COMPUTER MODELING OF A GASOLINE DIRECT INJECTION TWO-STROKE SNOWMOBILE ENGINE WITH IN-CYLINDER PRESSURE DATA ANALYSIS

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ABSTRACT

Following the win at the 2007 Society of Automotive Engineers' Clean Snowmobile Challenge collegiate event, the University of Idaho Clean Snowmobile Challenge team saw an opportunity and need to continue refining the design process. The Clean Snowmobile Challenge is an engine design-based competition focused on revising a production snowmobile. In past years, the design method has been a mixture of experience and multiple iterations of design, build, and test. The 2007 competition entry was no exception. However, with four years of work on the design, the snowmobile team was successful with a first place finish. Four years of development is not something that the team can afford to invest in every complete product. The next logical step in improving the design method is to incorporate computer-aided design.

Described in this work are the beginning stages of computer modeling for the current UICSC snowmobile engine, a Rotax 593 HO two-stroke engine retrofitted with a gasoline direct injection fuel delivery system. This includes the decision process for making the choice to use Optimum Power Technology's Automated Design software package and some details about this program, including expansion capabilities. The engine model is discussed in detail, including modeling methods, reasons, possible areas for improvement, and future research needed for continued model refinement. Results include a detailed discussion on in-cylinder pressure data gathering and analysis for characterizing the combustion process for use in modeling the engine. Continued research and development of this engine model will be required to provide the most benefit to the University of Idaho Clean Snowmobile team.

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DEFINITION OF TERMS

AD – Automated Design

AFR – Air-to-Fuel Ratio

BDC – Bottom Dead Center

CD – Coefficients of Discharge

CFD – Computational Fluid Dynamics

CPS – Crankshaft Position Sensor

CSC – Clean Snowmobile Challenge

CSV – Comma Separated Variable

CVT – Continuously Variable Transmission

EOC – End of Combustion

EMM – Engine Management Module

GDI – Gasoline Direct Injection

GUI – Graphical User Interface

log pV – Pressure vs. Volume Plot on logarithmic scale axis

MFB – Mass Fraction Burned

MSDS – Material Safety Data Sheet

OPT – OPTIMUM Power Technology

RAVE – Rotax Adjustable Variable Exhaust

RPM – Revolutions Per Minute

SAE – Society of Automotive Engineers

SDI – Semi-Direct Injection

SOC – Start of Combustion

TDC – Top Dead Center

UICSC – University of Idaho Clean Snowmobile Challenge

USDOE – United States Department of Energy

WOT – Wide Open Throttle

1. INTRODUCTION

Since 2000, the Society of Automotive Engineers (SAE) has hosted an annual collegiate competition called the Clean Snowmobile Challenge (CSC). This competition is designed to encourage the development of snowmobile engines while engineering a clean and quiet trail snowmobile. “Where do we go from here?” was a common sentiment from the University of Idaho Clean Snowmobile Challenge (UICSC) team following the win at the 2007 SAE CSC event. In past years the design method has been a mixture of experience and multiple iterations of design, build, and test. The 2007 competition entry was no exception. However, with four years of work on the design, the team was successful. With this design method, the likelihood of an optimized design is not high. The future will require a continually evolving design in order for the University of Idaho to continue to be competitive. These two aspects combined are a great motivation for the UICSC team to change its design method. Four years of development is not something that the team can afford to invest in every complete product. The next logical step is to go to a computer aided design base. This thesis describes the beginning stages of computer modeling of the current UICSC snowmobile engine, which will help in guiding design changes while providing tools to optimize the current design.

1.1. THE CLEAN SNOWMOBILE CHALLENGE

The specific goal of this competition is to develop a snowmobile engine and chassis package to be used in “…environmentally sensitive areas such as our National Parks or other pristine areas” (1). This goal is accomplished by reducing sound levels and harmful emissions, such as carbon monoxide and unburned hydrocarbons, without increasing emissions of oxides of nitrogen or hindering the snowmobile’s performance. While accomplishing this goal, the design must be reliable, cost effective, and practical. The engine choices are limited to a maximum displacement of $960cm^3$ for a four-stroke engine or $600cm^3$ for two-stroke or rotary engines (1). There are also very specific rules for what can and cannot be modified on the chassis of the snowmobile that limit the competition to a mostly engine design-based competition (1). Each snowmobile competes in several events which include: 100 mile fuel economy/endurance run, noise test, emissions test, handling, rider comfort, cold start, and acceleration events. Each of these events tests the durability and performance of the snowmobile design. The emissions test is performed using a five mode test as described in Table 1 where Mode 1 is the engine speed and torque at maximum power output. (2)

Table 1: Emissions testing mode points

Mode	1	2	3	4	5
Speed (% of mode 1)	100	85	75	65	Idle
Torque (% of mode 1)	100	51	33	19	0
Wt. Factor (%)	12	27	25	31	5

1.2. UICSC SOLUTION

The University of Idaho Clean Snowmobile Challenge (UICSC) team originally used a four-stroke engine as a solution for this challenge. The 2001 to 2003 UICSC team entries were an Arctic Cat snowmobile chassis retrofit with a four-stroke BMW motorcycle K75RT engine. Emissions were further reduced with use of a catalytic converter. This design strategy proved to be successful with back-to-back wins in 2002 and 2003. After that design strategy proved successful, the UICSC team decided to convert to a non-traditional design. This design strategy was to begin development of a two-stroke gasoline direct injection (GDI) engine. This design would not only clean up emissions of the notoriously “dirty” two-stroke engine, but would have the added benefit of two-stroke machines: a power-to-weight ratio unmatched by their four-stroke counterparts. In order to implement this design strategy, the UICSC team modified an Evinrude E-Tec outboard GDI system and retrofit it to a 2002 Polaris Liberty 600 engine. Most modifications and design changes were made through experience and educated guesses. In the 2007 competition year, after four years of research and iterations, the UICSC team finally constructed a competition-worthy package and once again won the

SAE CSC competition. This design needs to be refined further in order to compete in future years, and continue a tradition of innovative engineering.

1.3. RESEARCH GOALS

This research focuses on the beginning stages of modeling the UICSC team's 2007 SAE CSC snowmobile engine. This engine model was put together in Optimum Power Technology's Automated Design software, also referred to as Virtual 2-Stroke. The goal is to provide the UICSC team with a base to test engine design changes to more quickly and accurately determine a design path. The purpose of this paper is to present the details of this model, as well as suggesting future work to refine the model for accurate prediction of engine response to changes. Included in this paper is a description of the choice of Automated Design over alternatives, as well as a description of the capabilities and limitations of this engine modeling software package. The engine modeling methods, reasons, and areas for improvement will be discussed in detail. In-cylinder pressure data were taken to characterize combustion for the model and will also be presented in detail. Last to be discussed in this paper is the future work needed on the model, as well as uses for the model once complete.

2. BACKGROUND

2.1. TWO-STROKE GASOLINE DIRECT INJECTION ENGINE OPERATION

Every internal combustion engine must do four things: intake the combustion mixture components, compress the mixture, combust the mixture and obtain the power from combustion, and exhaust the resultant products. In a two-stroke engine it takes one revolution, or two strokes (axial motion from the top to bottom of the cylinder) of the piston, to do all four processes. There are several two-stroke engine types. However, this paper will focus on a reed valved, crankcase-compression charged, Schnürle-type loop scavenged, gasoline direct injected (GDI), spark ignition two-stroke engine (3). Figure 1 shows an example of such an engine. To explore the operation of this engine, we will follow an air charge through the engine from start to finish. Air comes into the engine via the throttle bodies, which are butterfly valves that meter the air to the engine and are controlled by the operator. The air then passes through the reed valves, which are check valves that allow air flow into the engine. The air is drawn in by the vacuum created in the crankcase from the piston's upward motion. After the piston reaches top dead center (TDC) it begins to come down. This pressurizes the crankcase, shutting the reed valves and compressing the air in the crankcase. During the piston's downward motion, it uncovers the transfer ports. This allows the now pressurized air in the crankcase to enter the cylinder. At this point, several things happen at the same time, the timing of which is highly dependent upon engine speed, fuel

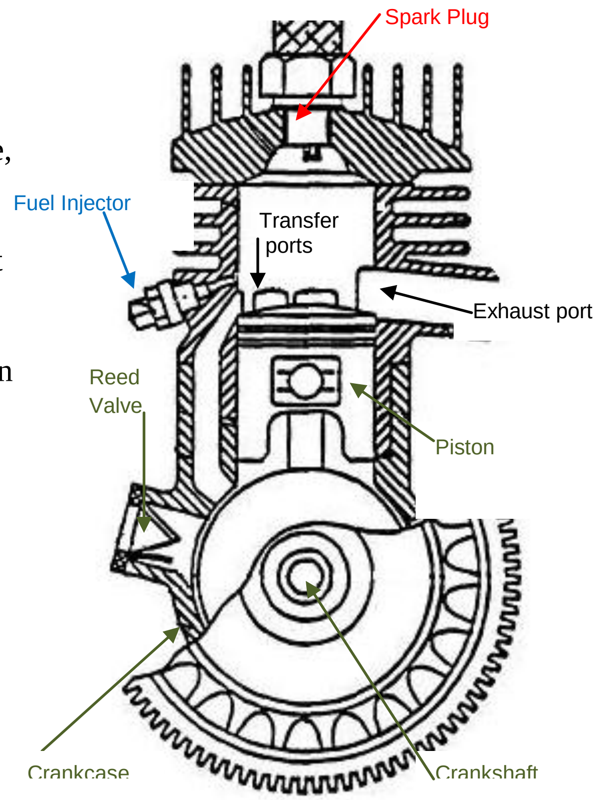

Figure 1: Cross section of a GDI two-stroke engine [modified from original (10)]

delivery and ignition timing. Fuel is injected directly into the cylinder when the piston is passing through bottom dead center (BDC) and/or while traveling upward. At the same time the incoming air is mixing with and displacing the outgoing combustion mixture. As the piston travels upward it compresses the air and fuel mixture in the cylinder. As this occurs, it is also creating a vacuum in the crankcase to intake the air. Just before the piston reaches TDC, a spark ignites the fuel-air mixture which increases the pressure in the cylinder and pushes the piston downward. This downward motion is converted to rotational motion by the crankshaft. As the piston moves downward, the exhaust port is uncovered, allowing the combustion mixture to escape through the exhaust system. Shortly after the exhaust port is uncovered, the transfer ports are uncovered, which allows for the intake air to displace the combustion products. The transfer ports work together to create a looped flow through the cylinder in an attempt to displace the combustion products with the new incoming air. The transfer ports are angled toward the boost port which is opposite the exhaust ports. The incoming air from the transfer ports goes toward the boost port, which is angled upward, causing the air to loop up to the cylinder head and back down to go out the exhaust. Figure 2 shows the general airflow through the cylinder in a Schnürle-type loop scavenging process.

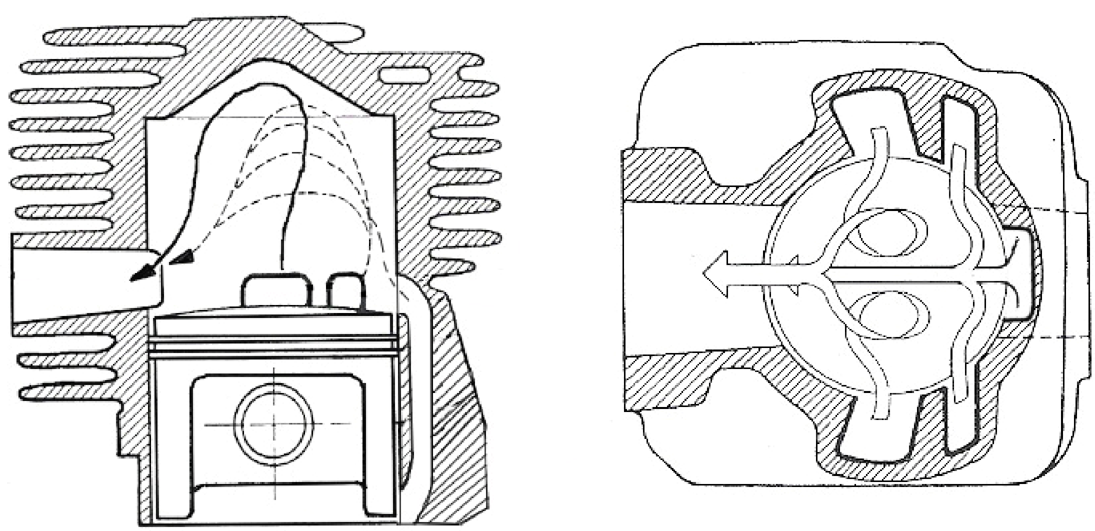

Figure 2: In-cylinder flow through a Schnürle-type loop scavenged engine (3)

Depending on fuel injection timing, fuel is injected directly into the cylinder during the scavenging process. Some of this unburned fuel air mixture escapes with the exhausted air into the exhaust system. This is called short-circuiting. The exhausted mixture exiting the exhaust port consists of unburned fuel, intake air, and combustion products. When the exhaust port opens, the exhaust mixture enters the tuned pipe and creates a pressure wave. This pressure wave is reflected off the converging portion of the tuned pipe back toward the cylinder. This reflected pressure wave can cause what is known as the plugging pulse. This creates a high pressure at the exhaust port which reduces the amount of short-circuited air and fuel mixture and increases the cylinder pressure just before the exhaust port closes, creating a “supercharging” effect. The tuned pipe is named due to this effect. The length and slope of the converging section are “tuned” for a specific engine speed band and efficiency of the effect.

It is important to note that the timing of the fuel injection can vary greatly depending on the desired outcome. Fuel is injected late at idle, just before or just after the exhaust port is closing on the piston’s upward motion. This late injection causes a stratified air-to-fuel mixture and reduces or eliminates the short-circuited fuel. At cruising speeds, the fuel is injected early, just before or after the upward stroke of the piston, creating a homogenous air-to-fuel mixture in the cylinder at the time of combustion. For further details on stratified and homogeneous combustion refer to the work of Johnson (4).

2.2. THE TWO-STROKE ENGINE IN A SNOWMOBILE

The typical snowmobile uses a continuously variable transmission (CVT) and chain case to connect the engine to the track. The track is turned by a drive axle which has two toothed sprockets. The drive axle is the output of the chain case, which is a chain driven gear set that reduces the countershaft input speed. The countershaft is driven by the CVT, which is driven by the engine. The CVT components include a primary pulley/clutch that is connected to the engine crankshaft, a secondary pulley/clutch that is connected to the countershaft, and a belt that connects the two. Both pulleys have two halves that move relative to each other as the rotational speed changes. The primary clutch has a spring that holds the primary pulley halves apart at low speeds, allowing the engine to idle and the belt to slip on the pulley so the snowmobile does not move. As engine speed increases, the clutch moves the halves of the primary closer together. This increases tension on the belt, transmitting power to the secondary pulley. As the vehicle speed increases and the secondary pulley's rotational speed increases, the secondary clutch begins to pull the two halves of the pulley apart. During acceleration, this combination creates a small effective pulley diameter on the primary and a large pulley diameter on the secondary, which is similar to low gearing. At higher vehicle speeds, the secondary effective diameter is reduced and the primary effective diameter is increased. Overall, the CVT maintains engine speed within a designed range while continually changing the gearing ratio from the engine to the track, increasing speed.

The two-stroke engine has a small band in the engine speed range where the tuned pipe is effective and other engine design parameters increase engine performance considerably. This engine speed range is called the power band. When coupled with a CVT, a two-stroke engine's power band can be utilized to increase overall snowmobile performance.

2.3. CURRENT UICSC SNOWMOBILE ENGINE

The 2007 UICSC team used a 2006 Ski-Doo MX Z chassis with a Rotax 593 HO carbureted engine base. The Rotax 593 HO is a 594.4cm^3 displacement, liquid-cooled spark ignition two-stroke engine. This engine has two cylinders oriented in-line with one another. The piston motion is 180° out of phase between the two cylinders,

Figure 3: Rotax 593 HO carbureted engine (7)

which means that when one is at BDC the other is at TDC. The engine was retrofit with an Evinrude E-Tec GDI system, shown in Figure 4.

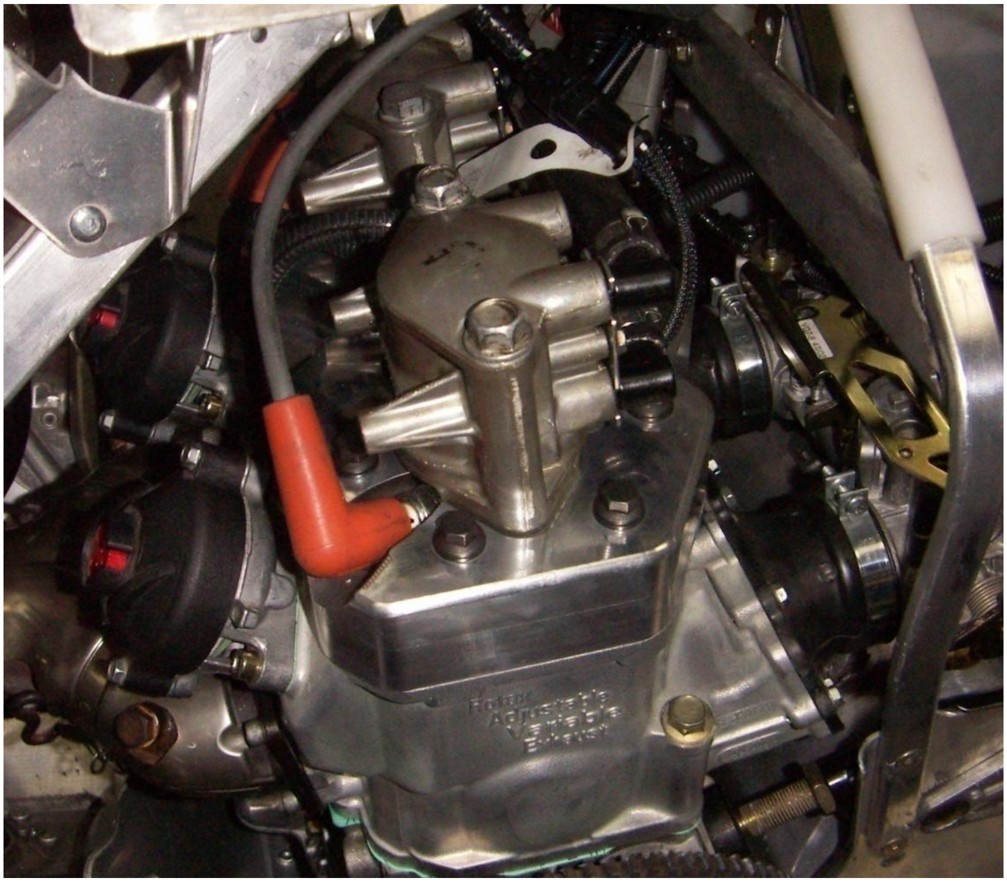

Figure 4: UICSC 2007 GDI Two-Stroke Engine

This retrofit required replacement of the carburetors with the throttle bodies used on the Rotax 593 HO semi-direct injection (SDI) engine, as well as a custom designed and built cylinder head and other miscellaneous parts. This engine produces stock power while reducing the emissions from the already clean two-stroke engine, the production Rotax 593 HO SDI engine. The engine intake system consists of two air boxes, used to silence the intake sounds, before going to the throttle bodies. To increase the sound damping efficiency of the air boxes, they are lined with high density foam. The throttle bodies, as mentioned before, are stock butterfly style Rotax 593 HO SDI throttle bodies with a small hole in them to meter air when the valve is shut. The cylinder layout is designed for Schnürle-type loop scavenging effects, with four transfer ports and one boost port, as described in the previous section. The exhaust port system consists of two auxiliary ports and one main port. The main port has the Rotax Adjustable Variable Exhaust (RAVE) power valve system (5). The RAVE system causes the exhaust port top to be lowered at lower engine speeds, which reduces emissions and noise, and provides better run characteristics. At higher engine speeds, particularly within the band for the tuned pipe operations, the power valve opens, increasing power output of the engine.

Beyond the power valves, both cylinders exhaust gasses combine together in the Y-pipe before entering the tuned pipe. This tuned pipe is tuned to increase engine performance at higher engine speeds (6000 to 8000 RPM range). In order to quiet the exhaust sounds exiting the engine, the stock muffler was used, with modification for a catalytic converter at the outlet. The stock exhaust muffler consists of four chambers. The first and third chambers are expansion volumes, and the second and fourth chambers are absorption chambers. The absorption chambers consist of a perforated tube running between the previous chamber and the next chamber, or outlet. Around this perforated tube is packing within a volume, used to absorb the sound. The catalytic converter is a metallic substrate oxidation catalyst from Aristo Catalyst, Inc. The catalytic converter was used to further reduce exhaust emissions.

2.4. DESIGN METHODS

While there are many different design methods, there is no one method that works well for every situation. However, there are some specific steps that should occur for a design to be successful: goals definition, design choice, the design implementation, and design testing to verify the goals were met. Each step can be simple or extremely difficult. In the past, the UICSC team has relied on experience and intuition to help choose a design. This experience was gained through multiple "design, build, test" iterations, where the design was either an implementation of current theories, or modification of previous designs. This method can work, although it is not very efficient. UICSC team overcame many of the initial hurdles to creating a GDI two-stroke snowmobile engine by using an experience based method since there was little in the way of design software or modeling methods capable of providing help in implementing the GDI fuel delivery system on a two stroke snowmobile engine. There was even less understanding on how the high engine speeds of a snowmobile engine would effect the GDI system design of the E-Tec system. Having solved the major problems with implementing a GDI fuel delivery system to a two-stroke snowmobile engine, the UICSC team is faced with a design method shift. No longer can the team rely on experience alone. With an infinite number of possible designs, it is difficult to choose which is best. Every small change in design can have huge effects on many different characteristics in the engine. Without a way to predict these effects, the team is forced to rely on intuition and past experience. If there were a method to predict the changes qualitatively, the design selection process would be more robust. This would allow for experience and intuition to be supported, or contradicted, before there are hundreds or thousands of dollars and hundreds of hours spent on a design. A computer model of the engine will increase design selection process efficiency. An accurate model will assist the UICSC team in making more informed decisions, and potentially optimizing the design much more quickly.

2.5. COMPUTER MODELING

The UICSC team has used computers in solid modeling, manufacturing, implementation, and in presentation. However the UICSC team has used computers sparingly in deciding what to solid model, manufacture, implement, and test. This area is the target of this research, to lay the groundwork to allow for this process. The problem in the past has been the front loading of work required to establish a model detailed enough to predict changes in engine performance. The amount of time spent on learning the program, modeling the engine, and beginning to implement the model in the design process is equivalent to that required to design, build and even test a design. Also, no program is perfect at predicting the results of design changes, and there is no question why there has not been a bigger push to create a computer model of the engine design. However, there has always been an interest in utilizing every resource in developing the two-stroke GDI engine. The UICSC team is now at a place where a computer model is needed.

No computer model will precisely predict engine performance. However, the more accurate the inputs to the model itself, the more accurate the results will be and will more closely resemble experimental data. Every engine design has built-in inaccuracies such as tolerances in manufacturing, wear, tolerances in fuel, and inconsistencies in ambient conditions. No two engines are exactly the same, which further complicates the modeling process. There are two goals in modeling: to match the experimental data and to predict future performance. Every program has correction factors, numbers that can be experimentally determined and input to the model, but they may not always work. These correction factors are a way to make the model match the current data, although this does not guarantee an accurate prediction of future data, especially when changes are made to the model.

With all this uncertainty, why bother making a computer model of an engine? A computer model, if constructed correctly, can give trends. For example, if an engine produces 100 horsepower (hp) and the computer model's results says the expected output is 125 hp in one configuration then we might say this model is off by 25%. However, if a change is made to the model, and it predicts that this change will now produce 150 hp (a 20% increase) and the modified engine now produces 119 hp (a 19% increase) we would say that that this model is accurate. This model may not be able to tell us with great precision what the expected power output would be, but if we ran several modifications through the simulation, it would be able to indicate which modification would be the best. Running a computer software program to find the best value for a given parameter before making the part would help tremendously. Not only would it save time, money and frustration, it would be able to provide solutions that would never have been tried, and see outcomes that would otherwise not have been predicted.

2.6. IN-CYLINDER PRESSURE

Changes in pressure and volume in the cylinder are related to the energy released from the combustion process. This information can be used to characterize the combustion process of the engine for a more accurate computer model. Some of the work performed on this project has been focused on gathering and analyzing in-cylinder pressure data from the UICSC snowmobile engine in order to provide the foundation for an accurate computer model.

2.6.1. PRESSURE GRAPHS

One of the first steps in determining the inputs for the computer model are to obtain pressure graphs the first of which will be a pressure versus volume (*pV*) graph. The *pV* graph can then be plotted with logarithmic scales on both axes creating a log pressure versus log volume (*log pV*) graph. The *log pV* graph allows for the expansion and compression portions of the cycle to be analyzed as polytropic processes which is characterized by the following equation (6):

$$pV^n = C$$

Equation 1

Where:

p is the pressure in the cylinder

V is the volume of the cylinder

n is related to the slope of the portion of the *log pV* graph that represents the compression and expansion processes

C is a constant

Also characterized by the *log pV* graph are the start of combustion (SOC) and end of combustion (EOC) points. These two points are represented as the departure from the straight line at the end of the compression process and the beginning of the expansion process respectively. The SOC and EOC points are important for later on, and are used as inputs for the model. Figure 5 shows an example two-stroke engine process on a *log pV* plot, with the SOC and EOC points labeled.

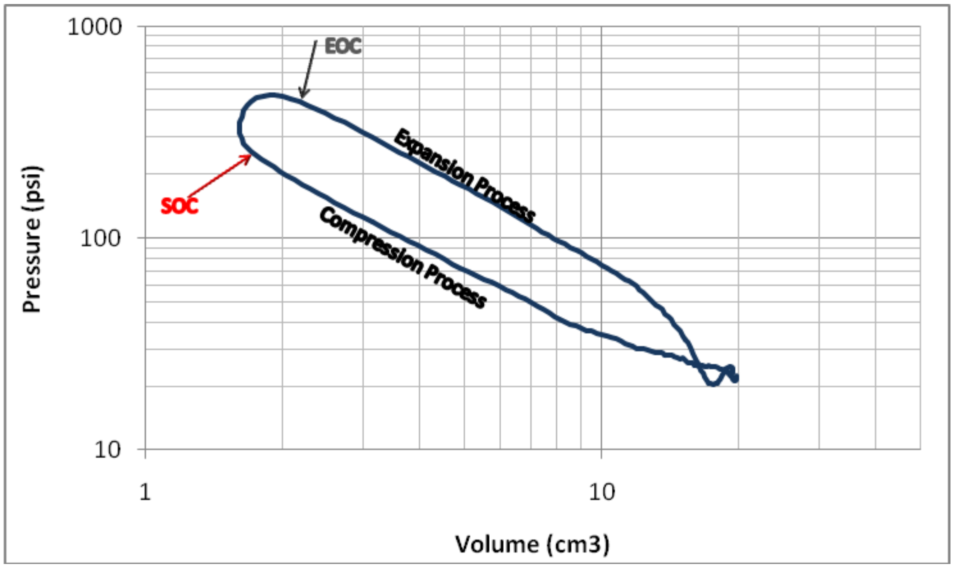

Figure 5: Typical two stroke pressure vs. volume curve plotted on log-log scale axes

2.6.2. MASS FRACTION BURNED

In order to determine the mass fraction of fuel burned, also known as the mass fraction burned (MFB), the technique developed by Rassweiler and Withrow will be used (6). This technique makes several assumptions: the effects of heat transfer are included in the analysis of polytropic exponent *n* only; the in-cylinder pressure raise due to combustion is proportional to the energy release from the combustion of fuel, not the mass of mixture burned; and the polytropic exponent *n* is constant during combustion. This method characterizes the MFB through the following equation (6):

$$x_b = \frac{p^{\frac{1}{n}}V - p_o^{\frac{1}{n}}V_o}{p_f^{\frac{1}{n}}V_f - p_o^{\frac{1}{n}}V_o}$$

Equation 2

Where:

x_b is the MFB

p, n, V are as described above

p_o is the in-cylinder pressure at SOC

V_o is the volume of the cylinder at SOC

p_f is the in-cylinder pressure at EOC

V_f is the volume of the cylinder at EOC

Figure 6 shows an example MFB curve. The values of the curve can be tabulated as inputs to the model.

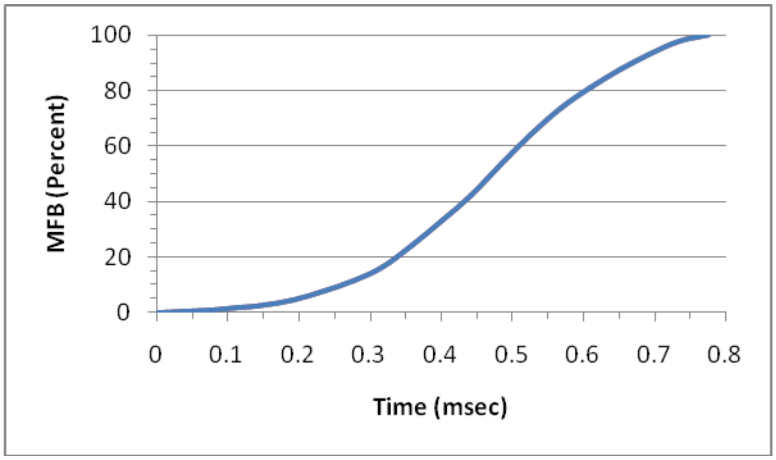

Figure 6: Typical mass fraction burned curve

3. SOFTWARE CHOICES

There are many modeling software packages on the market today. In choosing a software package, many factors must be taken into consideration: cost, program capabilities, program limitations, ease of learning the program, ease of operation, team reaction to program, and many more. It is often difficult to decipher the truth from embellishments in the advertisements for the product. Every program will claim it is accurate, and will give you proof of this fact, but that does not mean it is accurate for every engine, or able to accurately predict changes in performance due to changes in design. So the best method of choice is to speak with those who have used the programs, getting unbiased customer reviews. In searching for which product to use, two program names came up, and they could not have been more opposite. One was user-friendly, but not as flexible; the other was flexible, but not very user-friendly.

3.1. CHOICE OF PROGRAM

The two software packages that were researched for this project were the commercially available software packages from OPTIMUM Power Technology (OPT), and KIVA which is available through the United States Department of Energy (USDOE).

OPT's software packages are one-dimensional, unsteady-gas dynamic analysis programs. These software packages are in use in industry. This software package comes with everything required to input engine models, simulate the engine performance, and analyze simulation results.

KIVA is "…a transient, three-dimensional, multiphase, multicomponent code for the analysis of chemically reacting flows with sprays…" (7). KIVA is a non-commercial software package available through the USDOE for a fee. Since the KIVA software package is non-commercial, it is a source-code software package. This allows the end user to modify the code. The end users can make changes to the software to allow for more accurate modeling of their specific application, or to add features that are not available through the original software package. KIVA contains programs for inputting the model, running the computations, and analyzing the results. Results can be displayed in crude graphics using KIVA software or can be manipulated for use with other graphical software packages, such as EnSight. EnSight is a commercially available post-processing software package available to create three dimensional representations of KIVA software results.

3.2. COMPARISON OF PROGRAM PACKAGES

OPT's software packages have a relatively easy graphical user interface (GUI), which requires very little time to learn. Most GUI components of OPT's software packages are similar to several Windows file exploration programs. There is typically a tree view on the left and icons or listing on the right. KIVA, however, does not have a GUI supplied with the programming. There are ways to program one, but creating one requires a significant investment of time. Instead, KIVA uses a text input similar to computer programming languages to input the model. There are commercially available pre- and post-processing programs that make using KIVA much easier. KIVA is much more time-intensive to learn, and programming knowledge is a must, whereas OPT's software package is not as time intensive, and is fairly intuitive for the typical user. KIVA runs in a Linux or UNIX operating system, whereas OPT's software packages require Microsoft Windows as well as Microsoft Excel.

OPT's software packages are capable of modeling two- and four-stroke engine components, along with typical components including turbochargers, superchargers, intercoolers, and emissions catalysts. These capabilities, along with the quick computational speed, allow for these programs to be very useful in designing engines. Unfortunately, these software packages are limited in their capabilities. They are unable to model extremely complex geometries with great accuracy. However, OPT's software packages have the capabilities to interface with computational fluid dynamics (CFD) programs such as Fluent and Star-CD. This feature is limited to only one component in an engine model. This also requires having the software installed and licensing for other programs on the computer system used.

The KIVA software package is a much more intensive modeling program. It has the ability to do three-dimensional modeling for many different combinations of engine characteristics, as well as different combustion modes. Since KIVA is a source code, modifications to the program are possible. This means the software is limited only by the programming ability of the user and current research modeling capabilities. Due to the flexibility of this program, and the intensive user knowledge requirements, this program is not well suited for design. Instead, KIVA is a great tool for modeling, understanding, and predicting changes in very complex designs such as the GDI system interactions in the cylinders.

3.3. DECISION AND REASONS

Ultimately, the decision came down to three very straightforward factors. First and foremost was the perceived difficulty in learning the KIVA software package. Not only would this work have been impossible, due to time constraints, there would also have been difficulty transitioning the UICSC team from no engine model to such a difficult program. The KIVA software package is not easy to learn, nor easy to use once the program has been learned. OPT's software package did promise ease-of-use, and along with manuals, there is also technical support available to all customers. A close second, and always lingering issue, was the cost of the software. OPT's software packages are free, except a small administrative fee for licensing to the university, whereas KIVA costs considerably more for obtaining the source-code. Also with KIVA, there is an added cost for a computing system with a Linux or UNIX based operating system, something that is not readily available at the University of Idaho Mechanical Engineering Department at this time. The last factor considered was the software's capabilities and limitations, as discussed in the previous section.

Due to these three factors, the overall goals of this project, and the UICSC team goals, OPT's software package was chosen. OPT's software is better suited for design and quite capable of the initial research goals. KIVA is not a bad choice, nor should it be dismissed as a possible software package in the future. KIVA is much better suited for detailed analysis of the current design, and would be a great tool for future research into the particular detailed characteristics of the GDI two-stroke engine.

4. OPTIMUM POWER TECHNOLOGY'S SOFTWARE PACKAGE

4.1. VIRTUAL 2-STROKE VS. AUTOMATED DESIGN

OPTIMUM Power Technology offers several software packages, including Automated Design (AD), Virtual Engine, Virtual 4-Stroke and Virtual 2-Stroke. The differences among the software packages are the capabilities of each. While they all use the same GUI and simulation tools, each has its own software licensing. Virtual Engine, Virtual 2-Stroke and Virtual 4-Stroke components contain the same software. The only difference is the capability of modeling components for only two-stroke engines, only four-stroke engines, or both. All three packages use the same programs for modeling. Licensing "unlocks" the two- or four-stroke specific components, as well as other components. AD contains this software along with the ability to perform a parametric study on several parameters at once. All four software packages have programs that are used to analyze the simulation results. The differences among the software packages' analysis tools are the features available. These features include port and reed valve animation windows in the Virtual 2-Stroke Animate program (included in the AD software package).

The Virtual 4-Stroke software package will not model a two-stroke engine and is therefore eliminated from the choices. Virtual Engine is a combination of both Virtual 2-Storke and Virtual 4-Stroke, making it the same programming software as Virtual 2-Stroke in this application. AD contains Virtual 2-stroke, with appropriate licensing, and has the capability to do multiple parameter parametric study, which made the choice to use AD a natural one. This software was available through an agreement with SAE and OPTIMUM Power Technology for a small administrative fee.

4.2. DESIGN PROGRAM

AD contains the Design program, which is the pre-processing program, where each component's specifications are input, and testing procedures are defined. Design has the capabilities of modeling a multitude of components, some of which were not used in this modeling of the engine. The components used will be discussed in this section, including some limitations of each component and some parameters that are used to help more accurately model the actual components. Figure 7 shows the layout of Design program.

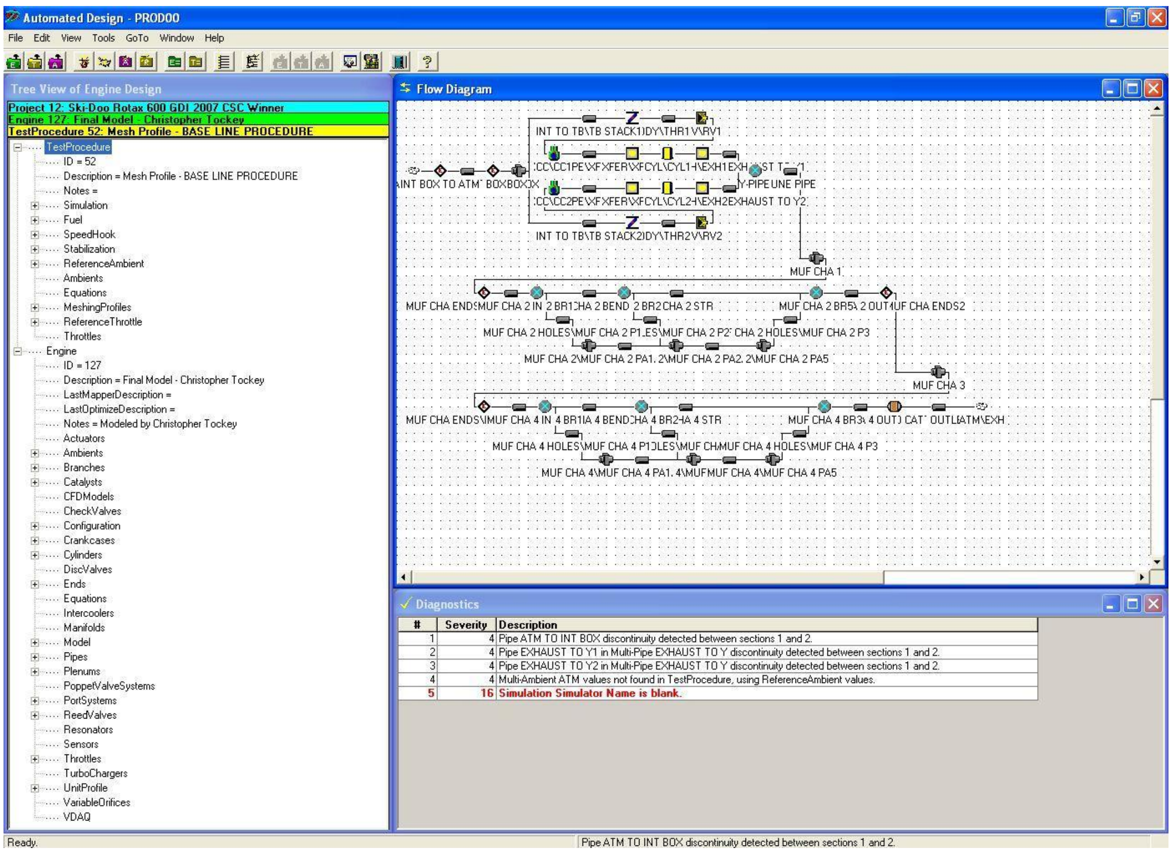

Figure 7: Screen Capture of the Design Program

4.2.1. ICONS

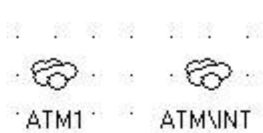

Figure 8: Single vs. Multi-component icons

Figure 8 shows the difference between single and multi-component icons. The icon on the left is a single component, the name of which is ATM1, while ATM/INT is the name of the multi-component. ATM/INT is named this way since it is a member of the ATM multi-component modeling group, and its name is INT. The difference between single and multi-components is that a multi-component can have several components with the same exact characteristics. If one is changed they all update. For the duration of this paper all components that are part of a multi-component modeling group will be referred to by their component name only. For example, ATM/INT will be referred to as INT only.

4.2.2. AMBIENTS

Figure 9: Ambients icon

Ambients are components that represent atmospheric conditions. These are used at the end of the intake or exhaust systems. Ambients allow atmospheric temperature, pressure, relative humidity and air purity (ratio of exhaust gasses to pure air in the gas mixture) to all be defined. Ambients can also be used to simulate forced induction and exhaust gas recirculation as well.

4.2.3. BRANCHES

Figure 10: Branches

Branches model junctions of three or more pipes by specifying angles between the pipes and defining the incoming and outgoing pipes. Branches have no physical mass; they are only a way to orient three or more pipes to each other. They also cannot model junctions beyond physical orientation of the components, which means pipes being joined must be modified slightly to allow for accurate modeling of joints.

4.2.4. CATALYSTS

Figure 11: Catalysts

Catalysts model the physical and chemical characteristics of a catalytic converter, including substrate type, physical dimensions, and overall reaction kinetic capabilities of the catalytic converter. One major limitation of this component is that a certain physical geometry is expected. Not all catalytic converters fit this, especially those used in smaller engines and custom applications. Therefore, in order to meet physical geometry requirements, modifications in the inputs of the component are necessary.

4.2.5. CRANKCASES

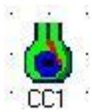

Figure 12: Crankcases

As described in the two-stroke operation section, this engine requires the pressurization of the crankcase to force the incoming air into the cylinder. That element is modeled by the Crankcases component. The Crankcases component not only models the volume used to pressurize the air, but also models the crankshaft of the engine, which includes important engine parameters such as stroke of the piston and crankshaft counterweight geometric properties.

4.2.6. CYLINDERS

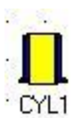

Figure 13: Cylinders

The Cylinders component models the physical characteristics of the piston, connecting rods, cylinder head, and cylinder itself. Within this component there are also inputs for modeling engine friction, timing, combustion characteristics, and scavenging characteristics. Friction is modeled using a curve to approximate the amount of parasitic losses on the engine. Combustion characteristic modeling uses inputs such as burn delay, burn duration, air-to-fuel ratio (AFR), ignition timing, mass fraction burned (MFB), and heat release or single/double Wiebe functions. The loop scavenging effects described in Section 2.1 are modeled using a scavenging curve.

4.2.7. ENDS

Figure 14: Ends icon

Ends are a way of describing how a pipe meets with another component. Such conditions can be a plain sharp cornered end, a bell mouth on a pipe or even a closed end. In modeling these components, there are inflow and outflow multipliers to a selected Coefficient of Discharge (CD) map. CD maps are coefficients used to modify the airflow for a given component's pressure drop and area change.

4.2.8. EQUATIONS

Components are sometimes mathematically interlinked, requiring a method of linking them together. Equations are used to mathematically describe this interlinking. Most component specifications can be used as a variable in equations and even several user-defined variables. One complication is that the names of the user-defined variables cannot be changed in VE, therefore making keeping track of them difficult.

4.2.9. PIPES

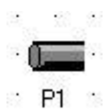

Figure 15: Pipes icon

As the name implies, Pipes model piping at any location in the engine. Pipe characteristics are input as circular in cross section, with pipe entrance and exit diameters, length and any bend information as the physical characteristics. In order to model noncircular cross section pipes, an effective diameter is calculated on the entrance and exit of the pipe and a shape factor must be specified. An effective diameter is the diameter of a circle with the same area as the cross section being modeled. The shape factor is a ratio of the pipe's actual surface area to the effective surface area. To allow for difficult pipe geometries, such as those present in the tuned pipe, a pipe can be broken down into sections, with the physical characteristics of each section defined separately.

4.2.10. PLENUMS

Figure 16: Plenums icon

Plenums model volumes in piping systems, such as intake air box, or exhaust chambers. In connecting plenums to pipes there is no way to easily specify the orientation of these connections.

4.2.11. PORTSSYSTEMS

Figure 17: PortsSystems

PortsSystems are components containing the specifications of each port in the cylinder. The ports are grouped together by their characteristics: intake, transfer, or exhaust. This is where the timing of the ports is specified, as well as the geometrical properties of the port. The orientation of the ports (direction they point, proximity to other ports) is not specified. Instead, the PortsSystems component models the ports as entrance and exit area of the cylinder. The effects of port orientation are included in the Cylinders component scavenging model.

4.2.12. REEDVALVES

Figure 18: ReedValves icon

The ReedValves component models the reed cages, specifying the reed, cage, and stop plate physical properties.

4.2.13. THROTTLES

Figure 19: Throttles icon

The throttle bodies are specified using this component; this is done by use of an area ratio, which is a ratio of uncovered area to overall area of the throttle.

4.2.14. TEST PROCEDURES

In order to run a simulation on an engine model, a testing procedure must be specified. There are two types of testing procedures in AD: SpeedHook and Mapper. Both are used to define the engine speed(s), the fuel used, and the tolerance on the simulation outputs. A SpeedHook is used to simulate the engine performance in a range of engine speeds or at one engine speed. However, a Mapper is used to vary a chosen parameter to several different values and run a simulation on all the different iterations of the engine. A SpeedHook is used to match the experimental data conditions for comparison, while Mapper is used to compare several different values of one component parameter. Part of the test procedure is the meshing profiles for the engine components. Meshing profiles can be defined globally, for one system, or for a single Pipes component. Meshing profiles cannot be defined for other components individually. AD requires that each component has at least three mesh points.

4.3. ENGINE SIMULATION RESULTS

AD has several methods for reviewing, analyzing and presenting simulation results. All simulation results can be saved as a comma separated variable (CSV) file, which can be viewed using spreadsheet software such as Excel. This allows the raw data to be used to create specific graphs or charts for review or presentation. In addition to raw data output, there are programs that are part of the AD software package that can be used to review, analyze and present simulation results. The programs are Analyze, DynoScope, and Animate. Analyze uses Excel to create graphs of engine parameters as a function of engine speed in revolutions per minute (RPM). This can be done to compare results from multiple design options. Multiple parameters can be placed on separate set of axes, or on the same axis for a comparison of the two, as demonstrated in Figure 20.

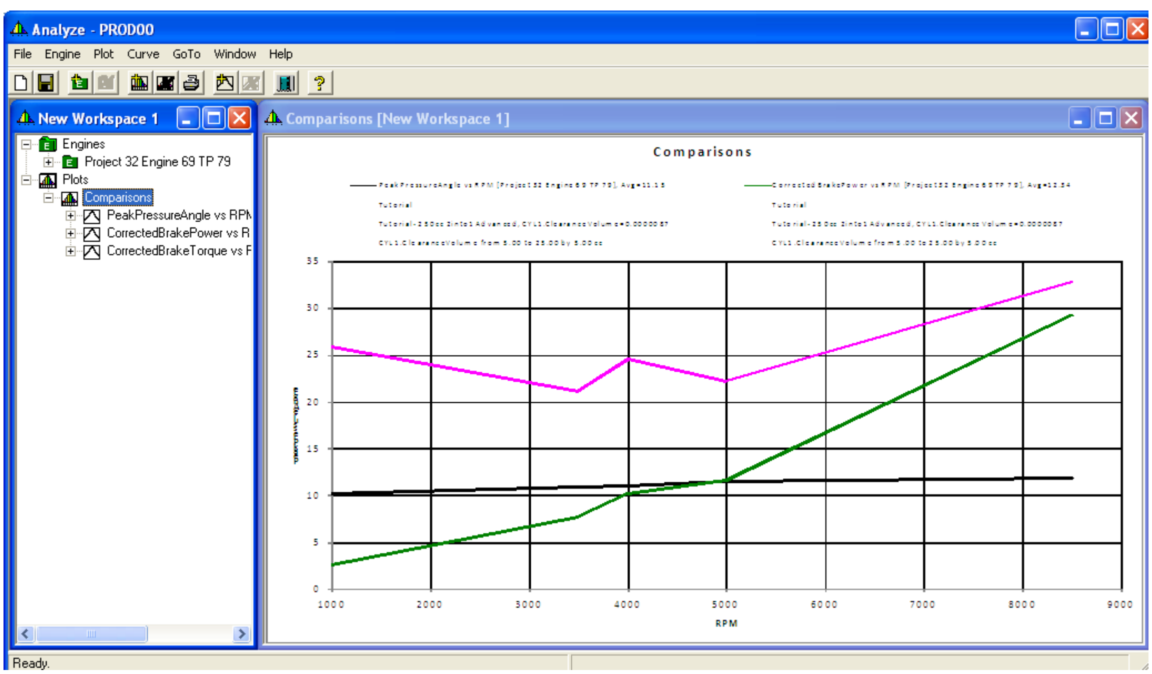

Figure 20: Screen capture of Analyze

DynoScope creates graphs of component parameters as a function of crankshaft angle. This can be used to see in-cycle fluctuations, or compare simulated results with experimental data such as in-cylinder pressure data. DynoScope can also show the parameters relative to cylinder timing events, such as ports opening or closing.

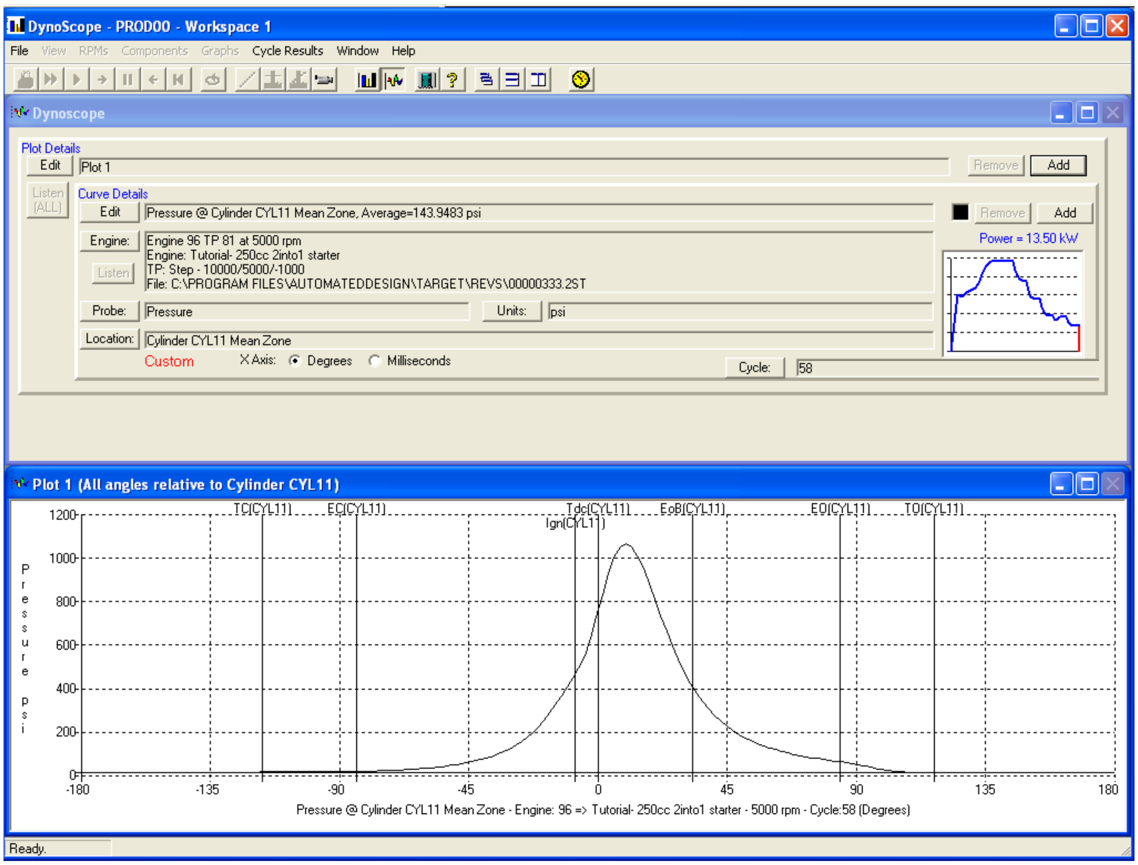

Figure 21: Screen capture of DynoScope

Another tool available is the Animate program, which provides a graphical representation of an engine cycle. What is graphically represented can be chosen to be a multitude of parameters, from temperatures and pressures to flow rates. Animate is a great tool to visualize the two stroke engine cycle. The animations can be used to see effects such as the plugging pulse described in Section 2.1. Along with a graphical representation of parameters, Animate can show animations of ports and reed valves, as shown in Figure 22.

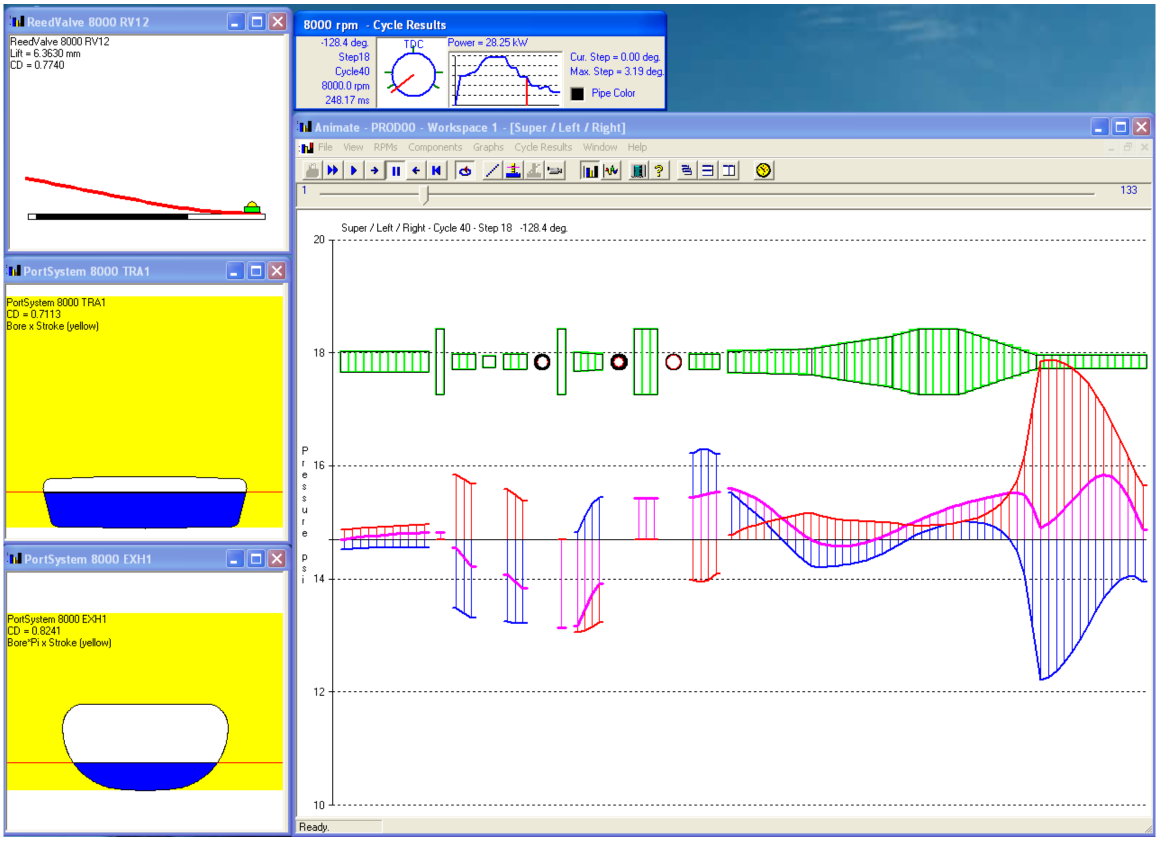

Figure 22: Screen capture of Animate with ports and reed valve animations

4.4. EXPANSION CAPABILITIES

Future work on the engine model is very important to this project. The basis of this project is founded on the idea that the engine model can be expanded and made more accurate in predicting results of future design changes. There are several methods in AD to facilitate this expansion. Built into the program are interfacing capabilities with Fluent and Star-CD which are commercially available CFD software packages. By putting a component into a CFD program with AD supplying the incoming conditions, and utilizing the outgoing ones, one-dimensional engine model is compatible with three-dimensional modeling software. This will give more accurate results for critical areas or areas that are difficult to model in AD alone. However, this feature is limited to only one component. Another major capability of AD is a built-in interface with MatLab, which allows AD to utilize MatLab capabilities including data acquisition. AD can utilize these capabilities to capture real time data for use in refining the model.

5. ENGINE MODELING

5.1. FINAL COMPONENT MODELING

Each engine component has a base model component that requires the input of different parameters depending on the type of component and method of modeling. Actual input values and estimated error on values are presented in APPENDIX A. This section will discuss the modeling of each component and includes a discussion on parameters that require more research to fully describe. Figure 23 shows the flow diagram for the final model, all components. ATM components represent atmospheric conditions, and were modeled as defaults: 20ºC, 1.0133 bar, pure air (no exhaust gasses in the mixture) and 50% relative humidity. They should be changed to allow the model conditions to match the conditions of the experimental data.

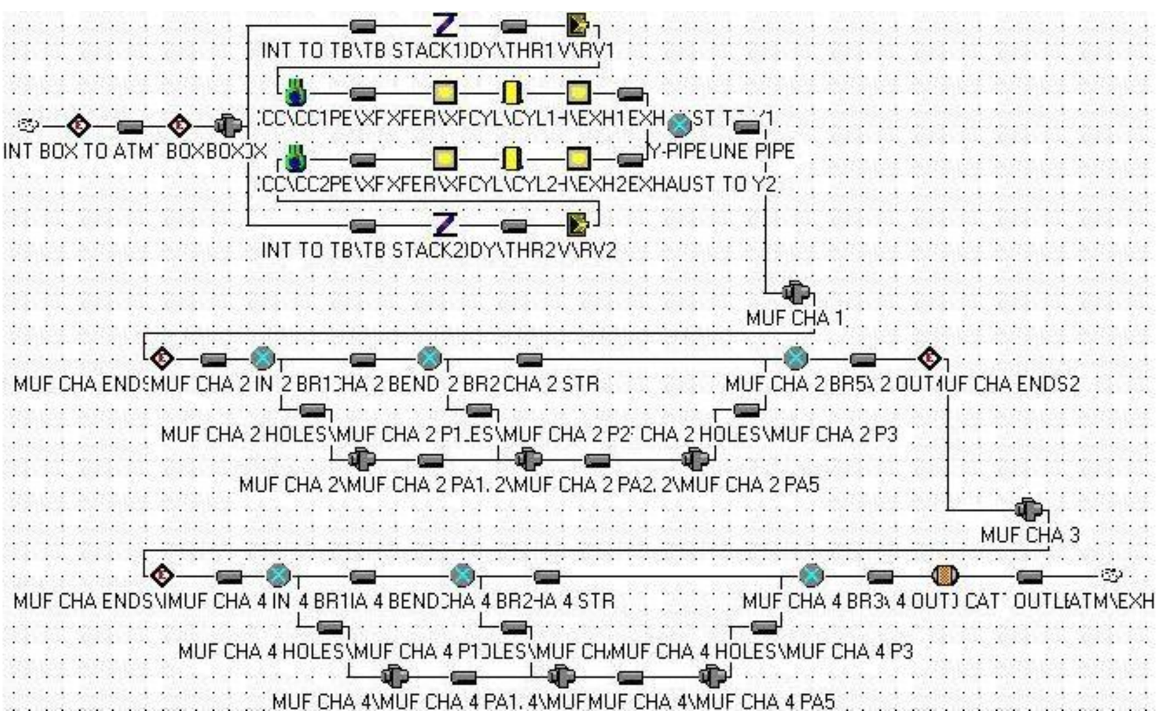

Figure 23: Full model flow diagram

5.1.1. INTAKE SYSTEM

The intake system includes the inner engine air box, the throttle bodies, and the reed valves, as shown in Figure 24. The snowmobile itself contains a second air box, used for intake silencing, though it is not feasible to have it attached to the engine during testing. For this reason, all engine data obtained will not have this component attached to the engine so the engine model does not contain this component. The air box intakes air from one location. However, there are two outputs, one for each cylinder. In order to model the intake system the following components must be used: Ambients, Ends, Plenums, Throttles, Pipes, and ReedValves. These components are connected as shown in Figure 25 for this model.

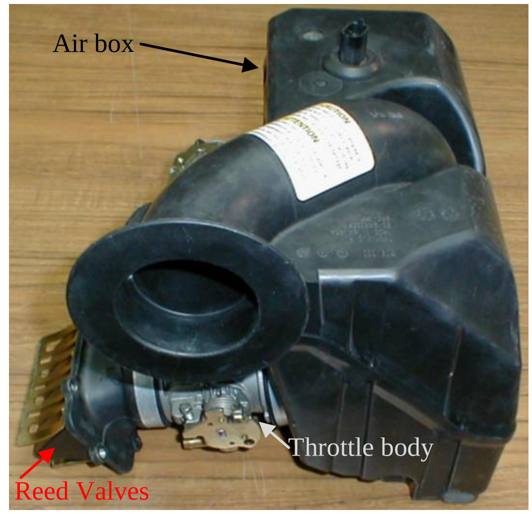

Figure 24: Intake system components

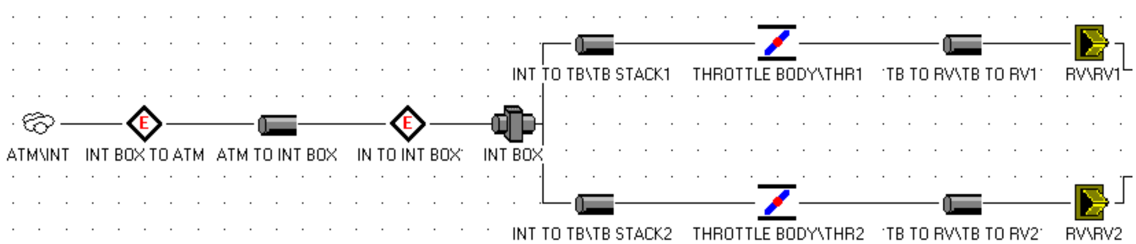

Figure 25: Intake system model interconnectivity

There are a few repeating parameters throughout the intake system. These parameters are initial gas temperature, initial gas purity, and initial pressure, which are assumed to be 20ºC, 1 (pure air, no exhaust gasses), and 1.0133bar, respectively. The initial gas temperature and initial pressure may be slightly different, especially at different ambient conditions. This is a starting point only, and will only cause the simulation to take longer if different conditions are used.

The first component in the intake system is the ATM Ambients component. As discussed previously, there are atmospheric conditions which are modeled with defaults only. Directly from the ATM is the INT BOX TO ATM End component, which is used to model the non-bell mouth condition at the intake box opening. This end condition is a plain style with the plain coefficient of discharge (CD) map provided in software package. For a more accurate model, experimentation should be done to determine a more accurate CD map.

ATM TO INT BOX is a Pipes component that is used to model the top portion of the intake box, which resembles a pipe that extends into the intake air box. The thickness of this component was measured at the entrance and exit, then averaged and assumed to be constant throughout. The entrance measured 2.5mm thick and the exit 2mm thick. The wall temperature was assumed to be approximately 20ºC, since the incoming air would cool the wall and the engine compartment would warm it up. This assumption is a starting number, and also highly dependent on ambient conditions. Research could be done to determine the correct value for a range of operating conditions. This would likely be ineffective at changing model accuracy in comparison with other parameters. The piping was divided into five sections for modeling. The sections were divided at a discontinuity in the pipe, such as at a step change in the diameter, or at the beginning or end of a bend. Each section's entrance and exit diameters and lengths were measured. The last section exits in a somewhat non-circular shape, which could be handled with the shape factor but, the out-of-roundness is so slight that it does not warrant a change in the shape factor. Errors in length and angles reflect the uncertainty of these divisions. It is difficult to obtain measurements on bends with little to no demarcation of where the bend becomes straight. This piping ends in a plain end, therefore the IN TO INT BOX End component was put in the model to account for this non-bell mouth termination of a pipe.

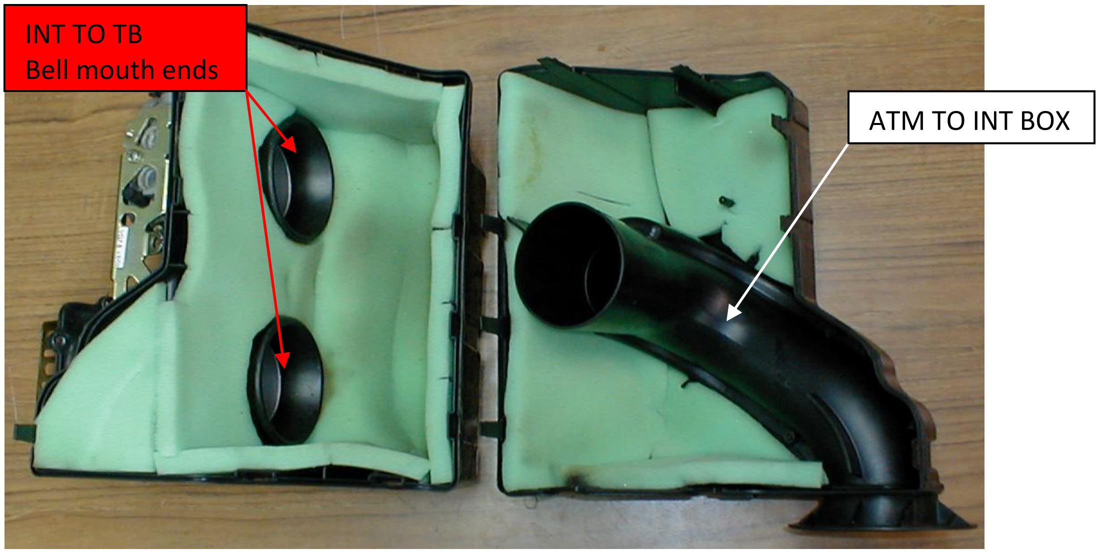

Figure 26: Inside the intake air box

The intake air box was modeled in SolidWorks to determine the surface area and volume of the Plenums component. INT BOX is a fixed volume type with a rough surface due to the lining to account for the acoustic and fluid dynamic properties. The intake air box is a good candidate for future work. This model does not delve into the intricacies of this component due to the orientations of the inlet piping to the outlet piping. A CFD model of this component would be the next step in creating a more accurate model of this component.

From the intake air box there are two bell-mouthed outlets, pictured in Figure 26. Due to the defaults of the program, these are already modeled as bell-mouths. The piping to and from the throttle bodies are modeled by INT TO TB and TB TO RV Pipes components. These components are very similar except for the dimensional values. The thickness of each pipe is estimated, since the thickness is continuously changing throughout the piping length. See Figure 27. This could be modeled more closely by use of layering on sections of the pipe. However, this would add a large number of

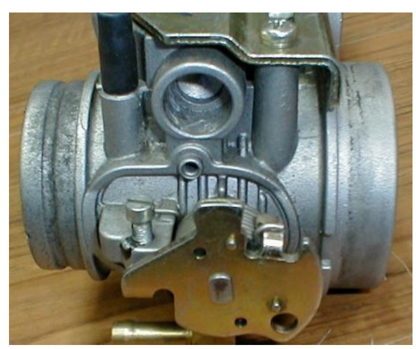

Figure 27: Throttle Body

sections to the piping and an overall complexity that would do little to make the model more accurate.

The throttle body is represented by THROTTLE BODY, which is a Throttles component. The throttle body is a butterfly style throttle, and the inflow and outflow CD map multipliers are left at unity for this model. Again, the CD map should be researched in more detail and appropriate inflow and outflow multipliers chosen. The area ratio is dependent on throttle position and ranges from 0.7786 at wide open throttle (WOT) to 0.0232 when the throttle is shut. Both numbers are derived from the throttle characteristics. At WOT the valve stem obstructs the flow, reducing the area in the throttle. The butterfly itself has a hole in it which allows air to pass when the throttle is shut, which contributes to the shut area ratio.

The last components in the intake system are the reed valves, which are modeled by ReedValves components. RV defines the shape of the reeds, ports, block, and stop. The reed pedals were assumed to be of glass fiber composition, which have default values for Young's modulus and density of $21.5 GN/m^2$ and $1.85 g/cm^3$ respectively. These values can be found experimentally or obtained form the manufacturer to improve the accuracy. RV also has the boundary conditions defined with a CD map. The software package did not come with default reed valve CD maps, therefore the global default map was used. The CD map for this component is one of the most important to determine.

5.1.2. TRANSFER SYSTEM

The transfer system consists of all components from the outlet of the reed valves up to and including the transfer ports. Figure 28 shows how this system is connected in the model. There are some repeating parameters throughout the transfer system. These parameters are initial gas temperature and initial gas pressure, which are assumed to be 30ºC, and 1.0133bar, respectively. The initial gas temperature is higher than the intake system initial temperature since it is very likely that the air would have heated up by this point in the engine.

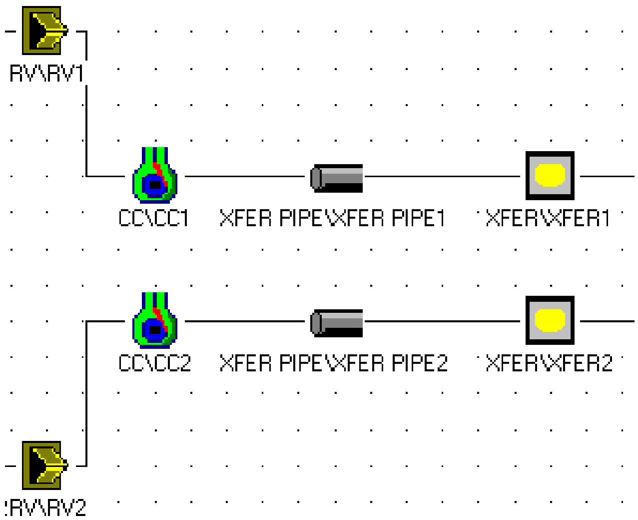

Figure 28: Transfer system model interconnectivity

The reed valve-to-crankcase transition is difficult to model without a multitude of extra components. The severity of the simplification in this model is unknown and should be researched further. Figure 29 shows the entrance into the crankcase, and the location of the reed valve. The air has several paths from the reed valve exit, as shown. However each path leads to the crankcase, therefore this area is modeled as part of the crankcase only.

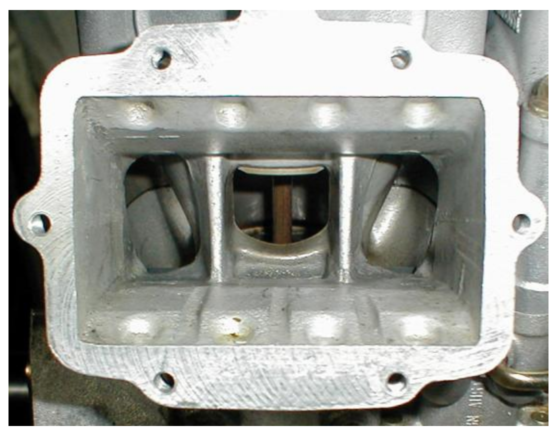

Figure 29: Intake entrance into crankcase

The two CCs shown in Figure 28 represent the Crankcases components, which includes the entire volume below the entrance of the transfer port piping and the crankshaft characteristics. Tolerances on the physical dimensions of the crankcase and crankshaft are representative of the difficulty in taking measurements. The engine can be disassembled further to verify the internal dimensions of the crankcase, as well as some particular dimensions of the crankshaft. The crankshaft clearance volume is defined as the volume up to the entrance of the transfer port piping and under the piston at BDC. This volume was measured with an oil of known specific weight, which was Valvoline

Multi-Purpose 2 Cycle Engine Oil, part number VV461. This oil was chosen due to the local availability, cost, and the readily available material safety data sheet (MSDS), which contains the density of the oil. In order to measure the volume, the container of oil was measured prior to and after filling the crankcase very carefully to prevent spilling or overfilling. The difference in the weight measurements was 14.3oz, which converts to an approximate 405.4g difference. This means that it took 405.4g of oil to fill the crankcase. With a density of 0.87g /cm^3 at 20˚C according to the MSDS (8), the crankcase clearance volume is 466cm^3. This measurement, however, did not account for the volume on the underside of the piston that was undoubtedly filled with a bubble. Therefore, a second measurement was completed on the underside of a piston, which resulted in approximately 71cm^3 more volume. Together, that creates a crankcase clearance volume of 537cm^3. The error on this measurement reflects the inaccuracy of the method, as well as the uncertainty of the density, given that ambient temperature at the time of measurement was 27˚C. However, this measurement was better than an approximation through other methods. If a more accurate measurement is desired, another method can be chosen or a more controlled measurement can be completed. The crankshaft inter-flywheel clearance value reported is used to approximate the dimension. The component is cast, and only some of the surfaces are machined to a specific dimension. The crankcase and crankshaft wall temperatures were approximated at 45˚C and 70˚C respectively. The crankcase wall temperature is lower due to coolant passages in the wall, which helps to cool it, whereas the crankshaft does not have coolant passages. These values will be higher than those of a typical carbureted two-stroke engine, due to the lack of cooling from fuel entrained in the air. These values could be experimentally found to increase accuracy of the model.

From the crankcase, there is transfer port piping and boost port piping which lead to the four transfer ports and the one boost port. In order to model these components, one Pipes component is created for both sides. XFER PIPE component is this Pipes component. In order to model the complex piping geometry, the pipes were modeled in SolidWorks, shown in Figure 30, and then broken into three sections: a straight section from the crankcase, a bend, and a small straight section into the cylinder. Each section’s entrance and exit diameters are effective diameters, based on the solid model characteristics. The length of each section is similar, as well as the bend characteristics of the second section. The tolerances indicate the uncertainty in measurements as well as the solid modeling characteristics.

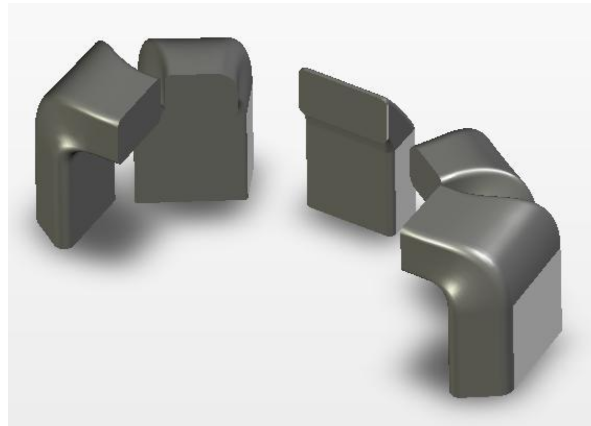

Figure 30: Solid model of transfer port piping

Each transfer pipe has similar features for each section lending to combining them into one pipe. However, they were not modeled with one pipe since the ports are modeled in a PortsSystems component that only allows one pipe to be connected with it. The XFER component is the four transfer ports and one boost port modeled together. The cylinder is symmetrical about a centerline drawn from the intake side to the exhaust side. This means that the four transfer ports are split into two of one set of physical dimensions and two of another set. Port number one represents the boost port, and due to the steep angle of entry it is much easier to measure the port piping to cylinder interfacing area and leave the entrance angle as ninety degrees. This does not affect the model, only the effective diameter and area of this port. Similarly, due to the shallow entrance angle, the first two transfer ports' physical dimensions are measured at the interface of the port piping and cylinder wall. On the third type of ports (the transfer ports closest to the exhaust side of the cylinder) it is easier to measure the actual pipe dimensions and adjust for the entrance angle. The effective areas of all ports combined determine the effective exit diameter of the XFER PIPE. The orientations of these components are critical in two-stroke engine operation. However, this program does not account for the orientation through physical geometry modeling. Instead, the entrance angles are used only to determine the effective area and diameter. The modeling of the effects of port geometry on combustion is discussed in the Cylinders modeling section following this section. The CD map used for the ports is a software package supplied map of side transfer ports from a Yamaha RS125 two-stroke engine. In order to further increase the accuracy of this model, a CD map analysis on the transfer ports should be done.

5.1.3. CYLINDERS

Among the more critical parts of this engine model are the cylinders. Experimental work reported here focuses on the Cylinders components. However, many of the input parameters for this component are assumed. The CYL component models the cylinder, cylinder head, and piston physical characteristics. In addition, the CYL component models friction, timing of combustion events, combustion, piston motion, and scavenging. The cylinder, cylinder head, and piston physical characteristics are a small portion of the inputs in the CYL component, and were obtained from the engine workshop manual (9), or from solid models from the 2006 University of Idaho senior design team head design (10).

In an attempt to categorize the combustion process, an in-cylinder pressure analysis was performed. The results of this analysis are used as inputs for the combustion characteristic modeling. The analysis, results, and inputs for the model are described in detail in Chapter 6. On a spark ignition engine, the timing of ignition can cause major changes in power output. Ignition timing was approximated using several points in the engine tuning map. Ignition timing was used in conjunction with the in-cylinder pressure data to determine the burn delay, or the time from when spark occurs to when combustion starts.

This leaves several parameters that are left unmeasured and must be assumed. These parameters are piston, liner, and head temperatures, friction, combustion efficiency, air-to-fuel ratio (AFR), fuel trapping efficiency, and scavenging. In-cylinder temperature values are difficult to measure or predict due to the volatile nature of the cylinder during combustion, and the inherent variability of these values. As such, the default program values were used: 150˚C for the liner temperature, 300˚C for the head temperature, and 250˚C for the piston temperature.

Friction in an engine is highly dependent upon the design, including the type and amount of fuel and oil used, and is a direct parasitic loss. In modeling friction for the engine, the software supplied two-stroke spark ignition model was used.

The AFR was assumed to be 13.2, which is slightly fuel-rich (an excess amount of fuel for the amount of air in the cylinder). This is done on two-stroke engines because measuring and monitoring AFR is extremely difficult, due to the short circuiting of unburned air-fuel mixtures, and it is much safer for the engine to run fuel-rich rather than fuel-lean. Fuel-lean conditions can cause extremely high combustion temperatures which may exceed the melting temperatures of internal components such as the aluminum pistons. Figure 31 is a “UI” sculpture made of pistons, some of which are great examples of what happens when the engine goes fuel lean.

Figure 31: "UI" sculpture made from ruined pistons

Combustion efficiency and fuel trapping efficiency are used to model the relationship between the amount of fuel put in the engine to the amount of energy created by the engine. Combustion efficiency was not changed from the default value of 85%, which is a reasonable value for this engine (11). Fuel trapping efficiency is typically related to scavenging on a carbureted engine. With direct injection that is not the case, and a fuel trapping efficiency must be specified. In order to make an assumption, the operation of the engine must be taken into account. At engine speeds less than approximately 2000 revolutions per minute (RPM), the engine operates in stratified combustion mode. During stratified combustion, the fuel is injected around the time the exhaust port closes. Therefore, the fuel does not have a chance to be short-circuited, which translates to a fuel trapping efficiency of unity. Above that, the engine operates in a homogeneous mode, which requires much earlier injection angles. As such, the fuel trapping efficiency would decrease considerably. Table 2 shows the assumed values for fuel trapping efficiency. The program interpolates for engine speeds not specifically defined.

Table 2: Fuel trapping efficiency assumptions

Engine Speed (RPM)	Fuel Trapping Efficiency
0	1
2000	1
9000	0.75

There are several scavenging choices that come with the software package, each of which is discussed in detail by Blair (11). Four choices stand out as most similar to the Rotax engine then any other: they are named YAM1, YAM6, YAM12 and YAM14. All are 250cm^3 cylinders with five ports designed for Schnürle-type loop scavenging. However, the YAM12, or cylinder number 12 of from Blair, most closely matches the orientation of the Rotax 593 HO engine ports (12). The scavenging curves can be found experimentally by following procedures outlined in Blair (12).

5.1.4. EXHAUST SYSTEM

The exhaust system includes the components from the exhaust ports out to the atmosphere. Figure 32 shows the connectivity of these components. For the exhaust system components the common assumptions are as follows: 250˚C for initial gas temperature, 0.2 for the initial gas purity.

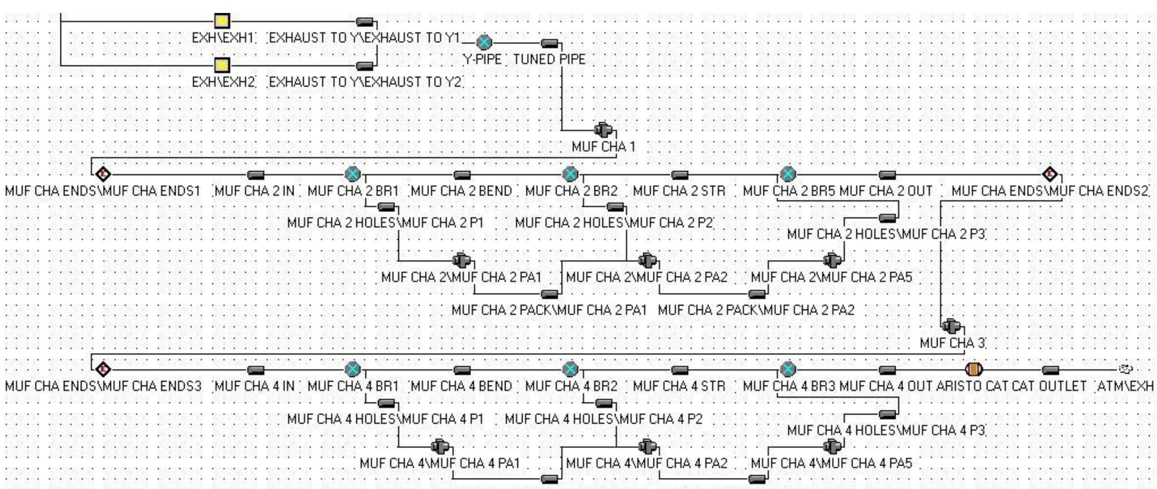

Figure 32: Exhaust system interconnectivity

Coming from the CYL components are the exhaust ports, EXH. They are modeled much like the transfer ports, using PortsSystems components. The exhaust ports consist of one large port and two symmetrical auxiliary ports. The auxiliary ports are modeled using the cylinder to auxiliary port pipe interface, and an entrance angle of ninety degrees due to the shallow angle on entrance. The large port has the RAVE power valve system (discussed in Section 2.3) associated with it, which at low engine speeds causes the exhaust port opening height to be lower than at higher engine speeds when the power valve is open. The engine speed at which the power valve opens is highly dependent on throttle position. As such, this model has the location of the power valve opening at 6500 RPM, which is approximately when it opens at WOT. This should be changed if the desired running characteristics are different than WOT. APPENDIX B shows the area of the port as the uncovered height increases for the power valve in the open and shut positions. The open heights are different based on whether or not the power valve is open or shut.

The exhaust flows from the exhaust port into piping; this piping is modeled by the EXHAUST TO Y Pipes component. This run of piping is broken into four sections, where the first section is shaped similarly to the exhaust port at the entrance and has a circular exit. The first section also accounts for the auxiliary exhaust port piping. The error associated with the length of this section is due to the curvature of the exhaust port, as well as the auxiliary exhaust ports effects on the section of the piping. The shape factor is estimated to account for the out-of-round shape and auxiliary exhaust ports. There is a discontinuity between sections one and two, due to the mating of the piping to the engine. The wall temperatures of this pipe vary significantly with engine speed and load, throttle position, and time. In order to accurately model this component, the wall temperature was input as a function of engine speed, shown in Table 3. These values closely follow the expected trend for exhaust gas temperatures under normal loading conditions.

Table 3: EXHAUST TO Y-Pipes component assumed wall temperatures

Engine Speed (RPM)	Wall temperature (˚C)
1000	150
6500	800
9000	1000

The exit diameter of section four is due to modeling the joining of the two cylinder exhausts into one. This component is known as the Y-pipe, and has a ninety degree angle between the two cylinder exhaust inlets and the angle is bisected by the outlet. The Y-pipe is modeled using a multitude of inputs. Most prominent is the Branches component Y-PIPE, which defines the relative angle between the pipes. In order to model the connection, the exit diameters of each inlet pipe must be adjusted to an effective diameter of the angled section of piping.

Exiting the Y-pipe is the tuned pipe, which is modeled as the TUNED PIPE Pipes component. These components dimensions are critical having a huge effect on engine performance, as well as the model's ability to predict the performance of the engine. The errors in the physical dimensions of each section account for dividing the complex geometry, the size and the inaccessibility of interior dimensions of the tuned pipe. The overall error is significantly less than the sum of all the individual sectional errors. The difficulty in modeling this component is that the wall temperatures of the pipe are critical to the tuned pipe operation; however, the wall temperature is highly dependent on engine speed, load, throttle position, and time. This creates a significant difficulty in modeling this component. In an attempt to closely approximate these values, the same variation with engine speed as the EXHAUST TO Y wall temperature was used, shown in Table 3.

With four chambers, the exhaust muffler is a complex part of the engine. Two of the chambers are just expansion volumes, and the other two have packing with a perforated tube for the exhaust gasses to traverse. To model this complex geometry, the muffler is broken down into the four chambers, with piping connecting each one. Each chamber was modeled approximately in SolidWorks to determine the volume and surface area of the chambers. The first chamber is only a volume, which was modeled using a

Plenums component and is named MUF CHA 1. The second chamber is the first with a perforated tube and packing material in it. To model the possible exhaust gas flow paths in this chamber, it was separated into three divisions. Each section has equal perforated pipe length and equal packing material volume. The perforated piping is split into three Pipes components of equal length; these components are named MUF CHA 2 IN, MUF CHA 2 BEND, and MUF CHA 2 STR. At the separation of each division, the piping has a branch which connects the two divisions' piping and the Pipes component representing the perforations, named MUF CHA 2 HOLES. MUF CHA 2 HOLES Pipes components are each connected to MUF CHA 2 Plenums components, which represent the volume of each division. Connecting the MUF CHA 2 Plenums components are Pipes components, which represent the restrictive characteristics of the packing material, named MUF CHA 2 PACK. The length of MUF CHA 2 HOLES is the thickness of the perforated piping. The effective diameter was approximated using the combined area of the holes in each division's perforated pipe. The number of holes in each section was approximated by estimating the surface area of each division and assuming they are approximately the same, and multiplying that by holes per unit area. The MUF CHA 2 Plenums components are assumed to be one-third the overall volume of the second chamber. The length of MUF CHA 2 PACK Pipes components is equal to that of the MUF CHA 2 HOLES. The diameter of MUF CHA 2 PACK is assumed to be related to several other component dimensions. The following equation represents that relationship (13):

$$d_p = \sqrt{\frac{d_h^2 - (d_{pp} + 2t_{pp})^2}{\rho_p}}$$

Equation 2

Where:

d_p is the diameter of MUF CHA 2 PACK

d_h is the diameter of MUF CHA 2 HOLES

d_{pp} is the diameter of the perforated pipe

t_{pp} is the thickness of the perforated pipe

ρ_p is the packing density coefficient

The packing density coefficient was assumed to be 6, which is its value for tight packing. The MUF CHA 2 OUT Pipes component was used to transition to the next chamber. Chambers three and four are modeled similarly to chambers one and two.

Attached to the outlet of the muffler is the catalytic converter. This component does not follow the typical catalytic converter physical characteristics as they are input into AD. Hence, several of the dimensions used to model the catalyst are different from the actual component to allow for the component to be modeled. The deviations are added piping to the entrance and exit and a slightly different entrance and exit diameter. Each deviation was minimized and has little to no effect on the model. Many of the physical dimensions of ARISTO CAT Catalysts component are approximated due to variations in dimensions. The CAT OUTLET Pipes component is used only to connect the catalyst to atmospheric conditions.

5.2. SIMULATION METHOD

Along with the engine model, a testing procedure must be modeled. The test procedure is defined dependent upon the desired simulation results. Test procedure options and capabilities are discussed in Chapter 4.

Of particular importance to this model is the meshing profile. As discussed in Chapter 4, AD requires at least three meshes for each component minimum. For example a pipe that is 1mm in length (as there are several in the muffler components) it is required to have meshes of 0.3mm length. This creates a number of meshes exceeding the program's limit when meshing larger components, such as the tuned pipe. In order to prevent this issue, several components need to have unique meshing profiles. The ARISTO CAT Catalysts component is not able to have its own meshing profile. Instead it uses the exhaust system meshing profile. This further compounds the existing problems with the required geometric properties of this component. In order to provide the mesh length needed for the Catalysts component, the exhaust systems mesh length was reduced to 6mm. This length would have created too many mesh nodes for the tuned pipe, so the TUNED PIPE and EXHAUST TO Y Pipes components have separate mesh profiles. Each Pipes component in the exhaust that is 1mm in length also has its own mesh profile. The intake system's mesh profile is reduced for several components, although none needed a separate mesh profile. The meshing profiles used in model are shown in APPENDIX C.

This mesh profiling method was implemented only to eliminate errors in the model. The mesh lengths possibly could be increased in certain components with little to no effect on the modeling accuracy while increasing the performance of the model by reduced simulation times. However, some mesh lengths may need to be reduced to more accurately model the component and the engine as a whole. More research is recommended in this area once the model is complete.

5.3. SIMULATION RESULTS

Every attempt at running a simulation on the engine model has ended in an error internally in the program. This is in part due to the complexity of the mode, and in part due to the values used for inputs, such as the CD maps. All items that created an error are cited in this chapter as needing further research. The engine model does provide outputs when components are removed. However, those components include the intake air box and the muffler both of which contain CD maps with the greatest uncertainties. These simulations, without the muffler and intake system, were used to discern issues with components such as the tuned pipe wall temperatures. With more research in the areas discussed in this chapter, the model will likely provide results which will be useful in further refinement of the model.

6. PRESSURE TESTS AND ANALYSIS

6.1. GATHERING DATA

In order to obtain the appropriate information for the combustion characterizing portion of the model, an analysis ending with a tabulated MFB must be performed. Experimental data are required for this analysis.

6.1.1. EQUIPMENT SETUP

In order to obtain pressure data, a Land and Sea dynamometer was attached to the crankshaft using their nine-inch dynamometer head specifically designed for smaller engines. The dynamometer was used to maintain a steady engine speed and the appropriate loading on the engine. For further discussion on the specifics of the dynamometer, see Johnson (4). The dynamometer was not used for data gathering, only to hold the engine at a set load and engine speed. Table 4 shows the operating points at which the data were taken, which correspond to the emissions testing mode points. Mode One engine speed and loading was not attainable at the time of testing, due to equipment issues with the dynamometer. The engine speed was held within 100 RPM of the target speed and the fluctuation in engine speed was due to dynamometer limitations. The torque output is approximate since the dynamometer was not appropriately calibrated. Five traces were taken at all points except 6800 RPM where six traces were taken due to signal noise. This was to ensure a good signal, and rule out the possibility of capturing an anomaly. These were taken approximately one minute apart.

Table 4: Test points for pressure data

Engine Speed (RPM)	Output Torque (ft-lbs)
6800	31
6000	22
5200	13
Idle	-

6.1.2. PRESSURE DATA

The pertinent data required are in-cylinder pressure and their corresponding crankshaft angles of rotation. In order to obtain these data, the work of a 2006 University of Idaho senior design team was consulted (10). This work specifically integrated a pressure sensor into the cylinder head design, allowing for the installation (permanent or temporary) of a pressure sensor in the cylinder head for the measuring of in-cylinder pressure data. Figure 33 and Figure 34 show the orientation of this pressure sensor within the combustion chamber. The pressure sensor used was a Kistler model 6052C pressure transducer which is capable of withstanding the range of temperatures and pressures seen in the combustion chamber of the engine, as it was designed for the engine combustion environment (14). The 6052C pressure transducer is a very small piezoelectric crystal type sensor, which produces a charge when strained (15). The overall dimensions of the pressure transducer are less than 17mm long and a little over 6mm in diameter at its widest point (14). The sensor is designed to reduce the engine vibration effects on the signal, while maintaining accuracy.

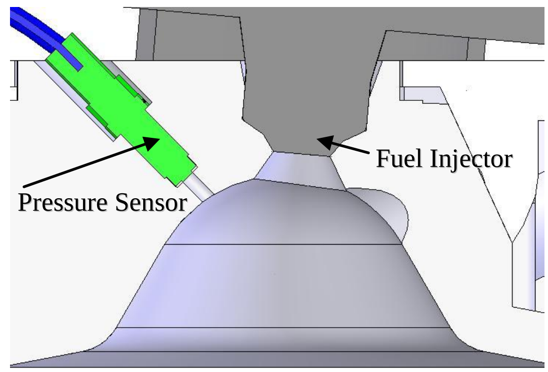

Figure 33: Cross-section of combustion chamber

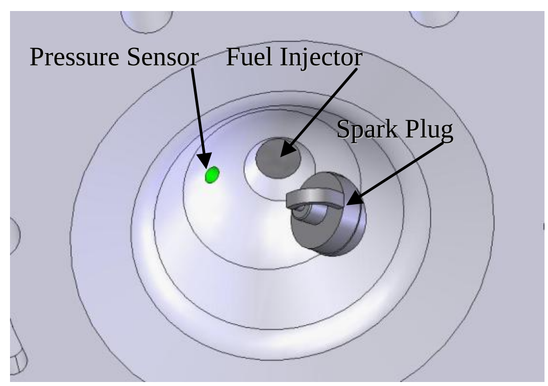

Figure 34: Underside of combustion chamber

A 1m long braided cable runs from the sensor to the charge amplifier. The charge amplifier used was a model number 422M96 from PCB Piezotronics, Inc. The charge amplifier output was routed to a power unit, model number 480B02 from PCB

Piezotronics, Inc. The signal was put into a two channel, 60MHz bandwidth oscilloscope. The oscilloscope was an Agilent 54621A, which has a floppy disk drive which saves signal traces in a CSV format. The CSV trace file contains signal conditioning characteristics and five hundred data points evenly spaced over the time span of the signal trace range. Each data point contains the voltage signal from both channels.

The pressure transducer was installed on the clutch side cylinder, for ease of installation. Each signal on the oscilloscope was carefully adjusted to ensure at least one entire cycle was captured. Figure 35 shows an example pressure trace.

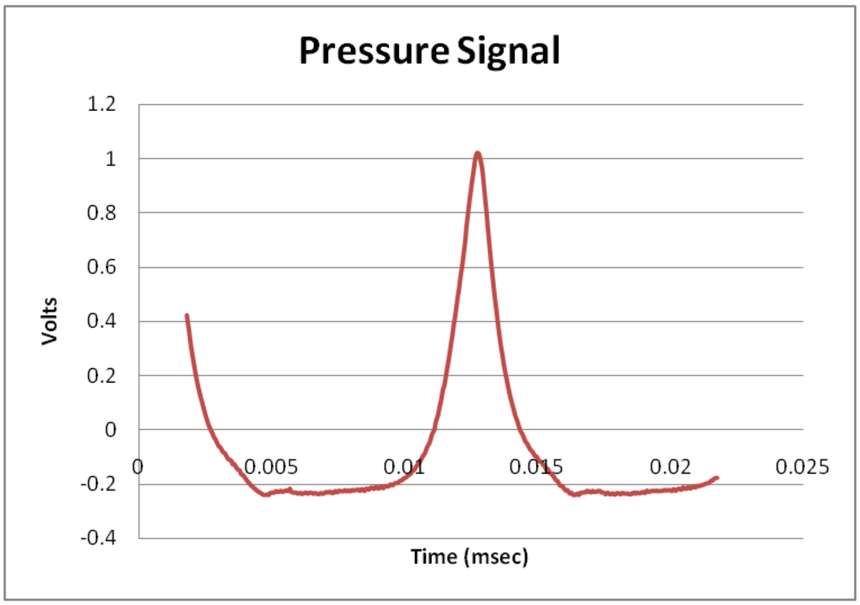

Figure 35: Example pressure trace, taken at 5200 RPM

6.1.3. VOLUME DATA

In order to obtain the volume of the cylinder at a given time, the crankshaft angle must be found. To find the crankshaft angle of rotation, the already installed crankshaft position sensor (CPS) was utilized. This device normally disrupts the signal to the engine control module (ECM) to allow the ECM to perform functions such as firing the injectors at the appropriate angle and signal ignition at the appropriate time for spark. The CPS works in tandem with a toothed flywheel, which is attached to the engine crankshaft. The toothed wheel has twenty-eight teeth, which vary in spacing between 7.5º and 15º depending on location (16). Due to the spacing of the teeth, it is assumed that the crankshaft angular speed is constant between the signals. This assumption itself introduces error due to the inherent inconsistent engine speed from combustion. Figure 36 shows an example of the signal from the CPS. The signal to the ECM is disrupted by a tooth resulting in a downward slope. The zero on this downward slope corresponds to the trailing edge of a tooth (16). For consistency, the zero on the downward slope marked "Tooth#1" in Figure 36 is defined as the number one tooth due to the relative ease with which it can be identified. The number one tooth's trailing edge passes the CPS 84º before TDC on the clutch side cylinder (16). The CPS signal was obtained via the second channel on the same oscilloscope used for pressure data.

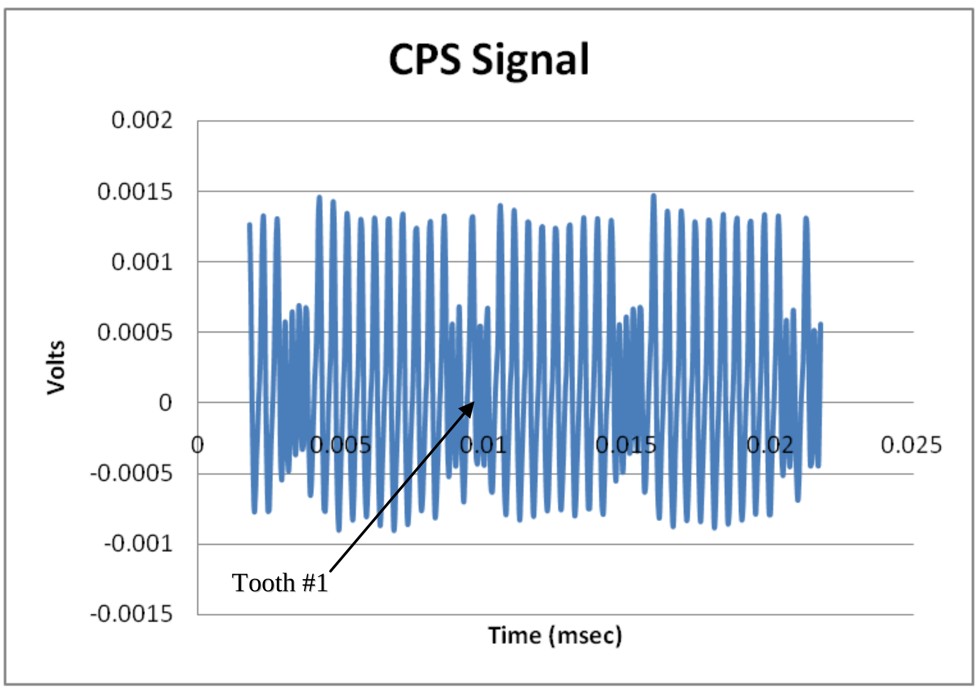

Figure 36: Example crankshaft position sensor signal, taken at 5200 RPM

6.2. ANALYSIS OF DATA

In order to define the inputs required for the model, the raw data must go through several steps. Given the format of the raw data (a CSV file), these steps were captured in an Excel spreadsheet. The Excel spreadsheet formulas are presented in APPENDIX D and the commented macro program needed to perform the calculations is included in APPENDIX E. The following sections describe the calculations performed.

6.2.1. VOLUME

The first step is to change the CPS signal to a crankshaft angle. To do this, each signal from a tooth must be identified and numbered. Once the teeth are identified and numbered, the corresponding crankshaft angle is then determined. The speed of the crankshaft is assumed to be constant between teeth and therefore the angle difference of each data point is constant between two teeth.

Once the crankshaft angle is determined for each data point received a volume of the cylinder can be calculated. This is done by the following geometric relationship:

$$V = \frac{\pi b^2}{4}\left[l - \sqrt{l^2 - \frac{s^2 \sin\theta^2}{4}} + \frac{s}{2}(1-\cos\theta)\right] + V_c$$

Equation 3

Where:

V is the volume

b is the cylinder bore of the engine

l is the connecting rod length of the engine

s is the stroke of the engine

θ is the crankshaft angle

V_c is the clearance volume of the cylinder at TDC

This calculation of volume is only an approximation of the volume at each data point. The time delay in the CPS and the assumption of constant engine speed between teeth contribute to the inaccuracy of the volume. However the error is minimal and only due to time delay effects which did not prove to be of any significance as the engine speed increased.

6.2.2. PRESSURE

Each data point has a pressure signal associated with it. This signal is a voltage and must be converted to a pressure signal. The charge amplifier has a 2.5 mV/pC nominal sensitivity, and the pressure transducer was calibrated from the factory to have approximately 21.33 pC/bar sensitivity at 200˚C (17). The sensor's size and location makes obtaining temperature readings of the sensor not feasible. As such, the factory calibration of 21.33 pC/bar at 200˚C was used in all calculations. The sensor has only a small amount of its face subject to the temperatures of combustion and coolant passages in close proximity, with coolant temperatures ranging from 38˚C to 71˚C during this testing. Those two effects lead to the assumption that this temperature is relatively close to actual sensor temperatures. The factory calibration for 23˚C temperature is a sensitivity of 21.57 pC/bar and at a temperature of 350˚C the sensitivity is 21.53 pC/bar (17). The sensitivity, while changing with temperature, is not effected significantly for the purposes of this data analysis.

The charge amplifier time constant creates an effect similar to AC coupling on an oscilloscope, which is an elimination of the DC signal. In this application, this is the pressure offset (15). Due to the operation of the charge amplifier, the offset from atmospheric is difficult to determine with complete certainty. There are multiple methods to overcome this effect. One such method is applying a known pressure value for a set crankshaft angle to determine the pressure signal offset. The difficulty in using this method for a two-stroke engine is that the pressure in the cylinder can vary significantly based on port geometry, engine speed, and crankcase geometry. For this data analysis, the pressure offset was determined through an approximation. The compression process of an engine is nearly a polytropic process which, as seen in Figure 5, is a straight line on the *log pV* plot (6). The pressure offset was approximated by determining the pressure offset which created the most linear line for the compression process. The linearity of the line was determined by a linear regression and maximizing the coefficient of regression (r^2) value. To prevent SOC or exhaust ports from effecting the linearization, the compression process was assumed to occur between 80˚BTDC to 25˚BTDC.

6.2.3. MASS FRACTION BURNED

Once the volume of the cylinder is characterized and the pressure signal is conditioned, the relationship between in-cylinder pressure and cylinder volume can be used to create the *log pV* graph. The compression and expansion processes can now be identified as the straight portions of the *log pV* plot. The SOC and EOC must be identified and the slope of the straight lines approximated.

The compression stroke was used to approximate the pressure offset, so the slope is easy to determine. However, the expansion side does not have nearly as linear a slope. This effect is magnified at low engine speeds, where thermal effects have a more significant effect on the pressure trace (6). For these reasons, the slope of the compression side is used to calculate the MFB values. The SOC manifests as a departure from a straight line at the end of the compression process. This departure is captured in the program by determining if the slope is continually diverging from the known linear slope. Similarly, the EOC signals the start of the straight line at the beginning of the expansion process. Due to the irregularity of the expansion process at varying engine speeds, EOC was found by determining when the slope returned to approximately the slope of the more linear compression process. Once SOC, EOC, and the slope of the compression process were defined, the MFB values were calculated using **Error! eference source not found.** at each data point between SOC and EOC.

6.3. RESULTS FOR USE IN MODEL

Due to the inherent irregularity of each engine cycle, the MFB curves can look significantly different, even for very similar conditions. Figure 37 shows the range of MFB curves determined at 6000 RPM engine speed. The variations can be caused by numerous effects. These effects can be typical engine effects such as misfires, incomplete burn, or inconsistent burn characteristics; or the effects can be caused by computational methods, such as the program's failure to capture SOC or EOC properly. The MFB curve that most closely resembled ideal, while still preserving the characteristics of the motor and minimizing electronic noise, were used as the input for the model at that engine speed.

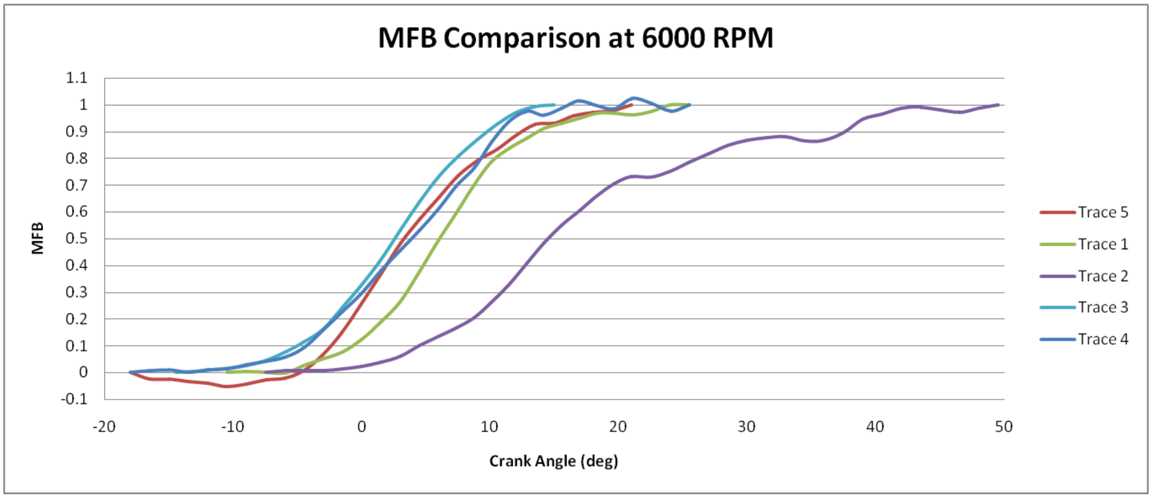

Figure 37: Comparison of mass fraction burned curves at 6000 RPM

The MFB tables, and plots, used for the model are shown in APPENDIX F. Each MFB table is modeled at a particular engine speed. The model will interpolate between the known engine speeds to determine the values for the unknown engine speeds. The errors in the MFB can be attributed to the errors in volume and pressure calculations, as discussed in the previous sections, and the phasing of the two. The phasing of the two signals was minimized by the use of a two channel oscilloscope. If the phasing were significant, the *log pV* graphs would have shown variations from the expected ideal, which was not the case for any of the data obtained. The errors in the MFB analysis are well within the expected variances between each combustion cycle.

7. FUTURE WORK

The work described in this paper is only the beginning of the model, and should be used as the basis for future modeling. There are many different changes that could be made to this model, depending on its intended use. No matter what changes are made, every verification opportunity should be attempted and documented for future use. In order for this research to benefit the UICSC team, further research is needed. The following is a synopsis of what can be done with this model as well as how to utilize the model in its current and future states.

7.1. MAKING THE MODEL QUICKER

For the purpose of quick qualitative analysis of a design idea, the model can be simplified. In order to do this, the number of mesh points can be considerably reduced. One easy way to accomplish this would be to remove the exhaust muffler, and replace it with a highly restrictive pipe that would approximate a flow hindrance comparable to the muffler. Note that when replacing a component such as the muffler, the simulation may not be able to accurately predict the engine behavior. However, for quick analysis this should not be significant. Another option would be to increase the mesh sizes in each component to the largest feasible. Experimentation can be done to determine when mesh size will hinder simulation performance significantly. Also note that each component is required to have a minimum of three meshes for AD to work. Making the model quicker will provide a very rough idea of engine performance changes during brainstorming of design changes, but should not be used as the basis of a design decision.

7.2. MORE ACCURATE MODEL

In order to make the model more accurate, several routes can and should be taken, such as detailed analysis of CD maps on the following components: Ends, ReedValves, PortsSystems, and Throttles. Each component’s inflow and outflow CD map multipliers can be determined as well. The supplied documentation from OPT provides detailed procedures on how to perform a CD map analysis. The AD software package even includes a program to convert the raw data obtained to an appropriate CD map for use in the model. Once detailed CD map analyses are completed, then the errors beyond the CD maps, if any, must be corrected.

Once the model simulation provides results without errors, many more opportunities for increasing model accuracy become available. Experimentation to measure temperatures and pressures at several locations in each component can be performed. These values can then be compared to the simulation's mesh values. Depending upon the location, parameter modeled, and parameter type, the measured value can be a steady state average or a transient measurement (compared to crank angle) such as an average exhaust gas temperature in the Y-pipe (steady state) or in-cylinder pressure (transient).

CFD modeling is another method of improving accuracy. One component can be replaced by a CFD model from Fluent software. This should be utilized on a component with complex geometries or extremely sensitive parameters. Two such components are the intake air box and the muffler. The entire exhaust system could be modeled, except for the catalyst, in a CFD model. CFD modeling requires that enough computing power, as well as either Star-CD or Fluent (both are commercially available CFD modeling software packages) be installed on the same computer as AD.

7.3. VERIFICATION OF MODEL

As discussed before, this model will not be of much use if it is not continually updated and changed. At every opportunity any engine design change must be modeled. Once this change is modeled, the simulation results should be compared to the experimental results. If a discrepancy is noted, then an analysis should be done to determine where the discrepancy lies and what can be done to reduce or eliminate it. There are several parameters in the software, while not discussed in this paper specifically, which can be used to correct items, such as assumed heat transfer rates.

These factors should not be overlooked but, changes to them must be justified. It is relatively easy to create a model that has simulation results matching experimental data, although it may not be able to predict a change in engine behavior, which is the overall goal of this model.

7.4. RECOMMENDATIONS FOR FUTURE WORK

After completing the model and verifying its accuracy, the UICSC team should use this model as a tool for predicting the performance effects of future modifications to the GDI two-stroke engine. Not only will this provide a solid basis for design change choices, this will also help to further refine the model. In order to facilitate this, all engine changes should be modeled and the prediction capabilities of this engine model verified. If the model predicts changes that are not seen in experimental data, then an analysis must be done to determine why. This will not only help to refine the model of the changes, but may also result in a base model change that makes the model more closely resemble the original engine as well as more accurately predict future modeling.

In order for the UICSC team to obtain the most benefit from the work presented in this paper, the following should be closely emulated. Once the model is complete any changes from the 2007 CSC entry should be modeled. The simulation results should be compared to experimental results from the changes. There will most likely be discrepancies. These should be documented in detail for use in future modeling. If the reason for the discrepancies is apparent then a change in the model should be performed and evaluated. If no reason for discrepancies is apparent then the data can be used at a later time to evaluate model changes. The overall goal is to find the source of differences in the simulated and actual engine response to changes, then correct those changes. In evaluating the corrections, every engine design revision should be evaluated to determine if multiple sources exist as explanations to discrepancies. Over time, this process will refine the model more and more, and small changes in design will provide the most significant amount of information concerning the sources of discrepancies.

8. CONCLUSION

The UICSC team has many years of successful engineering in its future. In order to facilitate this, all tools available must be implemented to help guide the decision making process in design changes. This project's overall goal was to lay the foundation for utilizing one such tool: computer modeling of the engine. The model described in this paper is the foundation of a model capable of predicting the engine's response to a design change.

The model was compiled in OPTIMUM Power Technology's Automated Design software package. This software was chosen based on conditions existing at the time of this project. As discussed in this paper, KIVA is another viable option in the future for use in more detailed research into the GDI engine, including fluid dynamics for in-cylinder characteristics.

Like many before and many after it, this model requires future work to ensure its successful implementation. As such, the recommended future work is not a suggestion on where to take this model. Instead, it directly points out areas that need further research in order to obtain a model adequately capable of predicting the engine's response to design changes.

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APPENDIX A – Model input values and estimated errors

All distance, area and volume values are in mm, mm^2, or cm^3 respectively, unless otherwise noted. Any value not stated in this list or in Chapter 5 is the program default.

Table 5: Model inputs and associated tolerances

Component	Sub	Parameter	Value	Tol
ATM TO INT BOX	Section 1	Length	25	1
		Entrance Diameter	63	1
		Exit Diameter	63	1
	Section 2	Length	55	5
		Entrance Diameter	60	2
		Exit Diameter	60	2
		Bend Angle	50	2
		Bend Radius	65	5
	Section 3	Length	50	2
		Entrance Diameter	60	2
		Exit Diameter	60	2
	Section 4	Length	100	5
		Entrance Diameter	60	2
		Exit Diameter	60	2
		Bend Angle	90	1
		Bend Radius	65	5
	Section 5	Length	70	1
		Entrance Diameter	60	2
		Exit Diameter	69	2
		Shape Factor	1	0.01
INT BOX	-	Volume	5,500	500
		Surface Area	210,000	2,000
INT TO TB	Section 1	Length	42	1
		Entrance Diameter	55	1
		Exit Diameter	46	1
TB TO RV	Section 1	Length	51	1
		Entrance Diameter	46	1
		Exit Diameter	48	1
	Section 2	Length	50	2
		Entrance Diameter	48	1
		Exit Diameter	57.7	2
		Shape Factor	1.2	0.2

RV	Petal	Length	43	1
		Width	21	1
		Thickness	0.5	0.1
	Port	Length	36	1
		Width	18	1
		Radius	1	0.5
		Position	4	0.5
	Block	Height	40	1
		Width	66	2
		Radius	5	1
		Angle	27	1
	Stop Plate	Height	15	1
		Length	37	1
		Spacing	0	0.1
CC	Crankcase	Diameter	135	10
		Width	60	2
		Clearance Volume	537	30
	Crankshaft	Stroke	73	0.5
		Diameter	120	5
		Width	21	1
		Clearance	11	2
XFER PIPE	Section 1	Length	30	2
		Entrance Diameter	56.7	2
		Exit Diameter	52.3	2
		Shape Factor	2.606	0.2
	Section 2	Length	10	3
		Entrance Diameter	52.3	2
		Exit Diameter	47.8	2
		Bend Angle	90	1
		Bend Radius	6	1
		Shape Factor	4	0.5
	Section 3	Length	5	2
		Entrance Diameter	47.8	2
		Exit Diameter	49.5	2
		Shape Factor	1.63	0.2
XFER	Port 1	Height	16	1
		Angle	90	5
		Width	28	1
		Open Fillet Radius	2	0.5
		Full Fillet Radius	2	0.5
	Port 2	Height	16	1
		Angle	90	2
		Width	23	1
		Open Fillet Radius	2	0.5

		Full Fillet Radius	2	0.5
	Port 3	Height	16	1
		Angle	65	2
		Width	27	1
		Open Fillet Radius	4	1
		Full Fillet Radius	2	0.5
CYL	Cylinder	Bore	72	0.5
		Head Surface Factor	1.272	0.1
		Clearance Volume	26.23	0.5
		Squish Clearance	0.5	0.5
		Con-Rod Length	132	1
	Piston	Height	77	1
		Compression Height	33	1
		TDC Clearance	1	0.5
EXH	Port 1	Height	40	1
		Angle	90	5
		Width	55	3
		Open Flat Radius	35	5
		Open Fillet Radius	15	5
		Full Flat Radius	30	5
		Full Fillet Radius	11	5
	Port 2	Height	15	1
		Angle	90	5
		Width	13	1
		Open Fillet Radius	2	0.5
		Full Fillet Radius	2	0.5
EXHAUST TO Y	Section 1	Length	60	5
		Entrance Diameter	47.7	2
		Exit Diameter	44	1
		Shape Factor	1.2	0.1
	Section 2	Length	15	2
		Entrance Diameter	45	1
		Exit Diameter	45	1
	Section 3	Length	25	5
		Entrance Diameter	45	1
		Exit Diameter	45	1
		Bend Angle	45	1
		Bend Radius	30	5
	Section 4	Length	60	5
		Entrance Diameter	45	1
		Exit Diameter	47.1	2
		Shape Factor	0.99	0.01
TUNED PIPE	Section 1	Length	75	3
		Entrance Diameter	66.6	2

		Exit Diameter	67	2
	Section 2	Length	32	5
		Entrance Diameter	67	2
		Exit Diameter	69	2
		Bend Angle	15	5
		Bend Radius	120	10
	Section 3	Length	45	3
		Entrance Diameter	69	2
		Exit Diameter	73	2
	Section 4	Length	75	5
		Entrance Diameter	73	2
		Exit Diameter	80	2
		Bend Angle	25	5
		Bend Radius	175	10
	Section 5	Length	20	3
		Entrance Diameter	80	2
		Exit Diameter	83	2
	Section 6	Length	40	5
		Entrance Diameter	83	2
		Exit Diameter	85	2
		Bend Angle	13	5
		Bend Radius	175	10
	Section 7	Length	40	5
		Entrance Diameter	85	2
		Exit Diameter	90	2
		Bend Angle	20	5
		Bend Radius	115	10
	Section 8	Length	50	5
		Entrance Diameter	90	2
		Exit Diameter	98	2
		Bend Angle	45	5
		Bend Radius	65	10
	Section 9	Length	140	5
		Entrance Diameter	98	2
		Exit Diameter	126	2
		Bend Angle	75	5
		Bend Radius	110	10
	Section 10	Length	150	5
		Entrance Diameter	126	2
		Exit Diameter	145	2
		Bend Angle	90	5
		Bend Radius	95	10
	Section 11	Length	50	3
		Entrance Diameter	145	2

		Exit Diameter	145	2
	Section 12	Length	55	5
		Entrance Diameter	145	2
		Exit Diameter	145	2
		Bend Angle	10	5
		Bend Radius	310	15
	Section 13	Length	100	5
		Entrance Diameter	145	2
		Exit Diameter	130	2
		Bend Angle	30	5
		Bend Radius	190	15
	Section 14	Length	130	3
		Entrance Diameter	130	2
		Exit Diameter	83	2
	Section 15	Length	50	5
		Entrance Diameter	83	2
		Exit Diameter	67	2
		Bend Angle	15	5
		Bend Radius	190	15
	Section 16	Length	60	3
		Entrance Diameter	67	2
		Exit Diameter	40	2
	Section 17	Length	60	3
		Entrance Diameter	40	2
		Exit Diameter	40	2
MUF CHA 1	-	Volume	5,000	200
		Surface Area	243,000	5,000
MUF CHA 2	-	Volume	1,230	50
		Surface Area	40,000	1,000
MUF CHA 3	-	Volume	3,900	100
		Surface Area	211,000	5,000
MUF CHA 4	-	Volume	1,120	50
		Surface Area	37,000	1,000
MUF CHA 2 IN	Section 1	Length	56	5
		Entrance Diameter	58	2
		Exit Diameter	58	2
		Bend Angle	55	5
		Bend Radius	60	5
MUF CHA 2 Bend	Section 1	Length	52	5
		Entrance Diameter	58	2
		Exit Diameter	58	2
		Bend Angle	50	5
		Bend Radius	60	5
	Section 2	Length	4	1

		Entrance Diameter	58	2
		Exit Diameter	58	2
MUF CHA 2 STR	Section 1	Length	56	2
		Entrance Diameter	58	2
		Exit Diameter	58	2
MUF CHA 2 OUT	Section 1	Length	1	1
		Entrance Diameter	58	2
		Exit Diameter	58	2
MUF CHA 2 HOLES	Section 1	Length	1	0.1
		Entrance Diameter	60	5
		Exit Diameter	60	5
		Shape Factor	33.7	5
MUF CHA 2 PACK	Section 1	Length	1	0.1
MUF CHA 4 IN	Section 1	Length	25	5
		Entrance Diameter	58	2
		Exit Diameter	58	2
	Section 2	Length	35	5
		Entrance Diameter	58	2
		Exit Diameter	58	2
		Bend Angle	35	5
		Bend Radius	60	5
MUF CHA 4 Bend	Section 1	Length	60	5
		Entrance Diameter	58	2
		Exit Diameter	58	2
		Bend Angle	60	5
		Bend Radius	60	5
MUF CHA 4 STR	Section 1	Length	60	5
		Entrance Diameter	58	2
		Exit Diameter	60	2
MUF CHA 4 OUT	Section 1	Length	1	1
		Entrance Diameter	60	2
		Exit Diameter	72.9	2
MUF CHA 4 HOLES	Section 1	Length	1	0.1
		Entrance Diameter	69.8	5
		Exit Diameter	69.8	5
		Shape Factor	33.7	5
MUF CHA 4 PACK	Section 1	Length	1	0.1
ARISTO CAT	-	Canister Length	61	5
		Canister Diameter	73	2
		Inlet Length	14	10
		Inlet Diameter	72.9	3
		Exit Length	14	5
		Exit Diameter	72.9	3
		Catalyst Length	60	2

		Catalyst Diameter	72.8	3
		Catalyst Dist	0.1	0.1
		Cell Density	105	10
		Void Fraction	0.68	0.2
		Shell Thickness	1.5	0.5
		Matting Thickness	0.1	0.1
		Substrate Thickness	0.165	0.1
		Wash Thickness	0.1	0.1
CAT OUTLET	Section 1	Length	1	1
		Entrance Diameter	72.9	3
		Exit Diameter	72.9	3

APPENDIX B – Area Table for RAVE power valve open

Table 6: Area table for RAVE open

Uncovered height (mm)	Uncovered area (mm^2)
0	0
1	11.11
2	31.28
3	57.21
4	87.7
5	122.01
6	159.67
7	200.29
8	243.58
9	289.2
10	336.74
11	385.9
12	436.41
13	488.05
14	540.64
15	593.99
16	647.96
17	702.4
18	757.15
19	812.09
20	867.09
21	921.97
22	976.57
23	1030.68
24	1084.1
25	1136.6
26	1187.97
27	1238.08
28	1286.81
29	1334.03
30	1379.62
31	1423.42
32	1465.26
33	1504.94
34	1542.22
35	1576.83
36	1608.42
37	1636.52
38	1660.45

39	1679.09
40	1689.37

Table 7: Area table for RAVE shut

Uncovered height (mm)	Uncovered area (mm^2)
0	0
1	11.11
2	31.28
3	57.21
4	87.7
5	122.01
6	159.67
7	200.29
8	243.58
9	289.3
10	337.24
11	387.22
12	439.09
13	492.29
14	544.8
15	596.16
16	646.27
17	695
18	742.22
19	787.81
20	831.61
21	873.45
22	913.13
23	950.41
24	985.02
25	1016.61
26	1044.71
27	1068.64
28	1087.29
29	1097.56

APPENDIX C – Meshing Profiles for Testing of Model

Table 8: Meshing Profiles

Meshing Profile	Mesh Length(mm)
INTAKE SYSTEM	10
TRANSFER SYSTEM	20
EXHAUST SYSTEM	6
EXHAUST TO Y	20
TUNE PIPE	20
MUF CHA 2 OUT	0.3
MUF CHA 2 P	0.3
MUF CHA 2 PACK	0.3
MUF CHA 4 OUT	0.3
MUF CHA 4 P	0.3
MUF CHA 4 PACK	0.3
CAT OUTLET	0.3

APPENDIX D – Excel Spreadsheet Equations for Pressure Calculations

Sheet 1 (“Data” sheet) –

- Columns A-C are the data from the CSV files
- Column F is inputs from user for specific application
- Tolerances used to determine specific characteristics

	A	B	C	D	E	F	G
1	x-axis	1	2		Engine Inputs:		
2	second	Volt	Volt		Bore:	7.2	cm
3	0.0018	0.4227969	0.001268		Stroke:	7.3	cm
4	0.00184	0.3927188	0.0008305		Con Rod Length:	13.2	cm
5	0.00188	0.3661563	4.922E-05		CC Volume	26.23	cm^3
6	0.00192	0.334125	-0.0004742		Ignition timing:	13	°BTDC
7	0.00196	0.3044375	-0.0007711				
8	0.002	0.2806094	-0.00065				
9	0.00204	0.2571719	-0.0003922				
10	0.00208	0.2349063	-0.0001695		Pressure Correction Factor:	21.33	pC/bar
11	0.00212	0.2095156	5.703E-05		Charge Amp Correction Factor:	2.5	mV/pC
12	0.00216	0.1888125	0.0002523				
13	0.0022	0.1700625	0.0006508				
14	0.00224	0.1517031	0.0011664				
15	0.00228	0.1356875	0.0013266		Tolerances:		
16	0.00232	0.1196719	0.0009203		Tooth:	3	
17	0.00236	0.1056094	8.828E-05		SOC slope:	0.2	
18	0.0024	0.0919375	-0.0004625		SOC Number:	10	
19	0.00244	0.0751406	-0.0007711		EOC Slope:	0.013	

Sheet 2 (“Calc” sheet) calculations only –

- The following table describes the formulas used in Calc
- # in row values means it is the same row number
- E2 is determined via the Calculations Macro for offset of first tooth
- I2 is determined via the Calculations Macro, as described in Section 6.2 for offsetting pressure
- P2 thru P5 are Engine Info converted to desired units from Sheet 1
- Column N is used for pasting the information on Sheet 5

Table 9: Calc sheet formulas

Column	Equation
A	=IF(AND(Data!C(#-1)>0,Data!C(#+1)<0),"Zero","Not")
B	=IF(AND(ABS(Data!C#)<ABS(Data!C(#-1)),ABS(Data!C#)<ABS(Data!C(#+1))),"Zero","Not")
C	=IF(AND(B#="Zero",A#="Zero"),Data!C#,"")
D	Determine from Calculations Macro (Described in APPENDIX E)
E	=IF(AND(A#="Zero",B#="Zero"),MOD(COUNT(E(#-1):E(#-1))-$E$2,28)+1,"")
F	Determine from Calculations Macro (Described in APPENDIX E)
G	=IF(F#="","",PI()/4*$P$2^2*(($P$3/2+$P$4)-(COS(RADIANS(F#))*$P$3/2)-SQRT($P$4^2-(SIN(RADIANS(F#))*$P$3/2)^2))+$P$5)
H	=IF(G#="","",LOG(G#))
I	=IF(G4="","",Data!B4/Data!$G$11/Data!$G$10*14503.77+$I$2)
J	=IF(OR(I#="",I#<=0),"",LOG(ABS(I#)))
K	=IF(F#="","",SLOPE(J(#-10):J#,H(#-10):H#))
L	=IF(TYPE(K#)=1,K#,"")
M	=IF(AND(NOT(F#=""),F#>=Outputs!$B$4,F#<=Outputs!$C$4),(I#^(1/ABS(Outputs!$B$6))*G#-Outputs!$B$3^(1/ABS(Outputs!$B$6))*Outputs!$B$2)/(Outputs!$C$3^(1/ABS(Outputs!$B$6))*Outputs!$C$2-Outputs!$B$3^(1/ABS(Outputs!$B$6))*Outputs!$B$2),"")
N	=Calc!M#

Sheet 3 (“CPS Info” sheet) –

- This sheet is used for referencing for CPS tooth number to angle conversion
- Column A is tooth number
- Column B is difference in angle between teeth
- Column C is angle of tooth (degrees after TDC, and all referenced off tooth number 1)

	A	B	C
1	1	15	276
2	2	7.5	283.5
3	3	7.5	291
4	4	15	306
5	5	15	321
6	6	15	336
7	7	15	351
8	8	15	366
9	9	15	381
10	10	15	396
11	11	15	411
12	12	15	426
13	13	7.5	433.5
14	14	7.5	441
15	15	7.5	448.5
16	16	7.5	456
17	17	15	471
18	18	15	486
19	19	15	501
20	20	15	516
21	21	15	531
22	22	15	546
23	23	15	561
24	24	15	576
25	25	15	591
26	26	15	606
27	27	7.5	613.5
28	28	7.5	621

Sheet 4 (“Graphs” sheet) –

- This sheet contains:
    - Graph of CPS and Pressure trace, as seen on the oscilloscope
    - Graph of Volume vs. crankshaft angle
    - Graph of Pressure vs. crankshaft angle
    - Graph of Pressure vs. Volume on log-log scale
- This sheet is used solely for verification that program is giving expected trends

Sheet 5 (“Outputs” sheet) –

- This sheet contains the expected output values for use in determining AD inputs
- B8=Data!F6, which is a user input
- B10=B4-B8
- All other values are from Calculations macro
- Column A and B rows 14 and up are copied from Sheet 2
- Included on this sheet is also a graph of MFB
- Units: Volumes are in^3, Pressures in psi; and angles in degrees

	A	B	C
1		SOC value	EOC value
2	Volume:	1.785108288	2.98635192
3	Pressure:	229.2283776	229.5435687
4	Angle:	-10.25	28.5
5			
6	n:	-1.186652113	
7			
8	Ignition timing:	-13	
9	Combustion Duration:	38.75	
10	Burn Delay:	2.75	
11			
12			
13	Crank angle	MFB	
14	-10.25	0	
15	-9	0.005180294	
16	-7.75	0.008081081	

APPENDIX E – Commented Calculations Macro (modified to fit on pages)

```
Sub Calculations()
' Calculations Macro
' This macro does all the calculations items in one step
' Keyboard Shortcut: Ctrl+m
' This section is for defining the Column D cells, which are the number of divisions
      between the known tooth cells
  Worksheets("Calc").Activate
  For Each C In Worksheets("Calc").Range("D3:D502").Cells
    upcount = 0  ' Variable Setup
    downcount = 0
    Do  'Do loop to determine # of cells until defined tooth in up direction
      If Not (C.Offset(upcount, -2).Value = "Not") And Not (C.Offset(upcount, -
      3).Value = "Not") Then
        Exit Do
      Else
        upcount = upcount - 1
      End If
    Loop
    Do  'Do loop to determine # of cells until defined tooth in down direction
      If Not (C.Offset(downcount, -2).Value = "Not") And Not (C.Offset(downcount, -
      3).Value = "Not") Then
        Exit Do
      Else
        downcount = downcount + 1
      End If
    Loop
    If upcount = 0 And downcount = 0 Then
```

```
            C.Value = 0  'Define the current cell value to be zero if it is a tooth
        Else
            C.Value = Abs(upcount) + Abs(downcount) 'Define the current cell value to be
            the total count
        End If
    Next

'This section will define the FIRST #1 tooth
    toothcount = 0 'Variable setup
    tolerance = Worksheets("Data").Range("F16").Value 'Tolerance on the mode
    testvalue = Worksheets("Calc").Cells(503, 4).Value - tolerance
    For Each C In Worksheets("Calc").Range("D3:D502").Cells
        If C.Value = 0 Then 'Determine if this cell is a tooth
            toothcount = toothcount + 1
        Else
            GoTo Line1
        End If
        If toothcount < 3 Then 'ignore the first two tooth (prevents comparing to numbers
            before the first tooth)
            GoTo Line1
        Else
            below = C.Offset(1, 0).Value
            above = C.Offset(-1, 0).Value
            twobelow = C.Offset(2 + below, 0).Value
            twoabove = C.Offset(-2 - above, 0).Value
            If above >= testvalue And twoabove < testvalue And below < testvalue And
            twobelow < testvalue Then
                Exit For
            Else
                GoTo Line1
```

```
            End If
        End If
Line1:
    Next
    Worksheets("Calc").Cells(2, 5).Value = toothcount - 1

'This section will assign the crank position angle to each cell "throwing out" before the
            first tooth and after the last tooth
    toothcount = 0
    For Each C In Worksheets("Calc").Range("F3:F502").Cells
        If Not (C.Offset(0, -3).Value = "") Then
            toothcount = toothcount + 1
            toothnum = C.Offset(0, -1).Value
            beforeangle = Worksheets("CPS Info").Cells(toothnum, 3).Value
            afterangle = Worksheets("CPS Info").Cells((toothnum Mod 28) + 1, 3).Value
            anglediff = afterangle - beforeangle
Mod360:
            If beforeangle >= 360 Then beforeangle = beforeangle - 360
            If afterangle >= 360 Then afterangle = afterangle - 360
            If anglediff < 0 Then anglediff = anglediff + 360
            If beforeangle >= 360 Then: GoTo Mod360
            If afterangle >= 360 Then: GoTo Mod360
            If anglediff < 0 Then: GoTo Mod360
            C.Value = beforeangle
        ElseIf toothcount = 0 Or toothcount >= Worksheets("Calc").Cells(503, 3).Value
            Then
            C.Value = ""
            GoTo Line2
        Else
            C.Value = C.Offset(-1, 0).Value + anglediff / C.Offset(0, -2).Value
```

```
        End If
Convert:
            If C.Value >= 180 Then C.Value = C.Value - 360
            If C.Value >= 180 Then: GoTo Convert
Line2:
    Next

' This section determines the best pressure offset by trying to linearize the compression
            stroke.
    Worksheets("Calc").Range("I2").Value = 0
    If Worksheets("Calc").Range("I503").Value < 0 Then 'Offset to prevent negative
            absolute pressures
        Worksheets("Calc").Range("I2").Value = Worksheets("Calc").Range("I2").Value -
            Worksheets("Calc").Range("I503").Value + 0.05
    End If
    num = 0
    For rownum = 3 To 502 'Define the range of cells that are the compression cycle for
            the first (or only) cycle
        If Worksheets("Calc").Cells(rownum, 6).Value > -81 And
            Worksheets("Calc").Cells(rownum, 6).Value < -79 And num = 0 Then
            first = rownum - 503
            num = 1
        End If
        If Worksheets("Calc").Cells(rownum, 6).Value > -26 And
            Worksheets("Calc").Cells(rownum, 6).Value < -24 And num = 1 Then
            last = rownum - 503
            num = 2
        End If
    Next rownum
    Worksheets("Calc").Range("J503:N507").FormulaArray = _
```

```
        "=LINEST(R[" + Format(first) + "]C:R[" + Format(last) + "]C,R[" + Format(first) +
            "]C[-2]:R[" + Format(last) + "]C[-2],TRUE,TRUE)"
Pressure:
    stepchange = 0.1 'Step change in pressure to optimize offset
    rsquarelast = Worksheets("Calc").Range("J505").Value
    Worksheets("Calc").Range("I2").Value = Worksheets("Calc").Range("I2").Value +
            stepchange
    If Worksheets("calc").Range("J505").Value < rsquarelast Then
        Worksheets("Calc").Range("I2").Value = Worksheets("Calc").Range("I2").Value -
            stepchange
    Else
        rsquarelast = Worksheets("Calc").Range("J505").Value
        GoTo Pressure:
    End If

' This section will add the pV diagram, as well as the log p-log V plot
    For rownum = 3 To 502
        If Worksheets("Calc").Cells(rownum, 10).Value = "" Or
            Worksheets("Calc").Cells(rownum, 10).Value <= 0 Then
            xstring1 = "=Calc!G" + Format(rownum + 1)
            ystring1 = "=Calc!I" + Format(rownum + 1)
        Else
            xstring = xstring1 + ":G" + Format(rownum)
            ystring = ystring1 + ":I" + Format(rownum)
        End If
    Next rownum
    Worksheets("Graphs").ChartObjects("Chart 6").Activate
    ActiveChart.SeriesCollection(1).XValues = xstring
    ActiveChart.SeriesCollection(1).Values = ystring
    Worksheets("Calc").Activate
```

```
'This section determines the Start of Combustion
   rownum = last + 503 'Start at 25 deg BTDC
   tol = Worksheets("Data").Range("F17") 'Tolerance on difference of slope compared to
         n
   counter = 1 'setup on counter
SOC:
   testslope = (Worksheets("Calc").Cells(rownum, 10) -
         Worksheets("Calc").Cells(rownum + 1, 10)) /
         (Worksheets("Calc").Cells(rownum, 8) - Worksheets("Calc").Cells(rownum + 1,
         8))
   If Abs(testslope - Worksheets("Outputs").Range("B6").Value) < tol Then
      counter = 1
      rownum = rownum + 1
      GoTo SOC
   Else
      counter = counter + 1
   End If
   If counter > Worksheets("Data").Range("F18") Then
      socrow = rownum - counter
      Worksheets("Outputs").Range("B2").Value = Worksheets("Calc").Cells(socrow,
         7).Value
      Worksheets("Outputs").Range("B3").Value = Worksheets("Calc").Cells(socrow,
         9).Value
      Worksheets("Outputs").Range("B4").Value = Worksheets("Calc").Cells(socrow,
         6).Value
   Else
      rownum = rownum + 1
      GoTo SOC
   End If
```

```
' This section determines the End of Combustion
    tol = Worksheets("Data").Range("F19") 'Tolerance on returning to compression slope
    n = Worksheets("Outputs").Range("B6").Value
EOC:
    testslope = Worksheets("Calc").Cells(rownum, 11).Value
    If Abs(testslope - n) < tol And testslope < 0 And Worksheets("Calc").Cells(rownum,
            6).Value > 0 Then
        eocrow = rownum - 11
        Worksheets("Outputs").Range("C2").Value = Worksheets("Calc").Cells(eocrow,
            7).Value
        Worksheets("Outputs").Range("C3").Value = Worksheets("Calc").Cells(eocrow,
            9).Value
        Worksheets("Outputs").Range("C4").Value = Worksheets("Calc").Cells(eocrow,
            6).Value
    Else
        rownum = rownum + 1
        GoTo EOC
    End If
' This final section is to ensure desired components are on the Outputs worksheet
    Worksheets("Outputs").Range("A14:B150").ClearContents
    Worksheets("Calc").Range(Cells(socrow, 6), Cells(eocrow, 6)).Copy _
        Destination:=Worksheets("Outputs").Range("A14")
    Worksheets("Calc").Range(Cells(socrow, 14), Cells(eocrow, 14)).Copy _
        Destination:=Worksheets("Outputs").Range("B14")
    Worksheets("Outputs").ChartObjects("Chart 1").Activate
    ActiveChart.SeriesCollection(1).XValues = "=Outputs!$A$14:$A$" + Format(14 +
            eocrow - socrow)
    ActiveChart.SeriesCollection(1).Values = "=Outputs!$B$14:$B$" + Format(14 +
            eocrow - socrow)
```

End Sub

APPENDIX F – Mass Fraction Burned Tables used in model

Table 10: Mass fraction burned for Idle

Angle after SOC (deg)	MFB
0	0
0.789474	0.002819
1.578947	0.002319
2.368421	0.009573
3.157895	0.012703
3.947368	0.016699
4.736842	0.026355
5.526316	0.034741
6.315789	0.062856
7.105263	0.089837
7.894737	0.127366
8.684211	0.173238
9.473684	0.229906
10.26316	0.292928
11.05263	0.361353
11.84211	0.43305
12.63158	0.501196
13.46491	0.570104
14.29825	0.633547
15.13158	0.691281
15.96491	0.738046
16.79825	0.783045
17.63158	0.817053
18.46491	0.853314
19.29825	0.878417
20.13158	0.90881
20.96491	0.927864
21.79825	0.948565
22.63158	0.967744
23.46491	0.981912
24.29825	0.993385
25.13158	1

Table 11: Mass fraction burned for 5200 RPM

Angle after SOC (deg)	MFB
0	0
1.25	0.006815
2.5	0.004908
3.75	0.000343
5	0.009713
6.25	0.01482
7.5	0.017679
8.75	0.032077
10	0.052749
11.25	0.078466
12.5	0.121978
13.75	0.154472
15	0.192418
16.25	0.23673
17.5	0.279145
18.75	0.3302
20	0.387849
21.25	0.448997
22.5	0.519363
23.75	0.583047
25	0.646249
26.25	0.699499
27.5	0.752467
28.75	0.795989
30	0.840139
31.25	0.88058
32.5	0.900955
33.75	0.919116
35	0.936838
36.25	0.954882
37.5	0.958501
38.75	0.964851
40	0.972351
41.25	0.974025
42.5	0.98959
43.75	0.99545
45	1

Table 12: Mass fraction burned for 6000 RPM

Angle after SOC (deg)	MFB
0	0
1.363636	0.004115
2.727273	0.009149
4.090909	0.017837
5.454545	0.025066
6.954545	0.045769
8.454545	0.076747
9.954545	0.113305
11.45455	0.16208
12.95455	0.241968
14.45455	0.327416
15.95455	0.42171
17.45455	0.531895
18.95455	0.638456
20.45455	0.734664
21.95455	0.807843
23.45455	0.870561
24.95455	0.927604
26.45455	0.973211
27.95455	0.994909
29.45455	1

Table 13: Mass fraction burned for 6800 RPM

Angle after SOC (deg)	MFB
0	0
0.833333	0.023893
1.666667	0.034457
2.5	0.046715
3.333333	0.085547
4.166667	0.096381
5	0.152739
5.833333	0.178002
6.666667	0.228791
7.5	0.264093
8.333333	0.309423
9.166667	0.376746
10	0.415219
10.83333	0.477295
11.66667	0.51693
12.5	0.563646
13.33333	0.598182
14.16667	0.65388
15	0.670793
15.83333	0.72888
16.66667	0.742863
17.5	0.775661
18.33333	0.784257
19.16667	0.827582
20	0.83888
20.83333	0.85385
21.66667	0.865933
22.5	0.899468
23.33333	0.880433
24.16667	0.919893
25	0.963822
25.83333	0.929997
26.66667	0.97019
27.5	0.948468
28.33333	0.956557
29.16667	1.020569
29.95614	0.993674
30.74561	0.997858
31.53509	1.019573
32.32456	1