essential that the designers understand that several Yamaha engine platforms have gone into the construction of the University of Idaho's custom platform. As such this work also includes the tabulated parts in Appendix A which is accurate at the time of this thesis and includes the year and model of the engine the component is originally from but notes when modification is necessary. Different engine components are used in the custom platform for several reasons, primarily being that the competition and the custom billet case required their use and the stock system was subsequently modified to accommodate. The modification of the components or system, the majority of the time, yielded benefits to the performance of individual systems, the overall vehicle's performance or the successful packaging of the system.

2.2 The Billet Case

The most significant and original modification from the stock system is the use of a custom billet engine case shown in Figure 1. It can be seen in Figure 1that the custom case has a rotated top end which is done for packaging reasons. The stock system is intended for the use in a motorcycle, when the custom platform was originally designed from 2007 to 2009 the top end is rotated back which allowed the engine to be pulled tighter behind the driver allowing for a tighter package. Moving the engine forward gives the vehicle design the option of moving the center of gravity forward and giving the vehicle a desirable weight distribution, similar to a mid-engine vehicle and a shorter wheel base for handling. To simply the powertrain layout and reduce weight of the system the limited slip differential is incorporated into the case such that brackets were not required for mounting. Traditionally differentials have an oil bath lubrication system; the custom platform uses the engine lubrication system to oil the differential to reduce maintenance complexity and the need for two different oils in the custom case and adequate separation of the two lubrication systems [1].

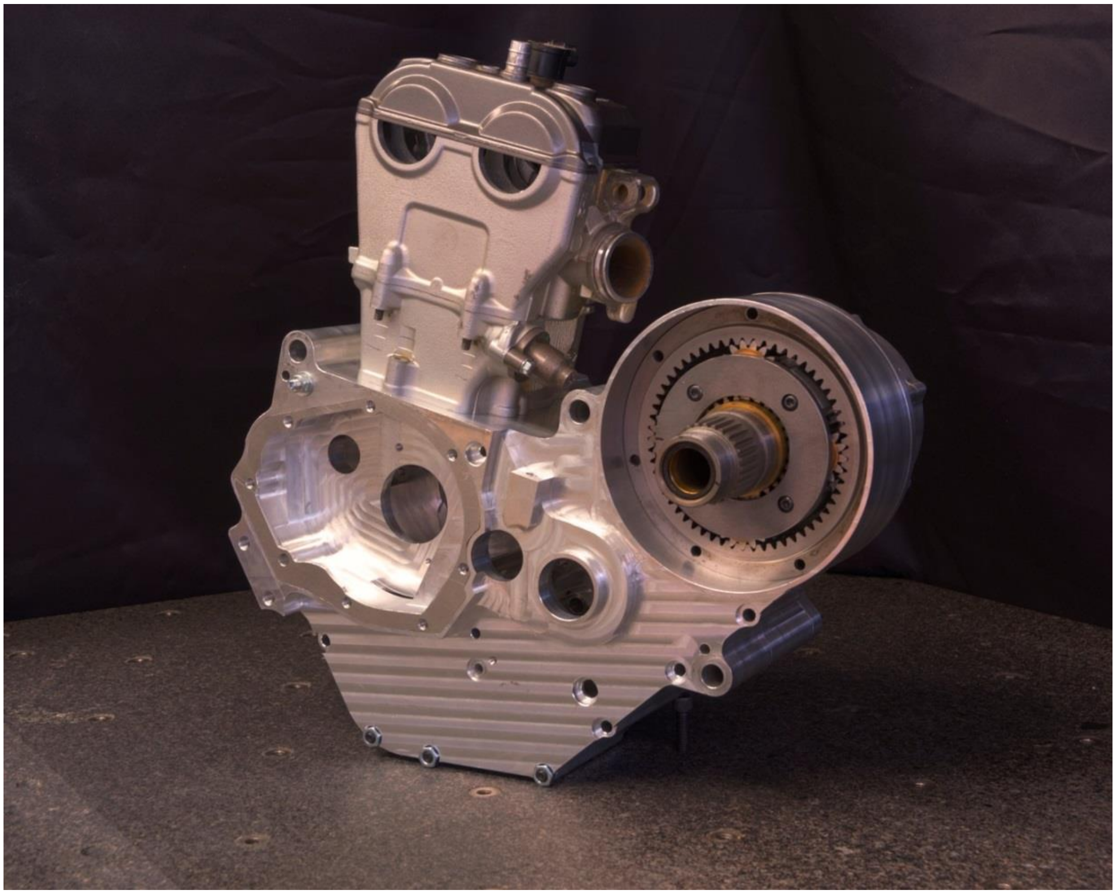

Figure 1 Custom Billet Case

2.3 EFI conversion

The most influential modification to the stock system is the EFI conversion that was done to the engine during its initial development. The conversion was done to improve the fuel economy was well as create more low and mid-range power from the engine. This occurs due to the higher amount of control available during the normal operation of the engine particularly at the low to mid-range of operation. To achieve this conversion from a carbureted system the engine was outfitted with two new sensors along with the modification of several engine components. The first sensor is the crank indexing position sensor located in the bottom end of the engine case near the fly wheel. The second is located at top of the engine and reads from the intake cam shaft and called a sync signal. The original EFI conversion was done for the engine platform used a different ECU but the hardware conversion is nearly identical as for more information regarding the EFI conversions of engine see [2].

2.3.1 Crank Indexing Position (CRIP) and Reference Signal

The addition of a CRIP sensor generate a reference signal allows the ECU to monitor and track the position of the crank relative to top dead center of the piston stroke. It is used for engine timing events. The sensor used is a magnetic sensor from a 2005 YZ250F and generates the reference signal from the trigger wheel on the fly wheel. The trigger wheel consists of a series of teeth around the perimeter of the flywheel with a signal missing tooth which gives reverence to the signal. To manufacture a trigger wheel a stock flywheel was turned down and had a trigger wheel pressed around the fly wheel. After the trigger wheel is pressed on the perimeter the flywheel was balanced. The stock flywheel was originally from a 2005 WR 250f. The modified flywheel is shown in Figure 2.

Figure 2: Modified WR250f Flywheel

The modified WR250f flywheel was turned down to remove the single tooth used in the WR system. In Figure 2 it can be seen that there is one missing tooth circled in white. This missing tooth as previously stated gives a reference point to the signal and is shown in the bottom of the Figure 2 and just to the left of center.

2.3.2 Sync Signal

The second signal needed to create the EFI system is the sync signal which is used primarily for starting the engine. The signal picks up upon the location of the intake cam and as such required the creation of a custom valve cover to hold the sensor such that is pick up as the intake cam moves past. The sync signal sensor comes originally from a 2005 R6 engine where the Hall Effect sensor was used for a cam pick up in the valve cover. From the Sync signal and the CRIP sensor the timing needed to properly operate the engine for spark and fuel injection timing is calculated. The custom valve cover is shown in Figure 3 where the sync hall effect sensor is shown in the bottom right of the figure circled in white.

Figure 3: Custom Valve Cover

2.3.3 Throttle Body

The final addition to the engine needed to achieve the EFI system is the use of a throttle body along with an appropriately sized injector for the horsepower rating of the engine size. Figure 4 shows the 2008 WR250x throttle body used on the platform.

Figure 4: 2008 WR 250X Throttle Body w/ Fuel Rail

This throttle body was used due to its original fitment to a similarly sized engine along with an injector intended for an engine platform that generates approximately the same power as the 2005 YZ 250F for which the engine’s top end is from. The WR250X throttle body has a 40 mm throat and is fairly small packaging along with little to no weight added to the platform. The WR250X throttle body was also a stock component that needed no modification in order to integrate into the system. The throttle body came with the necessary throttle position sensor needed to operation the EFI system with the aftermarket ECU.

2.4 Hybrid Architecture: Pre to Post-Transmission Conversion

Originally designed to be a pre-transmission hybrid architecture the engine platform was modified to be a post-transmission architecture while still maintaining a parallel coupling of the high voltage system and internal combustion engine. The main difference between the two designs is where power is applied from the electric motor to the power train. The names are fairly self-evident, but the location is fairly important do the modification of the Billet engine case to achieve the modification. The pre-transmission used the counter shaft of the engine which held the clutch and half of the gear sets of the transmission as shown in Figure 5, where the stock component is on the top half while the modified is featured just below [3].

Figure 5: Custom Counter Shaft for Pre-Transmission Architecture

This design allowed the electric motor to apply a greater torque to the wheels due to the multiplication of the constant mesh transmission allowing for greater performance of the vehicle. However during testing of the vehicle it was discovered that the vehicle was slower than expected due to simulation error when accounting for the shift times of the vehicle. With the electric motor coupled to the counter shaft the engine needed to be shifted during electric only mode of the vehicle to achieve higher speeds. The pre-transmission system also required a pulley system to match the speeds of the counter shaft to the speed range of the electric motor. This pulley system added considerable weight to the system for a diminished performance and as such was augmented in 2013 with the design and build of a new vehicle [4].

The modification came with removing the pulley system saving 50 lbs from the overall weight of the vehicle while coupling the electric motor to the output shaft of the engine with a one way bearing and flexible coupling. The stock shaft vs. the modified output shaft is shown in Figure 6; the long stinger on the custom shaft is where the electric motor is coupled. This coupling allows the electric mode to work independent of the transmission and apply torque directly to the differential improving electric acceleration and given the max speed of both the engine and electric motor there is no need for a gear reduction. This arrangement still allowed for the clutch to manage different operational speeds of both power systems. This was accomplished by the use of an aftermarket clutch that would only engage if the crankshaft would spin up the clutch to a certain speed which will be talked about in a later section.

Figure 6: Custom Output Shaft for Post-Transmission Architecture

2.5 Pingle Shifter

The pingle shifter is an electronic shifter that uses a solenoid to actuate the shift mechanism of the engine. The shifter is used commercially as a means of shifting for disable motorcycle riders. The shifter is a simple device that uses a timed delay to short the signal to the coil to prevent the ignition event to occur which unloads the engine allowing for an easier shift. The implementation of the pingle shifter was intended to allow for faster shifting times through small amount of movement from the driver to actuate the shifter through two buttons on the steering wheel. The pingle shifter gave opportunities to improve the control system of the hybrid system while seemingly shrinking the experience gap that can occur between different driver. This improvement to the control system comes from the concept of controlling the gear selection of the engine during operation similar to that of an automatic transmission, so that the load of the engine can be maintained to improve the fuel economy of the vehicle.

2.6 Rekluse Z-Start Pro Clutch

The Z-start Pro clutch is an aftermarket centrifugal clutch that originally was installed for its inherent functionality. The Z-start functions by the use of ball bearings as weights, which are arranged in around the center of the clutch. As the engine is spooled up the weights move along the troughs towards the outer perimeter of the clutch wedging themselves between the main pressure plate and the retraining plate. This motion applies pressure upon the main pressure plate which compresses upon the throw-out spring and bearing and applies the compressive pressure upon the clutch pack allowing torque to be carried through the clutch. The main pressure plate is shown in Figure 7 which also depicts the troughs that allow the system to function.

Figure 7: Rekluse Z-Start Pro Clutch

The z-start offers the ability to come to a complete stop in any gear. This function is a necessity with the use of the electronic pingle shifter as neutral is nearly impossible to find with the pingle, without which the vehicle would be continuously in gear and applying power to the transmission. The reciprocal of this inherent function also exists in that the clutch allows the operator to launch the vehicle in any gear. This system in theory works however as discussed further in Section 3.7 the design and implementation of the system held several issues.

2.7 Wet Sump Conversion

The stock engine uses a dry sump lubrication system with a Duocentric IC pump. The Duocentric oil pump is a non-crescent type that uses smooth rotors to seal the different section of the pump and uses the inner rotor to drive the outer rotor. The inner rotor of the pump moves eccentrically to the outer rotor which causes the pumping action as the volumes change during rotation of the two rotors [5]. The major modification to the oil pump is the removal of the primary rotor set which in the stocks system pumps oil up to the reservoir which feeds the second high pressure side of the pump which feeds the rest of the engine after the oil filter. The second major modification is the machining of the separation plate between the primary and secondary rotor set to allow oil to be pulled through the pick up from the bottom of the engine by the second rotor set.

There are several advantages to the modification and disadvantages. The advantages of the dry sump are that system can hold a large quantity of oil due to the oil reservoir allowing for longer oil life meaning fewer oil changes and the secondary pump requires less work as it is gravity fed from the reservoir. The wet sump system doesn’t require the oil reservoir and extra oil saving weight. The disadvantage of the wet sump system is that it is prone to oil starvation during high cornering forces as the oil is pushed up against the wall of the engine case away from the pickup. However given the design of the billet case a small compartment is used to ensure oil cannot move far enough away from the oil pick up during corner to cause oil starvation [6].

2.8 Resulting Vehicle Performance

The overall performance of the vehicle can be very difficult do to the large number of variables that can affect the outcome of a single repeated test. Given the issue with repeatability of vehicle performance it is best to break the vehicle into systems and try to isolate the testing towards individual systems or components. As of 2014 the vehicle did undergo testing to find the performance characteristics of the vehicle, particularly the power train systems. The testing was conducted to validate the TK solver math model created to test energy allocation and performance predictions for future development. The results show that the vehicle is capable of acceleration runs of 75 meters in 6.6 seconds for operating in ICE only mode, 6.4 in electric only and 5 seconds in hybrid mode with a passive split control system. The testing also focused on fuel economy of the engine showing that during ICE only mode the vehicle is capable of 18 mpg while in hybrid mode the vehicle is capable of 23.5 mpg [4].

This performance data shows that there is plenty of room or improvement in the fuel economy of the vehicle for both the ICE only mode and hybrid mode. This is improvement is not only possible but necessary as the competition that the vehicle is built for requires the vehicle to have at least a fuel economy of 27 mpg in hybrid mode to complete the endurance event. The possibility of the improvement is shown with the work done with a GT suite model created to simulate a new control system which predicted that the vehicle in hybrid mode is capable of 25 mpg with the passive system [7]. The calculated mpg determined from [7] is based off of a 2 Zone Heat Release Model developed in [8]. The model developed in [8] predicted that the engine would be capable of minimum brake specific fuel consumption (BSFC) of 290 g/kw-hr at approximately 8000 rpm. This model was also verified from data collected using a dynamometer which showed at most a relative error of 8.6% over the points collected [8]. These numbers are based off of the calibration of the engine which provides the largest component of the fuel economy and performance of the engine as of 2014.

Chapter 3: Engine Modifications Since 2013

The custom case has seen several modifications over the past two years to improve its reliability and allow for the proper functioning of several key sub systems. These modifications are each made to improve the reliability of the engine while ensuring that the engine will not prematurely wear out specific components which had been a frequent occurrence. The major modifications that have been performed to the custom are with respect to the crank shaft, the support structure for the chain drive and planetary gear reduction and the lubrication system.

3.1 Crankshaft Axial Play Modification

The crank shaft is as stated from a 2005 WR 250F which allows for a larger stator and starter integration. The use of this particular crankshaft facilitated several key design goals. The first as mentioned in chapter one, with some modification the crank indexing position and reference signal were created, the second is that the larger flywheel provided for a larger stator. The stator acts as a generator similar to that of an alternator in a common vehicle, however the stator requires a rectifier which converts the ac single to a dc signal and handles the recharging of the low voltage battery. The need for the larger stator allows for the needed power to operate the aftermarket ECU and ignition system which have a larger power draw then the stock systems. The last portion of the design is the incorporation of the starter assembly which is required by the rules of the competition [1].

The modification to the crankshaft assembly is concerned with the axial tolerances of the crankshaft and the custom case. Upon rebuilding the custom engine it was noted that the flywheel would not seat properly on the tapered section of the crank. This interference was caused by the large amount of axial play in the crank shaft which allowed the starter clutch assembly to come into contact with the chain guide at the bottom of the chain tensioner. Upon inspection it became clear that the crankshaft had an excessive amount of axial play that allowed the crank shaft to float and rub the starter clutching assembly, mounted on the back of the flywheel. The wear was minimal on the clutching assembly as it is a hardened steel part, the chain guide did have a fair amount of abnormal wear due to the contact.

Using the HAAS CNC mill and a dial indicator the amount of axial play was determined. This was done by placing the engine case with only the crank installed and a few bolts in the HAAS with a large c-clamp and two alignment blocks fitted in the t-slots of the range. The Dial indicator was attached to the spindle and using the manual controls of the mill the spindle was brought into place. The crankshaft is then push axially till its play is bottomed out and the dial indicator zeroed on the end of shaft. Then by gently pulling on the crankshaft the amount of axial play can be measured. This process was done twice once with the custom engine to determine to amount of axial play and a stock engine in order to bench mark allowable play in the crank. Using this method it was determined that the stock engine has, noting that is a used engine and tolerances can vary, of 0.014 inches. The custom engine showed an axial play of 0.0475 inches, well above the stock measurement. The testing set up with the engine itself is shown in Figure 8 were the dial indicator can also be seen.

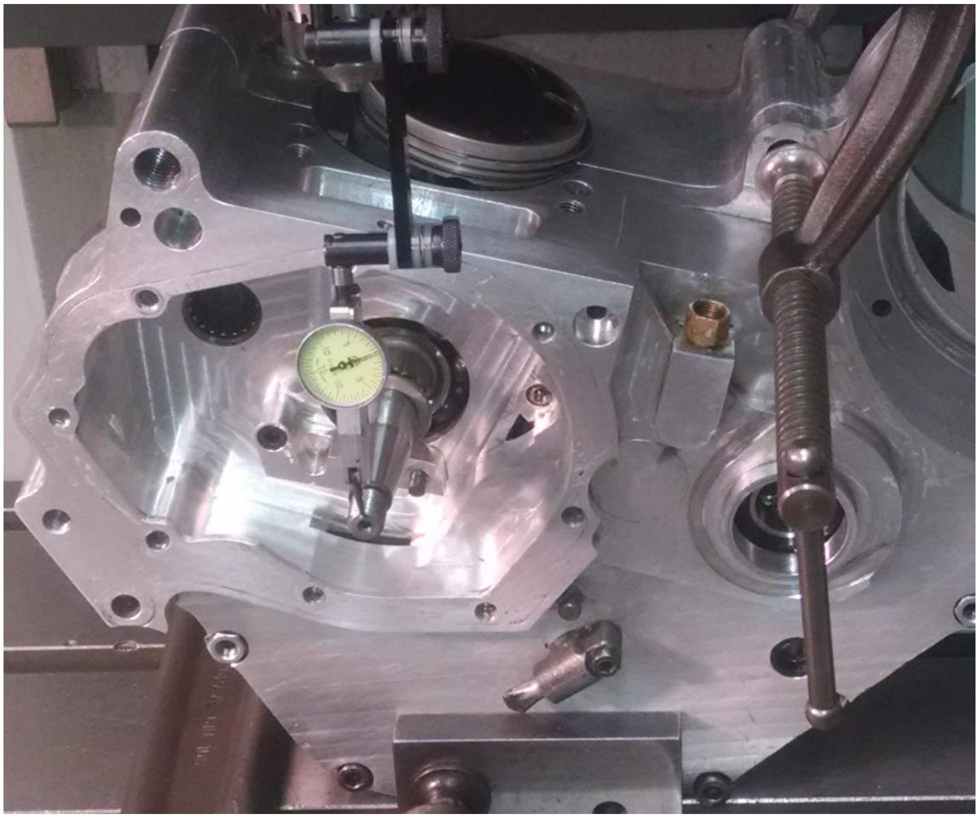

Figure 8: Crankshaft Axial Play Test Setup

Next, more measurements were taken on the case itself to determine the root cause of the axial play observed in the crankshaft. Using micrometers, measurements were taken on the depth of the bearing seats, the distance from bearing seat to the opposing seat, which were compared to the stock engine as well as the measurements of the crank shaft itself. Using these measurements and upon meticulous inspection of the bearing seats it was determined that the depth of the bearing seats were the cause of axial play.

To remedy this issue several solutions were considered the first solution brought forth was to simply machine the two mating surfaces of the case, in order to take up the axial play. The second solution was to create shims to be used between the crankshaft and the bearings to limit the movement of the crankshaft. The third solution brought up was to shim the bearings towards the crankshaft. The first solution of machining down the mating surfaces was determined infeasible because of the difficulty of machining down the mating surfaces, by way of ensuring that both halves would still align properly while not pinching any other shafts or components between the two halves. The second solution of using shims between the crankshaft and bearings was tried due to its relatively low cost and time to implement. Two 0.020 inch shims were made on the HAAS mill, this size was chosen because it would take up the majority of the play but allow for some thermal expansion of the parts during operation. Upon placing the shims and partially assembling the engine with just the crankshaft it was found that the shims were allowed to float between the radius of the crankshaft and bearings. It was determined that this solution may cause rubbing between the shims and bearings which could inadvertently ruin the oil pump should the metal particulate make its way through the pick up at the bottom of the engine.

The final option of using shims to take of the axial play out of the crankshaft was used. New shims that matched the out race of the bearings were made and placed between the bearings and their seats. The two shims were made from mild sheet steel with an individual thickness of 0.020 inches; mathematically these shims leave 0.0075 inches of play in the crankshaft. With the shims in place the axial play of the crank shaft is measured again using the HAAS set up shown in Figure 8 which showed that the crankshaft had 0.009 inches of play. The disparity of the measured value and the predicted is attributed to debris on the shims and bearing seats. Using a dial indicator the level of the bearing was also measured relative to the mating case halves. This measurement determined that the bearings were in fact seated parallel with the case halves and the 0.002 inch error was deemed acceptable. The final axial play of 0.009 inches was determined to be an acceptable play which allowed for the thermal expansion of the components. After several hours of run time on the engine the crankshaft play was checked again during another rebuild of the engine and the play was still 0.009 inches. With the axial play of the crankshaft seemingly unchanged after use shows that the modification held and did not wear or seat into the case the 0.002 inches observed in the shims placement.

It should also be noted that the axial play in the crankshaft also affected the custom fly wheel used in the engine. With 0.0475 inches of axial play the crank shaft was allowed to shift far enough to one side to push the fly wheel into the left side cover which holds the stator and starter gearing. The flywheel had cut a deep grove into the cover which naturally heated the fly wheel. Aside from the metal debris which would have quickly ruined the oil pump, the heating of the internal magnets used in the fly wheel were subsequently weakened [9].

A simple qualitative test is possible, take the worn flywheel and a similar sized flywheel and place both on to a vertical metal surface. The worn fly wheel will hardly stick to the metal while the new flywheel should only need a small amount of support but significantly less than the worn flywheel. The weakened magnets reduce the overall effectiveness of the electrical generation of the stator which is dependent upon the strength of the magnetic field. With the stator at a diminished capacity the engine does not charge the battery as effectively as it should which is why the battery needs to be charge constantly in between testing sessions. To resolve this issue a new custom flywheel is needed, and was not done due to the late timing of its discovery and time to the completion but it is highly recommended that it is fixed during the next rebuild of the engine.

3.2 Final Drive Support Redesign

The powertrain system operate using a chain drive for it primary gear reduction. The chain connects the output shaft to the planetary gear reduction and differential. Between the output and reduction shaft there is a support structure that ties the two shafts together. This structure lessens the cantilevered loads seen by the bushings used in the planetary gear reduction. Given that the planetary gear reduction is a one of kind component that cannot be easily replaced its vital role in the power train, its support structure’s integrity is of the upmost importance to ensure the service life of the differential.

The original component was made from 6061 aluminum and was salvaged from a previous FSAE vehicle using a GXSR 600 engine. The component used a 30 mm ID bearing for the output shaft and a thin walled SKF bearing, and in order for the component to fit the SKF a brass ring was pressed into the aluminum to take up the area meant for a larger bearing used on the previous engine. The previous design is shown in Figure 9 were the brass ring is installed into the differential side of the structure.

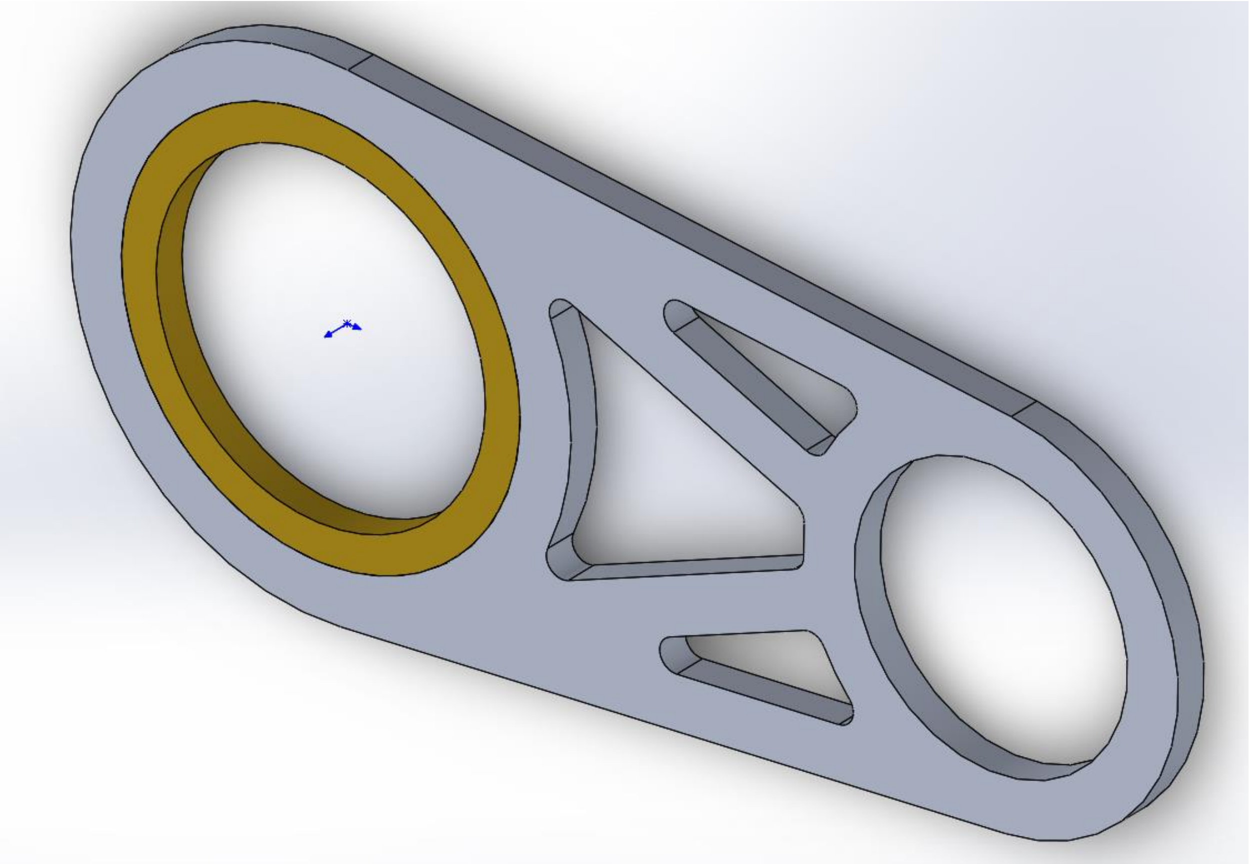

Figure 9: Original Structure Design

The original design came into question when upon removal during a rebuild when the SKF bearing showed severe wear and notching during hand operation. Upon further inspection the output shaft bearing was found to be wearing down the output shaft as it was only making contact in one location. Using the 3D coordinate measuring device coupled to solid works the center to center distance of the original part and the two shafts on the engine itself. These measurements found that the component was oversized with the support structures center to center distance for the two shafts was 5.28 inches were the two shafts measured 5.269 inches from center to center.

To remedy this design flaw a new component needed to be manufactured. While the effort was being made a new model was created in solid works using the measured center to center distance from the actual engine itself. The new model was created to improve the design by reducing the weight of the component and cost of manufacturing by using off brand bearings. The final design can be seen in Figure 10, were the material has been changed to the 7050 Aluminum series to reduce weight through strength of material. The part drawing can be found in Appendix B.

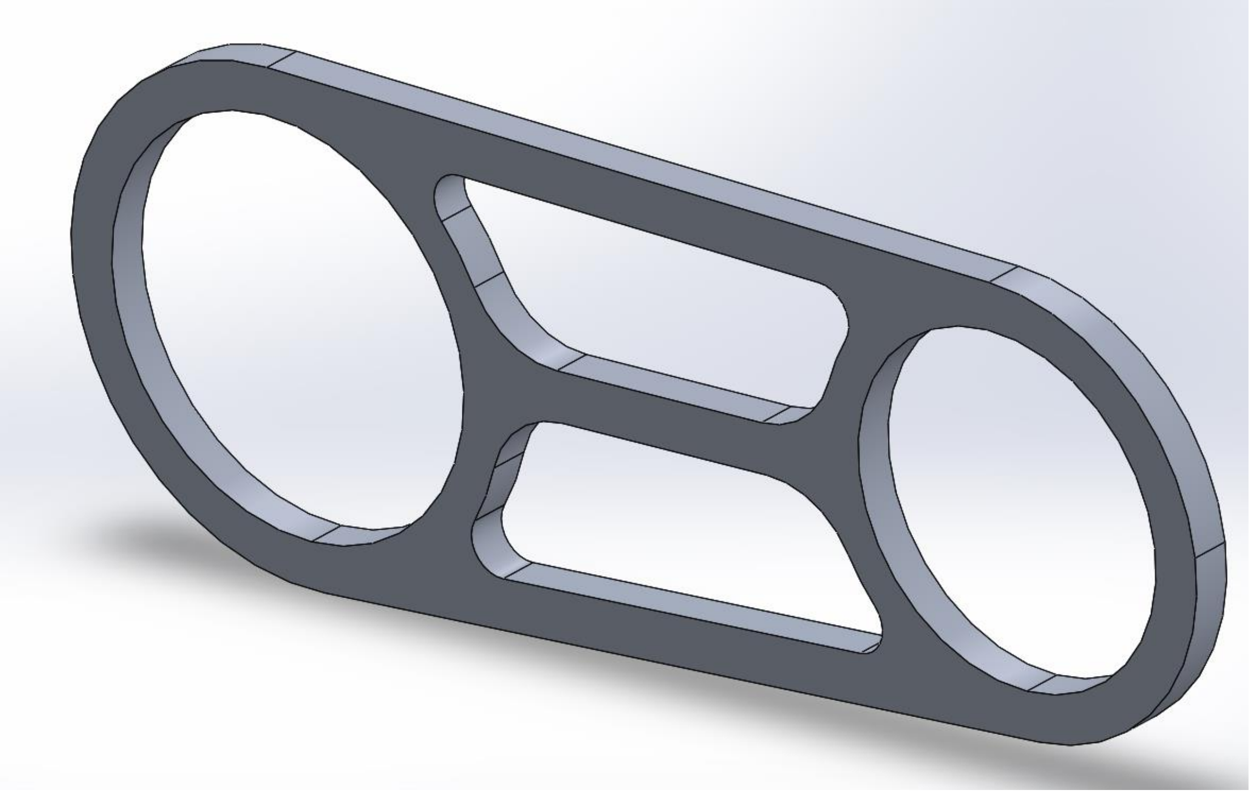

Figure 10: Render of New Support Structure

The brass ring for the thin walled bearing has been removed for weight, and the SKF bearing is now an off-brand bearing showing a cost reduction from $275 for the SKF to $15 for the off-brand. The weight of the component is also further reduced by the using the Trend Tracker function in Solid Works Simulation. The Trend Tracker function allows the recording of interactions of designs while recording properties and outputs from simulations. For this component weight, max Von-Mises stress and maximum displacement are all recorded per iteration. Plots in Figure 11 shows the different iterations of the design shown in Figure 9 where thicknesses of sections of the component are thinned up to a maximum deflection of 0.001 inches. This limit was chosen due to the important function the component plays, and the cost of the components it interfaces which are very expensive and extremely difficult to replace should the support fail. With the design work done the new part weighs 0.21 lbs while the original part weight, with the brass ring, was 0.5 lbs.

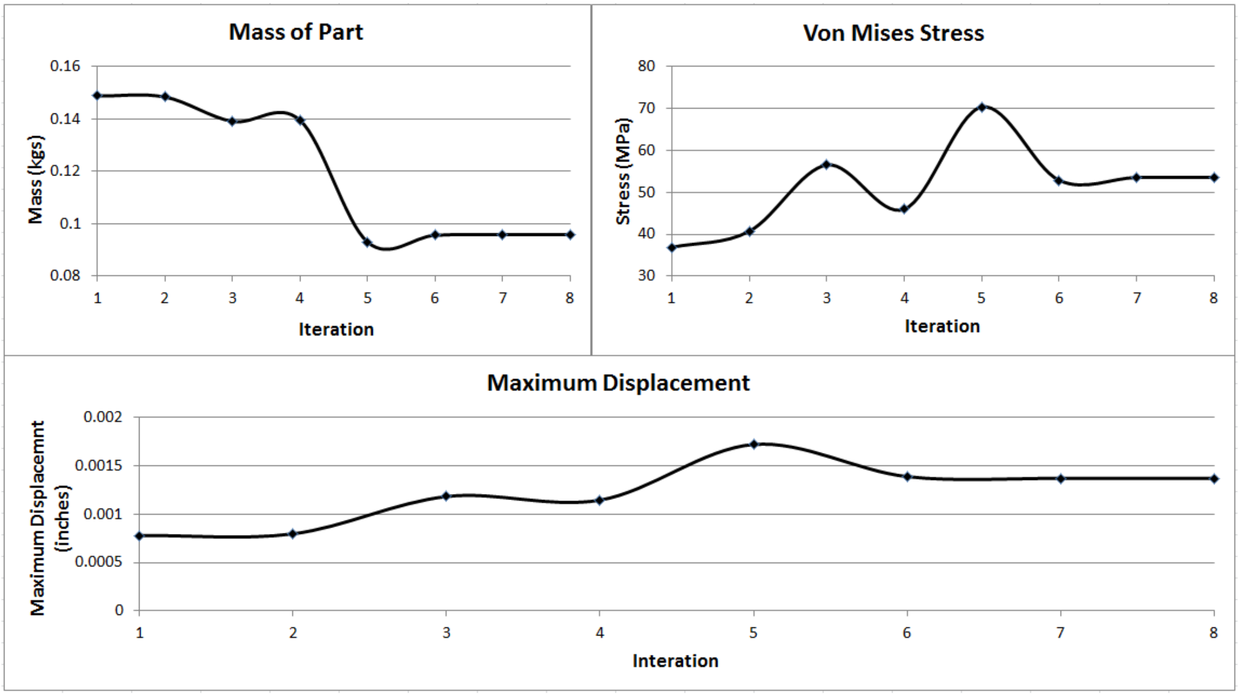

Figure 11: Results of Multiple Iterations of Design

3.3 Liquid Separation Issues

Initially the custom engine had a severe issue with mixing gasoline, water, and oil. This issue stemmed from several reasons. This issue was apparent when the engine had its oil changed which would appear milky white and had a slight granular texture it to and smelled distinctively of gasoline. To help determine what was being mixed the oil was collected during the oil change and placed in a glass jar, the jar was then left so that the individual substances would separate. After a day or two the mixture would separate out showing a white substance at the bottom of the jar, with engine oil next with two thin lines at the very top of the jar.

It was summarized that the two top layers were gasoline and water. With an inspection of the top end of the engine it was determined that the gasoline worked its way around the rings of the piston and cylinder which were both out of manufactures specifications. In later chapters it is determined that a portion of the gasoline may have come from the original calibration of the engine due to the large fuelling values used. The water was determined to originate from the water jacket, specifically where the cylinder was mated to the engine case. This was discovered when the stock case was visually inspected and found that is used an o-ring along with a hollow alignment dowel to seal the water jacket of the engine and cylinder at the base. The custom engine had never received the groove necessary for the o-ring to seat, nor had the engine received a hollow alignment pin for the water jacket.

The bottom section of the sample, the white substance, turned out to be the RTV gasket material used throughout the engine. This material is not a gasoline rated material, and as such would be dissolved by the gasoline in the oil. This realization explained the white substance in the oil as well as the persistent issue with oil leaks found on the engine during several rebuilds. To solve this issue a new gasket material was found that was gasoline resistant, Permatex Motoseal. This material also sets in 20 minutes and works exceeding well for the custom case as it is a non-hardening which makes the removal of the material faster and cleaner.

3.4 Oil System Modification

The oil system on the custom engine is one of largest modifications done since the pre to post-transmission conversion. The major modification included the addition of oil passages with the case to ensure adequate oil delivery to the specific components and the restriction of the differential oil passage. These modifications were intended to improve the longevity of the engine and performance of several key systems. These modifications brought the custom engine case closer to the stock system while still incorporating the differential into the lubrication system.

The original design of the engine case and hybrid architecture limited the oil routing primarily to the clutch and transmission. This oil passages could not be machined into the original design due to the custom counter shaft used in the pre-transmission hybrid architecture. The stock case oil routing, shown in Figure 12, takes oil from the secondary rotor set of the oil pump, passed the oil through the filter which then fed the oil to the crankshaft, the cams in the top end. The oil routing also took oil from the oil filter to the cross over tube, which wetted the constant mesh transmission and then carried oil to the output shaft, and counter shaft both of which have oil ports to supply oil to the bearing races and splines. The oil delivered to the counter shaft also continues through the shaft and exits into the wet clutch which through centrifugal force drives the oil through tiny holes to lubricate the friction plates.

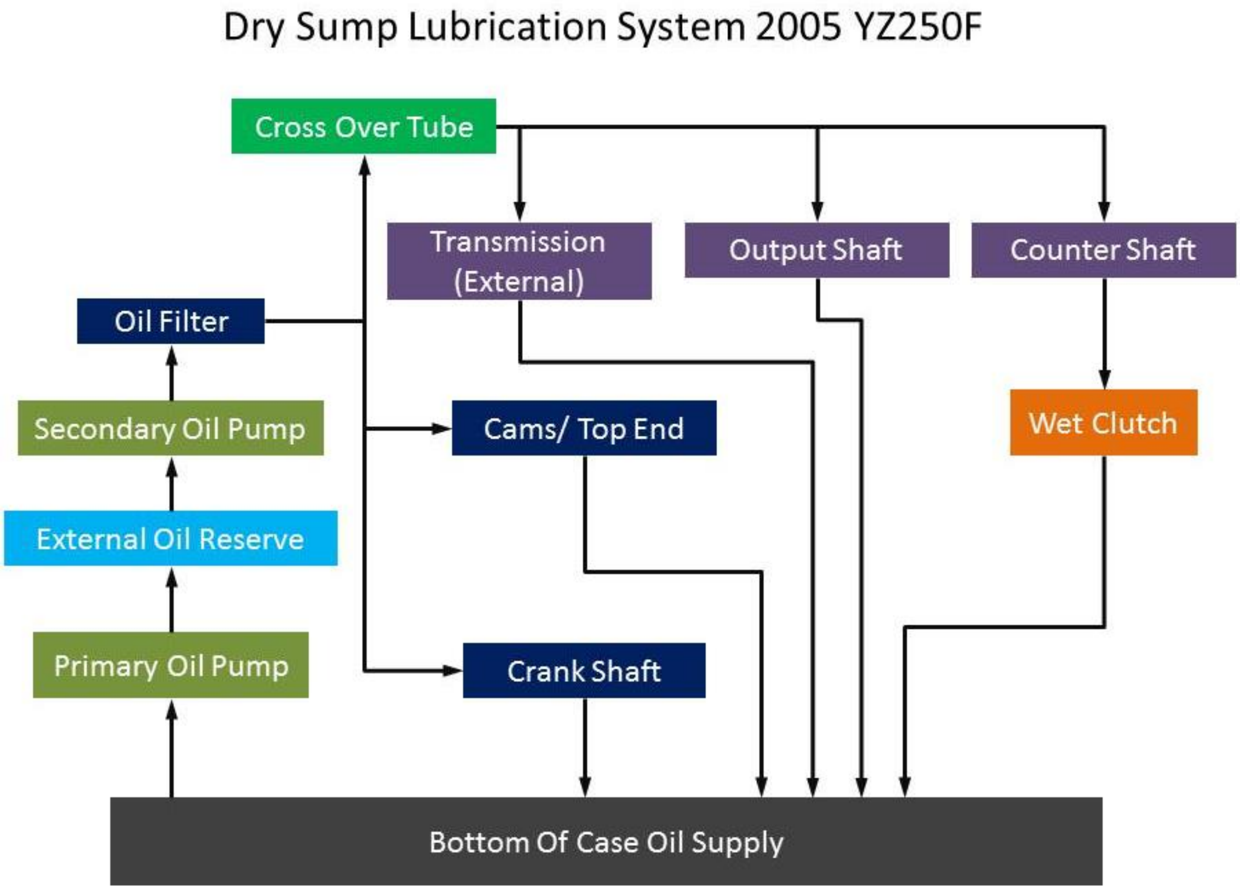

Figure 12 Stock Oil Routing Diagram

The custom engines oil routing is very much the same as the stock engine with some slight changes in the later stages of the routing and the inclusion of the oil line to the differential. The original custom case oil routing is shown in Figure 13 where the red lines reflect the changes and additions to the system. The oil line feeding the differential also received a restriction at the case to limit the oil flow to the differential. Originally the oil line was completely open which drastically reduced the oil pressure of the system which drastically reduced the longevity of several components specifically the cams which use a bearing surface, similar to a journal bearing rather than standard radial ball bearings.

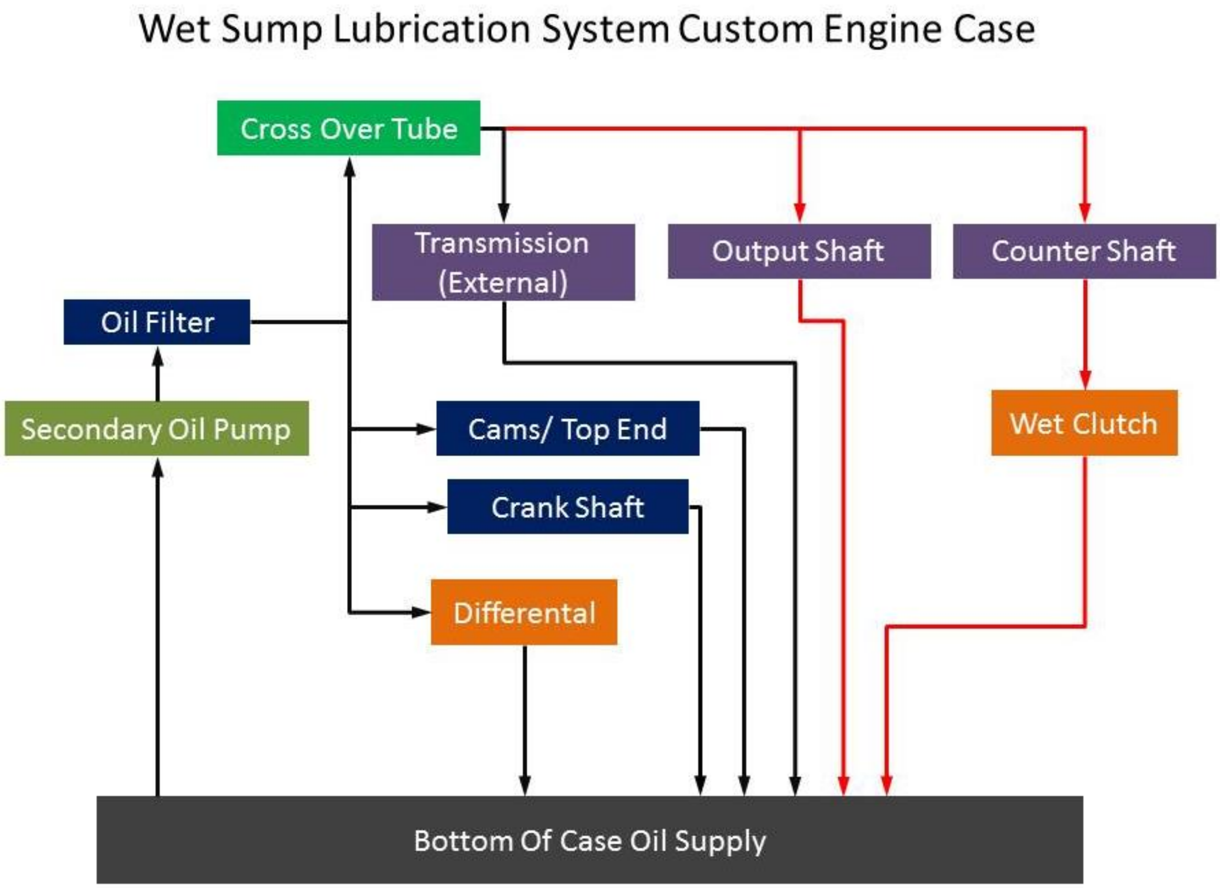

Figure 13: Custom Case Oil Routing Diagram

To create the new oil passages necessary several drilling operations were needed to move oil from one location to another through sections of the case to save weight and because in some instances it was the simplest way to provide oil the components. The main ports that needed to be drilled were the port to bring oil from the back of the countershaft to the output shaft and the new port for the cross over tube. The manufacturing of the port for the shafts is shown in Figure 14 while Figure 15 shows the new port for the cross over tube. The cross over tube received new porting because in the original state of the case the cross over port put oil to the cross over tube in the reverse direction of the stock system. This new porting was necessary as the oil galleys needed to move from the cross over tube to the counter shaft, and output shaft and eventually to the clutch.

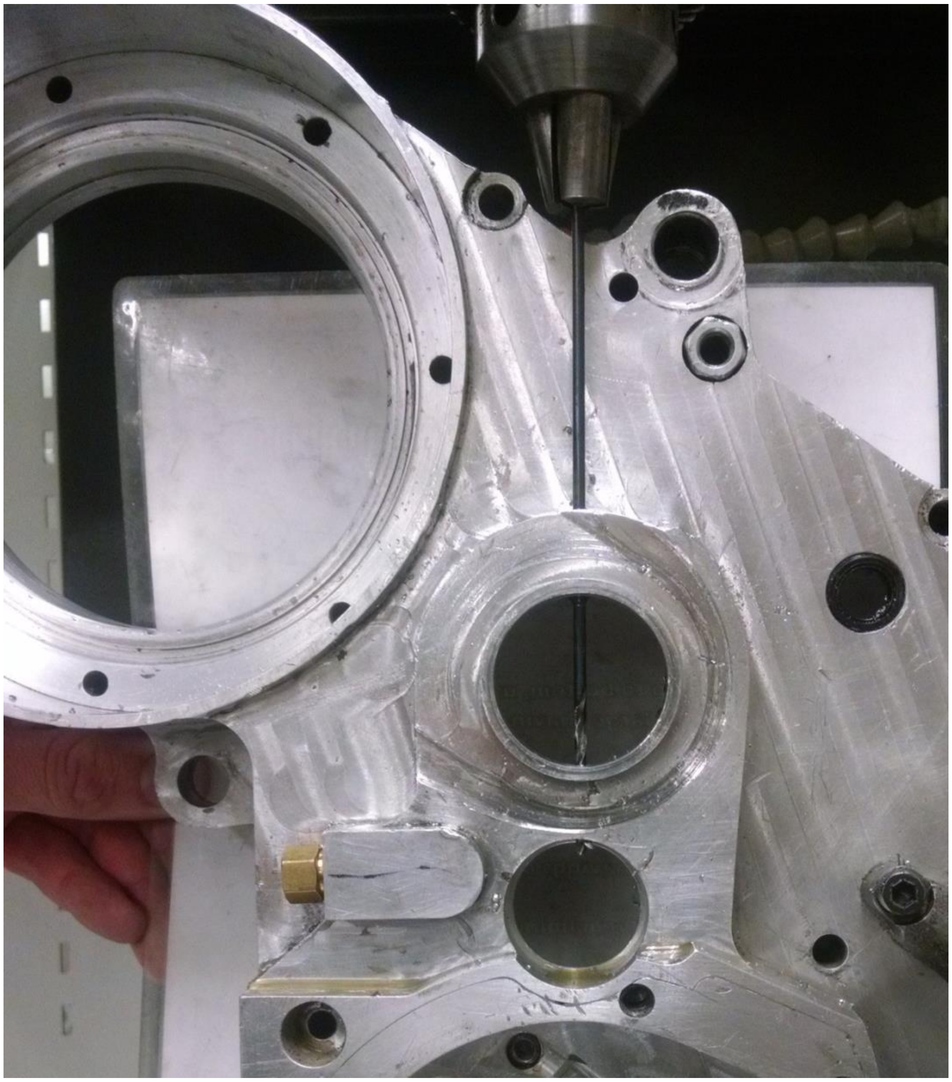

Figure 14: Drilling of Oil Galley from Counter Shaft to Output Shaft

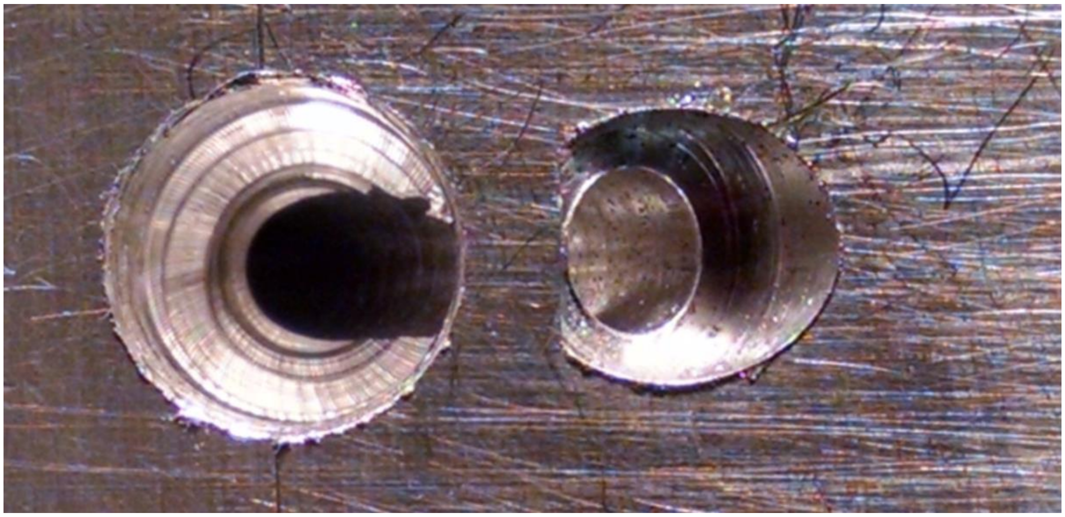

Figure 15: Port for the Cross Over Tube

In order to properly seat the fitting to be used in the cross over tube port the galley was drilled at 55 degree angle from the top surface of the case where the cylinder attaches to the case. The galley is shown on the right side of Figure 15 and was plugged using a small slug of aluminum pressed into place and the port was tapped for the flared fitting shown in Figure 16 which depicts the final porting for the cross over tube. On the right side of the flared fitting the slug was milled flush with the surface of the case as shown by the faint outline in Figure 16. The galley for the transmission shafts shown in Figure 14 was plugged using a similar method.

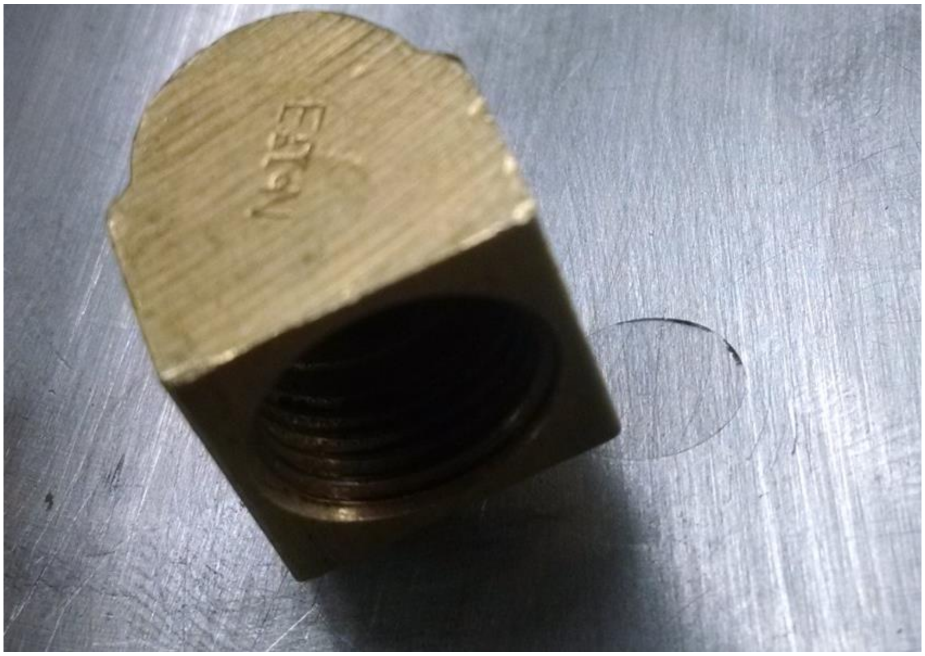

Figure 16: Finished Port for Cross Over Tube

3.5 Wet Sump Conversion

Aside from the new oil porting the wet sump conversion on the engine received an upgrade after it was discovered that it was faulting during a rebuild. The oil pumps used in the engines are typical pumps found in the power sports industry for their small packaging, low noise generation and can be driven off of the crankshaft or auxiliary systems [5]. Both engines use after some fashion the same pump from a 2005 YZ250f engine, however the custom engine has a modified pump for the wet sump conversion covered briefly in Chapter 1. The wet sump in the custom engine has one major drawback to its design, where it is extremely susceptible to becoming ruined should any metallic debris pass through the engine. The wet sump conversion that is on the custom engine saw the removal of the oil reservoir and the primary rotor set, and a small hole put into the separation plate between the primary and secondary set of rotors. This small hole allows for the secondary rotor set to pull oil through the sump at the bottom of the engine. The profile of the hole cut into the separation plate is shown in Figure 18 while Figure 17 shows the stock separation plate. The hole that is placed into the separation plate is done using the HAAS mill for its accuracy and consistency.

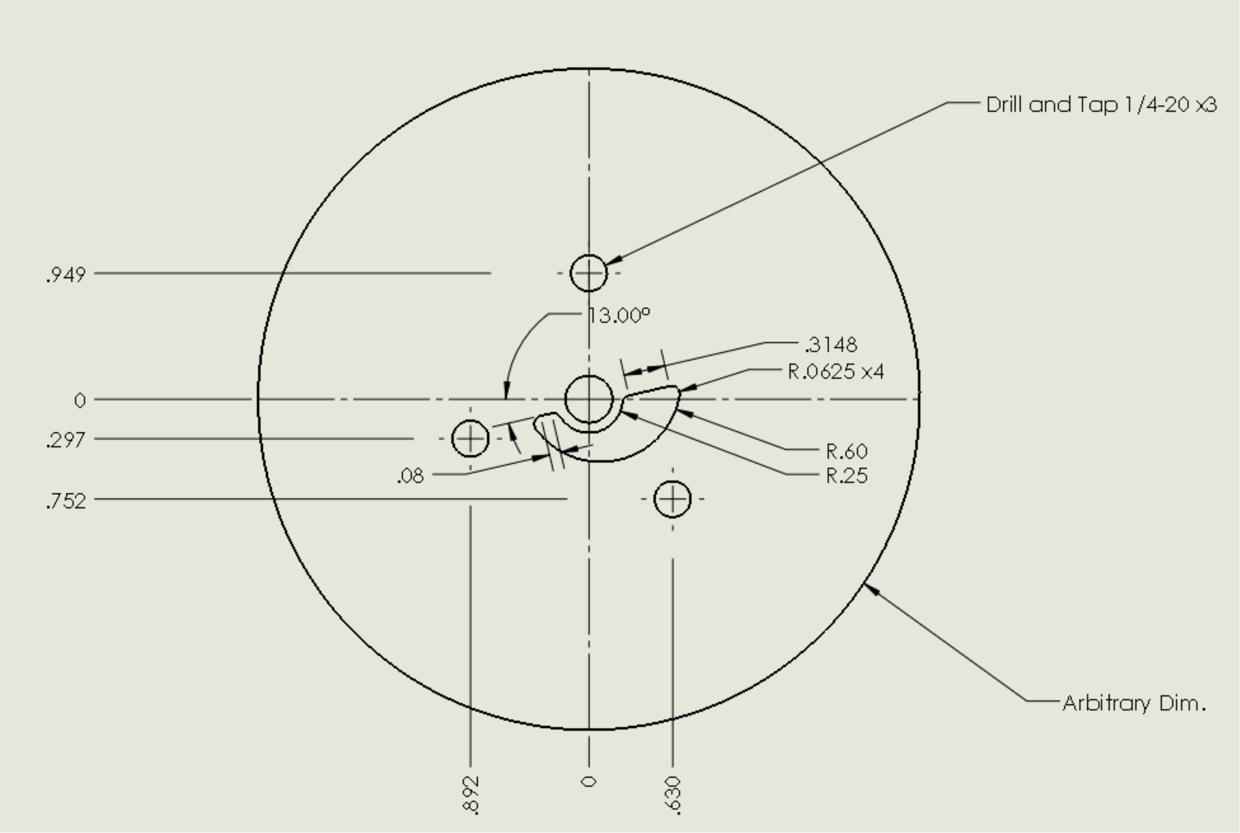

Figure 17: Wet Sump Conversion Fixture Plate

Take note that the stock part shown in Figure 18 is a used component, which is given away by the scoring that is shown in the surface of the part. This scoring incidentally renders the part useless as the pump requires a smooth surface for the secondary rotor sets to ride against to create suction in both the wet and dry system. The small scoring allows air and oil to move from one set of teeth with in the pump to the next reducing the effective pressure of the pump. Scoring from de-burring is also a possible failure of the pump which is why I have found that a very small file works best for the task with 320 grit sand paper.

Figure 18: Original Separation Plate

Should a small scratch happen to put into the surface of the part it is possible to save the part by using a high grit sandpaper to buff the scratch or score from the surface using a figure eight pattern while wet sanding. The best method found to test whether the part is satisfactory is using a finger nail, should you feel the scratch with a finger nail the part will not function properly. It is also worth noting that among the many failures of the engines over the years the majority of the them have a root cause with the oil pump being defective due to scoring caused by metal debris being passed through it and not being properly replaced.

In normal operation such as on a personal motorcycle the oil pump is relatively robust. This difference occurs simply because the majority of dirt bike riders do not push the engine to the extreme ends of its performance. Particularly on the dynamometer engine, the engines can easily rest at peak power for minutes at a time which is not a realistic occurrence in standard operation of the engine. This constant loading of the engine places extreme pressure on the lubrication system to be function correctly during the high load operation. Should the oil pump be less than its intended performance pre mature failure of components can and will result.

3.6 Crankshaft Sealing Modification

Typically the crankshaft will simply wear out over time, particularly with respect to the allowable play found in the connecting rod. This is caused by the large amount of stress that the connection is subjected to. However over the past several years nearly every crankshaft in the custom engine and dynamometer engine have been ruined due to a large heating issue, the end result of this shown in Figure 19.

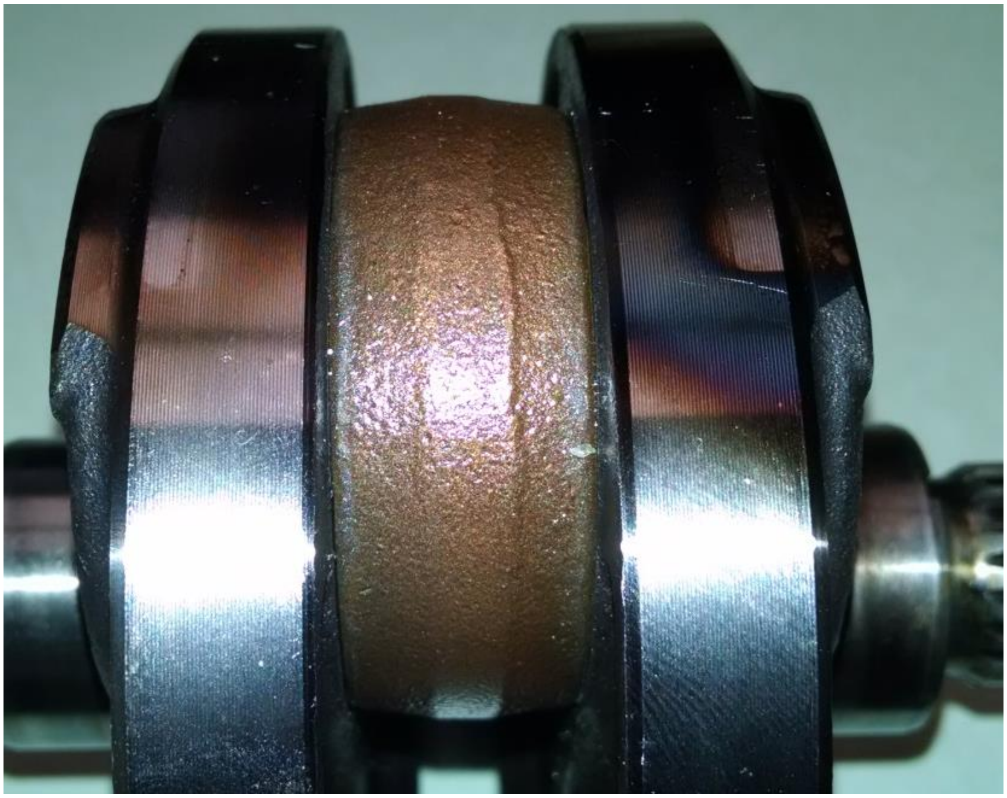

Figure 19: Burned Crankshaft

The crankshafts developed large heat affected zones where the pin used to attach the connecting rod met the counter balance of the crankshaft. These heat affected zones are caused by a large amount of friction between the connecting rod and the sides of the counter balance and the needle bearing used for the lower end of the connecting rod. It was reasoned that this large amount of heat could only have be caused by a lack of oil to the internal needle bearing used in the crankshaft. Originally it was believed that a faulty oil pump was to blame for the lack of oil. With an inspection the oil pump was in perfect working order. From this point and following the oil passages up the crank all junctions were a possible leak could occur were inspected. After a time is was discovered that there was a single seal used at the end of the crankshaft and the case allows for oil transfer while the crankshaft is in motion. This seal is a u-cup seal and was installed backwards. Being installed backwards the seal would not properly seal the crankshaft and not allow oil pressure to build within the system. As such under high load in the upper RPM band the crankshaft would receive less than adequate oil causing excessive heat to develop in the component at the needle bearing. The heat would eventually cause the component to stick or seize which in turn would ruin the cylinder and piston similar to a soft seize. The crankshaft for obvious reason would then have to be replaced cost significant time and money for parts.

The crankshaft seal was checked in all three engines, the custom, dynamometer and research engine which had no use up to that point and was recently bought online from a rebuild house. In all cases the crankshaft seal was installed backwards and were all corrected. Over several rebuilds the new crankshafts have yet to develop heat affected zones as shown in Figure 19. The dynamometer engine has seen significantly longer time in testing with no signs of excessive heat buildup as previously documented.

3.7 Clutching System

The engine was originally outfitted with the Rekluse Z-start pro clutch as discussed in chapter 2 section 6. The Z-start was intended as a way to allow the driver to launch the vehicle in any gear as well as simply the controls necessary to operate the vehicle. However, in [4] it is noted in the conclusions section that there existed an issue with the clutch disengaging until fully warmed up. It was determined that the clutch actually never disengaged and that clutch was in actuality slipping. During a rebuild of the engine it was noted that the throw-out spring and bearing were not actually being pressed into the main pressure plate shown in Figure 6. This occurred because the Z-start relies upon the clutch override found in the stock system to compress the clutch spring with the throw-out bearing which had not been designed to be used with the case due to the initial pre-transmission configurations custom counter shaft. To solve this issue a larger spacer was made that replaced the original spacer from Rekluse. This larger spacer was placed between the throw-out bearing and the main clutch spring as shown in Figure 20. This spacer forced the throw-out bearing to rest against the counter shaft and the compress the spring into the pressure plate. The addition of this spacer allowed the clutch to function properly such that at a low enough RPM the clutch would disengage and allow the driver to come to as stop or start in any gear as originally intended.

Unfortunately early in 2015 development year during a round of suspension and endurance testing the Rekluse Z-start pro clutch was severely damaged due to a lack of oil flow to the clutch plates. This lack of oil caused the fiction pads to degrade and become ruined which ruined the oil pump as well. To remedy the issue a new clutch was found by Rekluse that operates very similar to the Z-start pro clutch using centrifugal force to cause enough friction to allow torque to be transmitted to the powertrain. The new Rekluse Core EXP clutch using weights and small springs to apply pressure to the clutch after a certain RPM is exceeded. The new clutch also does not require the need for any custom parts given its adjustment method uses a new style of spacer. The EXP has been set the engage just above the idle point and since it was installed into the engine although at the time it has yet to be removed and inspected. The clutch has given no inkling of failure sense it was installed but at the next rebuild of the race engine the inspection should be carried out.

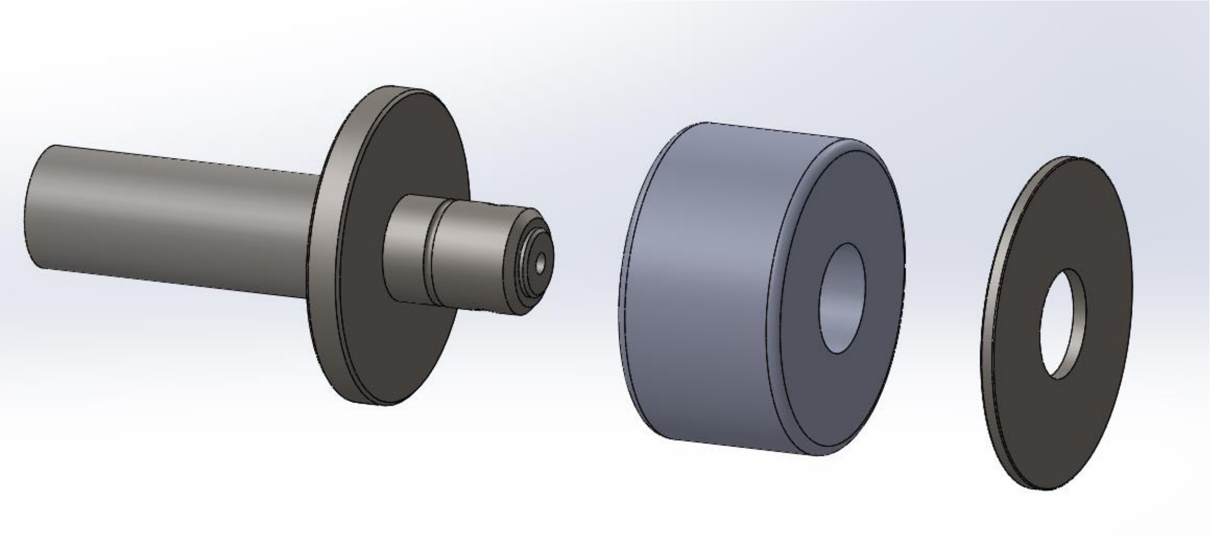

Figure 20: New Throw-out Bearing Spacer

Chapter 4: Engine Testing Setup

To properly calibrate the engine parameters an eddy current dynamometer is used along with a stock YZ 250F engine. An engine stand is used to hold the engine relative to the dynamometer through a coupling system. Both the engine stand and coupling system have seen significant modification to improve the reliability and range of the possible calibration points of the engine. The differences between the test engine and race engine are also examined and noted with a rationale behind the differences. The differences between the engines are considered small enough that simple changes in the calibration will suffice to allow for the base calibration to continue. Also discussed in the chapter is the setup parameters used for the dynamometer engine on the dynamometer software and the Motec ECU.

4.1 Equipment Stand and Set Up

The set up for a dynamometer engine can be critical to the successful calibration and testing of an engine. The previous set up for the engine development of the FHSAE vehicle was subsequently improved in order to gain a consistent and stable calibration as well as the testing needed for future development and model validation. The intent of the redesign was to reposition the engine relative to the dynamometer such that the rotational inertia is minimized. By minimizing the rotational inertia the operational area of the engine is improved, by allowing the engine to operate at lower RPM points. The lower RPM limit that can be achieved allows for a more complete static calibration closer to the idle point which is most difficult portion of the range to calibrate for a small single cylinder engine. The original setup used for engine testing was built and used in [2], were the original EFI conversion for the engine was pioneered.

The previous system used a modified love-joy coupling to connect the output shaft to the jack shaft used in the stand design. The love-joy connection was modified by using dowel pins pressed around the perimeter of one side of the coupling shown in Figure 21. The dowel pins were placed precisely such that the output sprocket could interface with the love-joy and allow torque to be transmitted. The love-joy coupling then connected to the jack shaft which was mounted to the stand using pillow block bearings. The opposing side of the jackshaft had a large flywheel used to bolt the main drive shaft used on the dynamometer. The full assembly is shown in Figure 22.

Figure 21: Modified Love-Joy Coupling [2]

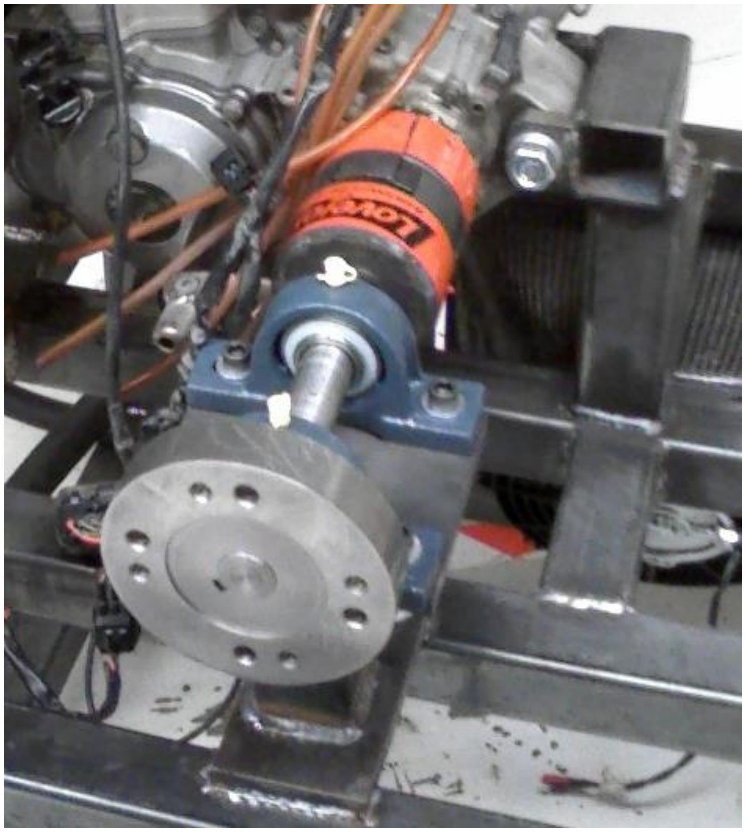

Figure 22: Full Coupling Assembly [2]

From Figure 22 the full assembly is shown with the full love-joy coupling on the stand along with the engine. Not shown is the main dive shaft used to connect the coupling system and the dynamometer itself. The new design has done away with the love-joy connection, the jack shaft and pillow block bearings and down sized the drive shaft all to reduce the rotational inertia of the system.

The new design of the coupling system use one unique part and removes the need for the love-joy, jack shaft and allows for a smaller drive shaft. The unique part was designed to be as small as possible and allow for a secure connection of the drive shaft to the engine and to operate safely in the confined space next to the engine. To save manufacturing time and cost a simple round of steel was made with the rough outline of the output sprocket milled into the back side. The output sprocket itself was modified by sanding down the teeth of the sprocket to fit into the machined pocket. With the sprocket in hand the pocket that the sprocket need to be fit to was machined out little by little until a desirable press fit was possible. The sprocket was then welded to the steel round to ensure that it would not accidentally work free from the press fit. The part was then installed on a stock output shaft and turned down to ensure that the adapter plate spins concentrically. Any eccentricity of the adapter plate may cause unnecessary vibrations in the system which could damage the system. The final product is shown in Figure 23 were the sprocket can be seen pressed into the round with the small weld around the perimeter.

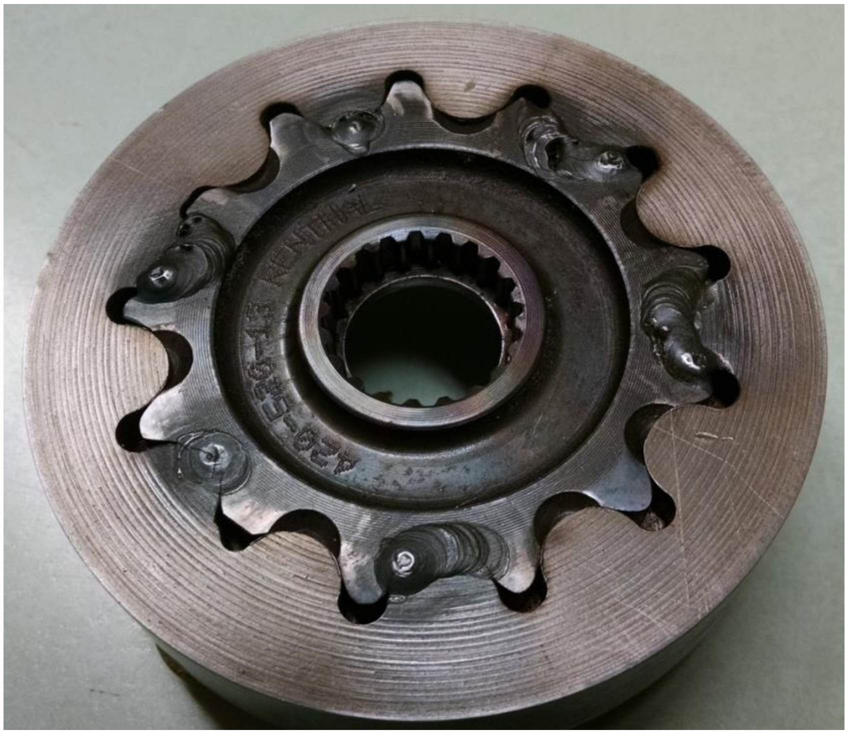

Figure 23: New Coupling System

By using the stock output sprocket in the design of the coupling there is no need to have internal splines cut into the part that would mesh with the stock output shaft. To ensure that the sprocket and metal round never separate a small flange was left in the center of the round. The flange was thin enough to allow the stock nut to be used with lock tight to hold the sprocket and metal round tightly together and on the output shaft. The adapter plate also had a 4 bolt pattern placed on the opposing side from the sprocket to attach the drive shaft. The drive shaft was downsized from the original which is used primarily for larger engines. The smaller drive shaft was the one typically used for engines capable of 100 horsepower and was previously used for testing done with a Yamaha R6 engine. Figure 24 shows the new coupling and drive shaft installed with the engine.

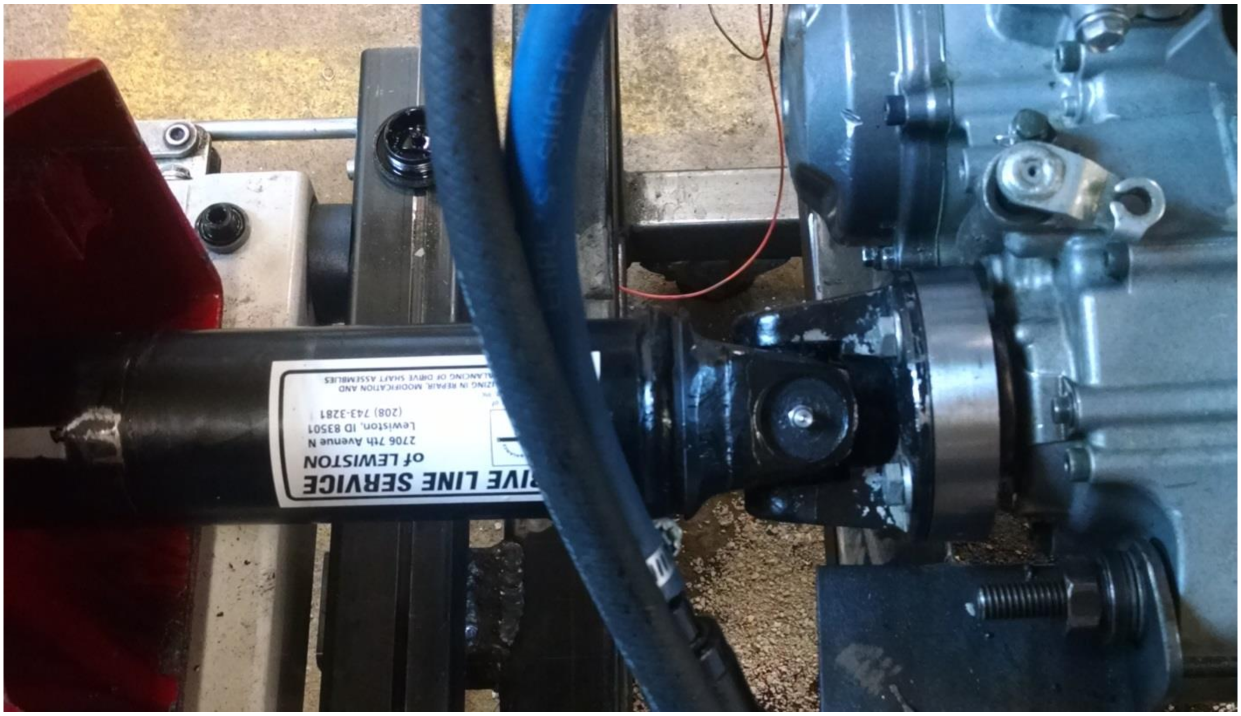

Figure 24: New Coupling and Drive Shaft

The engine placement was also move to accommodate the new coupling system as well as remove the elevation change that the old system exhibited. The new design allows for a nearly straight connection between the engine and the dynamometer. This straightened connection minimizes the amount of vibrations that can occur during operation of the engine as well as minimize the effects of resonance points should one be encountered during the operation of the engine.

The dynamometer stand also has been modified aside from the placement of the engine. The electronics were moved to a new platform placed above the engine along with the relocation of the oil reservoir. The electronics platform was moved from its previous location shown in Figure 25 on the far right of the figure, for two reasons. The first reason for the relocation being that the original location of the platform was mounted to a single beam that was cantilevered off of the main riser used on the stand. The location of the platform was susceptible to vibrations given off by the engine itself which would fatigue the electrical connections causing intermittent signals to the ECU. By placing the platform to its current location shown in Figure 26 the vibrations from the engine were minimized. The second reason is that by placing the platform up high it was away from a fluids the may be thrown off of the engine while placing the electronics in a safe location where the wiring to the engine could be easily managed while allowing for strain relief.

Figure 25: Old Platform and Engine Stand Configuration [4]

Figure 26: New Platform Location

4.2 Engine Setup

To improve the quality of calibration and testing it is important to set the engine up as close as possible to the custom engine used in the vehicle. To improve the calibration, the testing was done on a 2005 Yamaha YZ250F engine because the YZ is where the top end of the custom engine comes from. The YZ250F engine has the same bore and stroke and compression ratio as the custom engine meaning that the transition from one engine to the next should be extremely minimal. Ideally, the engines should be identical however give that the dynamometer engines experienced sever failures during testing the development of the calibration was slowed. As not to affect the development of the overall vehicle new component designs such as intakes and throttle bodies were fitted to the vehicles custom engine without proper calibration. The differences should be small enough not to affect the overall performance of the vehicle adversely although it should be noted that the engine needs to be calibrated with these changes to improve or make full use of the hardware on the custom engine.

4.2.1 Lubrication System

The dynamometer engine is set up with a dry sump lubrication system as shown in Figure 12 in chapter 2. The dynamometer uses this system for two specific reasons. The first being that the dry sump system allows for a larger oil capacity which means that the engine can run for longer periods of time on oil before requiring an oil change. The custom engine is limited to about 3 hours of run time before its oil should be changed the dynamometer engine is able to go up to 6 hours. The system is capable of 6 hours of operation on the same oil however it is a good practice particularly, if the engine is being testing under high loads, to change the oil every 4 hours to minimize the possibility of running burned up oil through the engine.

The second reason for the dry sump system is that there is no need to modify stock components to repair the engine when failures occur. When using the wet sump system if the oil pump should be ruined by some chance a new separation plate has to be prepared from the stock component as shown in Figure 17. Aside from minimizing the number of custom components, not using a modified separation plate for the wet sump is safer and cost effective. The issue with using the separation plate is that if a raised edge or the failure to notice a scratch on the separation plate can result in the failure of the oil pump. The oil pump may not be the only component affected either as discussed in chapter 2 the oil pump, particularly on the dynamometer engine has been the root cause of several thousand dollars of failures.

4.2.2 Throttle Body

Aside from the different lubrication system all the calibration and testing performed in this work used the 40 mm 2008 WR250X throttle body shown in Figure 3. The custom engine uses a Bosch 32 mm Electronic throttle body. The electronic throttle body allows the control system to operate the throttle of the engine and the high voltage system simultaneously when the vehicle is operated in hybrid mode. The WR250X throttle body was used for the calibration of the engine because it simplified the integration of the engine with the dynamometer. To integrate the engine and the dynamometer, throttle position is controlled using a servo. Currently the dynamometer doesn’t have a set up for an electronic throttle body to allow the Super Flo to control the throttle. It should also be noted that the smaller diameter of the throttle body will change the power output of the engine. However, given that the difference is only 8 mm the effect is considered negligible for this stage in the calibration process.

Aside from some small hardware set ups it is important to note that the dynamometer engines wiring is simpler than the custom engine for two reasons. The first is that dynamometer engine uses a few extra sensors that would make it redundant as the dynamometer collects the information already such as ambient air pressure were others were not applicable such as wheel speed or accelerometers. The lambda or 02 sensors was wired to an Innovate hand held used in the dynamometer cell. The hand held read out allowed the calibration process to be simplified by provided responsive feedback to the operators.

4.3 Dynamometer Engine Specifications

For the dynamometer to function properly and calculate the efficiencies from the measured values the correct information must be entered into the Super Flo program. This is done by entering information into the specifications sheet. For the most basic testing, three variables are required to start gathering the basic information from the dynamometer. The first two are bore and stroke which are used to calculate the displacement along with other efficiencies if enough information is given. The third is the gear ratio of the transmission and the primary gear reduction. This gear ratio is very important because it is used to calculate the crank speed which is used to determine the power produced by the engine.

In order to input the information with in Super Flo, open the specification sheet from the top ribbon. A window will open and allow information about the engine setup to be added to the software. The important information to be included in the specification sheet which is unique to the engine include the bore and stroke of the engine, the number of cylinders, whether the engine is a 4 stroke or 2 and any gear reduction used from the crankshaft to the dynamometer. Table 1 shows the full list of materials contained in the specification sheet along with their values. The majority of the parameters are self-explanatory with a few exceptions. The OvrRat parameter is the gear ratio from the crank shaft to the dynamometer the current value is 4.78. This value is used because the transmission is placed in third gear. Third gear is used to limit the speed of the engine while using near the maximum amount of the torque the facilities at the University of Idaho is capable of.

Table 1 Engine Specifications for Dynamometer

Parameter	**Value**	**Description**
Engine Bore	77 mm	Diameter of the Cylinder
Engine Stroke	53.6 mm	Length of connecting rod
Engine Cylinders	1	Number of cylinders
Engine Cycles	4	4 or 2 stroke
OvrRat	4.780	Gear Ratio from crank to dynamometer
Stoich	14.7	Stoichmetric value for chemical reaction of fuel
Puls/Rev	60	Number of pulses from dynamometers trigger wheel for speed sensing
Fuel SG	0.75 lb/gal	Specific Density for Fuel
Inertia	0.027	Inertia of dynamometer
6 inch	1	Turns on six inch intake turbine
Recpcl	60	
Cycle	30 sec	
Lower	1100 rpm	Lower value used for sweeps
Upper	5050 rpm	Upper value used for sweeps
Step TM	1 sec	Time step used during testing
StpSize	50 rpm	Step size over one time step during sweeps
Return	3000 rpm	Rpm value dynamometer will return to after sweep

Chapter 5: ECU Setup and Calibration

The calibration of the engine is one of the more difficult and a time consuming task to accomplish. To properly calibrate an engine it is imperative that one understands the ECU and calibration process before an attempt is made. Knowing what calibration method is to be used and which maps control what function and how they affect the engine is critical to calibration. Also it is important to know if the chosen ECU can handle the calibration method. Attempting to calibrate an engine before taking the time to fully understand the process or equipment can exponentially increase the time and cost of development.

5.1 Motec ECU

To achieve a decent calibration understanding the Motec ECU used with the engine is important. Without fully understanding what is happening when parameters are change with in the controller can be extremely detrimental to the performance, reliability and stability of the engine. Motec uses a map based system driven by at a basic level two main maps. These maps use throttle position and rpm to create an array of operating points where the operational parameters can be adjusted to affect the performance of the engine. The first main map used is the fuel map which dictates the amount of fuel injected into the port during operation. The second map dictates the time at which the spark plugs fires and is affected by the degrees before top dead center. With these two maps a basic running engine can be achieved with minor addition of compensations for engine temperature, exhaust temperature and other parameters to ensure the reliability and survivability of the engine. The fuel map visual representation is shown in Figure 27.

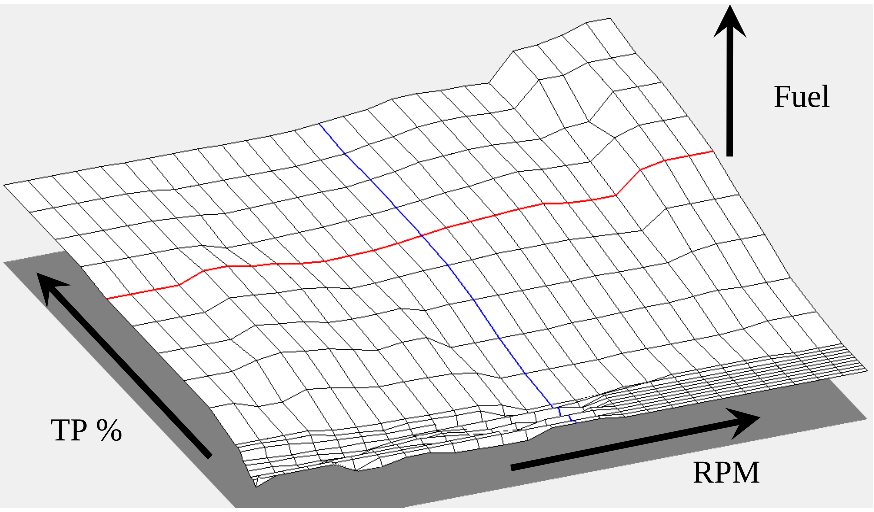

Figure 27: Fuel Map

5.2 General Setup

The general setup of the engine is where the basic information is given to the controllers which are determined form physical features found on the engine. The general setup is one of the first meshing points where the hardware meets the software. Under the adjust drop down menu you will find general setup. In general setup several options are shown such as the main setup, miscellaneous setup, fuel, ignition, rpm limit, firing order and others. The main setup is where the basic features of the engine are determined in the software such as the number of cylinders and how the operator would like certain values calculated such as the efficiency calculation which is used to compensation fueling values through a second set of sensors. The nominal values used in the controller are shown in Table 2.

Table 2 General Setup Parameters

Parameter	Value	Description
Number of Cylinders	1	Total Number of Cylinders
Efficiency Calc Method	4	Uses M.A.P. sensor to calculate efficiency point
Load Calc Method	1	Uses Throttle position for Ignition Map
Efficiency 2 Calc Method	0	Not in Use
Fuel 2nd Load Table Mode	1	Multiples main table by a percentage
Efficiency Z (4D)	0	Not in Use
Load Z (4D)	0	Not in Use
DBW Z (4D)	0	Not in Use

Under general setup is also miscellaneous setup which holds secondary information about the physical characteristics of the engine. This menu is where some of the basic information is given to the controller that dictates some of the startup and run-ability of the engine. The three really important features held in this menu are the acceleration source, the acceleration threshold and the run RPM. The acceleration source is the sensor that is used to determine the fuel and ignition compensations for accelerations such as throttle position or the manifold air pressure sensor. The acceleration threshold is used as a rate based limit which must be exceeded to engage the acceleration compensations for fuel and ignition. The run RPM value is the value which the ECU operates at while the engine is being turned over. The ECU has two different modes one where the engine is being turned over or started where the

ECU reads the parameters from the main maps called cranking and run where the sensors dictate the operational points which happens after the engine actually starts up. The parameters and values used in the miscellaneous setup are shown in Table 3 along with a small description.

Table 3 Miscellaneous Setup Parameters

Parameter	Value	Description
Acceleration Source	1	Uses the Throttle Position as a Source Channel
Acceleration Threshold	50	Sets Rate of Change Necessary for Compensations to Work
Run RPM	1500	Sets the Cranking RPM before
Full Throttle Timer TP	99	Specifies % of Throttle Position to start Timer
Full Throttle Reset Delay	1	Time Delay to Rest Full Throttle Timer below TP
Closed Throttle Timer TP	2	Specifies % of TP below which Closed Throttle Timer starts
Closed Throttle Reset Delay	0.5	Time below closed throttle timer TP will become active

5.2.1 Fuel Setup

Under the general setup there is also the fuel setup menu which dictates the necessary values for the fueling system used on the engine. In this menu the IJPU value is set and scales the percentage value used in the main fuel map. The fuel setup menu also dictates injector current, peak hold ratio and injection timing position. If new injectors are used on the engine for a modified intake this menu is where the values corresponding to the physical characteristics of the injector are integrated into the software.

One of the more important features this menu holds is the injection timing position which dictates when the injection of fuel either begins or stops. Currently, the injection timing position is set to the end of injection which means the value read from the injection timing map is the end of injections. For example if the map reads 270 degrees BTDC (before top dead center) the injection event will stop at 270 degrees of the crank from the top of the piston stroke. The reason for this is intuitively it is easier to dictate when the injection event ends and let the controller figure out when to start the event and knowing the end of injection provides a higher degree of control over the fuel delivery into the intake. The parameter values for the fuel setup menu are shown in Table 4.

Table 4 Fuel Setup Parameters

Parameter	Value	Description
Injector Scaling (IJPU)	10	Pulse Width (msec) for a 100% in the fuel map
Injector Current	0	Peak current for the injector set for a 12 to 16 ohm injector
Peak Hold Ratio	4.0	Not applicable for 12 to 16 ohm injectors
Injection Timing Position	0	Set to End of Injection
Fuel used Calibration	300	Total Injector flow rate
Fuel Acceleration Mode	0	Set to enrich on accelerations, and enlean on deaccel
Bank Trim Mode	0	Not applicable (breaks cylinders into banks)

5.2.2 Ignition Setup

Aside from fuel there is also the ignition setup for the engine which is the menu where the ignition system is relayed to the controller. In this menu the type of ignition, the number of coils and type of units (degrees or percent based) the ignition system uses for its main table and compensations. The type of ignition determines the ignition events trigger such as a falling or rising edge trigger. This comes from the use of magnetic or Hall Effect sensors which can be viewed as a digital signal. This is used because the signal is crisp and allows for accurate timing of events based upon either the sharp increase in the voltage or decrease. This because the sync signal uses a hall effect sensor which output a digital signal so when the signal rises or falls the ignition event can be triggered. The ignition acceleration mode is also determined in this menu. The ignition acceleration mode allows the operator to affect how the ignition timing is affected during acceleration event and deceleration events such as to advance the timing on accelerations and retard the timing on accelerations or vice versa. The ignition menu also limits the minimum advance which acts as a barrier to stack up from compensations which retard the timing of the engine in order to ensure the safety of the engine. The parameters used for the ignition set are shown in Table 5.

Table 5 Ignition Setup Parameters

Parameters	Value	Description
Ignition Type	1	Determines the trigger type used by the controller
Number of Coils	1	Determines the number of Ignition outputs used
Ignition Delay Time	50	Delay time of the Ignition system (spec from controller)
Ignition Percent/Degrees	1	Ignition values are read in Degrees or Percent
Ignition Acceleration Mode	0	Retard on Acceleration and Advance on Deacel
Ignition Minimum Advance	-50	Determines minimum number of degrees of retard timing
Ignition Current Source	1	Determines current amount needed by ignition system

5.2.3 RPM Limiter Setup

The RPM limiter menu is also found in the general set up and is used to determine the characteristics of the RPM limiter settings. The menu allows the operator to set the rpm limit at which point the controller will cut either the fuel delivery or the ignition system or both to drop the rpm below the limiter. This feature is titled RPM Limit type in the menu and depending upon operator preference and expectations of the engine to achieve a recovery or emissions regulations different methods may be used. The RPM recovery rate can also be determined in the menu which dictates the time based upon a percentage until the rpm limit type is returned to normal operation. The other important feature found in this menu si the RPM Limit Critical range which allows the engine to exceed the rpm limit without cutting the cylinders. This is a useful feature if the engine is capable of exceeding the rpm limit without cause damage to the engine. It should be noted that the RPM limit diag parameter should not be set lower than the critical range parameter unless the operator determines that a warning lower than the upper most limit is needed during testing events. The power recovery rate parameter found in this menu allows the operator to set a rapid recovery of RPM if the application of the engine warrants a sharp return to power however this may lead to premature engine failure if the driver continues to engage the rev limiter. The parameters used to determine the RPM limiter are shown in Table 6.

Table 6 RPM Limit Parameters

Parameter	Value	Description
RPM Limit	13500	Determines the RPM Limitation of the Engine
RPM Limit Ctrl Range	0	Additional RPM above limit where cylinders are cut
RPM Limit Type	5	Determines what limits the RPM
RPM Limit Randomizer	75	Used to randomly cut cylinders at RPM limiter
RPM Limit Diag	500	RPM value above limit to trigger error
Power Recovery Rate	50	Determines the rate of reduction of RPM cut
Low Limit Throttle Position	0	Not applicable
Low Limit Max Cut	0	Not applicable
Enable Rate of Change Factor	0	Not applicable
Enable Integral Factor	0	Not applicable

5.2.4: Sensor Setup

The sensors setup is straight forward task provided some information is known about the sensor and how to correctly input the information into the ECU. To setup a general sensor first go to adjust in the top ribbon and open up sensor setup. From the sensor setup menu open up the input setup which will open a new window that allows for sensors to be setup with a pin input on one of the ECU plugs as well as filtering for the sensor signal. The window allows the setup of several different types of generic sensors including Engine sensors such as throttle position, engine temp and lambda. The widow also allows for the setup and calibration of several other types of sensors from exhaust gas temperature sensors, speed sensor and user defined sensors such as accelerometers.

To set up a sensor first select the input pin that the signal will appear on such as Analog Voltage 1. Then once the input pin is determined the type of calibration can be set, Motec has several built in calibrations for standard automotive sensors such as pressure sensors and throttle position signals. Once the proper sensor calibration is selected a default value is added which is the value that the ECU will use if the signal faults out or is lost. After the calibration and default value is set up a filter can be used to smooth out any noise that may appear on the input signal a value of 1 denotes no filtering of the input signal while 10 is a medium filter and 50 is a heavy filter. Additional sensors not critical to the engine operation require a final step. For example to finish the setup for a speed sensor using a digital input pin go back under Adjust menu from the top ribbon and select the Digital Input Functions and go to the pin that was used in the sensor setup. From here a menu showing Function and Parameters, select Function. In the Function window press F1 and enter the value that corresponds to the type of function in this case a value of 1 is used for Speed Measurement.

Some sensors that are critical to the operation of the engine and are always used with engines such as throttle position, lambda sensors, and reference and sync signals have a special setup. To set these sensors up under Adjust, and Sensor Setup go the sensor desired for example the REF/SYNC Sensor Setup menu and select Setup. The setup menu will appear in the main window which allows for the input of the parameters that dictate the reference signal and the sync signal. The parameters needed for the reference and sync signal is shown in Table 7.

Table 7 Reference and Sync Sensors Parameters

Parameter	Value	Description
Ref/Sync Mode	5	Determines the number of missing teeth on the flywheel per cycle
Crank Ref. Teeth	12	Number of Teeth Including Missing tooth
Tooth Ratio	50	Ratio need to detect missing tooth
CRIP	493	Degrees from Missing Tooth to TDC
REF Sensor Type	2	Type of Sensor used
REF Sensor Edge Polarity	1	Determines when the sensor registers a tooth
REF Sensor Trigger Voltage	0.1	Accounts for Voltage offset on signal
Sync Sensor Type	1	Type of Sensor used
Sync Sensor Edge Polarity	0	Determines when the cam lobe registers
Sync Trigger Voltage	0	Accounts for Voltage offset on signal

The throttle position setup and calibration also uses a unique setup that differs from the generic setup discussed earlier. If a throttle body is being used such as used on the dynamometer engine, the sensor is easily calibrated. This is done by highlighting the throttle position closed parameter in the throttle position menu and pressing enter when the throttle is completely closed. The same is done for the open parameter by opening the throttle completely and pressing enter. If the engine is outfitted with a drive by wire system the method is the same with the exception that you need to calibrate the signal from the driver and the signal from the sensor on the electronic throttle.

5.3 Main Engine Maps

The Motec ECU functions using three basic maps the control the engines performance. These three maps dictate the fuel delivery quantity, the timing of the spark or ignition event and the timing of injection. Using these three primary events the calibration engineer can dictate to the controller how engine responds and operates. These three tables are found throughout all engine controller setups however the parameters used in the tables such as throttle position and rpm can change depending upon the control scheme used. The different control schemes will be discussed in a later section, currently the focus will be on the current setup of the maps. All three maps at a basic level operate using two control axes and the third dictates the quantity of a specific aspect or timing or said aspect.

5.3.1 Fuel Map

The main fuel map is based off of the throttle demand of the driver and the rpm of the engine. The throttle position is shown on the y-axis of the map and is broken down into a percentage of the throttle being open by the driver. The throttle position is used as it is in direct control by the driver and controls the load the engine experiences. The rpm is used on the x-axis of the map and is broken into division of 500 rpm intervals up to 12500 rpm. The fuel map is three dimensional maps where the third dimension is percentage upon the IJPU (injector pulse width unit) which correlates to injection quantity.

The IJPU is a parameter that can be adjusted in the setup of the controller through general setup under the fuel menu. The IJPU value parameter is the max time that the injector is open for the current setting is set to 10 msec. By setting the IJPU to 10 msec the math used to calculate the duty of the IJPU is simplified as the ECU take the IJPU value and multiplies it by the percentage read from the fuel map and determines the time that the injector is open for. If however a greater resolution is needed such as when the fuel values calibrated in the main table fall below 50 to 60 percent a smaller IJPU value can be used which will allow for larger percentages found in the main table.

5.3.2 Spark Map

The main spark map is very much the same as the fuel map with some small exceptions to its functionality and how it determines the operational point. The main spark map functions using RPM or speed of the engine along the x-axis of the engine but uses percent of load along the y-axis. The load on the engine can be determined through several different methods depend upon the application, and calibration method. The typical methods are throttle position, the use of manifold air pressure, and manifold air flow. There are other methods such as a differential pressure between atmospheric and manifold pressure. The type of load calculation is parameter that can be set in the main setup shown in Table 2.

The spark map controls the timing of the spark or ignition event and the third dimension is read in degrees before top dead center. As the map reads out in degrees before top dead center it must be realized that this value is the number of crank degrees before the piston reaches the top of its swept volume. The map can be reconfigured as shown in Table 5 to read out in a percentage rather than degrees however using a percentage based map would not be as intuitive as using degrees which most engines use degrees to time the engine mechanically.

5.3.3 Injection Timing Map

The injection timing map controls when the injector opens and delivers fuel to the intake system or into the cylinder if an engine is set up with direct injection. The injection timing is very similar to the spark map such that the map uses rpm and throttle position. Using the rpm and throttle position an array of operating parameters are created that can be calibrated for improved operation of the engine. The map uses values of degrees before top dead center just like the spark map. With proper injection timing the engine is able to operate at better efficiency while creating more power due the better mixing of fuel and air prior to combustion.

5.4 Engine Calibration

With the greater amount of control needed to operate contemporary engines a higher demand is placed upon the calibration engineer to enable the engine to perform safely and correctly while maintaining the performance goals. The calibration of engines can easily become a time consuming occupation due to the high number of free variables needed to achieve the performance goals of the engine while meeting the emissions regulations. There also the different calibration methods which all have pro and cons concerning the time to complete the calibration along with the ability of the engineer and the reliability concerns.

5.4.1 Calibration Methods

There are three basic calibration methods which are typically used. The choice on which method is used can depend upon the industry, application of the engine, and the needs of the consumer. The first method is the Alpha-N method, where N is the engine speed and the alpha is throttle position which refer to the two axes of the map used by the controller.

The second method of engine calibration is the Speed Density method which uses a manifold air pressure sensor and an air temperature sensor. The manifold air temperature sensor provides the pressure at the intake and the air temperature provides the ambient temperature which is used to determine the volumetric efficiency of the engine using the ideal gas law. Using built in tables of volumetric efficiencies the engine control unit then determines the amount of fuel correlating to the induced air flow into the engine to achieve a given lambda value at a given engine speed. The third method typically used is referred to as MAF calibration which primarily uses a manifold air flow sensor (MAF). This method like the speed density method determines the amount of fuel needed given the amount of air inducted into the cylinder. Unlike the speed density method which calculates the air inducted using ideal gas law and built in tables the MAF system directly measures the air inducted into the engine using a hot wire sensor.

Each method has its own positive and negative attributes that determine effectiveness of the calibration. Depending upon what demands are made on the calibration such as flexibility to engine modifications or high production demanding the ability to easily transfer to another engine a different calibration may be used. Some engineers also use a combination of the main calibration methods in order to supplement the performance of one particular method and achieve a higher performance from the engine.

5.4.2 Current Calibration Method

The current calibration method used on the FHSAE project at the University of Idaho is the Alpha-N method. This method is chosen for a several reasons. The first reason is that the Alpha-N is a very simple method of calibration that is relatively easy for students to start getting a feel for engine calibration. Although the method is time consuming and relies on the ability of the calibration engineer it functions as a great method to teach students how to calibrate, working with the ECU software and dynamometer through repetition. The repetition comes from the need to calibrate all viable points of operation due to the nonlinear nature of using the throttle position as a control axis. The number of repetitions also includes the need to move from one table to the next until the calibration point achieves the desired performance. The Alpha-N method is fairly intuitive in the terms of what the hardware interface of the engine is affecting as moving a butterfly valve.

To calibrate the base maps for the FHSAE engine with Alpha-N a simple model is followed for each point. This model consists of adjusting the three main maps of fuel, spark and injection timing at each point. To start a basic assumption is made about the viable operating points for idling the engine such that it will idle on the dynamometer. These values can be captured from a stock engine of similar bore and stroke or assumed if prior experience with a similar engine is known about the fuel and spark timing. From the base line assumptions a generic flow diagram for the calibration of an individual point is shown in Figure 28. The flow diagram shown in Figure 28 is not to be regarded as a set of rules that should be followed rigorously but more of guild lines due to the complexity of engines it is possible that the exhaust gas temperature or coolant temperature could become unacceptably high during any of the sweeps. Should this case arise the engine will need to be throttle back to idle where ideally the engine operates at a slightly richer state to help cool the exhaust pipe and or engine.

After a base line is established where the engine will idle comfortably on the dynamometer the rest of the fuel and spark map should by adjust to these minimum values while the injection timing is left as a rough approximation. From this point a relatively close point should be selected to begin the calibration process. Slight adjustment may be necessary to the fuel and spark values to get the engine to idle at the selected point. Once the point is achievable a fuel sweep should be conducted, which is done by adding or subtracting from the fueling value until the target lambda is found ideally a lambda of 1. After the lambda target is hit move to the spark map where a spark sweep is conducted while monitoring the exhaust gas temperature and torque output. At this point in the calibration of the point the goal is to produce the maximum torque the engine is capable of while not exceeding the exhaust gas temperature limit of the materials. To provide a safe margin in case of unexpected circumstances the engine is currently limited to approximately 1100 F. Once the peak torque is found and the EGT limit is not exceeded the lambda is checked to ensure the engine is does not run in a lean state such as above a lambda of 1.

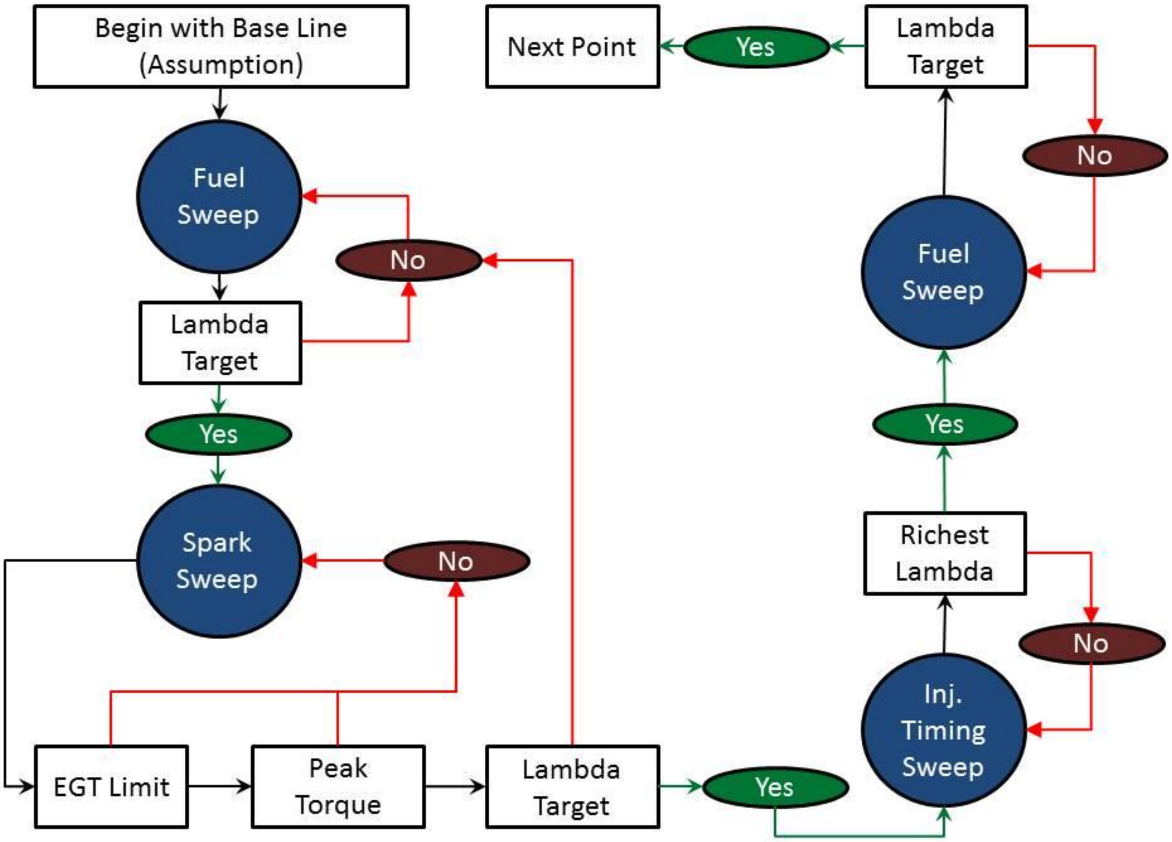

Figure 28: Generic Alpha-N Calibration Flow Diagram

If the lambda is checked for the spark values used the next step would be to ideally move to injection timing. However it is at times acceptable to move to the another operational point close to the first and repeat the calibration process as the injection timing should not cause a large effect on the engine particularly at the low end of the engine. Once the injection timing is to be calibrated the operation is identical to a fuel sweep but with injection timing where the lambda value is monitored. While adjusting the injection timing the lambda value should be moving if only slightly, the goal of this sweep is to find the richest lambda possible at the operational point. The idea behind finding the richest lambda reading is that when the reading is found it means that the injection timing is at a point where the majority of the fuel is inducted into the cylinder. After the injection sweep has found the richest lambda, a final fuel sweep is used to bring the lambda back to the target value to ensure the fuel efficiency of the point.

There are some concerns while calibrating an engine that should be noted throughout the calibration process. When performing a calibration it is best to have a third person to monitor the engine for oil leaks, or any issues that may occur with the engine while in operation. This caution is simply because the during a calibration there are many readings that must be taken into account to ensure the engine operates smoothly and the task of monitoring, lambda, exhaust gas temperature, coolant temperature, torque output and the dynamometers own sensors for coolant temperature can become taxing even for the mandatory two operators at all times. This leads to the next concern is that while operating the engine on the dynamometer always keep monitoring the EGT, coolant temperature, and pay close attention to the fuel temperature. The EGT and coolant temperatures should be monitored for obvious reasons and it will save considerable amounts of cost due to broken parts which has been a considerable cost for the past two years. The fuel temperature is monitored because if the temperature steadily rises the fuel density will change and when the engine is operated at a lower temperature the calibration will not be as intended and will cause issues.

5.4.3 Common Compensations

Aside from the basic calibration that is conducted on the dynamometer there are several different kinds of compensations that exist to modify fueling and spark values used from the main tables such as EGT and coolant temperature. These compensations also allow for variance in environmental conditions that the engine may be exposed to such as altitude changes, air temperatures. There are a few compensations currently in use with the calibration that ensure the reliability and longevity of the engine during repeated operation.

The main fuel compensations exist for coolant temperature and exhaust gas temperature. The coolant temperature compensations exist to add to the existing value read from the given operational points used in the main maps. For example if the coolant temperature should exceed 90 C the compensation table for coolant temperature will add a value of 55 units to the value read from the main fuel table. These values are referred to as a percent trim. This compensation not only kicks in at high temperatures which are built in to save the engine from overheating but the low temperature are also elevated to provide extra fuel for cold conditions and cold starts. This same kind of compensation exist for EGT which can cause catastrophic failures should it reach beyond the melting point of the materials used for the head of the engine. This compensation however only kicks in at high temperatures around 800 C. There is also the need at times to compensate the spark timing of the engine, such as during accelerations where the spark timing can be advanced to maintain the fuel efficiency of the engine while provide a sharper engine response.

Specific events also require complex compensations to meet the requirements of the engine. One in particular is the cold starting compensations used on the engine. The cold start compensations consist of two or more compensations which adjust fuel and spark at the same time to ensure reliably starting and if the engine is intended for certain industries emissions are a concern as well. Currently the cold start compensation consist of a cranking compensation and post start compensation which are functions of engine temperature which are a percentage trim of the coolant temperature.

5.4.4 Engine Model and Calibration Comparisons

Over the past several years several models have been created to simulate not only the engine but the vehicle as a whole as well. As this work is particular to the engine on the engine model created in MatLab will be examined. The model is a 2 Zone Heat Release Model created to simulate not only power and torque output of an engine but also the emissions [8]. As the calibration is ongoing the comparison between the experimental BSFC data and the experimental data collected during calibration process is all that will be considered. The theoretical BSFC map created by the model is shown in Figure 29 while the experimental results are shown in Figure 30.

From the data collected the model shows conservative estimates on BSFC values as the experimental data shows a slightly lower peak torque across the upper RPM band. The lower RPM band however is in good agreement with the torque production but the experimental data shows a much higher BSFC value. The location of the minimum BSFC value also shows good agreement between the model and experimental data. The differences between the experimental and theoretical could come from both the need to finish calibrate the model as well as the finishing the calibration for the engine using the Alpha-N method.

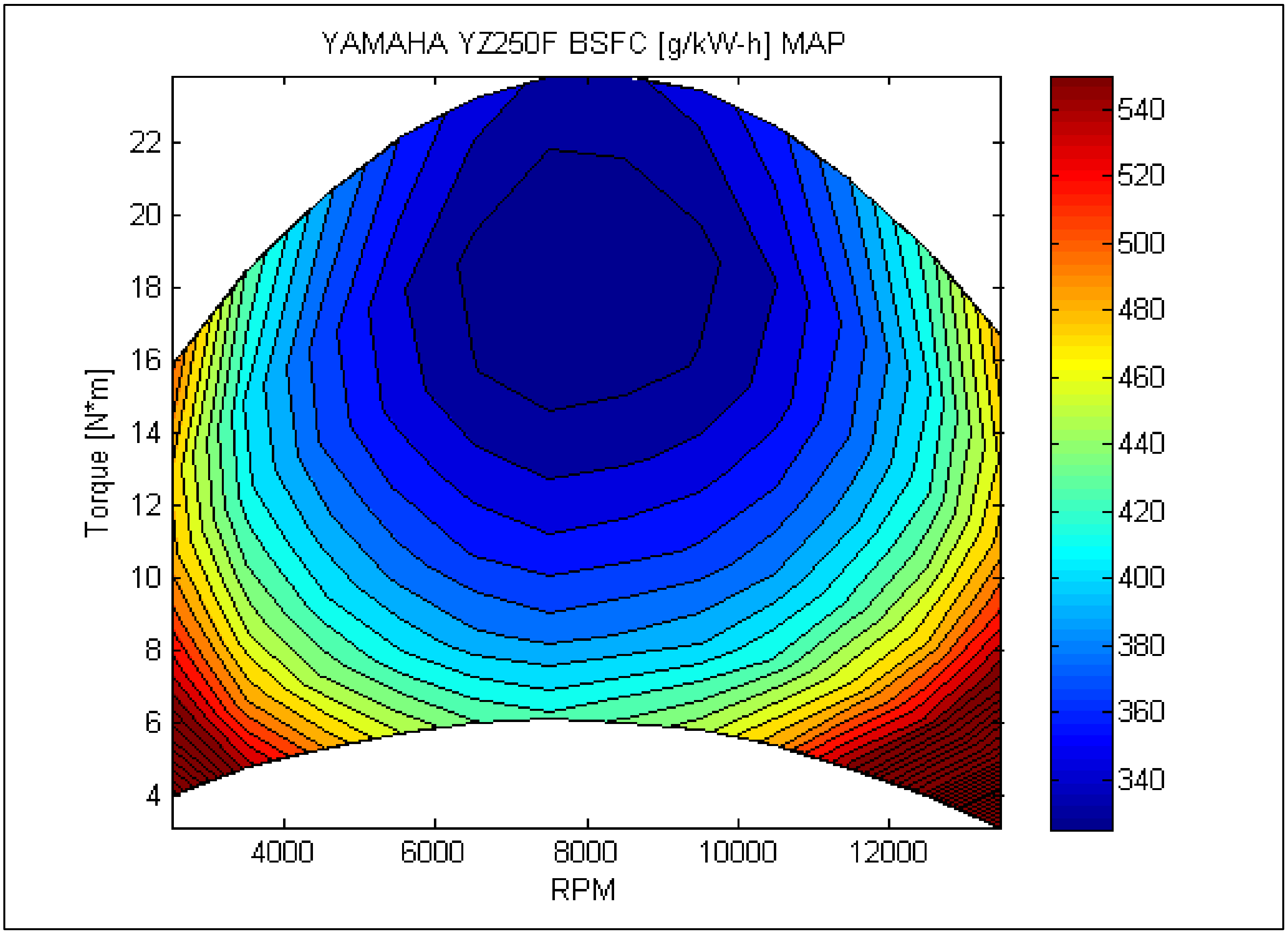

Figure 29: Theoretical BSFC Map [8]

Now comparing the old calibration data collected shown in Figure 31 originally used to calibrate the model in [8] the theoretical BSFC plot shown in Figure 29. This improvement is demonstrated by the widened out BSFC pocket containing the minimum BSFC which was originally centered around 7000 rpm to the new location between 7500 and 8000 rpm in Figure 30. The width of the minimum BSFC pocket has also been expanded from 6000 to 7500 rpm to approximately 6500 to 9000 rpm. From Figure 30 and 31 the quality of the calibration has shown significant improvement. This is demonstrated by not only the better BSFC values but the smooth nature of the BSFC values. In Figure 31 the BSFC plot shows discontinuities and seemingly random pockets of higher BSFC values which suggest a rough calibration. A small example of these random pockets of higher BSFC values is shown in Figure 30 by the sharp increase in the upper left of the plot at 6000 rpm and 16 N-m. The majority of the plot however indicates that the current calibration is significantly better than the previously tested calibration as it is mostly free of discontinuities.

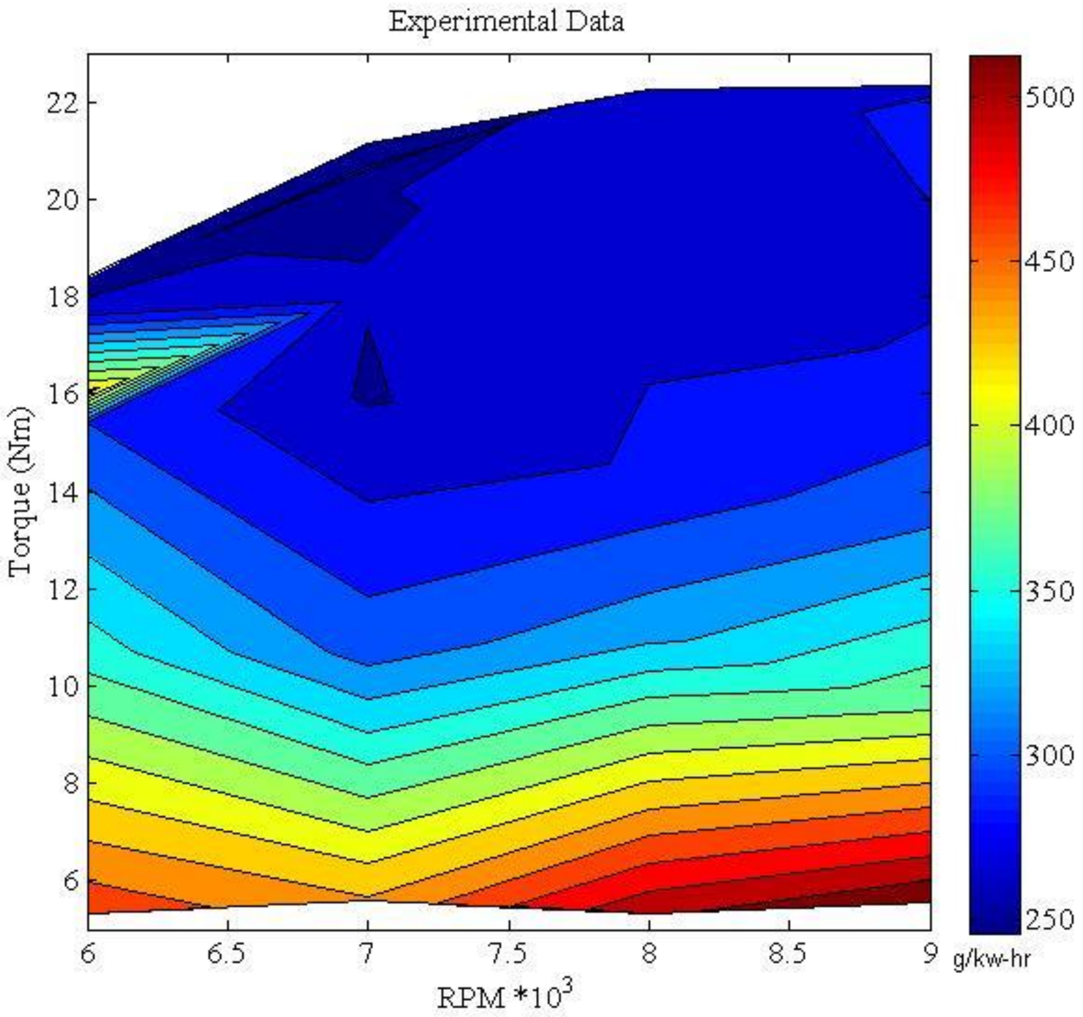

Figure 30: Experimental Engine Data

In order to add assurances to the data collected an error analysis was conducted on the calculations used to determine the BSFC values. This was done using the generic power equation using torque and speed. The error of the torque measurement taken by the dynamometer is considered the largest contributor to uncertainty in the calculated BSFC map shown in Figure 30. The uncertainty of the collected data is shown in Figure 32.

From Figure 32 the uncertainty of the BSFC values is relatively low averaging around 5 g/kw-hr in the minimum BSFC pocket. The uncertainty grows as the engine is subjected to lower loads where smaller torque is produced. This phenomenon occurs because the error of the dynamometer become more significant which is understandable as an error of 1.5 ft-lbs effects a torque reading of 19 ft-lbs greater then a reading of 65 ft-lbs. Fortunately the area that is of the most interest, the BSFC pocket shows a small amount of uncertainty of +/-5 g/kw-hr. This uncertainty analysis shows that in the upper rpm and load that the BSFC values are reliable and could be used in current engine and vehicle models developed for the FHSAE project.

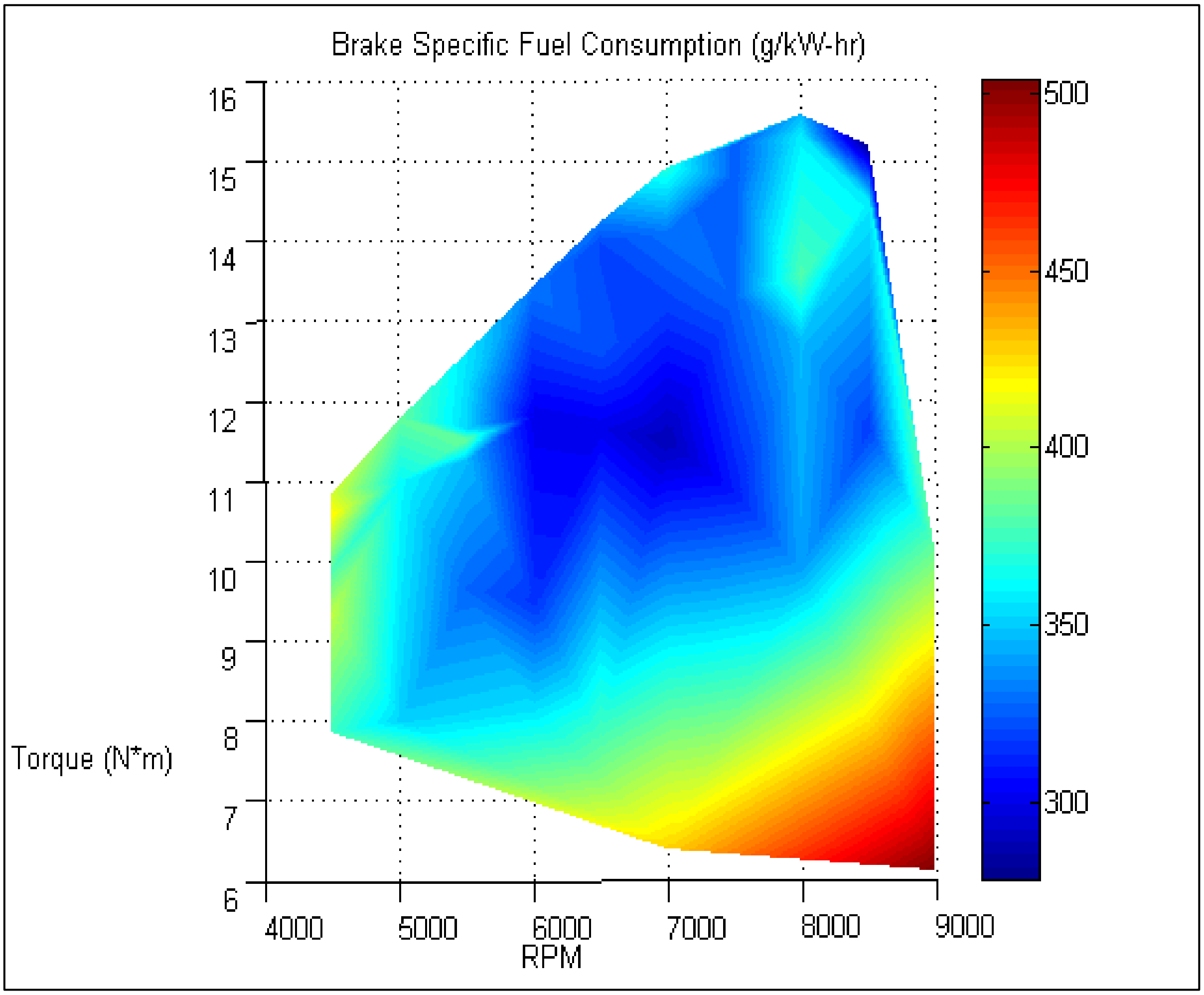

Figure 31: Old Calibration BSFC Map [8]

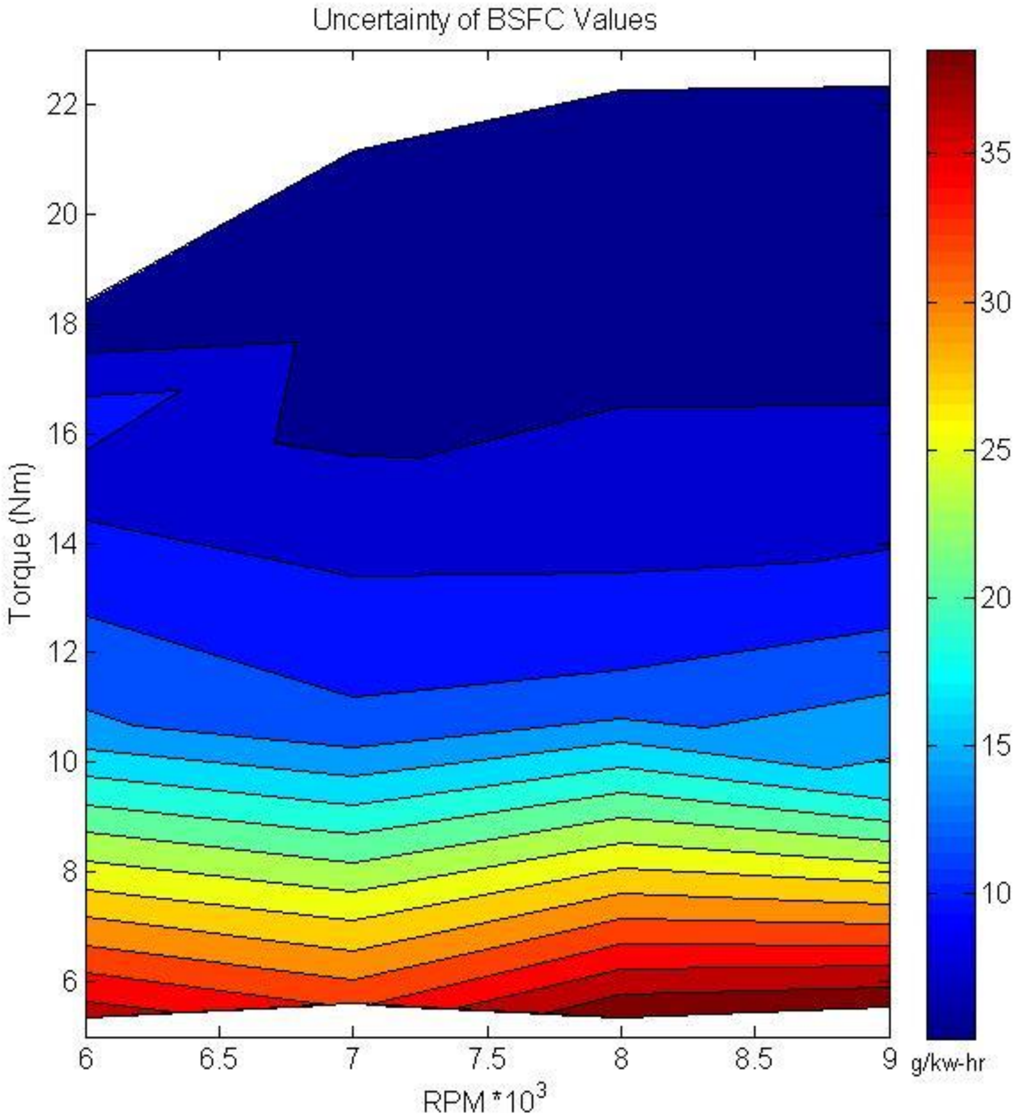

Figure 32: Uncertainty of BSFC Values

Chapter 6: Conclusions and Recommendations

In terms of a performance, reliability and longevity of the custom engine for the FHSAE project at the University of Idaho, the engine has never shown more promise. The performance of the engine has shown significant performance gains over the past two years in several areas. These areas of improvement include the power production, power delivery and the calibration especially has exceeded all expectations for the fuel economy of the engine.

6.1 Conclusions

The calibration of an engine can dictate the characteristics and reliability of an engine and the quality of a calibration will directly correlate to the reliability of the engine. To extract the most from engines, whether it is performance, fuel economy, or reliability or a measure of multiple characteristics compromises must be made. However the calibration currently in use is not to the point of making compromises. The reliability of the engine still needs significant amount of work and the full potential power that the engine is capable of is still yet to be determined. This means that the calibration needs to be fully finished as the current calibration has only be carried out for a significant portion of the operational range however more is needed.

Currently the calibration conducted so far covers 6000 rpm to 10500 rpm from twenty percent throttle to a hundred percent throttle for both fuel and spark. The calibration also covers a decent region of the lower rpm range down to 3000 rpm but only above fifty percent throttle opening. The calibration also includes an engine temperature and exhaust gas temperature compensations that adjust the main maps for improve starting and create barriers that protect the engine from overheating.

The 2 Zone Heat Release model with some work could help predict the emissions of the engine as discussed in [8]. Aside from the emissions from the engine it may be possible to try optimizing the calibration through advanced techniques. This optimization could be done using the model as a black box with in the optimization and validated though data collected using a dynamometer.

Another avenue of development is the continued use of the GT Suite model developed in [7]. By completing the calibration currently in development and using data collected from testing for BSFC and emissions a prediction could be made on the emissions over a drive cycle. This drive cycle could then be completed using the vehicle and calibrate and validate the GT Suite model. By going down this avenue a control system may be designed using the model with a shorting development timeframe which would improve the hybrid fuel economy as shown in [7].

6.2 Recommendations

For future modifications to the engine there are several areas that need to be addressed that would benefit the several characteristics of the engine and thereby the vehicle. These modifications would include work with the hot and cold side of the engine along with the validation of the current calibration. The future modifications should also include repairs to a few of the existing components and systems.

To improve the sound and performance of the engine a new muffler design should be developed. This muffler is a two box muffler designed by Gordon P. Blair which uses an absorption style muffler with a diffusing chamber. This muffler if properly designed should reduce the overall sound production of the exhaust system to comply with the rules dictated by the FHSAE competition. The muffler design also provides higher torque output almost equivalent to using a straight pipe on the exact engine [10].

To improve the reliability of the electric system which has shown issues with the low voltage system a new flywheel should be manufactured. Currently due to the damage done by the large amount of axial play that existed in the crankshaft the engine is incapable of recharging the low voltage battery adequately. To remedy this issue a new flywheel should be manufactured that can provide enough current to charge the low voltage battery. By manufacturing this new flywheel the vehicle should not require charging after every operational event.

Future modification to the calibration should also move towards a different calibration method. To decrease the amount of development time required to achieve a quality calibration and allow the engine to handle slight modifications without the need to recalibrate the entire engine. This new calibration should use the speed density method or some combination of a MAF and speed density. Moving forward with a new method of calibration will also bring the calibration done on the FHSAE project more in line with the current methods used in the automotive industry.

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[2] *R. T. Storms, "Fuel Injection and Intake/Exhaust Sizing of a YZ250F Engine for a Formula Hybrid Vehcile," MS Thesis, University of Idaho, 2013*

[3] *B. Brashford, "Electric Propulsion Subsystem for a Parallel-Drive Formula Hybrid Vehicle," MS Thesis, University of Idaho, 2013.*

[4] *D. Rinker, "Use of a TK Solver Performance Model in the Design and Testing of a Formula Hybrid Racecar," MS Thesis, University of Idaho, 2013.*

[5] *R.V. Basshuysen, and F. Schafer, "Internal Combustion Engine Handbook," SAE International, Warrendale, PA, 2004.*

[6] *P. V. Valkenburgh, "Race Car Engineering & Mathematics," Paul Van Valkenburgh, 2nd Edition, 1992.*

[7] *M. Asfoor, "Development & Optimization of a Rule Based Energy Management Strategy for Fuel Economy Improvement in Hybrid Electric Vehicles," PhD Dissertation, University of Idaho, 2014.*

[8] *J. Cuddihy, "A User Friendly, Two Zone Heat Release Model for Predicting Spark-Ignition Engine Performance and Emissions," MS Thesis, University of Idaho, 2014*

[9] *"Permanent Magnet Stability," Magma Magnetic Technologies.*

[10] *G. P. Blair, "Design and Simulation of Four-Stroke Engines," SAE International, Warrendale, PA, 1999.*

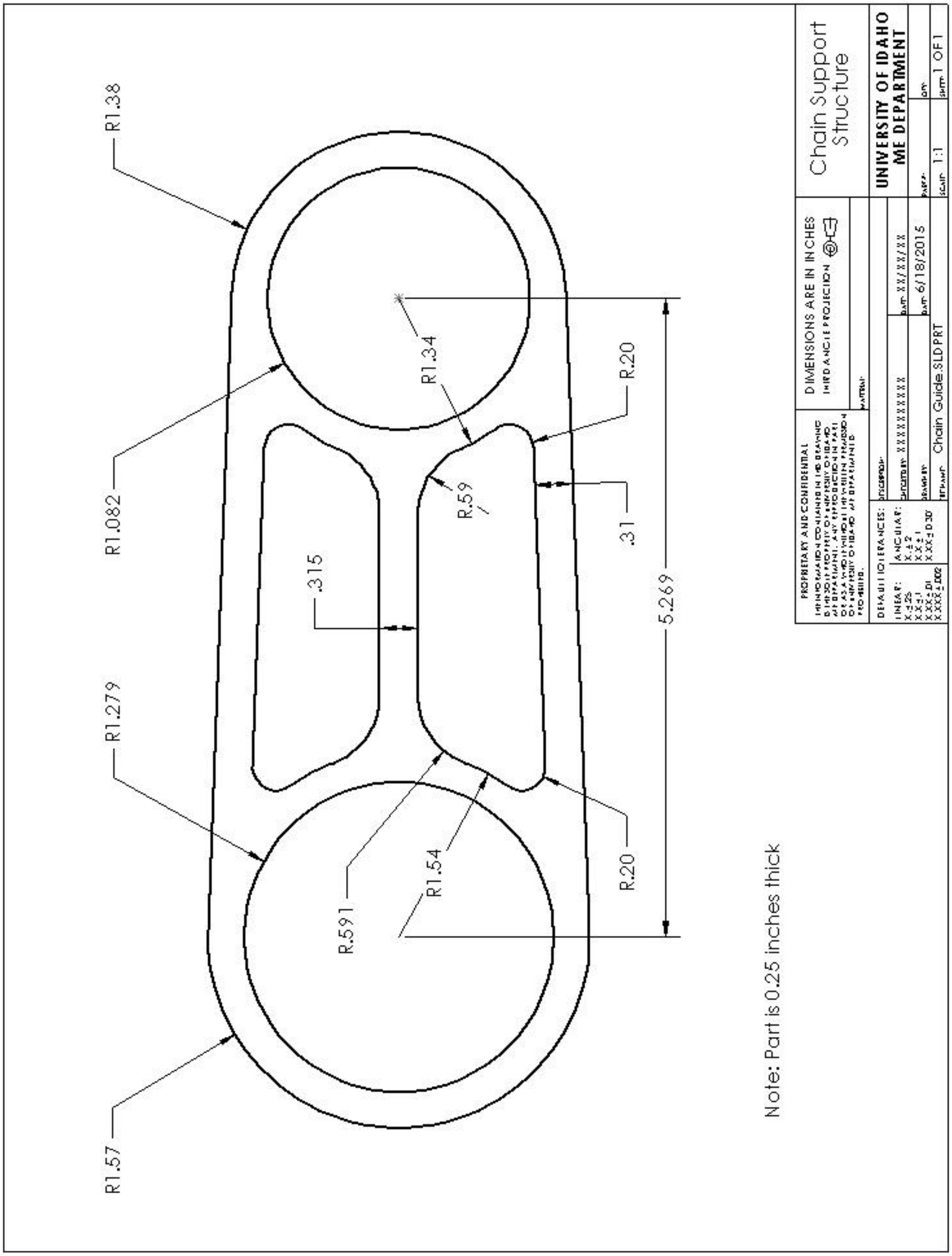
Appendix A: Tabulated Engine Parts

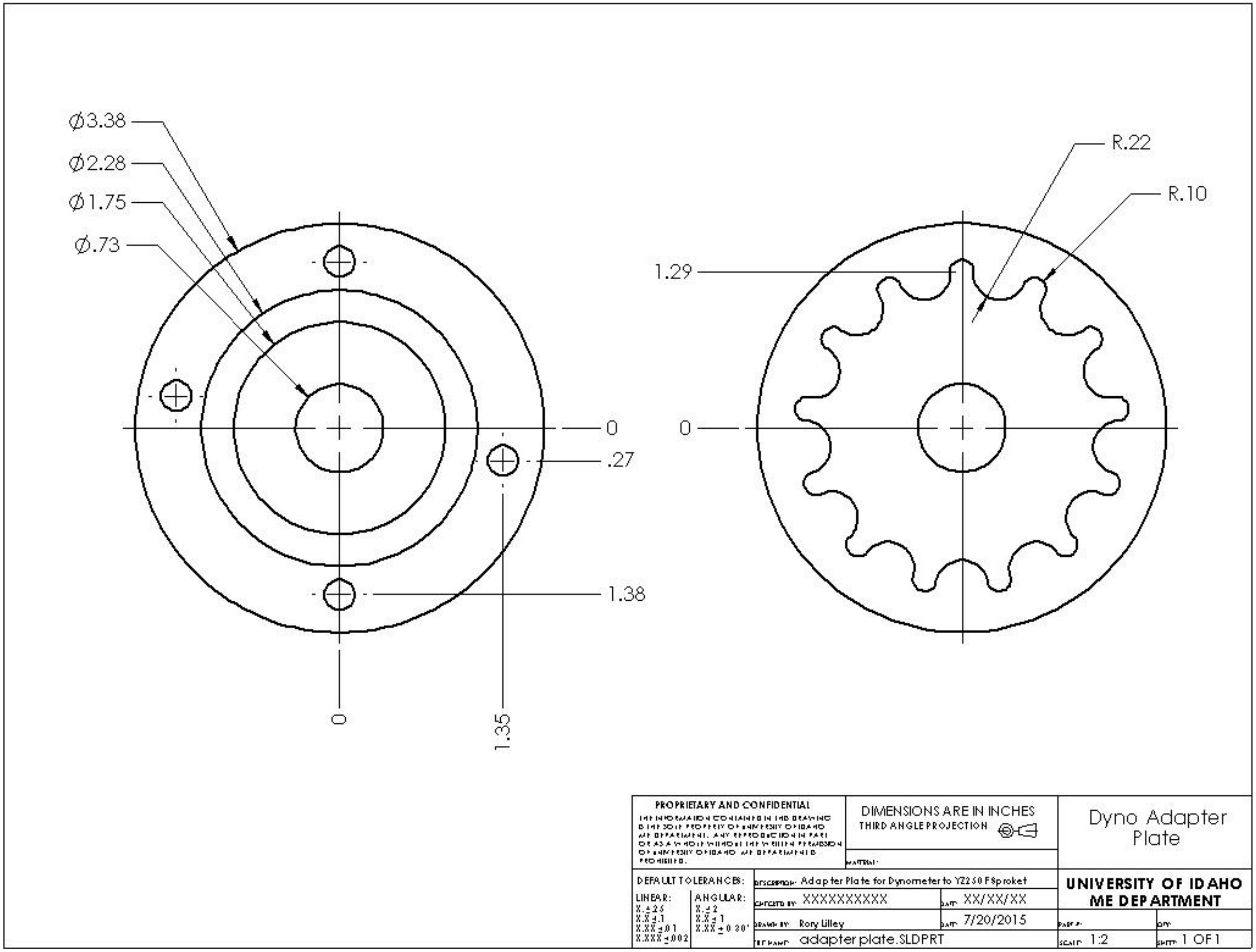
The table below shows the different engine components along with where specific components came from. The table also includes short descriptions of any modification that was performed on the components. This table shows all the major components that the engine uses but does not mention every component used on the engine when an entire system comes as a set of components such as the valve train.

Component	Modifications	Engine Donor
Valve Cover	Custom Build	N/A
Coil	YZ250F- 2005	None
Spark Plug	YZ250F- 2005	None
Head	YZ250F- 2005	None
Cams	YZ250F- 2005	None
Timing Chain/ Tensioners	YZ250F- 2005	Modified to not interfere with flywheel nearest crankshaft
Valve Train	YZ250F- 2005	None
Cylinder	YZ250F- 2005	None
Piston/rings	YZ250F- 2005	None
Connecting Rod	WR250F-2007	None
Crankshaft	WR250F-2007	None
Fly Wheel	WR250F-2007	Perimeter Turned down and Custom Trigger Wheel Pressed on
Counter Balance Shaft	YZ250F- 2005	None
Primary Gear Reduction	Custom Build	N/A
Clutch	Rekluse Core EXP	None
Clutching Mechanism	N/A removed	N/A
Oil Pump	YZ250F- 2005	Wet Sump Conversion and removal of primary rotor set
Oil Pump Cover	YZ250F- 2005	None
Water Pump	YZ250F- 2005	None
Water Pump Cover	YZ250F- 2005	Capped Original inlet and welded in new inlet nipple
Counter Shaft	YZ250F- 2005	None
Output Shaft	YZ250F- 2005	None
Gear Set 1	WR250X-2008	None
Gear Set 2	WR250X-2008	None
Gear Set 3	WR250X-2008	None
Gear Set 4	WR250X-2008	None

Gear Set 5	WR250X-2008	None
Shift Forks	YZ250F- 2005	None
Shift Drum	YZ250F- 2005	None
Kick Start Mechanism	N/A removed	N/A
Throttle Body	New Eagle	Extended Intake Runner and Injector Placement
Starter	WR250X-2008	None
Starter Gear Reduction	WR250X-2008	None
Starter Clutch	WR250F-2007	Used WR250F gear with internal clutch from WR250X
Stator w/ Rectifier	Aftermarket	None
Case Halves	Custom Build	Extensive Modifications See Section 3.4
Right Side Case Cover	Custom Build	None
Clutch Cover	Custom Build	Extended outward to allow for New EXP Clutch
Final Drive Support	Custom Build	None
Planetary Gear Reduction	Custom Build	None
Torsen Differential	Audi Quattro	None
CV Cups	Custom Build	None

Appendix B: Custom Part Drawings




Ø3.38
Ø2.28
Ø1.75
Ø.73
0
.27
1.38
0
1.35
R.22
R.10
1.29
0
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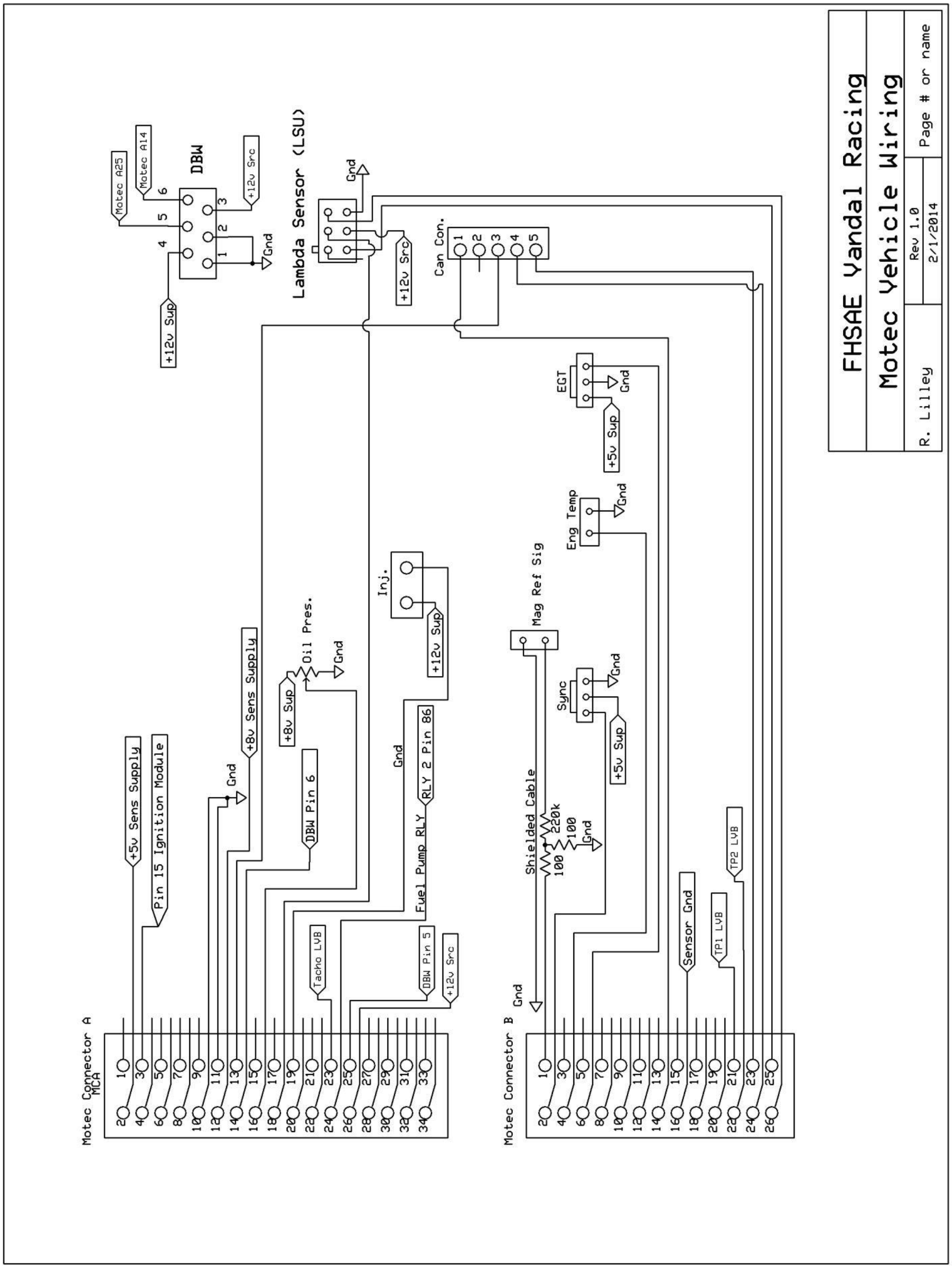
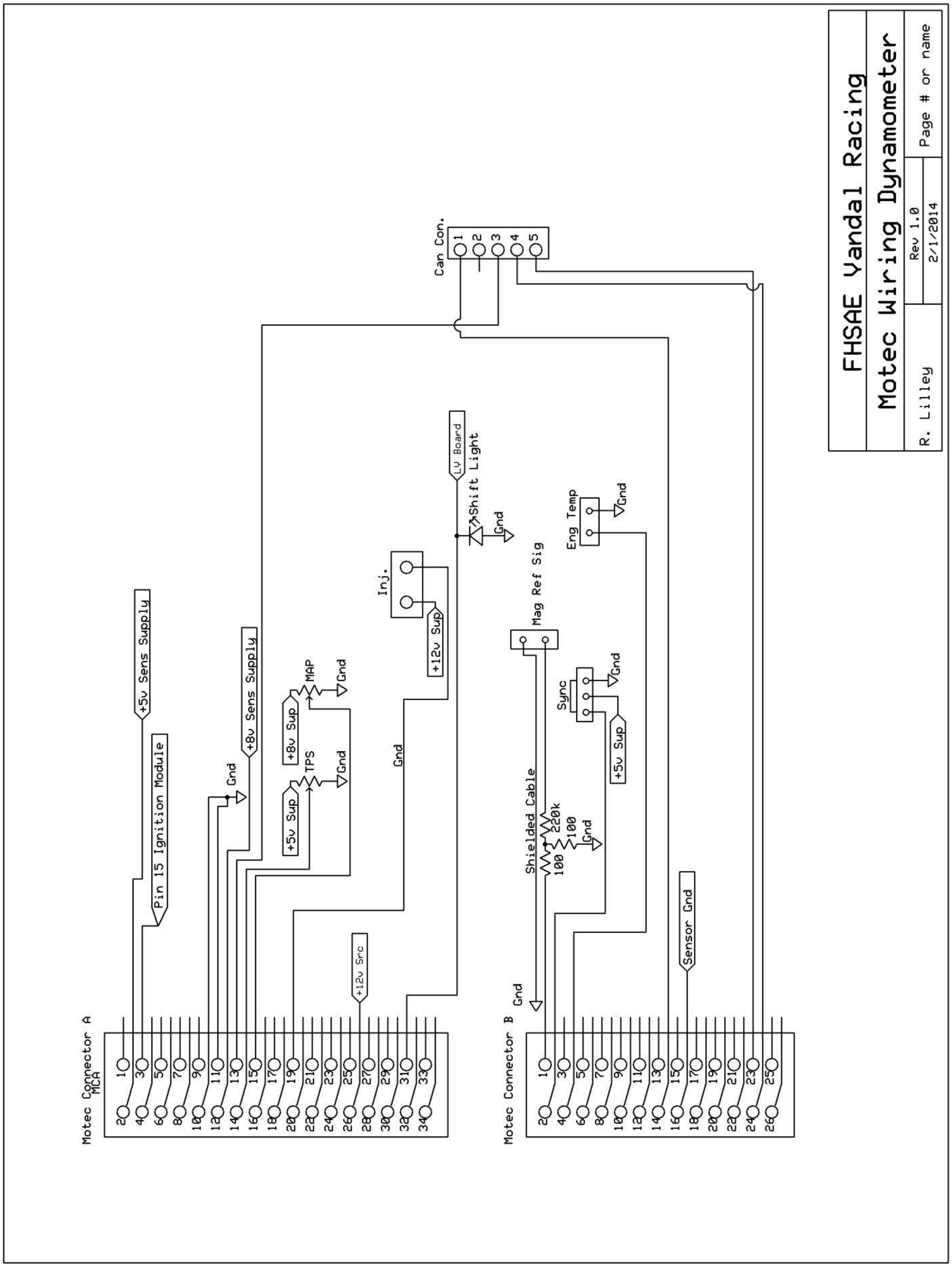
DIMENSIONS ARE IN INCHES
THIRD ANGLE PROJECTION
MATERIAL:
Dyno Adapter Plate
DEFAULT TOLERANCES:
LINEAR:
X.±.25
X.X±.1
X.XX±.01
X.XXX±.002
ANGULAR:
X.±2
X.X±1
X.XX±0 30'
DESCRIPTION: Adapter Plate for Dynometer to YZ250F Sproket
CHECKED BY: XXXXXXXXXX
DATE: XX/XX/XX
DRAWN BY: Rory Lilley
DATE: 7/20/2015
FILE NAME: adapter plate.SLDPRT
UNIVERSITY OF IDAHO
ME DEPARTMENT
PART #:
QTY:
SCALE: 1:2
SHEET: 1 OF 1

Appendix C: Vehicle Wiring Diagram



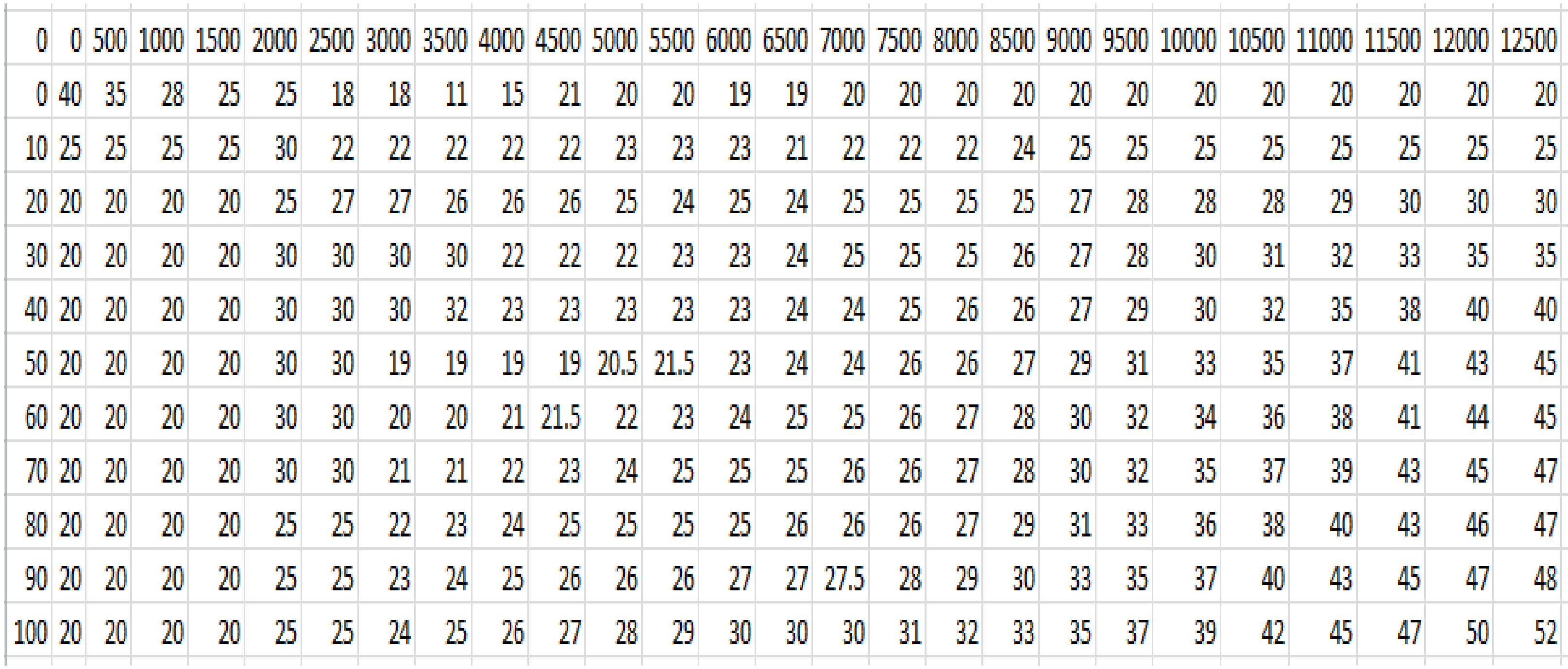
Appendix D: Test Cell Wiring Diagram



Appendix E: Main Fuel Map Values

0	0	500	1000	1500	2000	2500	3000	3500	4000	4500	5000	5500	6000	6500	7000	7500	8000	8500	9000	9500	10000	10500	11000	11500	12000	12500
0	30	30	30	30	23	23	26	26	23	22.5	22	22	27	27	27	25	25	25	25	25	25	25	25	25	25	25
1	30	30	30	30	21.5	27	27	27	27	27	27	27	27	27	27	25	25	25	25	25	25	25	25	25	25	25
2	21	27	27	27	27	27	27	27	27	27	27	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25
3	24	24	24	27	27	27	27	26	24	24	24	24	27	27	27	25	25	25	25	25	25	25	25	25	25	25
4	24	24	24	24	23	27	27	28	26.5	24.5	22.5	21.2	22.2	22.2	27	25	25	25	25	25	25	25	25	25	25	25
5	26	26	24	24	23	22	25	25	26.7	24.5	23.5	24.3	22	22	23	25	25	25	25	25	25	25	25	25	25	25
6	28	27	27	26	24	23	25	26	26	24.5	20	20	20	20	20.5	25	25	25	25	25	25	25	25	25	25	25
7	29	28	27	27	26	26	25	25	25	25	25	25	25	25	22	25	25	25	25	25	25	25	25	25	25	25
8	33	33	32	30	29	28	27	27	27	20	20	20	20	20	20	25	25	25	25	25	25	25	25	25	25	25
9	34	34	33	33	30	29	27.3	29	28.6	28	28	22	20	21	22	24	25	26	26	26	26	26	26	25	25	25
10	34	34	34	34	31.5	30.5	30	30	29.5	29.5	29.5	27	21	21.5	21.5	22	24	24	26	26	26	26	26	25	25	25
20	40	40	35	35	40	38	35	33	33.2	32.9	32.4	30	24	24	24	24	24	25	25	25	25	25	26	28	28	28
30	40	40	40	44	46	44	42.5	39	36	38	38.1	29	29	29	29	29	30	30	30.5	31	31	32	33	32	32	32
40	40	40	40	40	46	44	42.5	39	38.3	38.5	39.2	35	36	36	36	36	36	36	36.5	36	36	36	40	42	42	42
50	40	40	40	40	46	45	43.5	43	41	37.5	38	39	40	40.5	41	42	42	42	42	41	40	39	50	50	50	50
60	40	40	40	40	46	46	43	42	39	37.5	38	38.5	40	42	44	45	46	47	47.5	45	44	44	57	60	60	60
70	43	43	43	43	43	42	37.5	37.5	38	38.5	39	40	41	42.5	44	46	47	49	50.5	49	48.5	48	60	65	65	65
80	43	43	43	43	43	42	41	39	39.5	40	40.5	41	41	43	45	48	49	50	50	49	48	52	53	69	69	69
90	43	43	43	43	43	42	40	40	40	40	40	40	41	42	45	47	50	51	51	50	50	65	65	70	70	70
100	43	43	43	43	43	42.5	42.5	42.5	42.5	42	42	42	42.5	44	45.5	47	50.5	51	50	50	49	66	67	70	75	75

Appendix F: Main Spark Map Values



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0	40	35	28	25	25	18	18	11	15	21	20	20	19	19	20	20	20	20	20	20	20	20	20	20	20	20
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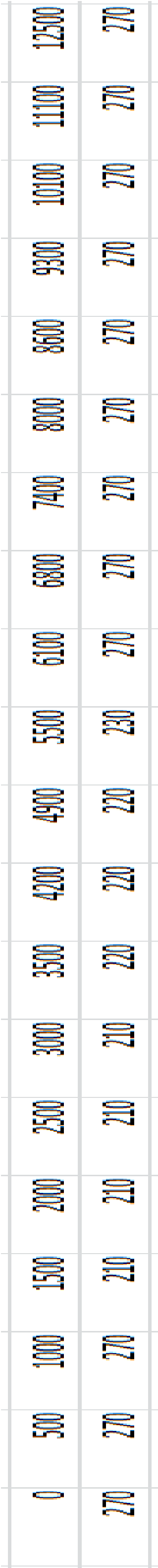
40	20	20	20	20	30	30	30	32	23	23	23	23	23	24	24	25	26	26	27	29	30	32	35	38	40	40
50	20	20	20	20	30	30	19	19	19	19	20.5	21.5	23	24	24	26	26	27	29	31	33	35	37	41	43	45
60	20	20	20	20	30	30	20	20	21	21.5	22	23	24	25	25	26	27	28	30	32	34	36	38	41	44	45
70	20	20	20	20	30	30	21	21	22	23	24	25	25	25	26	26	27	28	30	32	35	37	39	43	45	47
80	20	20	20	20	25	25	22	23	24	25	25	25	25	26	26	26	27	29	31	33	36	38	40	43	46	47
90	20	20	20	20	25	25	23	24	25	26	26	26	27	27	27.5	28	29	30	33	35	37	40	43	45	47	48
100	20	20	20	20	25	25	24	25	26	27	28	29	30	30	30	31	32	33	35	37	39	42	45	47	50	52

Appendix G: Injection Timing Values



0	500	1000	1500	2000	2500	3000	3500	4200	4900	5500	6100	6800	7400	8000	8600	9300	10100	11100	12500
270	270	270	210	210	210	210	220	220	220	230	270	270	270	270	270	270	270	270	270