COMPUTATIONAL INVESTIGATION OF ETHANOL AND BIFUEL FEASIBILITY IN SOLSTICE ENGINE

A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in Engineering

By

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I HEREBY RECOMMEND THAT THE THESIS PREPARED UNDER MY SUPERVISION BY Adam Michael Blake ENTITLED Computational Investigation of Ethanol and Bifuel Feasibility in Solstice Engine BE ACCEPTED IN PARTIAL FULLFILMENT OF THE REQUIREMENTS FOR THE DEGREE OF Master of Science in Engineering

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ABSTRACT


A Gasoline Direct Injection (GDI) engine enables an increased fuel efficiency and higher power output than a conventional Port Fuel Injection (PFI) system. By injecting pressurized fuel straight into each cylinder of an internal-combustion engine, the degree of fuel atomization is increased, as well as the fuel vaporization rate. In order to further harness the effects of direct injection, ethanol is implemented as a fuel. The cooling effect of ethanol fuel droplets changing to vapor inside the combustion chamber facilitates a higher compression ratio, thus increasing engine power and efficiency. Three dimensional computational simulation is used to investigate the feasibility of ethanol and gasoline-ethanol mixtures as a fuel over varying compression ratios in a GDI engine. ANSYS Workbench is used to build a dynamic mesh of the varying compression ratio models, in conjunction with SolidWorks modeling software. To simulate flow physics, fuel injection, and combustion in the engine, ANSYS Fluent is employed. A parametric study of the effect of spark timing and compression ratio under ethanol operation at cruise RPM is performed. Additionally, a dual-injector gasoline-ethanol setup is implemented for the GDI engine and the effects of injection timing and mixture fraction of fuel is analyzed. Both ethanol and bi-fuel operation settings are found to provide significantly higher horsepower than the stock GDI engine. The dual-injector, bi-fuel operation is found to provide a specific fuel consumption comparable to the stock engine
while providing substantially higher output. The results yield a promising fuel delivery strategy which can be appealing to many direct injection engine applications.
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Nomenclature

GDI- Gasoline Direct Injection
PFI-Port Fuel Injection
RPM-Revolutions Per Minute
DISI-Direct Injection Spark Ignition
CR-Compression Ratio
TDC-Top Dead Center
BDC-Bottom Dead Center
AFR-Air Fuel Ratio
CFD-Computational Fluid Dynamics
IMEP-Indicated Mean Effective Pressure
LTC-Low Temperature Combustion
ICE-Internal Combustion Engine
CAD-Crank Angle Degree
SFC-Specific Fuel Consumption
SL-Laminar Flame Speed
FAR-Fuel Air Ratio
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I. INTRODUCTION

Background

Ethanol as a Fuel

As American dependence on foreign oil increases, the need for viable alternative fuels is made increasingly clear. As reported by the Consumer Energy Report, 58% of US oil consumed in 2007 was imported from foreign countries [20]. At the current rate of increase in US oil consumption, the amount of imported oil is projected to increase to 64% by the year 2020 [16]. As the consumption of oil continues to grow and oil prices skyrocket, multiple facets of daily life for America is affected. Primarily, the transportation industry consumes 70% of all oil in the US [16], therefore it is essential to find alternative fuels in the automobile industry.

Currently, one of the leading candidates for an alternative fuel is ethanol. Many benefits are offered by ethanol, including ease of production in the US, decreased emissions, and higher safety than provided with fuels such as hydrogen. It can be implemented either as pure ethanol or as a gasoline-ethanol blend, often known as “flex-fuel”. The use of ethanol is not a new idea, with the US Department of Energy researching the possibility of converting the entire postal fleet to using ethanol as long ago as 1978.
Amongst the most attractive features of ethanol is its unique ability to resist autoignition, or engine knock. This is a phenomenon in which the fuel detonates uncontrollably, often causing catastrophic engine damage. This quality of ethanol has multiple practical benefits. First, it enables the use of a higher compression ratio engine, which is shown to deliver more power and efficiency [4],[7]. Ethanol has also shown to effectively increase the octane rating when blended with conventional gasoline. This enables safe use for turbocharged operation, as a higher octane rating is often needed to counteract the increased pressures experienced from additional boost.

In addition to performance benefits of ethanol, the emissions created by ethanol and ethanol blend fuels are greatly decreased. The study performed by Knoll et al. revealed that as the amount of ethanol in the fuel increased, the emissions including CO and NMHC were decreased [10]. This quality of ethanol has garnered much attention, as the government offers various incentives for the use of ethanol fuel. In the state of Ohio alone, four incentives are in place as of 2012 for the use of ethanol or support thereof.

Fuel Delivery Methods

While extensive research into the use of ethanol in most engines has been undertaken, current developments in internal combustion technology have made these efforts somewhat obsolete. Research by the DOE in 1978 addressed the use of ethanol in carbureted engines, whereas the research by Knoll addressed mostly port fuel injected engines [14], [10]. Current trends in internal combustion engines show the benefits of direct injection spark ignition (DISI) as compared with the conventional port fuel...
Direct injection gains improvement over PFI in multiple aspects of the engine cycle. PFI loses fuel efficiency as the fuel charge condensates in the intake valve area of the port, whereas DI ensures that all fuel injected is available for combustion. Additionally, cold starting for DI is enhanced due to higher injection pressure aiding in fuel atomization, which further aids vaporization. Another key benefit of DI is the ability to achieve higher compression ratio without knock, which is possible due to the charge cooling introduced in the cylinder by directly injecting fuel. With lower temperature combustion and more precise air to fuel ratio control, emissions are also reduced in DI setup. These factors contribute to the recent surge in Direct Injection technology in the automobile industry [22].
In addition to aiding fuel vaporization and increasing CR, volumetric efficiency is increased by direct injection [22]. However, this benefit is realized primarily when the injection period occurs during the intake stroke as opposed to when the intake valve is closed. This is explained by an increased density of the cool air/fuel mixture during intake, which increases the mass of fuel inducted into the chamber. An increase in volumetric efficiency is desired for increased performance [7].

Fuel delivery for DISI encounters many challenges not faced by carburetion or PFI systems. For direction injection engines, the fuel is injected directly into the combustion chamber, which requires a higher injection pressure and leaves less time for fuel mixing. This aspect of direct injection places an increased importance on the utilization of in-cylinder flow and turbulence to properly mix and distribute fuel droplets. Additionally, differing fuels show much different characteristics during injection, as highlighted by Min, et al.. Fuel properties of ethanol results in shorter penetration lengths than those typically experience by gasoline. This will greatly affect fuel distribution in the cylinder. For these reasons, an investigation of the optimal injection strategies for ethanol and ethanol blends is necessitated.

Combustion Cycle Theory

The 2.0 Liter Ecotec engine used in the Pontiac Solstice utilizes four-stroke operation, which is comprised of four major stages (compression, expansion, exhaust, and intake). This cycle is often analyzed with a PV plot which distinguishes the four stages. Since the combustion process is adiabatic, the work produced during the cycle can be found by calculating the area under the PV curve.
Starting from point 1, the combustion cycle can be summarized as follows. Stage 1-2 represents the compression stroke. During compression, the exhaust and intake valves are closed, and the piston moves toward TDC. Thus, the volume is decreased and the pressure is increased, which prepares a combustible mixture. Spark generally occurs at the end of compression stroke.

Stage 2-4 represents the expansion stroke, which is comprised primarily of two events. First, the combustion of the fuel occurs, represented in 2-3. Here, the pressure rapidly rises as fuel is burned, and the volume does not significantly change. The piston is moving toward BDC at this time, increasing the volume in the chamber. This stroke is important because it generates the pressure which exerts a force on the piston and crankshaft. In a non-ideal cycle, the combustion does not occur instantaneously, thus the two separate stages in 2-4 are not as easily distinguishable.
Stage 4-6 represents the exhaust stroke, which is again comprised of two events. First, 4-5 represents the opening of the exhaust valve, during which pressure is released from the cylinder. 5-6 shows the volume decrease associated with the cylinder moving back towards TDC. The exhaust valve is open, therefore no pressure is generated and exhaust gases are expelled.

Finally, 6-1 shows the intake stroke, during which the intake valves open as the cylinder approaches BDC. This is a crucial step for DI engines, as the injection often occurs as the fresh air is inducted into the cylinder.

Figure 2 represents the ideal “constant volume combustion” PV diagram. In reality, combustion occurs over a finite crank angle, thus 2-3 is not vertical [7]. This represents a desirable PV curve, however. A representative PV diagram of a realistic engine cycle is shown in Figure 3. It is important to note desirable traits which will increase performance. First, a high peak pressure will increase the work output. This can be accomplished by an increased compression ratio, increasing intake pressure via turbocharger/supercharger, an increase in fuel (assuming all fuel consumed), or a combination of these methods. Additionally, increased duration of combustion (i.e. longer pressure generation) will increase the power output of the cycle.
Proposed Research

This study will use computational combustion modeling to investigate the plausibility of operating a commercially available gasoline direct injection (GDI) engine using ethanol and ethanol blends. Availability of test data from collaborators at Wright Patterson Air Force Base has made the 2.0 Liter Ecotec engine from a Pontiac Solstice a clear choice for this study, as an extensive validation of the computational model will be possible. The Solstice is one of few stock engines to offer GDI capability, and due to being turbocharged it will offer a wide variety of operating conditions.

The preliminary study will investigate the Ecotec running solely on ethanol as fuel, using the injection parameters seen with standard gasoline operation. The fuel amount will be adjusted to the stoichiometric air-fuel ratio (AFR). To be studied is the effect of spark timing and compression ratio to optimize power for ethanol operation. These optimized parameters will form a platform for further study.
In addition to pure ethanol operation, a study will be performed investigating bi-fuel operation. Namely, a gasoline injection will occur in the cylinder, followed by a direct ethanol injection. This is expected to benefit combustion stability under high boost operation by effectively increasing the octane rating of the gasoline, thus decreasing tendency to cause engine knocking.

Methodology

In order to provide insight to the in-cylinder phenomena governing combustion, Computational Fluid Dynamics (CFD) is often used for internal combustion engine study. CFD allows users to quickly and easily vary parameters that would require much more time and effort for an experimental test. With this capability, CFD is a very efficient method of performing parametric study on the operating parameters of an internal combustion engine. For the scope of this study, CFD will be utilized as a tool in investigating the feasibility of operating a 2.0 Liter Ecotec engine on varying fuels. The first step is to develop a three dimensional solid model, generate a dynamic mesh of this model, and apply combustion modeling principles to develop a working simulation.

Crucial to the computational study is the validation of outputs from simulations. In order to achieve satisfactory validation, engine test data from collaborators at Wright Patterson AFB will be used to simulate operating conditions as well as outputs such as pressure and horsepower produced. With a validated model investigation can proceed to encompass different fuels, spark timings, and injection parameters.

Thesis Outline

This project is composed primarily of three sections. The first of which (Chapters 2 and 3) discusses the procedures underlying creation and validation of the computational
Solstice model. The second section (Chapters 4 and 5) discusses the feasibility of operating the Solstice engine on direct injection ethanol and multi-phase gasoline ethanol injection. Finally, Chapters 6 and 7 detail the conclusions drawn from the investigation and future work to be undertaken, respectively.

**Literature Review**

In order to gain adequate understanding of the current progress in the field of alternative fuels, a literature review was conducted. By performing this review, trends in performance with varying fuels is recognized, property data for use as inputs in simulations is gained, along with baseline information to explore the validity of the computational model.

Hara and Kimitoshi performed an in-depth experimental investigation of the combustion properties of varying fuels. In their study, comparisons were made between the laminar flame speeds of ethanol, iso-octane, and n-heptane. Additionally, an analysis of the effects of ethanol addition to both iso-octane and n-heptane was performed, using flame stability as a judgement of good performance. It was determined that the data regarding flame speed was well-validated against other researchers, which state that ethanol burns at a faster rate than both iso-octane and n-heptane. With a validated method of analysis flame properties obtained, Hara and Kimitoshi further determined that the addition of ethanol into n-heptane and iso-octane effectively stabilizes combustion.

Yajia et al. experimentally obtained data regarding the spray characteristics of ethanol, methanol, and gasoline in a Direct Injection Spark Ignition (DISI) engine. This
data was then compared to results obtained via numerical simulation with AVL FIRE software. Results produced penetration lengths at varying injection pressures and chamber pressures, as well as numerically obtained Sauter Mean Diameter (SMD) data for the three injections. It was found that fuel type has a dramatic effect on spray penetration, with ethanol having much lower penetration than gasoline. This data is helpful to validate spray models in ANSYS Fluent.

Gautam and Martin present an experimental investigation into the combustion characteristics of alcohol/gasoline fuel blends. These experiments are performed on a Waukesha Cooperative Fuel Research Engine, which enables the user to control factors such as compression ratio while maintaining a constant spark timing. This is key to the investigation because it enabled Gautam and Martin to analyze the indicated mean effective pressure, anti-knock index, and emissions between the varying blends at their practical utilization criteria. In order to systematically analyze the anti-knock properties of the varying blends, a “critical knock index” was analyzed, and a means of analyzing this from the derivatives of pressure traces established. This means of analysis allowed for very insightful data as to the anti-knock properties of fuel blends, their optimal spark timing and compression ratios for operation, as well as the best power outputs from each blend. This data is useful to provide a guideline for the optimal parameters under which to utilize fuel blends. It was found that the fuel which allowed the highest CR without knocking was that with the highest percentage of high-order alcohols blended in. This high CR operation also allowed for the best IMEP outputs, which theory of internal combustion principles widely suggests.
Pidol et al. offer an experimental investigation of the effects of ethanol blends in diesel fuels used in standard and Low Temperature Combustion Diesel cycles. Benefits of LTC are listed as “keeping below the soot-forming region and NOx creation zone”, in addition to a reduction in particulate emission by decreased flame temperature. Ethanol is chosen as the blend fuel because of its low cetane number and low boiling point. Results conclude that smoke is reduced due to oxygen in the fuel, and the increased volatility of ethanol aids the control of ignition timing. When combustion control is optimized, lower particulates and NOx emissions are found in addition to lower noise levels. A contained fuel consumption penalty equivalent to energy content decrease is present in all test runs.

Knoll et al. provide analysis on the emissions of vehicles operating on different ethanol blends of fuels. This investigation used statistical data to observe which types of cars are most popular in America, and chose a test fleet that represented this data. Conclusive trends showed that as the amount of ethanol in the fuel blends increased, the NMHC and CO emissions were decreased, while no significant change in NMOG and NOx were experienced. The effects of ECU power-enrichment strategy was also analyzed, yielding the result that vehicles employing Lean Fuel Trimming during operation will not experience a significant decrease in emissions compared to those that do not employ LFT.

Pefley performed a computational and experimental analysis of the conversion of a gasoline engine utilized by the USPS to run on ethanol and methanol fuels. The computational investigation provided insight to the effects of fuel types, compression ratios, rear axle ratios, and reduced cooling loads on multiple performance parameters, primarily efficiency and emissions. The study found that as the spark timing is retarded
(TDC or later) the emissions will decrease, due to the mixture being burned at lower pressures and temperatures. Additionally it was found that as the compression ratio increases, the power and efficiency are increased. The emissions results do not show as clear of trend however, as the NOx is increased whereas the aldehyde emissions are decreased. In addition to the performance discussion of varying fuels in the USPS vehicles, an enlightening discussion regarding the challenges faced during fuel conversion is presented; primarily being difficulty to cold start the engine as well as corrosion on gasoline fuel systems.

Turner et al. performed an investigation concerning the performance of gasoline-ethanol fuel blends. In their testing, a single cylinder engine was outfitted for direct injection occurring at varying timings. The effect of injection and spark timing on different fuel blends was optimized for increased MEP and decreased emissions. Combustion efficiency was determined based on the CO in the exhaust stream. It was found that the combustion efficiency for early injections (during intake stroke) increased as the percentage of ethanol increased. This was not the case for late injections (after intake stroke), which showed optimal combustion efficiency at close to 50% ethanol. These trends suggest that the time for the fuel and air to mix in-cylinder has a large effect on the combustion efficiency of direct-injected ethanol blends. The NOx emissions were also found to be minimum at near 85% ethanol, concluding that there is indeed an optimal mixture for reduction of harmful emissions.

Sementa et al. experimentally investigated the performance of gasoline vs. bio-ethanol in a high performance GDI engine with varying injection strategies. The combustion chamber was “optically accessible” in order to obtain imaging of fuel
injection and combustion phenomena, to provide visual correlation with IMEP data. Results obtained suggest that the flame propagation is faster for stratified charge operation rather than homogenous for both fuels. This produces a greater in-cylinder pressure, however and increased soot output due to wall impingement is a detrimental side-effect. Additionally, it was found that air motion and pressure at start of injection had a greater effect on gasoline spray distribution than that of ethanol. Finally, it was found that the stratified charge ethanol cases reduced emissions at an improved stability of combustion.

Moore et.al experimentally investigated improvements in efficiency of a direct-injection gasoline engine by modifying compression ratio and using fuels with various ethanol blend ratios. A compression ratio increase from 9.2:1 to 11.85:1 was achieved by piston modification, and the effective compression ratio was further increase with a modified valve train configuration. Blend ratios of E0,E10,E20,E50,E85,and EEE were tested with no intake boost. Results found that the higher ethanol blends effectively resist knock and reduce NOx formation, which enables use of greater valve overlap for increased performance. Additionally, the torque produced at low end operation was increased when E20 through E85 was used. The major detrimental effect of blended fuels was the increased soot produced with early injection, which is a side-effect of wall impingement from the fuel spray. This was countered by injection and valve timing adjustments.

Yamin and Dado performed a computational analysis of a variable compression ratio engine. An eight link rocker mechanism was employed to achieve a compression ratio which could be varied from 6.82:1 to 10:1. The model was validated with
experimental data on terms of cylinder heat loss, indicated specific fuel consumption, and combustion duration. The performance analysis of the engine under different compression ratios revealed that a maximum indicated power increase of 62% was achievable using the highest compression ratio and at optimal RPM. Added benefits of the increase in compression ratio were found in the decreased specific fuel consumption. Major challenges yet to be overcome are found in the emissions requirements, as the geometry associated with higher compression ratio induces wall impingement of the fuel spray.

Anand et al. performed in-depth analysis of fuel spray from a port fuel injector using laser backlight imaging. Of specific interest to the study was the effect of varying gas-ethanol blends on the spray patterns exhibited from the injector. Various injector pressures were used to explore the effects at differing operating ranges. It was found that ethanol generally exhibits a larger cone angle than gasoline under the same pressures. Additionally, it was found that nearly 15% more ethanol was injected during an identical pulse width, at the same pressure. Despite these viscosity-dependent differences, it was found that droplet size between the two sprays was not changed significantly between the two fuels. Although the injector is designed for PFI, it is very similar by design as the DI injector used in the 2.0 L Ecotec, therefore the information is pertinent to the thesis investigation.
II. COMPUTATIONAL MODEL OF SOLSTICE ENGINE

Modeling and Meshing

Solid Model

The first step in development of the computational model for the Solstice engine entails the use of 3-D modeling software to simulate the combustion chamber. Because the interest here lies in the fluid mechanics underlying the combustion cycle, the fluid volume in the chamber was modeled. Care was taken to verify accurate dimensions were used to produce a valid model. Of particular importance was the volume inside the combustion chamber when the piston is at top dead center (TDC). Any inaccuracies in this volume (known as the clearance volume) result in an incorrect compression ratio, which will cause variances in temperature and pressure during compression. A calculation of the compression ratio is shown below [7].

\[ CR = \frac{V_d + V_c}{V_c} \]

Equation 1. Compression Ratio

\[ V_d = \frac{\pi B^2}{4} (Stroke) \]

Equation 2. Displacement Volume

For the course of this study, three various compression ratios are to be used. To facilitate this, three separate computational models are needed. As is seen in the above equation, compression ratio may be varied by alteration of either the clearance or the
displacement volume. Upon observation of the equation for displaced volume, it becomes apparent that modifications to either the engine bore or stroke will change displacement volume. These changes however are not easily implemented, as a change in bore requires engine block modification, and a change in stroke requires many mechanical modifications (including crank radius). It is common practice to modify compression ratio by changing the piston shape by causing it to occupy more volume at TDC, thus decreasing clearance volume. As this is a more feasible and manufacturable approach to modifying the Ecotec, it will be followed here. A summary of dimensions at each compression ratio is shown in Table 1 below.

<table>
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<tr>
<th>Compression Ratio</th>
<th>Displaced Volume (cm$^3$)</th>
<th>Clearance Volume (cm$^3$)</th>
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<tr>
<td>9.2:1</td>
<td>499.5</td>
<td>60.90</td>
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<tr>
<td>11.0:1</td>
<td>499.5</td>
<td>49.95</td>
</tr>
<tr>
<td>13.0:1</td>
<td>499.5</td>
<td>41.63</td>
</tr>
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</table>

Table 1. Physical Dimensions of Model

Additionally, the piston bowl geometry plays an important role in flow characteristics inside the chamber. The Solstice engine employs a wall-guided direct injection system, meaning that the piston motion is utilized to guide the fuel towards the spark. In contrast, a spray-guided injection setup relies on the proximity of the spark plug to the injector for preparation of an ignitable mixture. For wall-guided setup, as the piston moves a certain amount of turbulence is generated, which is of paramount importance in the mixing, vaporization, and placement of fuel. Therefore, it is essential that the piston geometry is modeled accurately. Interaction of the piston geometry with air in the cylinder produces motion necessary to deliver the fuel to the spark. Using a three dimensional scanner the piston geometry was modeled under much tighter tolerances than would be possible with conventional methods.
Another area of the model worthy of much attention is the intake and exhaust port areas. The ANSYS help guide recommends typical geometry breakdown for ICE simulations, in order to facilitate simple and clean mesh geometries. This breakdown decomposes the intake and exhaust ports into three main sections, in order to reduce complexity and computation time when remeshing occurs. A geometrically accurate model of the fluid volume in the Solstice engine was decomposed into the recommended parts, with little variation. This facilitates ease in the meshing process.
Figure 6. Decomposed Intake Valves

Figure 7. Completed Fluid Volume
### Solstice 2.0 L Ecotec Specifications

<table>
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<tr>
<td>Compression Ratio</td>
<td>9.2:1</td>
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<tr>
<td>Bore x Stroke</td>
<td>86mm x 86mm</td>
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<tr>
<td>Maximum Valve Lift (Intake)</td>
<td>10.33mm</td>
</tr>
<tr>
<td>Maximum Valve Lift (Exhaust)</td>
<td>10.33mm</td>
</tr>
<tr>
<td>Injection Type</td>
<td>Direct Injection</td>
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<tr>
<td>Aspiration</td>
<td>Turbocharged</td>
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</table>

**Table 2. Ecotec Specifications**

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**Meshing**

Once a complete solid model of the fluid volume was obtained, meshing in ANSYS Workbench was undertaken. Factors under consideration when building the mesh included optimal accuracy at a manageable number of elements. This ensures that a computational model will produce repeatable results with reasonable computational time. In order to achieve this goal, many different meshing techniques were used on the varying parts of the Solstice model.

The chamber used a Patch Independent tetrahedral mesh of 51,854 elements. This enables a maximum cell skewness of 0.871. Skewness provides a good means of quantifying the quality of a mesh.
The intake and exhaust ports are of nearly identical mesh structure, the only difference being outer port diameter. As previously mentioned, the port was decomposed into three separate volumes, as recommended by ANSYS [8]. The outermost region comprising most of the port volume is known as the OB region. This region remains stationary throughout mesh motion; therefore a relatively fine quad element mesh is sufficient for high accuracy with reasonable computational cost. The next region is known as the IB, which is a more complicated geometry than the OB. Here, a hybrid quad/triangle mesh was employed to capture the difficult geometry. This volume moves with the valves during mesh motion, therefore an extremely fine mesh is problematic to the dynamic mesh motion. Hence, a moderate coarseness was chosen for the IB. Finally, the V-Layer comprises perhaps the most crucial geometry of the port decomposition. In this region the valve meets with the valve seat. Upon valve opening there is a large pressure differential in the chamber and the port, and all of the flow is forced through the V-Layer. Therefore it is critical that a fine mesh is created here to ensure proper resolution of the flow. This volume also moves with the valve. In order to complete such motion with a fine mesh, a quad mesh which layers as the volume grows is used. This is
less computationally involved than a hex or tri dynamic mesh, yet still retains sufficient accuracy in the region of interest. The total number of elements (per port) is around 11,500 on average, with a maximum skewness again of 0.871.

Figure 9. Decomposition of Valves and Meshed Valve
Due to the transient nature of an internal combustion engine, it is essential to model the motion of the components inside the chamber. This is accomplished by dynamic meshing techniques in ANSYS Fluent. With these tools, it is possible to model the piston and valve motion over the course of a combustion cycle, while maintaining mesh integrity and simulation accuracy.

In order to maintain mesh integrity with such motion, Fluent offers smoothing and remeshing options that allow the elements in a mesh to stretch, break up, and remesh as the cylinder volume increases and decreases. The user inputs parameters such as maximum/minimum cell size and maximum skewness that are evaluated at each timestep,
then smoothing or remeshing occurs when the cell size and skewness limits are exceeded. This enables a consistently accurate mesh throughout the range of motion encountered in an engine cycle.

Fluent offers “In-Cylinder” options for the simulation of Internal Combustion engines which greatly aid ease-of-use. These options allow the use to specify operating parameters such as engine speed, bore x stroke information, and crank information. This effectively defines the entire simulation in terms of crank angle rotation, which lends itself to easy visualization. It is also preferential to define events such as spark, injection, and valve events in terms of crank degrees rather than flow time. This also reduces likelihood of the user inputting inaccurate parameters.

<table>
<thead>
<tr>
<th>In-Cylinder Dynamic Mesh Settings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crank Shaft Speed (RPM)</td>
</tr>
<tr>
<td>Starting Crank Angle (Degree)</td>
</tr>
<tr>
<td>Crank Angle Step Size (Degree)</td>
</tr>
<tr>
<td>Crank Radius (m)</td>
</tr>
<tr>
<td>Connecting Rod Length (m)</td>
</tr>
</tbody>
</table>

Table 3. Dynamic Mesh Settings

To ensure a valid representation of the Solstice engine, care must be taken to ensure that the computational events are identical to the realistic events. This includes valve opening/closing time, maximum valve lift, piston travel, and engine speed (Revolutions per Minute). A summary of these events is shown in the table below.
A. Intake Stroke  
B. Compression Stroke  
C. Expansion Stroke  
D. Exhaust Stroke

Figure 11. Dynamic Mesh Motion
Computational Models in ANSYS Fluent

Flow Model

In order to accurately model the in-cylinder flow as the density varies throughout a cycle, a three-dimensional compressible Navier-Stokes solver is utilized in ANSYS Fluent. This enables realistic simulation of the effects of compressibility on the engine cycle, such as changing fuel injection trajectories as regions of various densities are encountered. This is coupled with a Realizable K-ε Turbulence model, which resolves turbulent flow based on turbulent kinetic energy as well as turbulent dissipation rate, therefore solving two transport equations. The K-ε model, proposed by Launder and Spaulding, offers good accuracy for many turbulent flow scenarios [17]. The K-ε has been used for various internal combustion engine simulations [13],[11] with good reported validations to experimental results.

The K-ε turbulence model is separated into three different subcategories: standard, RNG, and Realizable. Here, the Realizable model was chosen due to certain benefits over Standard and RNG, including a new method of calculating eddy viscosity recommended by Reynolds [15]. In this method, the eddy viscosity is resolved with a variable term to account for changing flow properties as the chamber conditions change. Additionally, a modified means of calculating the dissipation rate has been implemented in the Realizable model, which derives from the vorticity fluctuation [17].

Combustion Model

To simulate the combustion in the engine, the Partially Premixed Combustion model was chosen. This enables the simulation of "premixed flames with non-uniform
fuel-oxidizer ratios” [12]. This is appropriate for the simulation of direct injection engines, so that fuel injector parameters can be modeled and the effects analyzed systematically. The partially premixed model is a direct combination of the fully premixed model and the non-premixed model, which offer very different capabilities. Non-premixed is directed towards diesel simulation, in which the fuel and oxidizer are introduced in separate streams [12]. The fully premixed model accounts for fuel and oxidizer completely mixing before combustion, which is the phenomena encountered in port fuel injected engines. In partially premixed combustion, a fuel is introduced into and oxidizer, in terms of a mass fraction. This produces an equivalency which may be non-uniform depending upon in-cylinder flow. This is sufficient for modeling a direct fuel injection.

Combustion is modeled in terms of “progress”. The progress variable ranges from 0<\(c<1\), where ‘1’ signifies burnt mixture, ‘0’ signifies unburnt, and in-between is a linear combination of burnt and unburnt. As flame is introduced into the chamber, the progress is set to ‘1’, and this propagates throughout the fuel depending on the flame speed.

Flame speed is known to vary with temperature and pressure, therefore the flame speed should vary dramatically during the course of a combustion cycle. This is accounted for by employing a User Defined Function which recalculates the flame speed based on the pressure, temperature, and mixture fraction inside the cylinder at the current timestep. The UDF currently employed has been developed and validated by WSU FSRG, and offers inputs for various fuels. The correlation employs a medium order fit to experimental data.
Spark and Injection Model

In order to initiate combustion in Fluent, a spark ignition model is offered. This model enables the user to input the location desired as well as other key spark parameters. An initial radius is input which can be estimated by the typical gap between the electrode and ground on the spark plug. Additionally, spark duration is requested, which dictates the length of time that the spark is “on”. From data gathered by the Bosch spark controller during experimental testing at WPAFB Research Facilities, the spark duration was monitored to last approximately 35 crank angle degrees at 2000 RPM.

The spark model has direct relation to the combustion model in that “spark” is represented by introducing a non-zero progress variable. Effectively, the spark radius specified is set to a progress of ‘1’ which represents entirely burnt fuel. This progress (representing the flame) will either propagate throughout the chamber or be extinguished, depending on the ignitability of the fuel in the surrounding vicinity.
III. COLD FLOW STUDY AND MODEL VALIDATION

Cold Flow
In order to gain understanding of the flow phenomena that occur in the cylinder during the operating cycle, it is necessary to perform a cold flow simulation. In order to perform a cold flow, the engine mesh is moved through the entire combustion cycle neglecting injection of fuel and spark initiation. This enables the analysis of flow without the hindrance of fuel particles and vapor. Cold flow simulations serve multiple purposes; including verifying that the dynamic mesh is accurate, analyzing swirl and tumble inside the chamber, and analyzing intake and pumping performance. This offers insight into injection optimization in terms of utilizing turbulence to vaporize particles, as well as the amount of air inducted for calculation of a stoichiometric air-fuel mixture. To ascertain the effects of the three varying compression ratios on the amount of air inducted, a cold flow was performed with each mesh.

Boundary Conditions
To accurately simulate realistic operation, boundary conditions for the simulation were set to reflect conditions recorded by experimental data at 2000 RPM. Temperatures on the combustion chamber are fairly uniform, as well as the exhaust ports and valves. Temperature values set for intake valves are meant to reflect a transition from the high temperature combustion chamber to the room temperature intake. Pressure on the intake was recorded at 140,835 Pa (gauge). This is higher than atmosphere due to the use of a turbocharger. As RPM increase, it is expected that turbocharged pressure will increase.
<table>
<thead>
<tr>
<th>Zone</th>
<th>Temperature</th>
<th>Gauge Pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cylinder Wall</td>
<td>755 K</td>
<td></td>
</tr>
<tr>
<td>Cylinder Head</td>
<td>755 K</td>
<td></td>
</tr>
<tr>
<td>Piston Wall</td>
<td>755 K</td>
<td></td>
</tr>
<tr>
<td>Exhaust Valve Faces</td>
<td>755 K</td>
<td></td>
</tr>
<tr>
<td>Exhaust Valve IB</td>
<td>755 K</td>
<td></td>
</tr>
<tr>
<td>Exhaust Valve Stems</td>
<td>755 K</td>
<td></td>
</tr>
<tr>
<td>Exhaust Ports</td>
<td>755 K</td>
<td></td>
</tr>
<tr>
<td>Intake Valve Faces</td>
<td>600 K</td>
<td></td>
</tr>
<tr>
<td>Intake Valve IB</td>
<td>400 K</td>
<td></td>
</tr>
<tr>
<td>Intake Valve Stems</td>
<td>323 K</td>
<td></td>
</tr>
<tr>
<td>Intake Ports</td>
<td>323 K</td>
<td></td>
</tr>
<tr>
<td>Pressure Inlets</td>
<td>323 K</td>
<td>140,835 Pa</td>
</tr>
<tr>
<td>Pressure Outlets</td>
<td>755 K</td>
<td>120,000 Pa</td>
</tr>
<tr>
<td>Every Zone Initialized</td>
<td>323 K</td>
<td>140,835 Pa</td>
</tr>
</tbody>
</table>

Table 4. Cold Flow Boundary Conditions

Results

![Pressure Traces for Varying CR](image)

**Figure 12. Pressure Traces at Various Compression Ratios**

Cold flow simulations were completed for one entire power cycle (1080 CAD), using a mesh for each compression ratio. Figure 12 shows the pressure traces generated
for each compression ratio. As expected, the highest CR produced the highest peak power. When utilized without producing engine knock, this high peak pressure greatly benefits performance and efficiency. In addition to increased pressures at TDC, the temperature experienced is higher as well. For this reason, many high CR engines can cause auto ignition if the octane number of the fuel is not sufficiently high. Table 5 shows the peak pressures and temperatures from the various cold flow simulations.

<table>
<thead>
<tr>
<th>Compression</th>
<th>9.2:1</th>
<th>11:1</th>
<th>13:1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Peak Pressure (Mpa)</td>
<td>5.381</td>
<td>6.828</td>
<td>8.474</td>
</tr>
<tr>
<td>Peak Temperature (K)</td>
<td>884.949</td>
<td>932.920</td>
<td>988.179</td>
</tr>
</tbody>
</table>

Table 5. Cold Flow Summary

Another insightful observation from coldflow studies is the amount of air trapped in the cylinder after the intake stroke is completed. This value is key in observing volumetric efficiency in addition to calculating the amount of fuel to react with the given amount of air. Heywood describes pumping performance as a form of volumetric efficiency [7]:

\[ \eta_v = \frac{m_a}{\rho_{a,i} \cdot V_d} \]

Equation 3. Volumetric Efficiency from Heywood

The above equation describes a ratio of air inducted to air displaced by the piston under atmospheric conditions. Here, \( \rho_{a,i} \) is the inlet air density, which can be taken at atmospheric conditions to analyze overall volumetric efficiency or at intake manifold conditions to analyze pumping performance of the particular cylinder, valve and piston configuration [7].
<table>
<thead>
<tr>
<th>Compression Ratio</th>
<th>Air Induction (mg)</th>
<th>Pumping Performance (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>9.2:1</td>
<td>1292.84</td>
<td>99.25</td>
</tr>
<tr>
<td>11:1</td>
<td>1276.3</td>
<td>97.99</td>
</tr>
<tr>
<td>13:1</td>
<td>1250.52</td>
<td>95.97</td>
</tr>
</tbody>
</table>

Table 6. Cold Flow Induction Summary

As expected, the pumping performance decreases as the compression ratio increases. However, the overall volumetric efficiency is well above 100%, attributable to the use of turbocharged boundary conditions. This effectively forces the air into the chamber rather than relying on optimal inlet geometry and valve timing to maximize intake.

The intake stroke of the Ecotec engine is especially critical due to the timing of the direct fuel injection. To gain perspective on why this injection timing is particularly effective, the turbulence kinetic energy has been monitored over the course of the induction stroke. Figure 13 shows the TKE is most widespread and highest in magnitude at 450°, which corresponds with nearly halfway through the fuel injection pulse. Additionally, at this time step the flow rate through the intake is maximized. Shortly after closing of the intake valve, the kinetic energy due to turbulence is dissipated, and the remaining fuel mixture is to be evaporated due to the increasing temperatures in the combustion chamber during compression.
In addition to turbulence, the swirl and tumble inside the combustion chamber are fundamental to fuel/air mixing. Figure 14 shows the swirl generated on a plane normal to the axis of piston motion. Two particularly large regions of maximum swirl are immediately recognizable. This feature is unique to engines with two intake valves, such as the Ecotec. As the intake valves close, the swirl is dissipated. Thus, it is important for fuel injection to occur during intake to utilize this feature. Another important aspect in fuel/air mixing is seen in Figure 15. Here, the effect of the stock Ecotec piston on tumble
in the cylinder is made evident. This piston is effective at producing large vortices in the cylinder which greatly enhance fuel dispersion.

Figure 14. Swirl during Induction Stroke

Figure 15. Tumble During Induction Stroke
**Combustion Simulation**

Once the cold flow simulation is established, the necessary models to simulate a power cycle can be implemented. The necessary steps to complete a power cycle include fuel injection, fuel evaporation, spark initiation, combustion of gases, and finally exhaust. It is important to note how the turbulence and flow generated in cold flow study interact with fuel droplets and gases to create a combustible mixture. The computational models explained in Chapter II are implemented to account for these steps.

**Fuel Injection**

The 2.0L Ecotec engine employs a six orifice fuel injector shown in Figure 16 A which injects gasoline directly into the combustion chamber, interacting with the piston bowl to produce an ignitable mixture located near the spark plug. To simulate this accurately, the orientation and position of the injector are noted in the solid model, and duplicated within the simulation. The Fluent injection model also includes six separate plumes, injecting fuel at an average particle diameter of 11.2 microns. Flow rate and injection duration is controlled meticulously to ensure the proper amount of fuel delivery.

For a baseline combustion setup, it is noted that 1292 mg of fuel are inducted in the cold flow study for 9.2:1 CR. With a stoichiometric air to fuel ratio of 14.6:1 for gasoline, this requires 88.5 mg of fuel delivery. Experimental data provided suggests injection duration of 58 CAD (417°-475°).
Fuel Evaporation

Once the fuel droplets enter the chamber they interact with the flow and dynamics of the engine until they reach evaporation criteria. When the temperature, pressure, turbulence, or combination of above have met sufficient conditions, Fluent allows that particle to evaporate into a gas. For gasoline, the primary evaporation species is \( C_8H_{18} \), called iso-octane. To explain the computational evaporation process, the following figure is beneficial.
To model the ignition phase of the power cycle, Fluent offers a model to simulate spark ignition. Once a location and timing is input, a spark of user determined radius is initiated, setting the progress variable inside this diameter to ‘1’. Here, a progress of ‘1’
signifies completely burnt gas. Spark location is at the top apex of the cylinder head, as determined by the 3-D scan and then input to the solid model. Figure 18 A shows the equivalence inside the chamber before spark. Note the grey marker showing spark plug location in the chamber. Figure 18 B shows the equivalence at ‘0’ in the spark plug location, indicating consumed fuel. At the end of a good combustion event, equivalence of zero is desirable, as this indicates all fuel was used to generate power.

**Combustion**

Once spark is initiated, the progress propagates throughout the combustion chamber based on local equivalency of fuel, temperature, and pressure. The speed of propagation is determined largely in part by the calculated laminar flame speed (LFS). Higher LFS will result in faster propagation, which in most cases results in better combustion and more horsepower. Generally speaking, an equivalency of ‘1’ is most favorable for combustion, whereas rich (over one) will slow flame speed and lean (less than one) will overheat most engines. The figure below shows the propagation of flame by coloring the gradient of progress variable. High magnitudes indicate the progress variable is changing from ‘0’ to ‘1’. Propagation occurs radially away from the spark, as expected with a uniform distribution of fuel and a spherical spark.
Exhaust

Figure 19. Flame Propagation

Figure 20. Exhaust Velocity Vectors
**Validation**

In order to prove the accuracy of the computational model generated, it is necessary to make a comparison to realistic performance characteristics of the engine. The Society of Automotive Engineers offers a non-biased performance summary of engines on the public market in their “Certified Power Listings” publications [9]. For model validation, computational simulations are completed at three varying engine speeds and compared to the data from SAE to analyze agreement. For the scope of this study 2000, 3000, and 4000 RPM were chosen. Speeds lower than 2000 RPM are difficult to validate, as the engine operates much differently at idle. At 2000 RPM, it is assumed at wide open throttle (therefore the turbo is building to maximum 22 PSI) and the fuel pump is operating in high pressure mode (2200 PSI). At and above 3000 RPM, it is assumed that the turbocharger and fuel pump are operating at maximum capacity.

In establishing the validation, 2000 RPM was chosen as the starting point. Test data at this speed provided insightful parameters for establishing the injection model, spark timing, turbocharged pressures, and valve timings. Special consideration was given in determining valve timings as the RPM increased, as the Ecotec engine is equipped with variable valve timing and the valve overlap increases as engine load and RPM increase.
Figure 21. SAE Published Data for 2.0 L Ecotec [9]
<table>
<thead>
<tr>
<th>Engine Speed</th>
<th>1901 RPM</th>
<th>2000 RPM</th>
<th>3000 RPM</th>
<th>4000 RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Boundary Conditions</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inlet Pressure (turbocharged)</td>
<td>140,835 Pa</td>
<td>140,835 Pa</td>
<td>151,684 Pa</td>
<td>151,684 Pa</td>
</tr>
<tr>
<td>Inlet Temperature</td>
<td>323 K</td>
<td>323 K</td>
<td>323 K</td>
<td>323 K</td>
</tr>
<tr>
<td>Exhaust Pressure</td>
<td>120,000 Pa</td>
<td>120,000 Pa</td>
<td>125,000 Pa</td>
<td>125,000 Pa</td>
</tr>
<tr>
<td>Exhaust Temperature</td>
<td>810 K</td>
<td>810 K</td>
<td>810 K</td>
<td>810 K</td>
</tr>
<tr>
<td>Fuel Temperature</td>
<td>314 K</td>
<td>314 K</td>
<td>314 K</td>
<td>314 K</td>
</tr>
<tr>
<td><strong>Injection and Spark</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Spark Timing</td>
<td>15 deg bTDC</td>
<td>15 deg bTDC</td>
<td>15 deg bTDC</td>
<td>15 deg bTDC</td>
</tr>
<tr>
<td>Spark Duration</td>
<td>20 CAD</td>
<td>20 CAD</td>
<td>20 CAD</td>
<td>20 CAD</td>
</tr>
<tr>
<td>Start of Injection (SOI)</td>
<td>417</td>
<td>417</td>
<td>400</td>
<td>400</td>
</tr>
<tr>
<td>End of Injection (EOI)</td>
<td>475</td>
<td>475</td>
<td>500</td>
<td>524</td>
</tr>
<tr>
<td>Fuel Flow Rate</td>
<td>0.021265 Kg/Sec</td>
<td>0.016552 Kg/Sec</td>
<td>0.01575</td>
<td>0.025506</td>
</tr>
<tr>
<td>Fuel Density</td>
<td>705 Kg/m^3</td>
<td>685 Kg/m^3</td>
<td>685 Kg/m^3</td>
<td>685 Kg/m^3</td>
</tr>
<tr>
<td><strong>Valve Timing</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Intake Open</td>
<td>344</td>
<td>344</td>
<td>340</td>
<td>340</td>
</tr>
<tr>
<td>Intake Close</td>
<td>588</td>
<td>588</td>
<td>547</td>
<td>547</td>
</tr>
<tr>
<td>Exhaust Open</td>
<td>848</td>
<td>848</td>
<td>856</td>
<td>856</td>
</tr>
<tr>
<td>Exhaust Close</td>
<td>1092</td>
<td>1092</td>
<td>1080</td>
<td>1080</td>
</tr>
</tbody>
</table>

*Table 7. Model Parameters for Validation*
After running validation cases at 2000, 3000, and 4000 RPM, the accuracy of the computational model can be assessed. When compared to the SAE Data, a coefficient of determination ($R^2$) value is found to be 0.945. This is satisfactory for the course of this study, as the model shows sufficient correlation to both experimental data and to SAE Published data. It is noticeable from Figure 22 that the accuracy of the model is greatest at 2000 RPM, and the most error is found in the 4000 RPM case. This can be explained by the lack of experimental data at high RPM. The 2000 RPM case has defined valve timings, turbo pressure, fuel injection quantities, and spark timings from experimental data. At 4000 RPM, it is known that the valve timing changes, in addition to fuel injection timing and flow rates. Here, a reasonable approximation is drawn to make a
stoich mixture using the fully advanced valve timings outlined by Ecotec specifications. This assumption is reasonable as long as the Ecotec does not adjust equivalency at higher RPM.
IV. ETHANOL INJECTION STUDY
The Ecotec engine computational model developed and validated in the previous study is used to examine the feasibility of operation using ethanol as a fuel. A comparison between gasoline performance and ethanol performance at cruising speed is made. Three different models with varying compression ratios are studied, in addition to various spark timings. A comparison is made of performance at each compression ratio.

Model Settings
For the course of this study, 2000 RPM is used in order to simulate cruising conditions. The turbocharger pressure was assumed to be the same as that for the 2000 RPM gasoline validation case. Thus, the amount of air inducted is assumed to be the same when calculating stoichiometric conditions within the chamber.

In order to perform a realistic investigation of the feasibility of ethanol operation in the Ecotec engine, parameters for ethanol simulations were decided based on ease of implementation. The same fuel injector is used in the same location as the stock Ecotec engine. Additionally, fuel flow rate is assumed to be the same to ensure the fuel pump delivery rate is realistic. One of the considerable challenges associated with a DI ethanol setup is fuel delivery. As the stoichiometric AFR of ethanol is 9:1 as opposed to 14.6:1 for gasoline, approximately 38% more fuel is needed for an equivalent mixture. To accomplish the increased fuel demands, the injection duration is increased rather than the injection flow rate. This ensures that the stock fuel pump will be capable of the increased demands placed by using ethanol. Additionally, the higher latent heat of vaporization
makes the evaporation of ethanol more difficult than most fuels [19]. For this reason, high compression ratio engines are often fueled by ethanol.

As observed in Chapter III, 1292.8 mg of air was inducted for the turbocharged 2000 RPM cold flow with a 9.2:1 compression ratio. For a stoichiometric mixture, 87.9 milligrams of gasoline are needed whereas 143.6 mg of ethanol are required. This increased fuel requirement relates to over 60% increase in injection duration.

Additionally, as Yajia shows in [21], ethanol exhibits much less spray penetration in typical operation. Typically, Yajia reports an average of 20% increase in penetration lengths with ethanol when compared to gasoline. To compensate for this in the computational model, the WAVE breakup constant is adjusted 20%, to allow droplet breakup to occur earlier. Table 8 shows the injection parameters.

<table>
<thead>
<tr>
<th></th>
<th>Gasoline</th>
<th>Ethanol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injection Start</td>
<td>417</td>
<td>398.5</td>
</tr>
<tr>
<td>Injection Stop</td>
<td>475</td>
<td>493.5</td>
</tr>
<tr>
<td>Flow Rate</td>
<td>.003031 kg/sec</td>
<td>.003031 kg/sec</td>
</tr>
<tr>
<td>Particle Diameter</td>
<td>11.9 micron</td>
<td>11.9 micron</td>
</tr>
<tr>
<td>Breakup Constant</td>
<td>5.0</td>
<td>4.11</td>
</tr>
<tr>
<td>Total Fuel</td>
<td>88 mg</td>
<td>143.5 mg</td>
</tr>
</tbody>
</table>

**Table 8. Gasoline vs. Ethanol Injection Parameters**

**Comparison vs. Gasoline**

For a preliminary study an ethanol DI setup has been implemented using the same boundary conditions as the 2000 RPM gasoline validation case of Chapter III, with injection parameters as listed in Table 8. To provide equal comparison, spark timing is set to 705° for both cases. The gasoline validation case produced 28.3 horsepower at an equivalency of 1. Ethanol injection under the same equivalency produced 32.4
horsepower, a marked improvement. Additional benefits of ethanol operation are found in analyzing temperatures and pressure during the cycle.

<table>
<thead>
<tr>
<th></th>
<th>Gasoline</th>
<th>Ethanol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Horsepower</td>
<td>28.3</td>
<td>32.4</td>
</tr>
<tr>
<td>SFC (lbm/hp*hr)</td>
<td>0.340</td>
<td>0.580</td>
</tr>
<tr>
<td>Peak Pressure (MPa)</td>
<td>10.79</td>
<td>9.09</td>
</tr>
<tr>
<td>Peak Pressure Timing (°)</td>
<td>737</td>
<td>745</td>
</tr>
<tr>
<td>Peak Temperature (K)</td>
<td>2416</td>
<td>2102</td>
</tr>
<tr>
<td>Laminar Flame Speed at 700°</td>
<td>65.86 cm/sec</td>
<td>76.74 cm/sec</td>
</tr>
<tr>
<td>Unevaporated Fuel at 700° (%)</td>
<td>0</td>
<td>21.1</td>
</tr>
</tbody>
</table>

**Table 9. Gasoline vs. Ethanol Performance**

Table 9 shows some very insightful information as to benefits of ethanol operation. First, under operation at the same equivalency ethanol produced 4.1 more horsepower (14.5% increase). This increase comes with lower peak pressure and temperature, which is beneficial for engine longevity. The decreased peak values are expected with ethanol, as more cool fuel is injected into the hot chamber, thus decreasing the average temperature. Another beneficial aspect to ethanol is the increased flame speed, even at lower temperature. This ensures that the flame can propagate throughout the chamber effectively consuming all available fuel. Although it seems obvious that a higher peak pressure would result in more force on the piston thus producing more power, this is not always the case. This phenomenon can be explained by observation of the time at which peak pressure occurs for both cases. For the ethanol case, peak pressure is generated 8° later than for gasoline, which correlates to an increased combustion duration, thus a better power stroke. This is due to multiple factors; however the most likely contributor to increased combustion duration is the presence of liquid fuel at the time of spark. This ensures a lower temperature combustion, along with increased duration. This phenomenon is noted in Figure 23. In A, the gasoline particles appear to be completely evaporated inside the chamber, whereas B shows a significant amount of
ethanol left at 700 CAD. Observation of the P-V plots for each cycle also reflects the increased combustion duration of ethanol, evidenced by the wider curve in Figure 24.

![Figure 23. Liquid Particles at 700 CAD](image)

![Figure 24. P-V Comparison](image)

Based on the temperatures shown in Table 9, it is apparent that the temperature of the charge inside the chamber is lower for ethanol operation. This is due to the increased amount of fuel in the chamber for a stoichiometric mixture, in addition to the higher heat of vaporization of ethanol producing a charge cooling effect. Singh advises in [5] that for the low CO and hydrocarbon benefits of LTC to take effect, the charge temperature must be no greater than 2200 K. Table 9 shows that ethanol meets this qualification, with a
peak temperature of only 2102 K whereas gasoline operation reaches a peak temperature of 2416 K, well over the limit for LTC. This lower temperature is a key benefit for ethanol operation, as the benefits of low temperature combustion were studied in the Literature Review.

An expected downfall of ethanol operation comes in the form of increased specific fuel consumption. The stoichiometric air to fuel ratio for ethanol is 9:1, versus 14.7:1 for gasoline. This indicates that 63% more ethanol is needed to reach a stoichiometric mixture in the chamber. While this seems like a fundamental inefficiency, it should be noted that the spark timing and compression ratio have not been adjusted to maximize power and minimize SFC for the ethanol case. Further studies in this chapter aim to choose the best spark time and CR.

**Compression Ratio Study**

In the previous study, it was shown that ethanol operation results in lower peak temperature and pressure, while operating at higher horsepower. This contributes to the knock resistance of ethanol fuel, and may open the possibility of operation at higher compression ratios. In order to exploit the benefits of these aspects of combustion, ethanol DI is applied to the stock Ecotec with 9.2:1 compression in addition to modified computational models at 11:1 and 13:1 compression. To provide an equal comparison, spark timing is set to 690° (30 bTDC) for each case.

As indicated with the cold flow study of Chapter III, each compression ratio inducts a different amount of air, due to the geometry of the combustion chamber. This difference results in a separate amount of fuel required for a stoichiometric mixture at
each compression. Assuming an equivalency of 1.0 for operation conditions, the three compressions are compared as follows:

<table>
<thead>
<tr>
<th></th>
<th>9.2:1 Compression</th>
<th>11:1 Compression</th>
<th>13:1 Compression</th>
</tr>
</thead>
<tbody>
<tr>
<td>Horsepower</td>
<td>34.19</td>
<td>36.39</td>
<td>37.41</td>
</tr>
<tr>
<td>Fuel Injected (phi)</td>
<td>139.2 (.948)</td>
<td>140.827 (0.980)</td>
<td>141.836 (1.007)</td>
</tr>
<tr>
<td>Peak Pressure (MPa)</td>
<td>11.26</td>
<td>17.23</td>
<td>22.82</td>
</tr>
<tr>
<td>Peak Pressure Timing (°)</td>
<td>737.000</td>
<td>727.5</td>
<td>725</td>
</tr>
<tr>
<td>Peak Temperature (K)</td>
<td>2109</td>
<td>2148</td>
<td>2288</td>
</tr>
<tr>
<td>Laminar Flame Speed at 700° (cm/sec)</td>
<td>68.85</td>
<td>76.74</td>
<td>82.14</td>
</tr>
<tr>
<td>Unevaporated Fuel at 700° (%)</td>
<td>30.8</td>
<td>40.2</td>
<td>32.7</td>
</tr>
</tbody>
</table>

**Table 10. Performance at Varying CR**

As shown in Table 10, 13:1 CR showed the highest horsepower, with a 9.4% increase over 9.2 CR and a 2.8% increase over 11:1. As expected, peak pressures and temperatures increase as CR increases. This contributes to a higher LFS at each CR. Slight variations in equivalency are noted, especially at the lowest compression ratio. The induction of air used to predict fuel amounts is based on a cold flow simulation, whereas when injection occurs during intake the incoming air is cooled and density increases, thus allowing more air into the chamber. This process is known as charge cooling, and it results in a slightly lower equivalency in the chamber.

It is important to note the timing of maximum cylinder pressure for each case. Observation reveals that the lowest CR produced a much later peak pressure timing under the same spark timing. This is likely due to mixture preparation at 690°. Due to the lower temperature and pressures experienced, combustion is not favorable this early in the cycle, thus peak pressures are not developed until well after TDC. For higher compression cases, the pressures are sufficiently high to produce peak pressures shortly after TDC, enabling more use of the power stroke. Figure 25 shows P-V Diagrams at each
compression ratio. The gain in horsepower can be seen in the much higher pressure peaks of Figure 25 B and C. This enables more “area under the curve”, which translates to work and thus power.

9.2 Compression

11 Compression
Spark Timing Study

Previous cases have focused on comparing fuels and compression ratios using the same spark timing. Because data is not available for Ecotec performance when fueled by ethanol, nor is performance data available for higher CR operation, it is necessary to examine the properties of ethanol fuel on each CR at varying spark initiation times. To cover a wide range of timings, four cases were run at each CR. Fuel injection parameters were identical to those in above study, in order to ensure and equivalency of 1 for each case.
Figure 26 shows the power produced at four varying spark times for the stock 9.2 CR. It is noticeable that power depends strongly on spark timing. Test data from the Ecotec on gasoline indicate a factory spark timing of 705°, or 15° advance. However computations with ethanol show the performance is benefitted by more advance. An earlier spark time ensures that the rate of combustion is maximum when the piston is close to TDC, thus making better use of the power stroke.

Figure 27 shows the same study for 11:1 CR. Again, performance is benefitted by an earlier spark advance. Due to the increased amount of compression in the chamber, the fuel-air mixture is under more pressure before spark initiation. This results in a lower flame speed, as increased pressure will slow flame propagation. Issuing the spark further in advance of TDC allows time for the small flame front to heat the chamber. As the flame kernel grows, the rate of combustion will reach a maximum and pressure will be generated in time to maximize work on the piston.
Figure 27. Power vs. Spark Time, 11 CR

Figure 28. Power vs. Spark Time, 13.2 CR

Figure 28 shows that the highest CR tested is also benefitted by increased spark advance. As the pressures and temperatures experienced within the chamber are highest
in a 13:1 CR, it is expected that this model will show the most sensitivity to spark timing. It is far more crucial here for the spark advance to account for the increase in pressure to propagate flame effectively. Using a 2\textsuperscript{nd} order polynomial fit routine, a trendline was generated for each CR, and an optimum spark time for each was predicted.

<table>
<thead>
<tr>
<th>Compression</th>
<th>9.2:1</th>
<th>11:1</th>
<th>13:1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Optimal Time</td>
<td>692</td>
<td>695.5</td>
<td>695.75</td>
</tr>
<tr>
<td>Predicted HP</td>
<td>34.16</td>
<td>37.1</td>
<td>39.8</td>
</tr>
<tr>
<td>Goodness of Fit</td>
<td>.997</td>
<td>.967</td>
<td>.956</td>
</tr>
</tbody>
</table>

**Table 11. Optimal Predictions**

**Conclusions from Ethanol Study**

<table>
<thead>
<tr>
<th>CR</th>
<th>Spark Timing</th>
<th>Fuel Injected</th>
<th>Ignition Delay</th>
<th>SFC</th>
<th>Horsepower</th>
</tr>
</thead>
<tbody>
<tr>
<td>9.2:1</td>
<td>680</td>
<td>143.65</td>
<td>96.746</td>
<td>0.57</td>
<td>33.043</td>
</tr>
<tr>
<td></td>
<td>690</td>
<td>143.65</td>
<td>118.075</td>
<td>0.527</td>
<td>34.194</td>
</tr>
<tr>
<td></td>
<td>700</td>
<td>143.65</td>
<td>124.33</td>
<td>0.543</td>
<td>33.627</td>
</tr>
<tr>
<td></td>
<td>710</td>
<td>143.65</td>
<td>116.818</td>
<td>0.598</td>
<td>31.752</td>
</tr>
<tr>
<td>11.2:1</td>
<td>680</td>
<td>141.81</td>
<td>4.526</td>
<td>0.524</td>
<td>34.18</td>
</tr>
<tr>
<td></td>
<td>690</td>
<td>141.81</td>
<td>21.549</td>
<td>0.5</td>
<td>36.385</td>
</tr>
<tr>
<td></td>
<td>700</td>
<td>141.81</td>
<td>73.072</td>
<td>0.487</td>
<td>37.13</td>
</tr>
<tr>
<td></td>
<td>710</td>
<td>141.81</td>
<td>126.834</td>
<td>0.511</td>
<td>34.339</td>
</tr>
<tr>
<td>13.2:1</td>
<td>680</td>
<td>138.95</td>
<td>0.517</td>
<td>0.482</td>
<td>34.039</td>
</tr>
<tr>
<td></td>
<td>690</td>
<td>138.95</td>
<td>12.018</td>
<td>0.49</td>
<td>37.41</td>
</tr>
<tr>
<td></td>
<td>700</td>
<td>138.95</td>
<td>47.986</td>
<td>0.496</td>
<td>36.762</td>
</tr>
<tr>
<td></td>
<td>710</td>
<td>138.95</td>
<td>89.49</td>
<td>0.512</td>
<td>34.703</td>
</tr>
</tbody>
</table>

**Table 12. Ethanol Case Study Conclusions**
Table 12 shows a complete summary of the ethanol case study. Three compression ratios were tested, along with four spark timings at each compression ratio. Optimal spark timings were predicted at each compression ratio. Best spark timings for ethanol cases were found to be consistently earlier in the cycle than for gasoline. Horsepower was found to increase with compression ratio, in addition to a decreased SFC. A horsepower increase of 58.5% was found when comparing 13:1 ethanol operation to 9.2:1 gasoline operation. An approximation of ignition delay time was formulated to provide a realistic means of analyzing likelihood of autoignition. Ignition delay time was the least for the highest CR, however sufficient time is still available to complete combustion without the occurrence of engine knock.
V. BI-FUEL INJECTION STUDY

In the previous chapter it was demonstrated that fueling the Ecotec with ethanol under the same RPM and turbocharger conditions provides a significant increase in horsepower. The high CR models provided even further increased output. This knowledge is expanded upon to develop a bi-fuel injection computational model for 13:1 compression. The effect of various blend ratios of gasoline and ethanol from separate fuel injectors is studied. The relative injection timing of each respective fuel is examined. Results are compared to purely gasoline and purely ethanol operation.

Figure 29. Bi-Fuel Injection of Ethanol and Gasoline
**Model Settings**

For the course of this study, 2000 RPM is used in order to simulate cruising conditions. The turbocharger pressure was assumed to be the same as that for the 2000 RPM gasoline validation case. Thus, the amount of air inducted is assumed to be the same when calculating stoichiometric conditions within the chamber.

In order to simulate a realistic implementation of fuel injectors, a secondary injector is placed in the top of the cylinder head, directly opposing the stock injector. Injector properties such as number of orifices, injector pressure, fuel flow rates, and exiting particle diameter are assumed to be the same. Injector pulse width varies widely here, depending on what type of fuel and what mass fraction of fuel is being used. The fuel injection configuration is shown in Figure 29, with the gasoline injection (dark blue) occurring in the stock location and ethanol (light blue) occurring directly opposite. Here, both injection start at the stock injection time of 417 CAD.

**Flame Speed Correlation for Blends**

Flame speed is a critical aspect of accurately modeling combustion inside an engine. As previously shown, the flame speed of gasoline is significantly lower than that of ethanol. This affects performance and quality of combustion, and often necessitates different spark timing. Therefore, it is important to model flame speed as accurately as possible. Extensive publications on flame speeds for ethanol and gasoline at varying temperature, pressure, and equivalency exist, making accurate correlation somewhat simple. However, the lack of published data on various blend ratios calls for the use of a
predictive model of the flame speed. Z. Chen, Dai, and S. Chen [3] offer a model for blends of no more than two fuels, as follows.

\[ S_L = \frac{m}{\rho_u} \]

**Equation 4. Laminar Flame Speed**

Where \( S_L \) is the laminar flame speed, \( m \) is the mass flux, and \( \rho_u \) is the unburnt mixture density. Mass flux for a binary mixture is found in Equation 5, where \( Y \) represents the mass fraction of each fuel and \( c \) is a free parameter representing a ratio of the chemical heat release rate per unit mass of the fuels.

\[ m^2 = \frac{cY_1 m_1^2 + Y_2 m_2^2}{cY_{1,u} + cY_{2,u}} \]

**Equation 5. Mass Flux**

From these equations, it is expected that the flame speeds of the fuel blends will be within those of either pure ethanol or pure gasoline. This fact is evidenced in Figure 30, showing the correlation results compared to published data by Takashi [6]. It is seen that ethanol has the highest flame speed and accordingly the blends with the highest ethanol content have highest flame speeds. The plots for iso octane and ethanol are from published experimental results. It is noteworthy to mention that every blend ratio is in between these two published values, therefore it is reasonable to assume the predictive flame speed model is relatively accurate.
Equivalence Calculation for Blends

Overall equivalence ratio inside the combustion chamber is an important means of gauging the quality and characteristics of combustion during operating cycle.

Equivalence is calculated and stored at each time step using a “Custom Field Function” inside FLUENT. For all previous gasoline and ethanol cases, the fuel was delivered to reach an equivalence of unity. Thus it is necessary to monitor the overall equivalence for the bi-fuel cases, to ensure they are operating at the same equivalence ratio. Because gasoline and ethanol each have a very different stoichiometric air to fuel ratio (AFR), it is necessary to adapt the equation of equivalency to reflect this.

\[ AFR_s = 9 \times e_m + 14.6 \times (1 - e_m) \]

Where \( e_m \) is the mass fraction of ethanol in the total amount of fuel. With this adjusted stoichiometric AFR, the equivalence is calculated as follows:

\[
\Phi = \frac{FAR}{FAR_s} = \frac{MF_{eth}}{9 \times (1 - MF_{eth})} + \frac{MF_{gas}}{14.6 \times (1 - MF_{gas})}
\]

**Equation 7. Equivalence Ratio Calculation for Gasoline-Ethanol Blend**

**Experiment Setup**

<table>
<thead>
<tr>
<th>Case</th>
<th>Blend Ratio (MFR Gasoline/MFR Ethanol)</th>
<th>Injection</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.1</td>
<td>75/25</td>
<td>Both Standard</td>
</tr>
<tr>
<td>5.2</td>
<td>75/25</td>
<td>Late Ethanol</td>
</tr>
<tr>
<td>5.3</td>
<td>75/25</td>
<td>Late Gasoline</td>
</tr>
<tr>
<td>5.4</td>
<td>50/50</td>
<td>Both Standard</td>
</tr>
<tr>
<td>5.5</td>
<td>50/50</td>
<td>Late Ethanol</td>
</tr>
<tr>
<td>5.6</td>
<td>50/50</td>
<td>Late Gasoline</td>
</tr>
<tr>
<td>5.7</td>
<td>25/75</td>
<td>Both Standard</td>
</tr>
<tr>
<td>5.8</td>
<td>25/75</td>
<td>Late Ethanol</td>
</tr>
<tr>
<td>5.9</td>
<td>25/75</td>
<td>Late Gasoline</td>
</tr>
</tbody>
</table>

**Table 13. Bi-Fuel Case Setup**

For the bi-fuel injection study, nine cases are to be run. Design variables include fuel blend ratios (3) and injection timing strategies (3). Blend ratios are defined by mass fractions. Injection timings are defined as shown in Table 14. Start of injection is either at 417 CAD or 630 CAD (early or late). End of injection varies as the mass fraction of fuel changes, because the flow rate of the fuel injectors is assumed the same as validation cases and ethanol cases to provide a realistic injector model. Amount of fuel injected is defined such that the overall equivalency in the chamber is 1, as defined by Equation 6.

<table>
<thead>
<tr>
<th>Injection Label</th>
<th>Start of Injection (SOI) Gasoline</th>
<th>SOI Ethanol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Both Standard</td>
<td>417</td>
<td>417</td>
</tr>
<tr>
<td>Late Ethanol</td>
<td>417</td>
<td>630</td>
</tr>
<tr>
<td>Late Gasoline</td>
<td>630</td>
<td>417</td>
</tr>
</tbody>
</table>

**Table 14. Injection Label Definition**
Figure 31 shows the horsepower output from each case at the varying blend ratios and injection timings. The average output HP is highest for the 25% gas blend, and lowest for 75% gasoline. This is logical, as pure ethanol produced greater horsepower than gasoline. At each blend ratio, the injection timing showed significant and similar trends. A late ethanol injection was consistently the lowest power performance, with both standard and late gasoline offering higher power.

Investigation into the contours of equivalency at the time of spark reveals insight as to why late ethanol injection offers poor performance. As ethanol has a higher heat of vaporization, a late injection does not offer sufficient time for fuel evaporation. Thus, the equivalency in the chamber is lower than the ideal mixture of ‘1’. When gasoline is injected late, it is more likely to vaporize and become a combustible mixture.
Case 5.7
Both standard injections
Mass Fraction Gas=25
Mass Fraction Ethanol=75

At time of spark:
100% Gas Evaporated
84% Ethanol Evaporated

Case 5.8
Delayed Ethanol Injection
Mass Fraction Gas=25
Mass Fraction Ethanol=75

At time of spark:
100% Gas Evaporated
57% Ethanol Evaporated

Case 5.9
Delayed Gasoline Injection
Mass Fraction Gas=25
Mass Fraction Ethanol=75

At time of spark:
100% Gas Evaporated
89% Ethanol Evaporated

Figure 32. Phi at Spark Time for Bifuel Cases
Analysis of specific fuel consumption (SFC) shows a different conclusion for optimum setup. Again, both blend ratio and injection strategy showed significant impact on SFC. However, the 50/50 blend produced the lowest SFC for every injection setup. Late ethanol injection performed the worst, whereas the best performance was given by late gasoline injection. This agrees with trends for maximum horsepower analysis. The lowest SFC value was found for a 50/50 fuel mixture with late gasoline injection. This is a non-intuitive result, as it is not the location of the highest horsepower. This key aspect suggests that the interaction of the flame speed produced by a 50/50 mixture of fuel, in addition to the location of fuel at time of spark produces efficient combustion when utilizing a bi fueled injection strategy.
Conclusions

<table>
<thead>
<tr>
<th>Case</th>
<th>Blend Ratio (MFR Gasoline/MFR Ethanol)</th>
<th>Injection</th>
<th>Spark Timing</th>
<th>Ignition Delay</th>
<th>SFC</th>
<th>HP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ref</td>
<td>100/0</td>
<td>Standard</td>
<td>705</td>
<td>67.118</td>
<td>0.34</td>
<td>23.658</td>
</tr>
<tr>
<td>5.1</td>
<td>75/25</td>
<td>Both Standard</td>
<td>690</td>
<td>40.952</td>
<td>0.433</td>
<td>29.581</td>
</tr>
<tr>
<td>5.2</td>
<td>75/25</td>
<td>Late Ethanol</td>
<td>690</td>
<td>51.341</td>
<td>0.567</td>
<td>22.762</td>
</tr>
<tr>
<td>5.3</td>
<td>75/25</td>
<td>Late Gasoline</td>
<td>690</td>
<td>24.569</td>
<td>0.457</td>
<td>27.968</td>
</tr>
<tr>
<td>5.4</td>
<td>50/50</td>
<td>Both Standard</td>
<td>690</td>
<td>31.15</td>
<td>0.408</td>
<td>36.294</td>
</tr>
<tr>
<td>5.5</td>
<td>50/50</td>
<td>Late Ethanol</td>
<td>690</td>
<td>60.529</td>
<td>0.453</td>
<td>31.778</td>
</tr>
<tr>
<td>5.6</td>
<td>50/50</td>
<td>Late Gasoline</td>
<td>690</td>
<td>28.899</td>
<td>0.391</td>
<td>37.525</td>
</tr>
<tr>
<td>5.7</td>
<td>25/75</td>
<td>Both Standard</td>
<td>690</td>
<td>29.163</td>
<td>0.427</td>
<td>38.706</td>
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<tr>
<td>5.8</td>
<td>25/75</td>
<td>Late Ethanol</td>
<td>690</td>
<td>91.944</td>
<td>0.501</td>
<td>33.063</td>
</tr>
<tr>
<td>5.9</td>
<td>25/75</td>
<td>Late Gasoline</td>
<td>690</td>
<td>32.418</td>
<td>0.414</td>
<td>39.067</td>
</tr>
<tr>
<td>Ref</td>
<td>0/100</td>
<td>Standard</td>
<td>690</td>
<td>12.018</td>
<td>0.49</td>
<td>37.41</td>
</tr>
</tbody>
</table>

Table 15. Summary of Cases and Outputs

Table 15 briefly summarizes the outputs of the bi fuel injection cases, along with benchmark cases for pure gasoline and pure ethanol for comparison purposes. It can be seen that the horsepower is maximized when employing a 25% gasoline, 75% ethanol blend with a late gasoline injection. The output 39 HP here is greater than the output for the pure ethanol benchmark case. Because ethanol has a higher energy content, it seems logical that the highest horsepower would occur on the 100% ethanol benchmark, however this is not the case. Because of the late gasoline injection, combustion is promoted for a longer period of time during the power stroke. With pure ethanol, combustion duration is relatively short as the flame speed is higher and all of the fuel present is vaporized to combust. Employing a late secondary injection cools the chamber and introduces more particles to evaporate, which slows flame speed and extends combustion duration. Care must be taken not to introduce too much fuel late in the cycle however, as all the mixture may not evaporate. This is seen with many of the cases that employ a late ethanol injection.
SFC consumption is higher with all cases employing ethanol than for the benchmark case with pure gasoline. This is largely due to the fact that more ethanol is necessary for a stoichiometric mixture, evidenced by the 9:1 stoich AFR of ethanol vs. 14.7:1 for gasoline. The best SFC came from a 50/50 mixture with late gasoline injection, at 0.391. This is a 15% increase over the gasoline baseline, however it comes with a 58% increase in power. It is reasonable to assume that if the amount of fuel injected for the bi-fuel case is decreased to obtain the stock horsepower, the SFC would also be decreased, thus producing a more economical engine operation.
VI. FUTURE WORK

**Ethanol Injection**

In order to make pure ethanol operation a more feasible option for the 2.0 L Ecotec engine, additional research is needed. Currently, the best performing ethanol case from Chapter 4 shows power outputs of over 37 hp. This is a measured 13 hp more than the stock Ecotec at 2000 RPM. Future work involving the investigation of decreasing the turbocharger pressure would be beneficial to the research. At higher compression ratio, temperatures and pressures have proven sufficient to produce an ignitable mixture, therefore it is feasible to investigate use without turbocharged pressures. With less fuel injected, the specific fuel consumption of the engine will decrease to a more favorable range and the engine will produce a lower power more suitable for everyday driving.

Additionally, the research herein has been performed assuming the same fuel injection parameters as with the stock gasoline injector. Implementation of different injectors should be tested in both an experimental and analytical setting. Because ethanol shows different breakup droplet size and evaporation rates, it is reasonable to assume that a gasoline injector is not the optimum solution for ethanol fuel delivery. Use of all fuel injected is paramount to creating an efficient engine.

**Bifuel Injection**

In addition to studying the mixture ratio and injection timing strategy of the bifuel injection, a preliminary spark timing study is necessary to ensure maximum use of the
power stroke. As more injectors are introduced to the engine, more experimental variables become present that must be investigated in order to proceed effectively. As more experimental cases are run, the optimal injection timings can be further defined, and their interaction with spark timing and location analyzed.

Other variables introduced with the addition of a fuel injector are secondary injector placement and orientation, along with piston bowl geometry. As demonstrated in the validation and single injection cases, the piston bowl is crucial in delivering the fuel to the vicinity of the spark plug. This wall guide system is not designed for two injectors, however. Future work involving the design of a piston bowl and secondary injector interaction to effectively deliver fuel to the spark plug can prove very beneficial to engine performance.
VII. Conclusions

A computational model of the direct injection 2.0 L Ecotec engine is developed to examine the feasibility of operation with alternative fuel sources. The outputs of the computational model are compared to published experimental results, and a good correlation is seen. Models with increased compression ratios are then developed, utilizing the same boundary conditions. Ethanol delivery from the stock fuel injector model is investigated on the varying compression ratio models. Additionally, a dual-injection gasoline ethanol operation mode is investigated using the high compression ratio model.

After investigating the flow characteristics inside the chamber of the validated model, a parametric study of ethanol operation is performed and conclusions drawn from the results.

1. With stock compression ratio, spark timing, and injection timing, ethanol produces 4 additional horsepower per cylinder. However, not all ethanol is evaporated at time of spark, indicating the need for higher compression ratio.

2. With stock operating conditions, ethanol fuel produced a peak temperature 300 K lower than gasoline operation. This factor aides in resisting autoignition, which makes ethanol a good candidate for high compression applications.

3. A 58.5% increase in power compared to gasoline was found when ethanol operation was applied to a 13:1 compression ratio model. Peak temperature in the cycle was
significantly lower than those experienced during gasoline operation, thus indicating low risk of autoignition.

4. Spark timing investigation on each compression ratio model indicates an earlier spark benefits ethanol operation, as the predicted best timing lies around 25 degrees BTDC.

The aforementioned conclusions gave helpful insight to the implementation of a dual injector, bifuel operation mode in the high compression model. Baseline assumptions contributing to the dual-injector cases include implementing an earlier than stock spark timing, in addition to a 13:1 compression model.

1. Dual injection provides more precise control of the power stroke, as a late injection provides increased duration of combustion and further resistance to autoignition.

2. Injecting gasoline late in the compression stroke provides better results than its ethanol counterpart, as ethanol requires more time for the fuel to evaporate.

3. By using a 50% ethanol 50% gasoline mixture with a late gasoline injection, power is increased by 59%. This is accompanied by a mere 15% increase in specific fuel consumption, indicating substantial benefit in efficiency.

4. Greatest power is achieved with a 75% ethanol mixture, however the specific fuel consumption is also greatest with this mixture ratio, due to the high amount of ethanol.
APPENDIX

Bibliography


References


