2010

Investigation of EGR and Fuel Property Effects on Advanced Combustion Strategies Using In-cylinder Combustion Analyses

Jason D. Ice
West Virginia University

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Investigation of EGR and Fuel Property Effects on Advanced Combustion Strategies Using In-cylinder Combustion Analyses

Jason D. Ice

Thesis submitted to the
College of Engineering and Mineral Resources
at West Virginia University
in partial fulfillment of the requirements
for the degree of

Master of Science
in
Mechanical Engineering

Benjamin Shade, Ph.D., Chair
John Nuszowski, Ph.D.
Hailin Li, Ph.D.

Department of Mechanical and Aerospace Engineering
Morgantown, West Virginia

2010

Keywords: Advanced Combustion, EGR, Fuel Properties, In-cylinder Combustion, Diesel

Abstract

Investigation of EGR and Fuel Property Effects on Advanced Combustion Strategies Using In-cylinder Combustion Analyses

Jason D. Ice

Advanced combustion has progressively become a topic of interest as government regulations have stringently reduced the allowable limits of engine-out emissions from internal combustion engines. Advanced combustion can typically be classified as homogeneous charge compression ignition (HCCI) or premixed charge compression ignition (PCCI). In HCCI combustion, a well mixed fuel and air solution is spontaneously combusted, thus providing a homogeneous burn throughout the combustion process. By having an evenly distributed, homogeneous burn, fuel economy is not only significantly increased, but both oxides of nitrogen (NO_x) and particulate matter (PM) can drastically be reduced from that of conventional combustion. HCCI has the ability to incorporate the low PM emissions of a spark ignited (SI) engine as well as the high thermal efficiency of a compression ignited (CI) engine. Similar to HCCI, PCCI offers low NO_x and PM within a part load operating range with a quasi-homogenous fuel and air mixture. Other than identifying HCCI or PCCI as having very limited NO_x and PM production, in-cylinder combustion techniques can also be utilized to characterize these forms of advanced combustion.

An experimental study was performed at West Virginia University (WVU) in an attempt to achieve HCCI or PCCI operation by implementing a number of engine control strategies using a range of fuels with varying fuel properties. The test engine, a GM 1.9L, was incorporated with an ECU controller capable of controlling engine operating parameters such as rail pressure, start of injection (SOI), fuel quantity injected and exhaust gas recirculation (EGR) valve position. Piezoelectric crystal pressure transducers were introduced into each cylinder via glow plug adapters and a custom WVU data acquisition system was used to evaluate in-cylinder combustion parameters. Testing was performed at a 2100 RPM, 53 N-m, steady-state set point using an engine dynamometer. By implementing high levels of EGR, NO_x emissions were reduced nearly 90% due to lower cylinder pressures and increased ignition delay. PM emissions were also reduced by 15% within a specific EGR range due to increased mixing time available from a longer ignition delay as verified by in-cylinder pressure, heat release rate and mass fraction burned data. Fuel characteristics were compared at a single operating condition and at an optimized control strategy with increased thermal efficiency, for low NO_x and PM emissions that were determined from a range of EGR, pilot injection, main injection, rail pressure and fuel split sweeps. The cetane number (CN) of the fuel was found to be the contributing factor affecting the combustion process compared to fuel properties such as aromatics and volatility while the net heat of combustion per unit volume effected the brake specific fuel consumption. Low CN fuels resulted in a PM reduction of 30% but showed a 50% increase in NO_x emissions over the higher CN fuels at the same operating condition.

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Nomenclature

β	Ratio of Venturi Throat Diameter to Approach Pipe Diameter
θ	Crank Angle
ρ_i	Density at Inlet Conditions
ρ_s	Density at Standard Conditions
γ	Specific Heat Ratio
AF	Air-Fuel
ASTM	American Society for Testing and Materials
ATDC	After Top Dead Center
AVFL	Advanced Vehicle, Fuels and Lubricants
bmep	Brake Mean Effective Pressure
BTDC	Before Top Dead Center
BSFC	Brake Specific Fuel Consumption
CAFE	Corporate Average Fuel Economy
CAFEF	Center for Alternative Fuels, Engines and Emissions
C_d	Coefficient of Discharge
CFR	Code of Federal Regulations
CI	Compression Ignited
CO	Carbon Monoxide
CO ₂	Carbon Dioxide
CVS	Constant Volume Sampling
d	Venturi Throat Diameter
DAQ	Data Acquisition
DPF	Diesel Particulate Filter
ECU	Engine Control Unit
EERL	Engines and Emissions Research Laboratory
EGR	Exhaust Gas Recirculation
EOC	End of Combustion
FACE	Fuels for Advanced Combustion Engines

H ₂ O	Water
HC	Hydrocarbon
HCCI	Homogeneous Charge Compression Ignition
HDDE	Heavy Duty Diesel Engine
HECC	High Efficiency Clean Combustion
HEPA	High Efficiency Particulate Air (Filter)
HFID	Heated Flame Ionization Detector
HRR	Heat Release Rate
IVC	Intake Valve Closing
K _q	Venturi Flow Coefficient
LFE	Laminar Flow Element
LHV	Lower Heating Value
LTC	Low Temperature Combustion
	Mass Flowrate of Fuel
MEXA	Motor Exhaust Gas Analyzer
MFB	Mass Fraction Burned
	Number of Cylinders
N ₂	Diatomic Nitrogen
N ₂ O	Nitrous Oxide
NDIR	Non-Dispersal Infrared
NI	National Instruments
NO	Nitric Oxide
NO _x	Oxides of Nitrogen
O ₂	Diatomic Oxygen
OEM	Original Equipment Manufacturer
ORNL	Oak Ridge National Laboratory
	Pressure at Intake Valve Closing
	Pressure at Specified Crank Angle
P	Cylinder Pressure

ΔP	Pressure Drop across Venturi Throat
PCCI	Premixed Charge Compression Ignition
PM	Particulate Matter
Q	Heat Released
Q_s	Standard Volumetric Flowrate
R	Universal Gas Constant
ROHR	Rate of Heat Release
RPM	Revolutions per Minute
RTD	Resistance Temperature Device
SCR	Selective Catalytic Reduction
SI	Spark Ignited
SOC	Start of Combustion
SOF	Soluble Organic Fraction
SOI	Start of Injection
SSV	Subsonic Venturi
	Temperature at Intake Valve Closing
	Temperature at Specified Crank Angle
T50	Temperature at 50% Burn
T90	Temperature at 90% Burn
T95	Temperature at 95% Burn
TDC	Top Dead Center
THC	Total Hydrocarbons
	Volume at Intake Valve Closing
	Volume at Specified Crank Angle
V	Cylinder Volume
VCR	Variable Compression Ratio
VG	Variable Geometry Turbocharger
WVU	West Virginia University
Y	Expansion Factor

1 Introduction and Objectives

1.1 Introduction

Advanced combustion has progressively become a topic of interest as government regulations have stringently reduced the allowable limits of engine-out emissions from internal combustion engines. A recent global proposal imposed by the Environmental Protection Agency (EPA), proposed a 20% reduction in both greenhouse gas emissions (GHG) emissions and fuel consumption by 2018 [1], leading advanced combustion techniques as an alternative to hybrid vehicle technology as a means to achieve the initiative.

Advanced combustion can typically be classified as homogeneous charge compression ignition (HCCI) or premixed charge compression ignition (PCCI). In HCCI combustion, a well mixed fuel and air mixture is spontaneously combusted, thus providing a homogeneous burn throughout the combustion process. By having an evenly distributed burn, fuel consumption is not only significantly decreased, but both oxides of nitrogen (NO_x) and particulate matter (PM) are also drastically reduced compared to that of conventional combustion. HCCI has the ability to achieve the low PM emissions of a spark ignited (SI) engine and the high thermal efficiency of a compression ignited (CI) engine [2]. Similar to HCCI, PCCI offers low NO_x and PM within a part load operating range with a quasi-homogenous fuel and air mixture.

A great deal of research has been performed on ways of controlling and sustaining HCCI-like events during steady state and transient engine operation. Simultaneous reductions in NO_x , PM and fuel consumption are mainly dependent on engine design, fuel properties and engine operating parameters such as exhaust gas recirculation (EGR) fraction and injection strategies. It is often important, in advanced combustion studies, to examine both the advantages and disadvantages of different fuels over a range of operating conditions and their effect on engine performance characteristics. The main disadvantage of advanced combustion, especially HCCI, is the difficulty of controlling the combustion phasing on a transient basis. Other than identifying HCCI or PCCI as very limited NO_x and PM emissions, computational in-cylinder combustion analyses such the heat release rate and mass fraction burned curves, can be utilized to characterize these forms of advanced combustion.

1.2 Objectives

The main objective of this research was to experimentally examine the effect of EGR and fuel properties on advanced combustion strategies through in-cylinder pressure analyses. The secondary objectives of this research were to improve brake thermal efficiency without compromising the integrity of the engine and to determine the characteristics of a fuel that affect the achievable advanced combustion operating range of the test engine.

2 Literature Review

This section examines the concept of advanced combustion operation and strategies associated with lowering engine-out emissions without the use of aftertreatment systems. Section 2.1 describes different advanced combustion techniques along with the operating parameters and engine geometries that may affect the quest for a simultaneous reduction in both NO_x and PM from diesel engines. Section 2.2 examines in-cylinder pollutant formation and analysis techniques used to identify advanced combustion or predict engine-out emissions. Section 2.3 describes various fuel properties that may affect the pursuit of advanced combustion and the consequence they have on pollutant formation. Finally, §2.4 examines a number of EGR configurations and how they are used to meet the very stringent regulations set forth by government agencies.

2.1 Advanced Combustion in Diesel Engines

2.1.1 Introduction

Diesel engines have historically been used in many production applications including heavy duty vehicles and numerous commercial products. Not only do they have high thermal efficiencies, but diesel engines are also very durable. A common dilemma with the diesel engine is the excess emissions of NO_x and PM. NO_x is formed in the high temperature regions of combustion while PM is formed mainly within the fuel-rich regions of combustion. With emissions regulations becoming increasingly stringent, a simultaneous reduction in both NO_x and PM is highly sought after. There are many ways to reduce both NO_x and PM through advanced combustion methods, but HCCI is seemingly the most promising.

2.1.2 Homogeneous Charge Compression Ignition

The concept of HCCI has been around for nearly two decades. First introduced as a way to reduce unburned hydrocarbon emissions and enhance part-load performance of a 2-stroke SI engine [2], HCCI has now become a method of simultaneously reducing both NO_x and PM emissions without decreasing the thermal efficiency of diesel engines. The goal of HCCI is to eliminate locally rich or lean mixture regions through combusting a homogeneous mixture of fuel and air. By obtaining a homogenous fuel and air mixture, in theory, combustion will occur spontaneously throughout the combustion chamber as seen in Figure 1.

Figure 1: HCCI Combustion versus Conventional Diesel Combustion [3]

In order to have an effective HCCI combustion event, engine operating parameters and fuel characteristics must be optimized to enhance homogenous mixing. Typical engine operating parameters that must be controlled include injection timing, injection pressure, EGR, and valve timing [4]. Fuel characteristics such as cetane number, aromatics and fuel volatility also play a major role in the development of a homogenous mixture. Once HCCI combustion has occurred, it is very difficult to maintain and control throughout various engine speeds and loads. Due to the complexity of controlling HCCI, various strategies have been developed to obtain stable HCCI under various operating conditions.

2.1.3 Effect of Operating Parameters on HCCI

HCCI is difficult to control and maintain but can be achieved by optimizing engine operating parameters. Specific engine parameters such as EGR, start of injection, boost pressure, rail pressure, etc. can be controlled through the use of open source software such as Drivven or dSPACE controllers connected directly to an OEM wiring harness. The primary complexity in controlling HCCI is control of combustion near TDC without the occurrence of knock [2]. Injection timing must generally be advanced for a longer ignition delay to achieve true HCCI, however advancing the injection timing typically leads to wall wetting, low combustion temperatures and high HC emissions. In this regard, injection timing must be optimized with other parameters such as injection pressure and intake air temperature to avoid misfire and knock. Engine knock is often associated with advanced combustion regimes but can be detrimental to an engine's structural integrity, especially during single injection events. Engine knock can be quantified by a ringing intensity and corresponding "knock index". The intensity of knock can be identified based on an acoustic wave generated by pressure oscillations within the cylinder, often measured with a knock analyzer [5]. To suppress engine knock and limit fuel impingement on the cylinder wall, a dual injection or split-injection strategy is often used for advanced combustion studies. In a split-injection strategy, a pilot injection precedes the main injection by a specified duration to limit pressure rise rates and suppress HC emissions by allowing an extra stage of heat release prior to the main injection.

To compensate for the low combustion temperatures, intake air temperature is generally raised to increase the chance of auto-ignition. Increasing intake air temperatures reduce PM and HC emissions by elevating combustion flame intensity due to enhanced vaporization, as well as shifts the start of combustion closer to TDC for a more homogeneous mixture [2]. EGR also plays an important role for determining homogeneity within the combustion chamber. By implementing EGR, a reduction in combustion temperature is observed due to EGR products entering the combustion chamber. Components of EGR, especially CO_2 and H_2O , have a higher specific heat ratio than air thus act as a heat sink. The use of EGR, with proper EGR rates, can phase combustion near TDC and when combined with

optimized injection timing, can lead to a homogenous mixture ultimately promoting HCCI operation.

Another important operating parameter to promote HCCI operation is equivalence ratio. The equivalence ratio is defined as the stoichiometric AF ratio divided by the actual AF ratio. The equivalence ratio has the largest effect on local combustion temperatures during the combustion process. Operating toward stoichiometric conditions can lead to rapid combustion that can be detrimental to an engine's integrity. For advanced combustion it is important to operate very lean by utilizing a higher AF ratio (lower equivalence ratio) than conventional CI combustion as seen in Figure 2.

Figure 2: Local Equivalence Ratio versus Local Temperature for Various Advanced Combustion Strategies [6]

2.1.4 Effect of Engine Geometry on HCCI

The compression ratio of an engine plays a crucial role for the development of HCCI. As compression ratio increases, auto-ignition of the fuel and air mixture occurs much before TDC. At low compression ratios, better efficiencies and power output are observed by shifting ignition timing toward an optimized location [2]. In order to optimize the compression ratio of an engine and to promote an HCCI combustion event, various operating parameters must be controlled over a range of compression ratios, hence a variable compression ratio (VCR) engine. Since VCRs are expensive and generally only used as single cylinder research engines, the combustion chamber is often modified to promote HCCI operation. The combustion chamber

can have a significant effect on heat release rate due to turbulent mixing and mean velocity within the piston bowl. Generally, the heat release rate in direct injection (DI) and SI engines is proportional to turbulence and mean velocity; however in HCCI the opposite is observed [7]. A study at the Lund Institute of Technology compared a square bowled combustion chamber to a flat shaped combustion chamber and found that the peak combustion rate varied greatly between the two configurations. The square bowled configuration showed a significantly higher combustion rate, a much faster combustion duration and increased turbulence compared to the flat shaped combustion chamber. It was determined that the low combustion rate and high combustion duration in the square bowled chamber is a direct result of larger heat losses within the design, ultimately lowering combustion efficiency [7]. Figure 3 shows the turbulence, rate of heat release (ROHR) and mean velocity of the flat combustion chamber over a certain crank duration while Figure 4 illustrates the effect of the square combustion chamber on turbulence, ROHR and mean velocity over the same crank duration.

Figure 3: Mean Velocity, Turbulence and ROHR of Flat Combustion Chamber [7]

Figure 4: Mean Velocity, Turbulence and ROHR of Square Combustion Chamber [7]

2.1.5 FACE and the Effect of Fuel Characteristics on Advanced Combustion

The properties of a fuel can have a large impact on the characterization and operation of the advanced combustion regime, making the selection of a proper fuel crucial for advanced combustion methods. Two of the most important fuel properties when attempting to maintain advanced combustion are the cetane number and aromatic content. Lower cetane fuels typically require heating the intake air to achieve ignition due to combustibility characteristics. Higher cetane fuels generally exhibit a strong low temperature heat release rate as a result of early combustion phasing [8]. To develop a standard of fuel references for advanced combustion research, Fuels for Advanced Combustion Engines (FACE) were developed. The FACE goal is to provide a statistically engineered set of nine different fuels to enable researchers to quantify fuel effects on advanced combustion techniques. Fuel property ranges for the nine fuels were set according to the matrix seen in Figure 5.

Figure 5: FACE Design Matrix [9]

Studies performed at ORNL using FACE fuels with a “high efficiency clean combustion” (HECC) strategy revealed that the operating range of injection timings was far more advanced for low cetane fuels than high cetane fuels. Low cetane fuels also resulted in nearly zero soot PM emissions with comparable NO_x and BSFC of the high cetane fuels due to increased time for mixing from the longer ignition delay – favorable for advanced combustion methods. It was also revealed that fuels with lower aromatic content produce fewer NO_x , PM and HC emissions at the same injection timing [10].

2.1.6 Premixed Charge Compression Ignition and Low Temperature Combustion

Premixed charge compression ignition (PCCI) is collectively used as a term to describe a partial HCCI regime. Since obtaining and maintaining HCCI is so difficult, PCCI is simply a term used to describe an attempt toward reaching a homogenous mixture by pre-mixing the air/fuel prior to a combustion event. As seen in Figure 2, PCCI operates within a locally hotter region with a higher equivalence ratio compared to HCCI. Most studies typically investigate PCCI rather than HCCI due to its relative ease of achievement in CI engines, especially at part load operating conditions [11]. A study at ORNL revealed the traditional trade off of low NO_x with higher soot PM and HC emissions. Figure 6 shows the improvement of lower NO_x emissions with PCCI compared to conventional combustion. Low temperature combustion (LTC) is another advanced combustion regime that encompasses both HCCI and PCCI. From Figure 2,

LTC can be seen to operate at a much lower temperature than traditional HCCI and PCCI regimes at similar equivalence ratios.

Figure 6: Engine-Out Emissions of Conventional versus PCCI Combustion [12]

2.1.7 Summary of Advanced Combustion in Diesel Engines

Although the concept of HCCI has existed for decades, it has recently become a topic of interest as government regulations on NO_x and PM become increasingly stringent.

- HCCI can simultaneously reduce both NO_x and PM emissions.
- Engine operating parameters such as injection timing, rail pressure and EGR ultimately determine advanced combustion.
- Advanced combustion regimes must be operated with higher AF ratios than conventional CI combustion.
- Lower cetane fuels are more suitable for advanced combustion due to longer ignition delay periods.

2.2 Fuel Properties and Characteristics

2.2.1 Introduction

The new emissions standards imposed by the Code of Federal Regulations (CFR) are set to significantly limit NO_x and PM emissions from diesel engines. On-road heavy-duty diesel

emission standards have decreased NO_x from 6.0 g/bhp-hr in 1990 to 0.20 g/bhp-hr in 2010. PM emissions went from 0.60 g/bhp-hr in 1990 to 0.01 g/bhp-hr in 2007. In order to achieve such low emission requirements, various fuels and additives have been developed since the imposition, each one with various macroscopic properties and characteristics. This section describes some of the most important fuel properties that effect emissions such as cetane number, aromatic content, sulfur content, energy content, volatility and viscosity.

2.2.2 Cetane Number

The cetane number of a fuel is a measure of ignition quality or an ability to auto-ignite. There are various ways to determine the cetane number of a fuel such as the Cetane Index method (ASTM D4737) [13] or the ASTM D6890-04 Ignition Quality Tester [14], but the most reliable and current method to determine cetane number is the ASTM D613 method. The ASTM D613 method uses a VCR CFR certified engine and is based on the percentage of n-cetane and heptamethyl nonane in a proprietary blend that produces the same compression ratio to that of a test fuel [14]. An increase in cetane number typically reduces ignition delay and ultimately lowers cylinder temperatures and pressure rise rates. Due to the reduction in ignition delay, HC and PM emissions are higher but NO_x is reduced as a result of low cylinder temperatures [10].

2.2.3 Aromatic Content

The aromatic hydrocarbon content is related to the amount of heat needed to initiate combustion and is determined by ASTM test method D1319 [16]. Hydrocarbons consist of carbon molecular rings of singular or double bonds. A single ring is referred to as a monocyclic aromatic while a double bond is considered a polycyclic aromatic. The D1319 test method uses a fluorescent indicator absorption technique. In this method, a sample of fuel is placed in a glass column packed with silica and fluorescent gel. Alcohol is then introduced into the glass column to separate out the hydrocarbons. The hydrocarbons are then visible under ultra-violet light as a layer whose volume percentage corresponds to the aromatic content of the fuel [17]. Studies have shown that NO_x may increase as aromatic content rises due to higher cylinder

temperatures needed to break the polycyclic bonds [18, 19]. An increase in aromatic content is also known to lower the cetane number [19].

2.2.4 Sulfur Content

Since 2007, ultra-low sulfur fuel has been required for heavy-duty on-road diesel engines to initiate aftertreatment devices such as catalytic DPFs. Along with the regulated 0.01 g/bhp-hr PM in 2007, sulfur content was set to a maximum of 15ppm in fuel and consequently reduced in lube oil as well. The low sulfur fuel has proven to lower PM emissions by reducing sulfates formed during combustion [20]. A study by Navistar revealed that brake specific particulates increased by 0.025 g/bhp-hr for an increase of 0.1% sulfur content [21]. A catalytic process known as hydrodesulfurization is commonly used to separate the sulfur from heavy oils. In this process, a fraction of crude oil is mixed with hydrogen and passed over a catalyst bed at very high temperatures and pressures [22].

2.2.5 Energy Content

The energy content of diesel fuel is defined as the amount of energy per unit mass/volume released when combusted. Low-density fuel will have higher energy content per unit mass compared to a high-density fuel which will have higher energy content per unit volume. The specific energy content of a fuel can significantly affect engine power output unless a fuel quantity optimization injection strategy has been developed. Studies have shown an increase in fuel consumption and CO₂ for lower energy content fuel [23].

2.2.6 Volatility

The volatility of a fuel is often referred to as the distillation temperature and usually given as T50, T90 or T95. The distillation temperature is the point at which 50%, 90% or 95% of the fuel, by volume, has condensed into a separate sample. Low volatility is then quantified by elevated T95 temperatures [24]. Some studies have suggested that an increase in volatility will increase PM emissions while others contribute the PM increase to fuel density [25]. Since volatility is a function of other fuel properties such as viscosity, cetane number and density, it is often difficult to determine the effect that volatility has on emissions.

2.2.7 Viscosity

The viscosity of a liquid or gas is a property that refers to an ability to resist shear. Viscosity is mainly dependant on temperature while pressure has minimal effect. At high temperatures, viscosity for a liquid has a low resistance to shear and is therefore easier to flow. The opposite occurs for a gas, as temperature increases, viscosity also increases. The difference between the two lies within the molecular structure of the substance. Viscosity mainly affects fuel injection systems, especially the spray pattern [26], as low viscosity fuels can leak through injector seals.

2.2.8 Summary of Fuel Properties and Characteristics

Fuel properties can have a large effect on achieving advanced combustion and initiating aftertreatment devices, especially on modern on-road heavy-duty diesel engines. Depending on the fuel property, NO_x and PM emissions can greatly be affected. Table 1 summarizes an increase in some of the common fuel property quantities and their corresponding effect on engine out NO_x and PM emissions. Table 2 illustrates the NO_x and PM emissions as a result of decreasing fuel property quantities. Note Table 1 and Table 2 were developed from research studies utilizing typical diesel combustion without aftertreatment devices.

Table 1: General Effect of Increasing Fuel Property Quantities on NO_x and PM Emissions

	NO _x	PM
Increased Cetane Number	↓	↑
Increased Aromatic Content	↑	↑
Increased Sulfur Content	0	↑
Increased Volatility	↑	↑

Table 2: General Effect of Decreasing Fuel Property Quantities on NO_x and PM Emissions

	NO _x	PM
Decreased Cetane Number	↑	↓
Decreased Aromatic Content	↓	↓
Decreased Sulfur Content	0	↓
Decreased Volatility	↓	↓

2.3 In-cylinder Pollutant Formation and Analysis Techniques

2.3.1 Introduction

CI engines typically operate much leaner and use expensive aftertreatment devices to reduce NO_x and PM emissions while SI engines operate at stoichiometric conditions and use catalytic converters to reduce NO_x , HC and CO. Products of combustion such as CO, CO_2 and HC that form during the combustion process of CI engines are generally less concerning due to the much lower concentration compared to that of SI engines. Understanding how pollutant emissions form develops a basis for knowing how to control emission reducing devices or advanced combustion techniques. In-cylinder combustion analyses, such as the heat release profile, can often be used to quantify emissions and characterize fuel performance on advanced combustion regimes.

2.3.2 NO_x Formation

Due to the higher combustion temperatures of a CI engine, the formation and development of oxides of nitrogen (NO_x) are ultimately inevitable. NO_x is produced when normally stable N_2 molecules dissociate into monatomic nitrogen (N) and react with oxygen. The dissociation that occurs is highly temperature dependent with excess NO being produced at temperatures of 2500-3000K [27]. NO_x consists of both nitric oxide (NO) and nitrogen dioxide (NO_2) with NO comprising 70-90% of the total NO_x [28]. At temperatures less than 1200K, NO_2 constitutes the remaining species of NO_x emitted following the combustion process. Three formation mechanisms are generally responsible for the formation of NO_x : thermal, nitrous oxide (N_2O) and prompt. The first and most significant is the thermal mechanism. The thermal mechanism, also referred to as the extended Zeldovich mechanism, accounts for the dissociation of N_2 and O_2 into their monatomic states (Equation 2-1, Equation 2-2, Equation 2-3). The N_2O mechanism is less temperature dependant and occurs when the molecular N_2 and O_2 form N_2O , which then react with oxygen to form NO. Prompt NO_x , also less temperature dependent, is formed at the early stages of combustion with N_2 reacting with fuel radicals that are then oxidized to form NO_x [28]. Potential health problems associated with NO_x include a

number of respiratory diseases as well as toxic compounds being formed when NO_x reacts with organic compounds.

2-1

2-2

2-3

2.3.3 PM Formation

Diesel PM consists mainly of solid carbon soot particles as well as other harmful pollutants such as sulfates, ash and soluble organic fractions. PM is often seen as black smoke emitted from the exhaust under vehicle acceleration. The solid carbon particles are formed as a result of incomplete combustion in fuel-rich zones with maximum PM emissions observed when the engine is at maximum load [27]. Sulfate formation is generally both lube oil and fuel dependant, but due to modern government regulations, ultra-low sulfur fuel is now required for all on-road diesel engines. Fuel additives are often put into the ultra-low sulfur fuel due to lubricating problems causing abnormal cylinder surface wear and sticking of injectors [27]. A diesel particulate filter (DPF) is typically used to control or virtually eliminate traces of PM on modern production vehicles while retrofit DPFs are implemented on older vehicles.

2.3.4 CO Formation

Carbon monoxide (CO) is formed from fuel oxidation during combustion and is contributed to insufficient time and oxygen available for the complete oxidation of CO to CO_2 . At near stoichiometric conditions, the formation of CO is highly non-linear with respect to the equivalence ratio. CO can form at the flame-front as well as from fuel interactions with cylinder walls. Since CI engines typically operate much leaner than SI engines, CO emissions are much lower and far below government regulations.

2.3.5 CO_2 Formation

Carbon dioxide (CO_2), which is considered a greenhouse gas, is a product of combustion that involves a hydrocarbon fuel. Increases in CO_2 emissions usually indicate higher fuel

consumption and higher brake specific PM. The best way to reduce CO₂ emissions is to use higher thermal efficiency engines that ultimately consume less fuel [27].

2.3.6 HC Formation

Hydrocarbon emissions form due to an insufficient localized AF ratio (too high or too low) for auto-ignition of the fuel to occur. HC particles are condensed onto the surface of solid carbon soot due higher boiling and condensing temperatures of diesel fuel [27]. Two injection strategies lead to excess HC emissions in diesel engines; over-leaning and under-mixing. An over-leaning injection strategy involves a very lean AF ratio so that ignition delay is increased and complete combustion is limited [28]. Under-mixing involves fuel rich zones of combustion due to incomplete mixing with air. HC emissions can also form as a result of wall wetting or wall quenching due to fuel impingements on the cylinder wall and rapidly cooling prior to combustion. HC emissions have also been known to form as a result of trapped fuel in cylinder rings, excess fuel emitted after an injection event and from engine oil in the combustion chamber.

2.3.7 Heat Release Rate

The heat release rate (HRR) during combustion can reveal a number of characteristics when attempting to classify advanced combustion or predict NO_x emissions. The HRR can be derived using the first law of thermodynamics combined with a few engineering assumptions. By utilizing the first law and assuming uniform pressure, uniform temperature and an ideal gas, the gross HRR can be written as a combination of specific heat ratios, instantaneous pressure and cylinder volume as well as crank angle (Equation 2-4). The net HRR (Equation 2-5) accounts for heat transfer to the cylinder walls (—) using the well known “Woshni” method.

$$\frac{dQ_{gross}}{dt} = \frac{\gamma}{\gamma - 1} \frac{P}{R} \frac{dV}{dt} \quad (2-4)$$

$$\frac{dQ_{net}}{dt} = \frac{dQ_{gross}}{dt} - \frac{dQ_{loss}}{dt} \quad (2-5)$$

In conventional combustion the HRR can be idealized by two different regions on a plot of HRR versus crank angle (Figure 7). The first region, the premixed burn portion, is typically used to predict NO_x formation and can be denoted by the spike at the beginning of the combustion event. The second region is the diffusion portion, or rate controlled combustion, which occurs after the premix and comes to a gradual end ATDC. In an advanced combustion regime, a more rapid HRR can be observed with the premixed (first stage or low temperature reaction from Figure 8) combustion event occurring over a longer duration and an abrupt end to diffusion combustion event ATDC. Figure 8 depicts an advanced combustion HRR for both a single and split injection strategy.

Figure 7: Heat Release Rate Curve for Conventional Combustion [29]

Figure 8: Advanced Combustion Heat Release Rate Curve for Single and Split Injection Events [30]

As previously mentioned, the HRR curve can reveal a number of characteristics about NO_x formation. An increase in the heat release during premixed combustion alludes to a rapid increase in cylinder pressure and temperature, resulting in higher NO_x emissions [30]. Similarly, if there is a decrease in the heat release during premixed combustion, NO_x is elevated due to an increased amount of burning at the flame front [31, 32]. NO_x may also become increased as the HRR shifts more toward TDC as a result of higher pressure and temperature. Typical HCCI operation has been associated with a longer ignition delay thus allowing more efficient mixing and overall lean equivalence ratio [33].

2.3.8 Mass Fraction Burned

The mass fraction burned (MFB) is analogous to the combustion process and can be used to evaluate fuel consumption over the crank duration. The MFB reflects engine design and overall performance based on the percentage of fuel that is burned at a specific crank angle [34]. Common values of the MFB occur at 10%, 50% and 90% durations. Values of MFB from the start of combustion to 50% indicate the intensity of the premixed combustion while the remaining duration indicates the intensity of the diffusion burn. Equation 2-6 represents the MFB as derived from the first law of thermodynamics.

2.3.9 Brake Specific Fuel Consumption

The brake specific fuel consumption (BSFC) is a measure of engine fuel efficiency. BSFC is a function of speed and load and is therefore independent of engine size, making it a useful tool for the comparison of various internal combustion engines. Equation 2-7 represents the determination of BSFC, where $\dot{m}_f$ is the mass flow of fuel to the engine and P_b is the brake power calculated from brake torque and engine speed.

$$\text{BSFC} = \frac{\dot{m}_f}{P_b} \quad \text{2-7}$$

2.3.10 Brake Thermal Efficiency

An important parameter when evaluating the effectiveness of an engine is the brake thermal efficiency (BTE). BTE can be thought of as a ratio of work produced by the engine to the energy of the fuel needed to produce the work within a specific time. Ideally the BTE of an engine would be 100% but due to mechanical and heat losses within the engine; the typical BTE of a conventional diesel engine is only 25-40%. The BTE of an engine can be calculated as a function of BSFC and the lower heating value of the fuel, Equation 2-7.

$$\text{BTE} = \frac{P_b}{\dot{m}_f \cdot \text{LHV}} \quad \text{2-8}$$

2.3.11 In-cylinder Pressure

In-cylinder pressure measurement provides an approach to obtain and evaluate both the HRR and MFB to establish real-time combustion characteristics. In-cylinder pressure can also be used to evaluate the effects on structural integrity of an engine by examining maximum cylinder pressures and determining maximum pressure rise rates. The in-cylinder pressure is typically obtained by a pressure sensor mounted in the cylinder head. Other methods include measuring the cylinder pressure via the force exerted on cylinder head bolts during operation [35]. Due to the cost of the sensors and instrumentation involved, in-cylinder pressure measurement is generally restricted to research specific engines. In this study, in-cylinder pressure was obtained via piezoelectric pressure transducers housed within custom glow plug adapters.

2.3.12 Summary of In-cylinder Pollutant Formation and Analysis Techniques

The importance of understanding how pollutants are formed and ways to characterize emissions based on in-cylinder combustion measurements are crucial for the development and establishment of modern emissions reducing devices. Harmful emissions, especially NO_x , PM, and HC have become the focus of modern regulations involving both on-road and off-road diesel engines. These pollutants and in-cylinder techniques to identify these pollutants were researched with the following conclusions.

- NO_x is formed through three formation mechanisms: thermal, N_2O and prompt
- Thermal NO_x has the greatest effect on NO_x formation
- PM is a result of incomplete combustion
- HC emissions can result from unburned fuel in crevices or due to wall wetting
- An increase in heat release rate during premixed combustion results in higher NO_x emissions
- Decrease in heat release rate during premixed combustion results in higher NO_x emissions due to increased burning at the flame front
- HCCI is more likely to be achieved with longer ignition delay due to sufficient mixing
- Mass fraction burned gives the percentage of fuel burned over a particular crank angle

2.4 Exhaust Gas Recirculation in Diesel Engines

2.4.1 Introduction

Unlike spark ignited engines, compression ignition engines operate leaner with a much higher combustion temperature, as they rely on compressing a gas to achieve combustion. NO_x is generally formed during combustion by the nitrogen, from the air and fuel, reacting with oxygen, from the intake air, at very high temperatures in the combustion chamber, typically above 2000K. Since diesel engines operate with such high combustion temperatures, NO_x formation is inevitable. PM can vary with EGR rates but is generally controlled by increasing injection pressure and using a better quality fuel, while HC and CO are secondary to NO_x and PM formation. Over the years, stringent emissions regulations have forced manufacturers to

produce cleaner diesel engines. Figure 9 depicts the increasing emission regulations set forth by the governments of Europe, Japan and the United States upon heavy duty diesel engine manufacturers.

Figure 9: On Road Heavy Duty Diesel Engines Emission Regulations Requirements [36]

One of the leading ways to reduce NO_x in diesel engines is through the use of EGR. EGR allows a portion of the exhaust gas to be re-circulated into the intake manifold of the engine prior to entering the cylinder. By introducing a diluting gas into the intake stream, the intake charge of the engine is diluted and the adiabatic flame temperature is significantly reduced. A typical problem in utilizing EGR is that if the EGR rate is too high, combustion temperature decreases so much that combustion may become very erratic. Although EGR reduces NO_x formation, there is generally a trade off with the formation of PM. At higher EGR rates, PM becomes more apparent as combustion is becoming inherently less efficient. Many studies are being examined to optimize EGR rates to have less of an effect on PM formation [37].

With the introduction of EGR into the intake, three effects work in conjunction to modify combustion and ultimately lower NO_x production: dilute, thermal and chemical. The dilute effect is achieved by reducing the amount of oxygen available for combustion by replacing the incoming fresh air with exhaust gas. Studies have shown that an oxygen reduction for combustion reduces NO_x, increases PM, and produces a significantly longer ignition delay [38]. The thermal effect takes place due to the increasing heat capacity of the intake charge from the high concentrations of CO₂ and water vapor in the re-circulated exhaust gas. This increased heat capacity correspondingly causes a lower combustion (peak) cylinder

temperature that ultimately reduces NO_x production. The chemical effect is a direct result of dissociation of CO_2 and water vapor at very high temperatures. Dissociation absorbs thermal energy from the combustion process to lower peak cylinder temperatures.

By comparing each of the three effects, CO_2 has a higher dilute effect than the water vapor due to its high specific heat in the exhaust gas. The thermal effect is higher for water vapor than CO_2 since water vapor has a higher specific heat capacity. The chemical effect is higher for CO_2 than water vapor since water vapor requires higher combustion temperatures to dissociate [39].

2.4.2 Constraints Placed on NO_x , PM and Fuel Consumption

With the growing demand for cleaner, more efficient engines comes the declaration of government impositions for lower NO_x and PM emissions as well as increased fuel economy. All emissions regulations are imposed under the Code of Federal Regulations and are classified in Tiers for light duty engines. Tier 1 standards are imposed for all light duty vehicles from 1994-1997. Tier 2, which is a more stringent regulation, are set for all light duty vehicles produced from 2004-2009. In 2004 NO_x production for light duty vehicles was regulated at approximately 0.20 g/mile while PM was regulated at 0.01 g/mile. Growing concerns between the connection of global warming and vehicle emissions pushed the NO_x production limit to 0.07 g/mile under the new Tier 2 Emissions Standards and PM to 0.01 g/mile. Heavy duty diesel engines were regulated from 6.0 g/bhp-hr NO_x production in 1990 to just 0.20 g/bhp-hr in 2010. PM emissions went from 0.60 g/bhp-hr in 1990 to 0.01 g/bhp-hr in 2007. The fuel economy of vehicles is set by the Corporate Average Fuel Economy (CAFE). In 1990, light duty vehicles were regulated at a combined 20.5 mpg while in 2009 fuel economy is regulated at 23.1 mpg [40].

2.4.3 External EGR

EGR is a fairly simple concept that allows exhaust gases to re-enter the intake manifold to decrease NO_x production by lowering peak cylinder temperatures. As previously mentioned, EGR has had advancements throughout the years, but the most common EGR system consists of an EGR valve, flow tubes, heat exchanger and a pulse width modulation control unit. The external EGR process begins at the exhaust manifold. After in-cylinder combustion has

occurred, the exhaust valve is opened and gas is introduced into the exhaust manifold. Once in the manifold, the desired amount of exhaust gas is driven through a tubing network and into a tube and shell heat exchanger. Although an EGR system can be implemented without a heat exchanger (hot EGR), studies have shown that cooled EGR can be up to 30% more efficient in reducing NO_x emissions [41]. Post heat exchanger, an EGR valve is electronically controlled to regulate exhaust flow, while a variable diameter orifice plate regulates the flow rate of exhaust gas. After a proper dosing of EGR has been introduced into the intake, the dilution process occurs. The intake charge temperature depends on a number of factors. The most important factor affecting intake charge temperature is the gas temperature exiting the heat exchanger. To insure proper temperature and ultimately EGR rate, coolant can flow to and from the heat exchanger to either increase or decrease the intake charge temperature. EGR rate is determined, via the engine ECU, from a number of factors including engine temperature, engine speed and manifold air pressure, which are all monitored electronically. Studies have been examined to achieve optimum EGR rate by balancing the reduction in NO_x and the formation of PM.

2.4.4 Internal EGR

As opposed to external EGR, internal EGR traps a portion of exhaust gas within the combustion chamber after a complete cycle has taken place. The trapped exhaust gas is then mixed with incoming air to reduce cylinder temperatures. The advantage of an internal EGR system is that it requires no additional hardware to regulate EGR flow and is far less complex than the external EGR system but may require VVT. The main disadvantage of an internal EGR system is excessive valvetrain wear due to high soot concentrations in the oil as a result of high EGR rates.

2.4.5 High Pressure EGR

High pressure EGR systems route a portion of the exhaust gas taken before the turbocharger, relying on a differential pressure to supply EGR back into the intake. High pressure control is either monitored through the VGT or a back pressure valve located downstream of the turbocharger. A typical complication of the high pressure system is that

particulate matter or soot that remains in the exhaust gas upon entering the intake stream. By introducing “polluted” exhaust gas, engine wear rate may also be significantly increased [42]. High pressure EGR systems are typically implemented into many production vehicles compared to low pressure EGR systems.

2.4.6 Low Pressure EGR

As opposed to the high pressure EGR system, the low pressure EGR system is often used in retrofit applications where NO_x and PM emissions need to be significantly reduced on vehicles not originally equipped with an EGR system. The low pressure EGR system takes a portion of the exhaust gas after the turbocharger and any aftertreatment devices. By taking a sample of exhaust gas downstream of the turbocharger and aftertreatment devices, PM or soot re-entering the combustion chamber is virtually eliminated. Another advantage of a low pressure system is that it can be implemented on virtually any engine without engine modifications. Many retrofit, low pressure EGR systems offer NO_x reductions of 25-50% and a reduction in PM of over 90% along with proper aftertreatment systems [43].

2.4.7 Hot EGR System

Unlike a cooled EGR system, the hot EGR system does not require a sophisticated control scheme or the use of a heat exchanger. Instead of routing the exhaust gas through an air to water heat exchanger, a hot EGR system simply routes the hot exhaust gas back into the combustion chamber. Although peak combustion temperatures may not be as low compared to a cooled EGR system, the hot EGR system does significantly reduce NO_x and PM emissions however, especially in high pressure retrofit applications.

2.4.8 Cooled EGR System

Cooled EGR systems have been employed into production heavy duty diesel engines since late 2002. A cooled EGR system contains a heat exchanger as described in §2.4.3 to ultimately lower the temperature of the re-circulated exhaust gas. Depending on the application, most cooled EGR systems divert anywhere from 5-30% of the engines’ exhaust through an air to water heat exchanger [44]. Although a cooled EGR system requires a more

sophisticated control system, all of the sensors and electronics are monitored by the engines' ECU. Cooled EGR systems, implemented on most modern production diesel engines, are typically combined with the high pressure systems. Figure 10 shows the effect of hot and cooled EGR on NO_x emissions versus EGR rate. A study at Tsinghua University found that NO_x decreased 70-80% by using EGR within a certain operating range [45]. The effect of EGR in direct injection diesel engines has been extensively investigated and led research to conclude that the rate of EGR plays the biggest role in NO_x emissions and PM production. The study revealed that at lower rates of EGR, higher injection pressure and a reduction in EGR orifice diameter was needed to effectively lower NO_x and PM emissions. At higher EGR rates, lower NO_x was observed at the expense of high PM concentration. It was concluded that an EGR rate of 40% achieved lowest NO_x emissions without the tradeoff of high PM.

Figure 10: Effect of Hot and Cooled EGR on NO_x Emissions vs. EGR Rate [45]

As seen from the figure, NO_x emissions drastically decrease with an increase in EGR rate. Note cases A, B and C are tests performed at different engine loads. Since hot EGR brings heat into the cylinder, an increase in in-cylinder temperatures and ultimately the thermal charge of the engine occurs. For hot EGR systems, low NO_x emissions are typically counteracted by EGR. From Figure 10 it can be determined that cooled EGR results in a much greater NO_x reduction in a common rail injection system compared to a hot EGR system. Following a similar trend, Figure 11 illustrates the PM emissions for a cooled EGR system compared to a hot EGR system. From Figure 11 it is clear that cooled EGR produces a significantly less amount of PM compared to the hot EGR system, especially at higher EGR rates.

Figure 11: Effect of Hot and Cooled EGR on PM Emissions [45]

2.4.10 Effect of EGR on Fuel Consumption

Fuel consumption is heavily influenced by the amount of EGR entering each cylinder. If EGR rates are minimal or too great, fuel consumption may increase due to incomplete combustion. To reduce the chance of incomplete combustion, optimization of engine operating parameters is often maximized to provide desired fuel consumption. A study at Central Engineering Laboratories discovered that fuel consumption steadily increased at EGR rates of 10-20% under light loads [46]. A reduction in pumping losses and cooling losses (due to decreased combustion temperatures) were found to be the main contributors of the increase in fuel economy. With a demand from the EPA to produce near zero emissions and meet fuel economy criteria, many manufacturers are moving to a modern SCR-EGR system.

2.4.11 Effect of EGR on Engine Wear

A common conception is that an EGR system will decrease an engine's life and lead to excessive engine wear. It is thought that the soot generated from EGR can absorb anti-wear additives in oil. A study at Loughborough University examined the effect of high pressure EGR on piston rings, cylinder liner and engine oil. The study showed that the wear on piston rings and cylinder liner increased as the level of EGR increased. "At full load, EGR rates above 10% increased engine wear. Engine wear also increased with oil age" [47]. The rate of oil aging also increased as the EGR level increased. The study also concluded that high pressure EGR increased component wear at high loads but at low loads, component wear was very low. The

wear increase from EGR can be contributed to the increase in carbon content as seen in Figure 12.

Figure 12: Component Wear and Carbon Content versus EGR Rate [47]

2.4.12 Summary of EGR in Diesel Engines

EGR has been used in diesel engines since the early 1970's as a method to control harmful emissions such as NO_x and PM. Since the introduction of EGR in consumer available vehicles, EGR technology has become increasingly advanced.

- Low Pressure EGR systems can reduce NO_x and PM emissions by, 25-50% and 90%, respectively, in retrofit applications.
- Cooled EGR systems can be up to 30% more efficient than hot EGR systems.
- NO_x and PM can fluctuate with EGR rate in DI engines.
- Component wear of an EGR equipped vehicle increases with EGR rate.
- Engine wear becomes significant at EGR rates above 10%.
- The rate of engine oil aging increases as EGR rate is increased.

3 Experimental Setup and Test Equipment

3.1 Introduction

All testing discussed in this document was conducted at West Virginia University's Center for Alternative Fuels, Engines and Emissions (CAFE) and performed in the Engines and Emissions Research Laboratory (EERL). The EERL implements a full-scale dilution tunnel thought to accurately simulate the mixing of exhaust gas with ambient air conditions to replicate a "real-world" environment. The test engine was purchased for testing incorporated by the project sponsor and was intended to replicate the engines used by ORNL and Argonne National Laboratory in similar testing platforms. No internal parts of the engine were altered to enhance or initiate an advanced combustion strategy throughout the test period.

3.2 Test Engine

Experiments were conducted on the engine shown in Figure 13. Throughout the testing performed at WVU, a much larger EGR cooler from a heavy duty diesel engine (HDDE) was used to replace the OEM EGR cooler which was unable to provide sufficient cooling of the exhaust gas under high EGR percentages (see Appendix). A heated intake element was also added to the test engine to assist the combustion process during cold starts with low cetane fuels by elevating intake temperatures and enhancing any advanced combustion regimes through timing advancement and EGR rates. Figure 79 shows the test engine with the oversized EGR cooler and heated intake element.

Figure 13: WVU Test Engine

The engine was attached at the driveshaft, to the dynamometer, through a custom flywheel/clutch assembly. A custom flywheel was designed and implemented onto the test engine due to an insufficient dual-mass flywheel setup on the stock configuration. Thermocouples were added at each intake and exhaust port to record temperatures at these locations. Couplings were fabricated into the intake and exhaust manifold to measure oxygen concentration. Existing laboratory thermocouples were used at other locations of significance such as intake air, exhaust, fuel, coolant and oil to measure temperature. Table 3 shows specifications of the test engine used by WVU.

Table 3: Test Engine Specifications

Type	CDTi Diesel Engine
Manufacturer	General Motors
Model	Z19DT
Intake System	VGT
Valve Configuration	2 valves per cylinder
Year	2005
Configuration	In-line 4 Cylinder
Displacement	1.9 L
Bore	82 mm
Stroke	90.4 mm
Compression Ratio	18:1
Injection System	Common Rail
EGR	Cooled, External
Rated Power	88 kW @ 4000 RPM

3.3 In-cylinder Pressure Measurement

In order to determine the in-cylinder pressure during engine operation, Kistler 6056A41 piezoelectric crystal pressure sensors (Figure 14) were implemented into each of the four cylinders of the engine. Each sensor was adapted into the engine cylinder via custom glow plug adapters that replaced the stock glow plug, as seen in Figure 15. The manufacturer specifications for the pressure sensors are provided Table 40. Due to the accumulation of carbon on the pressure sensors, cleaning was crucial in order to obtain accurate combustion data. Cleaning was periodically performed on each sensor with a special solvent, while the adapters were cleaned with traditional oven cleaner. Each end of the sensor wire, which connects to the charge amplifier, was also cleaned with special solvent to remove any accumulation of oil and surface debris. Figure 16 shows the Kistler 5010B1 charge amplifiers which were used to convert the dynamic charge from the pressure sensor, to an amplified voltage that was easily read by the DAQ system.

Figure 14: Kistler 6056A41 Pressure Sensor

Figure 15: Pressure Sensor Glow Plug Adapter

Figure 16: Kistler 5010B1 Charge Amplifiers

A custom analysis software and data acquisition system, developed by Dr. John Nuszowski of West Virginia University, was used for the combustion analysis. The custom software allowed for real-time pressure, temperature, HRR and MFB plots to be accessed during testing to verify engine operation and/or misfiring events. Numerous derived and calculated combustion characteristics such as the maximum pressure, pressure rise rates, start of combustion, end of combustion, maximum temperature, etc. were also available in real-time during testing. Figure 17 depicts a real-time screenshot captured during a testing event. The combustion acquisition system used a stand-alone computer located in the test cell for ease of access. Due to computational limitations, the combustion acquisition system was only associated with the laboratory DAQ system through PM triggering i.e. to start and stop data collection.

Figure 17: In-cylinder Combustion Software

3.4 Measured and Calculated Combustion Parameters

Although most of the combustion parameters were calculated using in-cylinder pressure measurements from pressure sensors, exhaust temperature was recorded as separate parameter. Exhaust temperature was recorded from a K-type thermocouple, installed in the exhaust manifold of the engine, while engine speed was measured by a BEI shaft encoder connected to the engine crank via a helical coupling.

3.4.1 In-cylinder Pressure

As mentioned in §3.3, the in-cylinder pressure was obtained from a piezoelectric crystal pressure sensor signal which was then converted to a voltage by a charge amplifier. By measuring the in-cylinder pressure directly, the maximum in-cylinder pressure and location of maximum in-cylinder pressure can be calculated. Determining these parameters is useful for identifying potential engine integrity issues associated with high pressures and can also be used

to determine in-cylinder temperatures. The dynamic pressure signal from each sensor was referenced by assuming a constant polytropic constant to give an absolute pressure measurement [29].

3.4.2 Heat Release Rate

Determining the heat release rate can offer predictions of NO_x emissions, as higher HRR is generally associated with higher NO_x [48]. From the heat release data obtained by Equation 2-5, parameters such as maximum HRR, location of maximum HRR, net heat released during combustion, gross heat released during combustion and fuel flow can be determined. Equations 3-3 and 3-4 show the calculation for determining net and gross heat released during combustion from start of combustion to end of combustion (SOC-EOC).

3-3

3-4

Once the gross heat released during combustion is determined, total fuel flow can then be calculated using the LHV of the fuel then multiplied by the number of cylinders, see Equation 3-5. Note this equation is valid for 4-stroke engines only with 100% combustion efficiency.

3-5

3.4.3 In-cylinder Gas Temperature

Since NO_x and other pollutants are a function of temperature, the in-cylinder gas temperature is an important parameter to evaluate during combustion. For the work presented in this document, in-cylinder gas temperature is presented as the mean of the gas temperature; however local temperatures may be higher or lower. The in-cylinder temperature can be determined by assuming an ideal gas at intake valve closure with constant molecular weight. Consequently location of the maximum temperature can then be determined. Location of the maximum temperature is thought to influence NO_x formation regardless of the diesel fuel type [48]. Equation 3-6 depicts the rearranged ideal gas equation with temperature at any crank angle on the left, while Equation 3-7 shows the pressure at intake valve closure

and temperature at intake valve closure in terms of Equation 3-6. Combining Equations 3-6 and 3-7, the temperature at any crank angle can be determined.

$$\text{---} \tag{3-6}$$

$$\text{---} \tag{3-7}$$

$$\text{---} \tag{3-8}$$

3.5 Control of Engine Operating Parameters

As discussed in §2.1.3, the ability to control engine operating parameters is crucial for obtaining advanced combustion. Accessing the engine’s ECU allows for full control of operating parameters such as SOI, EGR rate, fuel injection quantity, fuel duration, rail pressure, etc. To access such operating parameters, WVU purchased an open engine controller based on NI hardware developed by Drivven, Inc (Figure 18).

Figure 18: Open Engine Controller Purchased from Drivven, Inc

A custom wiring harness, constructed by Ross Ryskamp of West Virginia University, allowed the Drivven controller to connect directly to the OEM wiring harness as seen in Figure 19. The custom wiring harness also allowed for the OEM ECU to be connected directly to the

OEM wiring harness for test evaluation and comparison to the Drivven controller. Additionally, the Drivven controller was pre-programmed with “baseline” operating parameters relative to the test engine as purchased by WVU.

Figure 19: Custom WVU Wiring Harness

The Drivven controller used LabVIEW software for the user interface which was initiated on a stand-alone computer outside of the test cell. The software allowed for real-time observation and full control of quantities based on voltage signals received by the controller during engine operation. Operating parameters of concern such as the SOI, EGR valve position, rail pressure, fuel quantity, fuel duration and pre-injection quantities were easily controlled by the user-friendly interface seen in the Appendix.

3.6 Engine Laboratory

Testing of the engine was performed at the EERL on the WVU campus. The EERL is a unique test laboratory with a full-scale dilution tunnel designed to the guidelines outlined in Title 40 CFR Part 1065. The laboratory includes a three dynamometer test cell, gaseous emission measurement devices, particulate matter sampling systems, Class 1000 clean room, DAQ systems and intake air conditioning devices.

3.6.1 Dynamometer

The test engine was run by an Eaton alternating current dynamometer capable of delivering up to 400hp. The engine was connected to the dynamometer through a clutch system that was attached to a custom driveshaft. Throttle is generally controlled using a proportional integral derivative technique to allow the dynamometer to measure, and ultimately match, the desired engine torque. However, throttle was controlled using the Drivven controller for the testing presented in this report. Torque was measured using a strain gauge on a lever arm located on the side of the dynamometer. To maintain quality assurance, the dynamometer was calibrated using specific weights that hung from a basket on the end of the lever arm. No less than 10 points were taken to generate a calibration curve in accordance with 40 CFR §1065.310 (c) [49]. Once the calibration curve was established, coefficients were entered into the DAQ system to correspond to the measured torque of the dynamometer.

3.6.2 Full-Scale Dilution Tunnel and Sampling System

Emissions testing for the work presented in this report were conducted using the full-scale dilution tunnel seen in Figure 20. The dilution tunnel allowed the exhaust gas to mix and react with ambient air. By mixing with ambient air, the exhaust gas is perceived to replicate the reaction with the atmosphere in real-world conditions, outside of the test cell, and to allow for proper particulate formation. Dilution air was obtained from ambient air and filtered using an approximately 99.99% efficient HEPA filter to remove any particulate matter. The air was maintained at an approximate dew point of 14.5°C and conditioned/humidified to approximately 25°C ± 5°C. To account for CO, CO₂, NO_x and THC in the dilution air, a sample of the air was collected in a Tedlar bag during each test, known as the background bag. At the end of each test, the bag sample was sent to analyzers for determination of the quantity of each pollutant in the dilution air.

Figure 20: EERL Full-Scale Dilution Tunnel

In accordance to Title 40 CFR Part 1065, a constant volume sampling system (CVS) was used for measuring engine out emissions. A blower was located downstream of the sampling system to pull the diluted exhaust through the dilution tunnel. A subsonic venturi (SSV), located downstream of the sampling plane, was used to measure and sustain the flow rate of the diluted exhaust. For the testing performed in this study, the tunnel flow rate was set to a constant 400scfm and calculated using Equation 7-1 in the Appendix. The sampling plane, located upstream of the SSV, consisted of probes and heated transfer lines that connected to a Horiba exhaust gas analyzer (MEXA) to begin the gas measurement process. Samples of each gas (CO, CO₂, NO_x and THC) were maintained at a specified temperature compliant to 40 CFR §1065.145 [49]. To maintain quality assurance of the volumetric flow rate, propane injections were periodically performed on the dilution tunnel. During a propane injection, propane is injected into the tunnel where the engine exhaust mixes with the dilution air. After the injections occur over a period of 300 seconds, the recovered quantity of propane is then compared to a known injection quantity within a specified tolerance. For consistency, three consecutive tests were run with no more than 2% deviation.

3.6.3 Exhaust Gas Analyzer

Gaseous emissions were analyzed using the Horiba MEXA 7200D seen in Figure 21. The MEXA features an automated calibration system with the capability of simultaneously measuring and recording CO₂, CO, NO_x, and THC species. Real-time emission concentrations, calibrated using gases of known concentrations, were available on a user-friendly interface.

Prior to each test, the MEXA was automatically zeroed and spanned to account for analyzer drift. The methods of measuring gaseous emissions via the MEXA 7200D are further discussed in this section.

Figure 21: MEXA 7200D

3.6.4 Carbon Monoxide and Carbon Dioxide Analyzer

CO and CO₂ emissions concentrations were measured using a non-dispersive infrared (NDIR) method. The NDIR method utilizes the conversion of light energy to vibrational energy to detect and measure CO and CO₂ species in the form of infrared radiation absorption. Concentrations of CO and CO₂ are then proportionally related to the degree of absorption [50]. Prior to entering the NDIR, the exhaust gas was sent through a chiller to minimize the interference of existing water vapor. Additionally, intake CO₂ was measured separately using a Horiba AIA-220 gas analyzer to determine EGR rate. Neglecting background CO₂ concentrations, EGR rate was then calculated using Equation 3-9, where “DR” is the dilution ratio obtained by dividing the tunnel flowrate by intake air flow. To account for analyzer drift, both the MEXA and intake CO₂ analyzers were zeroed and spanned between tests. Analyzer response was obtained by generating a calibration curve of no less than 10 points prior to testing.

3.6.5 Oxides of Nitrogen Analyzer

NO_x was measured using a chemiluminescence technique. In chemiluminescence, light is emitted when NO reacts with ozone (O_3). Since NO_x consists primarily of NO_2 and NO in a diesel engine, the gas is passed through a NO_2 to NO converter, with a known efficiency, prior to the reaction with ozone. During the reaction, photons emit light that is proportional to the concentration of NO. Similar to the NDIR analyzer, the chemiluminescence device was also zeroed and spanned prior to each test. In accordance to 40 CFR §1065.370 [49], a calibration curve was generated prior to testing to ensure the analyzer response was linear.

3.6.6 Hydrocarbon Analyzer

Gaseous hydrocarbon concentrations were measured by an ionization process using a heated flame ionization detector (HFID). During this process, a sample of the exhaust gas is passed through a hydrogen flame causing the gas to ionize and produce heat energy. Concentrations are then determined by relating a current that is proportional to the number of carbon atoms during the ionization process, to a corresponding voltage output. To ensure minimal drift, the HFID was zeroed and spanned prior to each test. In accordance to 40 CFR §1065.360 [49], a calibration curve was generated to ensure linearity.

3.6.7 Oxygen Concentration

Both intake and exhaust oxygen concentrations were measured by a Horiba MEXA 720 analyzer located in the intake manifold and exhaust pipe, post VGT, respectively. For a consistency check, a Rosemont 755R Paramagnetic oxygen analyzer was used to verify the intake oxygen content. The paramagnetic analyzer works by drawing a sample of gas over a magnetodynamic sensor. The sensor consists of a “dumbbell” shaped glass that is filled with an inert gas such as nitrogen. The dumbbell is suspended from a platinum wire within a non-uniform magnetic field and allowed to rotate when a sample gas is introduced. The corresponding rotation is caused by the attraction from the oxygen molecules to the stronger magnetic field. Optical systems are used to measure the dumbbell rotation and oxygen

percentage is then correlated to the amount of current it takes to stabilize the rotating dumbbell [51].

3.6.8 Particulate Matter Sampling System

Once the exhaust gas was diluted in the dilution tunnel, particulate matter was then collected on a filter using a gravimetric measurement method. Located near the sampling plane, a stainless steel tube diverted the exhaust gas into a custom WVU PM box. The PM box, pictured in Figure 22, was maintained at 115°F (46°C) with a filter face temperature of $47 \pm 5^\circ\text{C}$ to eliminate the possibility of condensation on the filter face. Prior to entering the filter, the gas was passed through a cyclone with a specific cut size at variable flowrates. The PM was then collected on a TX-40 47mm filter, housed within the filter holder. Prior to each test, a filter was pre-conditioned and pre-weighed in the EERL clean room. After a test, the filter was then post-conditioned and post-weighed to determine the amount of PM that was produced during the test. A digital mass flow controller was used to control the flowrate of exhaust gas that entered the filter face. Note this system accounts for PM deposited on the filter which may also include soluble organic fraction (SOF). In some cases raw fuel (SOF) may be deposited on the filter which cannot be classified as “soot based” PM. In order to directly measure soot based PM, an AVL Micro Soot Sensor was installed in the exhaust pipe.

Figure 22: Custom WVU PM Collection Box

3.6.9 AVL Micro Soot Sensor

In order to determine the soot based PM and continuously measure the PM concentration in real-time, an AVL micro soot sensor, seen in Figure 23, was placed in the exhaust prior to the dilution tunnel.

The micro soot sensor measures the diluted elemental carbon concentration in the exhaust gas from a photoacoustic technique. With this measurement technique, a sample of the “black” exhaust gas is exposed to modulated light. A corresponding sound wave is then recorded due to the expanding and contracting from the periodic cooling and heating of soot particles. Since clean air does not produce a signal, the sound wave signal increases proportionally to the concentration of soot within the measured volume [52]. The signal is then translated into a real-time carbon concentration.

Figure 23: AVL Micro Soot Sensor

Table 4: AVL Micro Soot Sensor Specifications [52]

Measured Value	Concentration of Soot (mg/m ³ , µg/m ³)
Measuring Range	0-50 mg/m ³
Display Resolution	0.001 mg/m ³
Detection Limit	~ 5 µg/m ³
Turndown Ratio	1 : 5.000
Data Rate	Digital = 10 Hz, Analog = 100 Hz
Rise Time	≤ 1 sec
Operation Temperature	5°C to 43°C
Probe/Bypass Flow	~ 2 ± 2 L/min
Interfaces	RS232, Digital I/O, Analog I/O, Ethernet
Laser Class	Class 1 laser product

3.6.10 EERL Clean Room

As discussed in §2.6.4, each filter was pre-conditioned and pre-weighed prior to testing, then post-conditioned and post-weighed after each test to determine the amount of PM that was produced during the specific test. The WVU Class 1000 EERL clean room utilizes a Sartorius SE2-F micro balance with a repeatability of 0.25µg and a readability of 0.1µg to weigh PM filters. To ensure proper filter weights by accounting for buoyancy corrections, the clean room temperature was maintained at 22°C ± 1°C with a dew point of 9.5°C ± 1°C, compliant to 40 CFR §1065.190 [49]. All personnel that entered the clean room wore booties to reduce the chance of tracking in contaminants.

3.6.11 Intake Air

The test engine was provided with conditioned intake air received from the ambient atmosphere. Prior to entering the engine, the ambient air was passed through a HEPA filter to remove any particulates. The air was then sent to handling units where moisture was removed. To ensure repeatability, the air was then re-humidified and maintained at 25°C ± 5°C with a relative humidity of approximately 50% and a NO_x correction factor of 1.00 ± 1%. The volumetric airflow rate entering the engine was determined from a laminar flow element (LFE) device. The LFE used for the test engine was a 4 inch Meriam Instruments 50MC2-1 capable of

386.13 scfm at a differential pressure of 9in. H₂O. A differential pressure sensor by Omega and a resistance temperature device (RTD) was used to determine the differential pressure and inlet air temperature of the LFE, respectively. By knowing the differential pressure over the LFE and air temperature entering the LFE, a volumetric airflow rate was then calculated.

3.6.12 EERL DAQ System

The National Instruments SCXI 1001 data acquisition system (DAQ), pictured in Figure 24, was used to measure various laboratory and engine parameters such as temperatures or emission concentrations. The DAQ system works by receiving current or voltage signals and converting it to engineering units that are proportional to the input signal. Such received current or voltage signals may come from the MEXA, thermocouples or pressure transducers.

Figure 24: EERL NI DAQ System

4 Discussion of Results

4.1 Introduction

The following sections will provide test data from each fuel and corresponding in-cylinder combustion parameters. All testing was performed at a steady state 2100 RPM and a bmep of 3.5 bar (53 N-m). The steady state tests were created with 240 seconds of stabilization time and 180 seconds of data acquisition. Stabilization time was created to allow for stable engine temperatures and to provide sufficient time for the EGR fraction to sustain a desired percentage. Since the percentage of EGR is calculated as a function of the dilution ratio, intake CO_2 and diluted exhaust CO_2 , a lag time is associated with the actual amount of EGR entering the intake of the engine. For ease of demonstration, EGR fractions have been rounded to the nearest whole number i.e. 50.8% becomes 51%. Also note that only Cylinder 1 data are presented in this document. For pilot sweeps, a 20% fuel split indicates 20% of the fuel is delivered on the pilot injection while the remaining 80% is supplied during the main injection. Combustion data are presented in this section while the corresponding emissions data are presented in the Appendix. Throughout the series of testing, misfire, especially in cylinder 2 was observed at specific operating conditions. Misfire was then classified as a visual, erratic change in the real-time pressure versus crank angle data, resembling a motoring pressure curve, in at least one of the cylinders and present in one more cycles as verified by high HC emissions. Also note that the presented pressure curve data is averaged for 200 cycles.

4.2 Fuel A

Fuel A was designed with a very high cetane number (CN), high aromatics and high volatility. Specific properties for Fuel A are listed in Table 39 from the Appendix. From Table 1, the high aromatic and high volatility may increase both NO_x and PM but the extremely high CN may promote a reduction of NO_x . Due to the high CN, intake heating was not implemented for each series of testing. The first series of testing consisted of performing EGR sweeps to identify an optimum range of EGR fractions that resulted in the lowest NO_x and PM emissions. Table 6 shows the emission concentrations of the EGR sweeps. Single injection sweeps were also performed but resulted in extremely high HC and PM emissions, see Table 41. As seen in the

table, BSFC decreases 27% from 50% EGR to 60% EGR. This decrease may be contributed to the instability in load as seen in the variability of BMEP. For the following EGR sweeps, main SOI and pilot SOI were fixed at 0° and 40° BTDC respectively with a 50% injection split at 800 bar rail pressure. Figure 25 illustrates the in-cylinder pressure as a function of crank angle for the EGR fraction sweeps.

Figure 25: In-cylinder Pressure versus Crank Angle of EGR Sweeps for Fuel A

As seen in Figure 25, increasing EGR reduced the peak cylinder pressure and retarded the phasing when peak cylinder pressure was observed. When operated with 50% EGR, the peak pressure of 6284 kPa was observed at 1.75 °ATDC while at 65% EGR a peak pressure of 5424 kPa was observed at 3.25 °ATDC. The reduction in pressure can be correlated to the increasing heat capacity of the intake charge as EGR fraction is increases, resulting in lower cylinder pressure and ultimately lower cylinder temperatures. At a 65% EGR fraction, raw fuel was deposited on the PM filter along with elevated HC and NO_x emissions, as seen in Table 6, possibly due to incomplete combustion. Figure 26 shows the HRR as a function of crank angle for the EGR fraction sweeps.

Figure 26: Gross HRR versus Crank Angle of EGR Sweeps for Fuel A

From Figure 26, an increasing EGR fraction reduces the peak of first stage heat release and increases ignition delay, indicating the importance of CN. This trend was also observed in a study at Lund Institute of Technology and is thought to be due to an increase in heat lost to the cylinder wall as EGR increases [53]. The conditions during the first stage or low temperature heat release event appear to have a strong impact on NO_x formation with increasing EGR as seen in Table 6. At 50% EGR, the first stage heat release peak is approximately 0.012 kJ/deg, while at 65% EGR is 0.01 kJ/deg. An EGR fraction of 55% was thought to be the optimum operating point due to lower NO_x and PM compared to 50% and 65% EGR. Figure 27 shows the mass fraction burned as a function of crank angle for the EGR sweeps.

Figure 27: Mass Fraction Burned versus Crank Angle of EGR Sweeps for Fuel A

As illustrated in Figure 27, an increasing EGR fraction retards combustion. This trend is due to lower cylinder temperatures as EGR increases and reduced oxidation rate due to the increased CO_2 and reduced O_2 . The effect of the pilot injection can also be identified from the MFB curve as the initial spike around TDC. Additionally, a crank angle (CA) 10-90% was calculated to determine the approximate duration of combustion, see Figure 28. The CA 10-90% burn duration was calculated by subtracting the location of mass fraction burned at 10% from the location when 90% of the mass fraction is burned. Although the CA 10-90% burn duration is a good indicator of how fast a combustion event occurs, for split injection events it may not be as accurate due to the extended first stage combustion duration. From Figure 28, an EGR fraction of 55% results in the longest burn duration, 40.25° , while 50% EGR results in the shortest burn duration, 31.75° . The decreased burn duration at 50% and 55% EGR can be related to the increased slope during the first stage heat release as denoted from Figure 26. At the 55% EGR fraction the 10-90% burn duration is longest due to the increased length of first stage heat release as seen in Figure 26. Figure 28 also shows the maximum pressure rise rates for the EGR sweeps.

Figure 28: CA 10-90% Burn Duration and Max PRR of EGR Sweeps for Fuel A

As shown in Figure 28, the maximum pressure rise rate (PRR) decreases linearly with increasing EGR fraction. The decrease in PRR may be due to the lower oxygen content available for combustion as EGR and ultimately ignition delay is increased. The reduction in PRR can also be correlated to the much longer 10-90% burn duration at the highest EGR fraction (refer to Figure 28). Ideally a maximum pressure rise rate of less than 10 bar/deg is sought during advanced combustion events based on research performed at ORNL [10], any higher and the engine's integrity may be compromised. An EGR fraction of 50% results in a maximum PRR of approximately 2.5 bar/deg, well within the limits of typical advanced combustion regimes. Figure 29 shows the corresponding brake thermal efficiency (BTE) and brake specific fuel consumption (BSFC). The decrease in BTE at the high EGR fraction is a result of a larger quantity of fuel i.e. increased BSFC, being needed to sustain the operating point. BSFC at 50% and 55% EGR is significantly lower than the 65% EGR fraction but PM emissions were elevated at 50% EGR as seen in Table 6.

Figure 29: BTE and BSFC of EGR Sweeps for Fuel A

Table 5 shows the calculated in-cylinder pressure parameters for the EGR sweeps. As depicted by the pressure traces in Figure 25, the location of maximum pressure and temperature retards as EGR fraction increases, namely due to the increased ignition delay. Also depicted in the table, and as shown in Figure 27, the start of combustion (approximated as 5% mass fraction burned) retards as EGR fraction is increased, at 55% EGR the start of combustion is 7.75°BTDC compared to 11.5°BTDC at 50% EGR. At 65% EGR, the trend appears to dissociate as the start of combustion is 10.5°BTDC, which may be due to a lower BMEP for this specific test (3.0 bmep).

Table 5: Calculated In-cylinder Combustion Parameters of EGR Sweeps for Fuel A

EGR Fraction	-	50%	55%	65%
Maximum Pressure	kPa	6284	5965	5424
Location of Maximum Pressure	deg ATDC	1.75	2.75	3.25
Maximum Pressure	kPa	6284	5965	5424
Location of Maximum Pressure	deg ATDC	1.75	2.75	3.25
Maximum Pressure Rise	bar/deg	2.60	2.06	1.67
Location of Maximum Pressure Rise	deg ATDC	-6.0	-4.25	-14.25
Maximum Average Bulk Mixture Temperature	K	1255	1254	1219
Location of Maximum Temperature	deg ATDC	20.25	21.5	24.25
Location of 10% Mass Fraction Burned	deg ATDC	-4.5	-6.5	-5.0
Location of 50% Mass Fraction Burned	deg ATDC	12.25	14.0	16.75
Location of 90% Mass Fraction Burned	deg ATDC	27.25	33.75	34.0

Table 6: EGR Sweep Emissions Data for Fuel A

EGR	HC (ppm)	NO _x (ppm)	HC (g/kW-hr)	NO _x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
50%	1484	21.14	4.566	0.211	262.9	368.6
55%	1911	13.66	5.216	0.118	261.4	319.2
65%	4076	8.210	11.44	0.064	300.4	332.5

Pilot injection sweeps were performed from 20-60°BTDC at 1000 bar rail pressure, 50% EGR and 20% fuel split. As seen in Figure 30, retarding the timing of the pilot injection was found to advance combustion phasing due to the initiation of a later combustion event. An increase in NO_x was observed as the pilot injection occurred closer to TDC, as a result of a less homogenous combustion event, indicating an increase in cylinder temperature as the pilot injection is retarded, see Figure 31. The effect of advancing the pilot on the HRR can be seen in Figure 31.

Figure 30: In-cylinder Pressure versus Crank Angle of Pilot Injection Sweeps for Fuel A

Figure 31: Gross HRR versus Crank Angle of Pilot Injection Sweeps for Fuel A

As seen in Figure 31, a more advanced pilot injection results in a lower peak HRR and a longer ignition delay, ultimately retarding the timing of the combustion event. Although the

20° pilot injection produces a more rapid HRR curve, misfiring established from extremely high HC (~2000 ppm) emissions limit it from an optimization strategy. Similarly, at 60°BTDC HC and gravimetric PM emissions were elevated, thought to be due to wall wetting. Figure 32 shows the BTE and BSFC for the respective pilot injection sweeps.

Figure 32: BTE and BSFC of Pilot Injection Sweeps for Fuel A

From Figure 32, an extremely low BTE is observed for the pilot injection sweeps. At 20°BTDC, BTE is just 22.5% while the highest BTE occurs at 40°BTDC (~27.9%). Since the LHV of the fuel remains constant, the low BTE is a direct result of incomplete combustion due to wall wetting and increased BSFC. Rail pressure sweeps were also performed from 800 to 1800 bar, with a main SOI at TDC, a 40°BTDC pilot, 50% fuel split and 55% EGR with 800 bar producing the lowest HC and PM emissions, see Table 7. Upon completion of the discussed engine sweeps, the optimum strategy of the data set for Fuel A is summarized in Table 8 while Table 9 shows the emission concentrations at the respective optimized strategy. Note soot based PM is unavailable for Fuel A, as the AVL soot sensor was not installed at this time. Table 10 shows the calculated in-cylinder combustion parameters at the optimized strategy.

Table 7: Rail Pressure Sweep Emissions Data for Fuel A

Rail Pressure (bar)	HC (ppm)	NO _x (ppm)	HC (g/kW-hr)	NO _x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
800	2176	11.58	5.981	0.101	268.5	319.3
1200	3977	8.61	10.218	0.065	276.8	444.6
1600	3931	7.984	9.67	0.059	350.0	358.8
1800	3135	9.945	8.332	0.0787	275.9	355.3

Table 8: Optimized Strategy for Fuel A

EGR	55%
Rail Pressure	800 bar
Fuel Split	50%
Pilot Injection	40 °BTDC

Table 9: Optimum Operating Strategy Emissions Results for Fuel A

NO _x (ppm)	13.6
HC (ppm)	1911
Soot Based PM (mg/kW-hr)	N/A
Gravimetric PM (mg/kW-hr)	319.2
BSFC (g/kW-hr)	261.4
BTE (%)	31.8

Table 10: Calculated In-cylinder Combustion Parameters at Optimized Strategy for Fuel A

Heat Released	kJ	0.606
Maximum Heat Release Rate	kJ/deg	0.029
Location of Maximum Heat Release Rate	deg ATDC	12.75
Maximum Pressure	kPa	5965
Location of Maximum Pressure	deg ATDC	2.75
Maximum Pressure Rise	bar/deg	2.06
Location of Maximum Pressure Rise	deg ATDC	-4.25
Maximum Average Bulk Mixture Temperature	K	1254
Location of Maximum Temperature	deg ATDC	21.50
Location of 10% Mass Fraction Burned	deg ATDC	-1.75
Location of 50% Mass Fraction Burned	deg ATDC	14.0
Location of 90% Mass Fraction Burned	deg ATDC	33.75

4.3 Fuel B

Fuel B was designed with a medium CN, low aromatics and high volatility. Specific properties for Fuel B are listed in Table 39 from the Appendix. From Table 1, the low aromatic content may decrease both NO_x and PM but may be counterbalanced by the increased volatility. Due to a high CN, the intake heating element was not implemented for each series of testing. The first series of testing consisted of performing EGR sweeps to identify an optimum range of EGR fractions that resulted in the lowest NO_x and PM emissions, refer to Table 11 for emission concentrations. For the following EGR sweeps, main SOI and pilot SOI were fixed at 0° and 40° BTDC, respectively, with a 50% injection split at 800 bar rail pressure. Figure 33 illustrates the in-cylinder pressure as a function of crank angle for the EGR fraction sweeps.

Figure 33: In-cylinder Pressure versus Crank Angle of EGR Sweeps for Fuel B

At an EGR fraction of 40%, the highest peak pressure is observed at 7880 kPa, while the highest EGR fraction (65%) resulted in the lowest peak pressure of approximately 5749 kPa. The effect of ignition delay becomes apparent at EGR fractions greater than 50% as the corresponding decrease in cylinder pressure led to misfire. NO_x was also significantly decreased with increasing EGR due to lower cylinder temperatures, but HC and PM significantly increased

at high EGR fractions, possibly due to incomplete combustion and misfires, see Table 11. Soot based PM was only established at 40% and 50% EGR but show that approximately 30% of PM is soot based. Figure 34 shows the HRR as a function of crank angle for the EGR fraction sweeps.

Table 11: EGR Sweep Emissions Data for Fuel B

EGR	HC (ppm)	NO _x (ppm)	HC (g/kW-hr)	NO _x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
30%	1376	92.25	6.314	1.416	311.7	370.5
35%	942.8	79.48	4.227	1.207	311.6	313.7
40%	973.4	76.29	4.729	1.249	311.0	364.1
50%	1589	39.92	5.741	0.470	285.5	330.7
65%	4690	11.52	13.18	0.103	302.0	1104

Figure 34: Gross HRR versus Crank Angle of EGR Sweeps for Fuel B

EGR fractions from 30-40% appear to have similar HRR curves while the increase in ignition delay can be observed at 50% and 65%. At 30% EGR, the first stage heat release is approximately 0.02 kJ/deg while at 65% the peak HRR in the first stage of combustion is 0.015 kJ/deg and retarded approximately 10°. From Figure 34, a 50% EGR fraction may represent an advanced combustion event more closely than the 30-40% EGR fractions based on the minimal

peak of first stage heat release and a rapid diffusion combustion. The 50% EGR fraction was thought to be the optimum operating point due to lower NO_x and HC emissions. Figure 35 shows the mass fraction burned as a function of crank angle for the EGR sweeps.

Figure 35: Mass Fraction Burned versus Crank Angle of EGR Sweeps for Fuel B

As illustrated in Figure 35, an increasing EGR fraction further retards combustion near TDC and the effect of ignition delay becomes apparent. EGR fractions of 30-40% result in a similar mass fraction burned as opposed to 65% EGR where misfiring became apparent. Additionally, the CA 10-90% was calculated to determine the approximate duration of combustion, see Figure 36. An EGR fraction of 65% resulted in the longest burn duration (~35°) while a 50% EGR fraction resulted in the longest burn duration (30°). Figure 36 shows the corresponding pressure rise rates for the EGR sweeps.

Figure 36: CA 10-90% Burn Duration and Max PRR of EGR Sweeps for Fuel B

EGR fractions of 30-40% provide a maximum pressure rise rate of 4 bar/deg while the 50-65% EGR fractions result in 2 bar/deg. Figure 37 shows the corresponding BTE and BSFC of the EGR sweeps. As seen from the figure, BTE and BSFC are generally consistent around 27% and 310 g/kW-hr with EGR fractions of 30-40% respectively, while a 50% EGR fraction provides the highest BTE at 29.5% and lowest BSFC at 285 g/kW-hr. From the in-cylinder pressure analysis and calculated emissions concentrations, it was determined that the optimum EGR fraction for Fuel B was 50%.

Figure 37: BTE and BSFC of EGR Sweeps for Fuel B

Table 12 shows the calculated in-cylinder parameters for the EGR sweeps. As seen in the table, the location of maximum pressure retards as EGR fraction increases until erratic combustion at 65% EGR. Maximum temperature is greatest at the optimum EGR point of 50% but occurs later (22.5°ATDC) during the combustion phase. Also depicted in the table, and as shown in Figure 35, the start of combustion retards as EGR fraction is increased until 50% EGR where the maximum pressure rise rate occurs much earlier, similar to Fuel A.

Table 12: Calculated In-cylinder Combustion Parameters of EGR Sweeps for Fuel B

EGR Fraction	-	30%	35%	40%	50%	65%
Heat Released	kJ	0.503	0.542	0.533	0.515	0.546
Maximum Heat Release Rate	kJ/deg	0.028	0.031	0.030	0.040	0.034
Maximum Pressure	kPa	7612	7843	7880	6351	5749
Location of Maximum Pressure	deg ATDC	1.75	2.25	2.0	3.5	4.75
Maximum Pressure Rise	bar/deg	4.03	3.95	4.04	2.10	1.68
Location of Maximum Pressure Rise	deg ATDC	-7.5	-7.0	-7.0	-3.25	-12.75
Maximum Temperature	K	1289	1329	1326	1322	1341
Location of Maximum Temperature	deg ATDC	19.5	20.0	19.75	21.0	24.0
Location of 10% Mass Fraction Burned	deg ATDC	-7.5	-7.25	-6.25	-1.75	1.0
Location of 50% Mass Fraction Burned	deg ATDC	10.5	10.75	11.25	13.0	15.75
Location of 90% Mass Fraction Burned	deg ATDC	23.0	23.25	25.0	28.5	35.75

Pilot injection sweeps were performed from 40-80°BTDC at 1200 bar rail pressure, 50% EGR and 30% fuel split. As seen in Figure 38, in-cylinder pressure was observed to decrease with advancing pilot injection. At a pilot injection of 40°BTDC, soot based PM was elevated but NO_x and HC were significantly lower compared to the 50-80°BTDC injection sweeps, see Table 13. At a pilot injection of greater than 60°, erratic combustion was observed due to wall wetting and can be confirmed by the extremely elevated HC and gravimetric PM emissions. Figure 39 shows the effect of advancing the pilot injection on the gross HRR.

Figure 38: In-cylinder Pressure versus Crank Angle of Pilot Injection Sweeps for Fuel B

Figure 39: Gross HRR versus Crank Angle of Pilot Injection Sweeps for Fuel B

Figure 39 indicates that early pilot injections retard the start of combustion and lowers the maximum peak HRR. Although the more advanced pilot injections result in more HCCI-like

HRR curve, ultimately higher emissions due to wall wetting limit them from an optimized strategy. Figure 40 shows the BTE and BSFC for the respective pilot injection sweeps.

Figure 40: BTE and BSFC of Pilot Injection Sweeps for Fuel B

From Figure 40, a drastic decrease in BTE is observed at the more advanced pilot injections while BSFC is increased. At a 40° pilot injection BTE is approximately 26.8% while at 80° the BTE is 26%. BSFC is increased 3% from 312 g/kW-hr at 40°BTDC to 322 g/kW-hr at 80°BTDC. The increase in BSFC is a result of the increase in fuel flow to maintain the 3.5 bmep due to erratic combustion from wall wetting at the more advanced pilot injections, also seen by the increase in gravimetric PM but from Table 13.

Table 13: Pilot Injection Sweep Emissions Data for Fuel B

Pilot °BTDC	HC (ppm)	NO _x (ppm)	HC (g/kW-hr)	NO _x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
40	791.6	73.20	3.345	1.050	312.6	232.7
50	1259	64.11	5.223	0.879	321.8	302.5
60	1392	78.97	5.909	1.112	321.9	451.4
70	2128	73.33	8.686	0.986	318.3	836.3
80	2689	60.74	10.54	0.779	321.3	1111

Fuel split sweeps were performed with similar decreases in cylinder pressure as the EGR and fuel split sweeps performed for Fuel A, refer to Figure 41. Soot based PM increased as the fuel split increased while gravimetric PM increased 30% from a 20% to 50% fuel split (Table 14).

Figure 41: Fuel Split Pressure Trace for Fuel B

Table 14: Fuel Split Emissions Data for Fuel B

Fuel Split	HC (ppm)	NO _x (ppm)	HC (g/kW-hr)	NO _x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
20%	1411	74.32	5.552	0.981	305.2	381.9
30%	1449	87.11	6.018	1.215	321.3	509.7
40%	1663	61.23	6.624	0.799	319.5	465.1
50%	1797	62.82	7.364	0.844	330.3	560.9

Rail pressure sweeps were also conducted in an attempt to reduce PM emissions but produced minimal differences in the pressure curve as seen in Figure 42. Figure 43 shows the effect of varying rail pressure on HRR. As expected, maximum peak HRR occurs at the highest rail pressure (1600 bar) although a reduction in PM is observed at 1600 bar, see Table 15.

Figure 42: Rail Pressure Sweep Pressure Trace for Fuel B

Figure 43: Gross HRR versus Crank Angle of Rail Pressure Sweeps for Fuel B

Table 15: Rail Pressure Sweep Emissions Data for Fuel B

Rail Pressure (bar)	HC (ppm)	NO _x (ppm)	HC (g/kW-hr)	NO _x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
800	1547	33.59	5.512	0.393	291.6	390.9
1200	2145	27.15	7.116	0.291	306.0	419.4
1600	2313	25.57	7.138	0.249	323.5	361.5

Upon completion of the discussed engine sweeps, the optimum strategy of the data set for Fuel B is summarized in Table 16 while Table 17 shows the emission concentrations at the respective optimized strategy. Table 18 shows the calculated in-cylinder combustion parameters at the optimized strategy.

Table 16: Optimized Strategy for Fuel B

EGR	50%
Rail Pressure	1200 bar
Fuel Split	30%
Pilot Injection	40 °BTDC

Table 17: Optimum Operating Strategy Emissions Results for Fuel B

NO _x (ppm)	73.2
HC (ppm)	791.6
Soot Based PM (mg/kW-hr)	57.00
Gravimetric PM (mg/kW-hr)	232.7
BSFC (g/kW-hr)	312.6
BTE (%)	26.9

Table 18: Calculated In-cylinder Combustion Parameters at Optimized Strategy for Fuel B

Heat Released	kJ	0.611
Maximum Heat Release Rate	kJ/deg	0.048
Location of Maximum Heat Release Rate	deg ATDC	10.75
Maximum Pressure	kPa	7529
Location of Maximum Pressure	deg ATDC	2.00
Maximum Pressure Rise	bar/deg	3.63
Location of Maximum Pressure Rise	deg ATDC	-7.75
Maximum Average Bulk Mixture Temperature	K	1471
Location of Maximum Temperature	deg ATDC	21.00
Location of 10% Mass Fraction Burned	deg ATDC	-7.25
Location of 50% Mass Fraction Burned	deg ATDC	11.5
Location of 90% Mass Fraction Burned	deg ATDC	23.75

4.4 Fuel C

Fuel C was designed with low CN, low aromatic content and low volatility. Specific properties for FACE C are listed in Table 39 from the Appendix. As suggested from Table 1, a low CN may result in increased NO_x but decreased PM. Due to the low aromatic content and volatility, both NO_x and PM may be significantly reduced. Because of the low CN of FACE C, the custom intake heating element was implemented for assisted starting and promotion of elevated cylinder temperatures during testing. The first series of testing consisted of performing EGR sweeps to identify an optimum range of EGR fractions that resulted in the lowest NO_x and PM emissions, see Table 19. By implementing the heating element, intake manifold temperature was increased approximately 20°C but was found to have minimal effect on in-cylinder pressure and heat release rate. For the following EGR sweeps, main SOI and pilot SOI were fixed at 0° and 40° BTDC respectively with a 50% injection split at 800 bar rail pressure. Misfiring was observed for EGR fractions above 45%. Figure 44 illustrates the in-cylinder pressure as a function of crank angle for the EGR sweeps.

Figure 44: In-cylinder Pressure versus Crank Angle of EGR Sweeps for Fuel C

As seen from Figure 44, an increasing EGR rate drastically decreases in-cylinder pressure, similar to the trends seen for Fuels A and B. At the lowest EGR fraction (20%), the highest peak pressure is observed at 8706 kPa while the highest EGR fraction (50%) resulted in a peak pressure of 6067 kPa. Consequently, NO_x was significantly decreased with increasing EGR fraction but HC emissions were elevated as seen in Table 19. Figure 45 shows the HRR as a function of crank angle for the EGR sweeps. As noted from Figure 45, an increasing EGR not only phases the first stage HRR toward TDC, but also reduces peak HRR over 50% from 0.053 kJ/deg at 50% EGR to 0.034 kJ/deg at 20% EGR. Figure 46 shows the corresponding mass fraction burned as a function of crank angle for the EGR fraction sweeps.

Table 19: EGR Sweep Emissions Data for Fuel C

EGR	HC (ppm)	NO_x (ppm)	HC (g/kW-hr)	NO_x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
20%	956.2	170.0	4.671	2.81	298.0	321.5
30%	1076	115.8	4.712	1.72	290.2	236.9
40%	1672	60.55	6.300	0.746	282.8	209.1
50%	6150	32.18	22.21	0.358	320.2	659.1

Figure 45: Gross HRR versus Crank Angle of EGR Sweeps for Fuel C

Figure 46: Mass Fraction Burned versus Crank Angle of EGR Sweeps for Fuel C

As illustrated in Figure 46, an increasing EGR fraction significantly retards combustion due to the increased heat capacity and lower oxygen content of the intake mixture. A CA 10-90% was calculated to determine the approximate duration of combustion, see Figure 47. A longer 10-90% burn duration was observed at 40% EGR while the shortest duration was at 50% EGR. Figure 47 also shows the corresponding pressure rise rates for the EGR sweeps.

Figure 47: CA 10-90% Burn Duration and Max PRR of EGR Sweeps for Fuel C

A linearly decreasing maximum pressure rise rate is depicted in Figure 47 as EGR fraction increases, similar to the results for Fuels A and B. An EGR fraction of 20% resulted in a maximum pressure rise rate of 5.5 bar/deg compared to just 1.7 bar/deg at 50% EGR. Also as EGR fraction increases, the pressure rise rates become retarded with 40% EGR having a maximum pressure rise rate closer to TDC (1°BTDC), refer to Table 20. Figure 48 shows the corresponding BTE and BSFC. As seen from the figure, BTE increases up to 40% EGR then suddenly drops at 50% EGR, thought to be due to misfire, with BSFC decreasing to 40% EGR and suddenly increasing at 50% EGR. The increase in BTE is directly related to BSFC, thus with an increase in BSFC, a decrease in BTE can be observed, and vice versa. From the in-cylinder pressure analysis and calculated emissions concentrations, it was determined that the optimum EGR fraction for low NO_x , PM and BSFC was 40-45%.

Figure 48: BTE and BSFC of EGR Sweeps for Fuel C

Table 20 shows the calculated in-cylinder parameters for the EGR sweeps. As seen in the table, the location of maximum pressure, pressure rise rate and temperature are phased further away from TDC as EGR fraction increases, as seen in the case for Fuels A and B. Also depicted in the table, and as shown is Figure 46, the start of combustion advances as EGR fraction is increased until misfire at 50% EGR.

Table 20: Calculated In-cylinder Combustion Parameters of EGR Sweeps

EGR Fraction	-	20%	30%	40%	50%
Heat Released	kJ	0.634	0.626	0.570	0.609
Maximum Heat Release Rate	kJ/deg	0.034	0.037	0.049	0.053
Maximum Pressure	kPa	8706	7972	6893	6067
Location of Maximum Pressure	deg ATDC	1.75	2.75	4.25	14.75
Maximum Pressure Rise	bar/deg	5.52	4.43	2.36	1.7
Location of Maximum Pressure Rise	deg ATDC	-7.0	-4.25	-1.0	12.75
Maximum Average Bulk Mixture Temperature	K	1461	1460	1478	1504
Location of Maximum Temperature	deg ATDC	19	19.5	20.5	22.5
Location of 10% Mass Fraction Burned	deg ATDC	-7.25	-4.75	0.5	0.75
Location of 50% Mass Fraction Burned	deg ATDC	10.0	10.75	12.5	12.5
Location of 90% Mass Fraction Burned	deg ATDC	21.5	24.5	30.0	28.5

Pilot injection sweeps were performed from 30-40°BTDC, at 40% EGR, 50% fuel split and 800 bar rail pressure. A longer burn duration was observed at 35°BTDC while the shortest burn duration occurred at 40°BTDC as seen in Figure 49. At 30°BTDC, NO_x was elevated but HC was significantly lower compared to the 35°BTDC pilot injection. However at 40°BTDC NO_x was minimal but HC emissions were extremely elevated due to incomplete combustion, see Table 21. It was determined that a pilot injection of 35 °BTDC provided the best tradeoff of NO_x, PM and HC emissions.

Figure 49: CA 10-90% Burn Duration and Max PRR of Pilot Injection Sweeps for Fuel C

Table 21: Pilot Injection Sweep Emissions Data for Fuel C

Pilot (°BTDC)	HC (ppm)	NO _x (ppm)	HC (g/kW-hr)	NO _x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
30	797.1	87.47	2.804	1.075	273.7	422.6
35	1345	54.76	4.724	0.635	281.2	225.9
40	2277	44.07	8.143	0.509	290.7	266.8

Rail pressure sweeps were also conducted to evaluate engine performance and attempt to reduce PM emissions. The effect of varying the rail pressure from 800–1600 bar and setting the

EGR fraction to 45% can be observed in Figure 50 while Figure 51 shows the corresponding HRR.

Figure 50: In-cylinder Pressure versus Crank Angle of Rail Pressure Sweeps for Fuel C

Figure 51: Gross HRR versus Crank Angle of Rail Pressure Sweeps for Fuel C

As seen in Figure 50, peak pressure is observed at 1600 bar while 800-1400 bar have similar peak pressures. Gravimetric PM was reduced at 1400 bar but an increase in PM was observed at 1600 bar, see Table 22. The HRR shown in Figure 51 indicates the effect of rail pressure on peak HRR. At 800 bar peak HRR in the first stage heat release is approximately 0.027 kJ/deg while at 1600 bar the peak is observed at 0.03 kJ/deg. The diffusion combustion event shows similar trends but a rail pressure of 1400 bar shows the highest peak HRR at 0.058 kJ/deg. Figure 52 shows the BTE and BSFC for the respective rail pressure sweeps.

Table 22: Rail Pressure Sweep Emissions Data for Fuel C

Rail Pressure (bar)	HC (ppm)	NO _x (ppm)	HC (g/kW-hr)	NO _x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
800	1131	60.15	4.21	0.752	289.7	240.0
1000	1477	58.44	5.54	0.725	284.7	213.9
1200	1441	67.61	5.49	0.854	294.1	203.6
1400	1814	51.34	6.30	0.586	283.1	203.5
1600	1976	60.20	7.40	0.742	292.5	225.0

Figure 52: BTE and BSFC of Rail Pressure Sweeps for Fuel C

Fuel split sweeps were performed from 40-60% at 1200 bar rail pressure, 40-45% EGR and a pilot injection of 35°BTDC with similar results to Fuels A and B. Figure 53 depicts the in-cylinder pressure traces of the respective fuel split sweeps. As seen from Figure 53, maximum in-cylinder pressures are slightly increased as the fuel split is increased. At a 40% fuel split, the main injection appears to have a much steeper slope ATDC while the 60% fuel split appears to have similar first stage and diffusion combustion events, as noted by the two similar peaks in the pressure trace. Figure 54 shows the gross heat release rate of the corresponding fuel split sweeps.

Figure 53: In-cylinder Pressure versus Crank Angle of Fuel Split Sweeps for Fuel C

Figure 54: Gross HRR versus Crank Angle of Fuel Split Sweeps for Fuel C

As seen from Figure 54, the HRR of the 40% and 50% fuel split are suggestive of an advanced combustion event as denoted by a minimal premixed combustion with a rapid diffusion combustion slightly ATDC. Although a 40% fuel split provided the more rapid combustion event, NO_x emissions were elevated due to higher cylinder temperatures. It was determined that a 50% fuel split provided the lowest NO_x and HC emissions tradeoff with similar PM emissions to 40% and 60% fuel split. Pressure rise rates were also minimal at a 50% fuel split while 40% and 60% fuel splits had similar pressure rise rates as seen in Figure 55. Figure 56 shows the brake thermal efficiency and brake specific fuel consumption of the fuel split sweeps while Table 23 shows the respective emissions concentrations.

Figure 55: Maximum Pressure Rise Rate of Fuel Split Sweeps for Fuel C

Figure 56: BTE and BSFC of Fuel Split Sweeps for Fuel C

Table 23: Fuel Split Sweep Emissions Data for Fuel C

Fuel Split	HC (ppm)	NO _x (ppm)	HC (g/kW-hr)	NO _x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
40%	1137	81.37	4.381	1.060	286.9	123.6
50%	1342	74.48	5.220	0.972	292.6	215.7
60%	1271	78.53	4.948	1.028	305.4	786.6

Upon completing the discussed sweeps for Fuel C, the optimum strategy of the data set for decreased NO_x and PM is summarized in Table 24 while Table 25 shows the emission concentrations at the respective optimized strategy. Table 26 shows the calculated in-cylinder combustion parameters at the optimized strategy.

Table 24: Optimized Strategy for Fuel C

EGR	40-45%
Rail Pressure	1200 bar
Fuel Split	50%
Pilot Injection	35 °BTDC

Table 25: Optimum Operating Strategy Emissions Results for Fuel C

NO _x (ppm)	104.1
HC (ppm)	904.3
Soot Based PM (mg/kW-hr)	14.3
Gravimetric PM (mg/kW-hr)	232.8
BSFC (g/kW-hr)	268.7
BTE (%)	31.3

Table 26: Calculated In-cylinder Combustion Parameters at Optimized Strategy for Fuel C

Heat Released	kJ	0.670
Maximum Heat Release Rate	kJ/deg	0.052
Location of Maximum Heat Release Rate	deg ATDC	9.75
Maximum Pressure	kPa	7221
Location of Maximum Pressure	deg ATDC	11.50
Maximum Pressure Rise	bar/deg	3.64
Location of Maximum Pressure Rise	deg ATDC	-2.50
Maximum Average Bulk Mixture Temperature	K	1616
Location of Maximum Temperature	deg ATDC	20.50
Location of 10% Mass Fraction Burned	deg ATDC	-2.25
Location of 50% Mass Fraction Burned	deg ATDC	10.25
Location of 90% Mass Fraction Burned	deg ATDC	23.25

4.5 Fuel D

Fuel D was designed with high a CN and volatility but low aromatic content. Specific properties for FACE D are listed in Table 39. From Table 1, a high CN may result in lower NO_x but increased PM and the low aromatic content may decrease both NO_x and PM emissions. Due to the high CN, the intake heating element was not implemented for each series of testing. The first series of testing consisted of performing EGR sweeps to identify an optimum range of EGR fractions that resulted in the lowest NO_x and PM emissions, see Table 27. For the following EGR sweeps, main SOI and pilot SOI were fixed at 0° and 40° BTDC respectively with a 50% injection split at 800 bar rail pressure. Note misfiring was observed for EGR fractions above 60%. Figure 57 illustrates the in-cylinder pressure as a function of crank angle for EGR fraction sweeps.

Figure 57: In-cylinder Pressure versus Crank Angle of EGR Sweeps for Fuel D

Similar to Fuels A, B and C, an increasing EGR fraction drastically decreases in-cylinder pressure. At the lowest EGR fraction (20%), the highest peak pressure is observed at 9300 kPa while the highest EGR fraction (70%) resulted in a peak pressure of approximately 5703 kPa. Due to very low cylinder pressures, erratic combustion was observed at 70% EGR while 60%

was on the threshold of misfire, denoted from HC and PM emissions. Consequently, NO_x was significantly decreased with increasing EGR fraction but HC and PM were elevated at high EGR fractions as seen in Table 27. Figure 58 shows the HRR as a function of crank angle for the EGR fraction sweeps.

Table 27: EGR Sweep Emissions Data for Fuel D

EGR	HC (ppm)	NO_x (ppm)	HC (g/kW-hr)	NO_x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
20%	546.8	181.3	3.017	3.513	330.7	576.2
30%	733.2	107.5	3.587	1.813	312.4	492.5
40%	992.6	59.57	4.093	0.829	295.5	499.6
50%	1297	35.29	4.701	0.422	298.8	530.6
60%	1658	22.63	5.258	0.230	286.2	611.7
70%	2950	11.73	7.943	0.092	298.5	906.2

Figure 58: Gross HRR versus Crank Angle of EGR Sweeps for Fuel D

As EGR fraction increases, the first stage heat release peak decreases and phases toward TDC. The peak in the first stage HRR, especially denoted at 20% EGR could be a result of auto ignition from the fuel since EGR quantities are minimal. At 20% EGR, the first stage peak HRR is

approximately 0.035 kJ/deg while at 70% the peak HRR in the premixed zone is just 0.01 kJ/deg. Although the 60-70% EGR fraction may represent an advanced combustion event more closely than the 20-50% EGR fractions, ultimately misfiring and resulting elevated PM and HC limit the operation of extremely high EGR fractions. An EGR fraction of 45-50% was thought to be the optimum operating point due to lower BSFC and NO_x and HC emissions. Figure 59 shows the mass fraction burned as a function of crank angle for the EGR sweeps.

Figure 59: Mass Fraction Burned versus Crank Angle of EGR Sweeps for Fuel D

As illustrated in Figure 59, an increasing EGR fraction phases combustion near and after TDC. The CA 10-90% burn duration was also calculated, see Figure 60. An EGR fraction of 70% results in shortest burn duration due to incomplete combustion while a 20% EGR fraction results in the longest burn duration at 37° CA. Figure 60 also shows the corresponding pressure rise rates for the EGR sweeps.

Figure 60: CA 10-90% Burn Duration and Max PRR of EGR Sweeps for Fuel D

A linearly decreasing maximum pressure rise rate and 10-90% burn duration is depicted in Figure 60 as EGR fraction increases, also seen from the results obtained from Fuels A, B and C. The increase in burn duration observed at 70% EGR could be a direct result of incomplete combustion of the fuel. An EGR fraction of 20% resulted in the highest maximum pressure rise rate at approximately 5.8 bar/deg compared to just 1.7 bar/deg at 70% EGR. Figure 61 shows the corresponding BTE and BSFC. As seen from the figure, BTE generally increases with elevated EGR fraction while BSFC generally decreases. From the in-cylinder pressure analysis and calculated emissions concentrations, it was determined that the optimum EGR fraction for low NO_x , PM and BSFC was 45-50%. Table 28 shows the calculated in-cylinder combustion parameters for the EGR sweeps. As seen in the table, the location of maximum pressure and temperature are being phased further away from TDC as EGR fraction increases until incomplete combustion at 70% EGR.

Figure 61: BTE and BSFC of EGR Sweeps for Fuel D

Table 28: Calculated In-cylinder Combustion Parameters of EGR Sweeps for Fuel D

EGR Fraction	-	20%	30%	40%	50%	60%	70%
Heat Released	kJ	0.609	0.573	0.552	0.526	0.514	0.484
Maximum Heat Release Rate	kJ/deg	0.034	0.027	0.027	0.030	0.033	0.028
Maximum Pressure	kPa	9300	8563	7860	7236	6512	5703
Location of Max Pressure	deg ATDC	1.25	1.5	1.75	2.5	3.25	4.25
Maximum Pressure Rise	bar/deg	5.77	4.99	4.25	3.41	2.36	1.74
Location of Maximum Pressure Rise	deg ATDC	-12.75	-10.5	-8.25	-6.25	-3.75	-13.0
Maximum Average Bulk Mixture Temperature	K	1336	1334	1330	1316	1317	1294
Location of Maximum Temperature	deg ATDC	18.75	19.5	20.25	21.0	22.25	25.75
Location of 10% Mass Fraction Burned	deg ATDC	-13.75	-11.5	-9.0	-6.75	-3.25	2.25
Location of 50% Mass Fraction Burned	deg ATDC	10.25	10.25	10.75	11.5	13.0	17.5
Location of 90% Mass Fraction Burned	deg ATDC	23.0	22.5	24.25	26.0	29.0	35.5

Pilot injection sweeps were performed from 30-55 °BTDC at 1200 bar rail pressure, 50% EGR and 50% fuel split. As seen in Figure 62, in-cylinder pressure was minimally affected by the

pilot injection sweeps. At 30 °BTDC, NO_x was elevated but HC were significantly lower than the 40-55 °BTDC injection sweeps. At a 45, 50 and 55°BTDC pilot, erratic combustion was observed and can be confirmed by the extremely elevated HC and PM emissions denoted by Table 29.

Figure 62: In-cylinder Pressure versus Crank Angle of Pilot Injection Sweeps for Fuel D

Table 29: Pilot Sweep Emissions Data for Fuel D

Pilot (°BTDC)	HC (ppm)	NO _x (ppm)	HC (g/kW-hr)	NO _x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
30	572.0	122.5	2.416	1.862	306.2	372.9
35	729.6	60.34	2.944	0.837	303.4	379.8
40	1269	58.52	5.367	0.831	328.9	736.7
45	1489	44.84	5.711	0.573	302.3	734.5
50	1691	48.08	7.217	0.680	333.7	1083
55	2031	41.12	8.189	0.547	335.9	1455

It was determined that a pilot injection of 35 °BTDC provided the best tradeoff of NO_x, PM and HC emissions. Although soot based PM was extremely elevated for Fuel D, the

optimum strategy for decreased NO_x and HC is summarized in Table 30. Table 31 shows the emission concentrations at the respective optimized strategy of the data set while Table 32 shows the calculated in-cylinder combustion parameters at the optimized strategy.

Table 30: Optimized Strategy for Fuel D

EGR	45-50%
Rail Pressure	1200 bar
Fuel Split	50%
Pilot Injection	35 °BTDC

Table 31: Optimum Operating Strategy Emissions Results for Fuel D

NO _x (ppm)	60.3
HC (ppm)	729.5
Soot Based PM (mg/kW-hr)	83.42
Gravimetric PM (mg/kW-hr)	379.7
BSFC (g/kW-hr)	303.4
BTE (%)	27.7

Table 32: Calculated In-cylinder Combustion Parameters at Optimized Strategy for Fuel D

Heat Released	KJ	0.620
Maximum Heat Release Rate	KJ/deg	0.039
Location of Maximum Heat Release Rate	ATDC	9.75
Maximum Pressure	kPa	7697
Location of Maximum Pressure	deg ATDC	2.00
Maximum Pressure Rise	bar/deg	4.23
Location of Maximum Pressure Rise	deg ATDC	-8.00
Maximum Average Bulk Mixture Temperature	K	1452
Location of Maximum Temperature	deg ATDC	20.0
Location of 10% Mass Fraction Burned	deg ATDC	-7.0
Location of 50% Mass Fraction Burned	deg ATDC	10.75
Location of 90% Mass Fraction Burned	deg ATDC	24.25

4.6 Fuel Characteristic Effects at a Specified Operating Condition

To examine the effect of fuel properties such as CN, sulfur content, volatility, etc, in-cylinder combustion analyses were used to develop trends at a single operating condition and correlated to the measured emissions concentrations from Figure 35. The operating condition listed in Table 33 was performed at 2100 rpm and a constant bmep of 3.5 bar. For specific fuel properties, refer to Table 39. Figure 63 shows the in-cylinder pressure at the single operating condition listed in Table 33. Note soot based PM was unavailable for each fuel at this operating condition.

Table 33: Operating Condition to Examine Fuel Characteristic Effects

EGR	%	50
Rail Pressure	bar	800
Fuel Split	%	50
Pilot Injection	°BTDC	40
Main Injection	°BTDC	0

Figure 63: In-cylinder Pressure versus Crank Angle at Specified Operating Condition of Each Fuel

As seen from Figure 63, Fuel A was observed to have the lowest peak pressure at 6284kPa while Fuel D had the highest peak pressure of 7235kPa, a 15% difference while Fuels B

and C had a 1% difference in peak pressure. Although Fuels A and D have a similar CN, the net heat of combustion per unit mass of Fuel A is much higher than Fuel D. However since Fuel A has a low specific gravity, it ultimately has a lower heat of combustion per unit volume. The decrease in peak pressure of Fuel A may be contributed to the lower net heat of combustion per unit volume since fuel was injected on per volume i.e. less fuel, lower BSFC and lower cylinder temperatures as verified by Figure 67 and NO_x emissions in Table 34. Figure 64 illustrates the gross HRR for each fuel at the specified operating conditions.

Table 34: Emissions Data at Specified Operating Condition

	HC (ppm)	NO _x (ppm)	HC (g/kW-hr)	NO _x (g/kW-hr)	BSFC (g/kW-hr)	Gravimetric PM (mg/kW-hr)
Fuel A	1484	21.14	4.566	0.211	298.1	368.6
Fuel B	1581	31.51	5.423	0.352	284.7	398.0
Fuel C	2068	44.30	7.778	0.543	320.2	249.5
Fuel D	1297	35.29	4.701	0.422	298.8	530.6

Figure 64: Gross HRR versus Crank Angle at Specified Operating Condition of Each Fuel

The increased ignition delay and higher HRR associated with lower CN fuels is verified by Figure 64 and Figure 66. As seen from the figure, Fuel C has a much longer ignition delay as well as a higher peak HRR during the first stage and diffusion combustion events, indicating elevated cylinder temperatures. Figure 65 shows the effect of CN on ignition delay. Note ignition delay was calculated by taking the difference of 10% mass fraction burned from the first noticeable heat release event. As expected, ignition delay is reduced with a higher CN fuel. The effect of the increasing peak HRR during the first stage heat release and diffusion combustion events on cylinder temperature can directly be related to a decreasing CN as seen in Figure 67.

Figure 65: Ignition Delay versus CN at Specified Operating Condition

Figure 66: Mass Fraction Burned versus Crank Angle at Specified Operating Condition

Figure 67: Maximum Average Bulk Mixture Temperature from Each Fuel at Specified Operating Condition

As the CN increases i.e. from Fuel A to C, the maximum in-cylinder temperature also increases. The increase in cylinder temperature can also be correlated to the NO_x plot in Figure

70. The effect of fuel properties on the CA 10-90% burn duration and maximum pressure rise rate becomes less apparent as seen in Figure 68.

Figure 68: CA 10-90% Burn Duration and Max PRR of Each Fuel at Specified Operating Condition

The decreasing PRR trend from Fuel A to C may be contributed to a decreasing T90 distillation temperature (increasing volatility) however the effects of the CN outweigh the contribution of volatility on combustion as established from Figure 64. Fuel characteristics become less apparent on the CA 10-90% burn duration as there seems to be no relation to the fuel properties listed in Table 39 and is more of a function of the HRR curve. Figure 69 shows the BTE and BSFC at the specified operating condition.

Figure 69: BTE and BSFC at Specified Operating Condition of Each Fuel

As seen in Figure 69, BTE decreases linearly while BSFC increases linearly. Fuel A has the highest BTE at 31.7% while Fuel D has the lowest BTE at 28.1%. The decrease in BTE may be attributed to the increasing quantity of fuel being injected to maintain the desired load as the net heat of combustion is reduced. The same trend can be correlated to BSFC i.e. with less fuel BSFC is higher from Equation 2-7. Table 35 shows the calculated combustion parameters at the specified operating condition for each fuel.

Table 35: Calculated In-cylinder Combustion Parameters of Each Fuel at Specified Operating Condition

EGR Fraction	-	Fuel A	Fuel B	Fuel C	Fuel D
Heat Released	kJ	0.424	0.498	0.569	0.526
Maximum Heat Release Rate	kJ/deg	0.029	0.038	0.052	0.034
Maximum Pressure	kPa	6284	6729	6657	7236
Location of Max Pressure	deg ATDC	1.75	3.25	4.5	2.5
Maximum Pressure Rise	bar/deg	2.60	2.45	1.91	3.41
Location of Maximum Pressure Rise	deg ATDC	-6.0	-3.5	0.75	-6.25
Maximum Average Bulk Mixture Temperature	K	1255	1334	1444	1316
Location of Maximum Temperature	deg ATDC	20.25	21	21	21
Location of 10% Mass Fraction Burned	deg ATDC	-4.5	-1.5	1.75	-6.5
Location of 50% Mass Fraction Burned	deg ATDC	12.25	13.0	13.0	11.5
Location of 90% Mass Fraction Burned	deg ATDC	27.25	27.0	27.0	26.0
Brake Thermal Efficiency (BTE)	%	31.69	29.58	28.80	28.11
Brake Specific Fuel Consumption (BSFC)	g/kW-hr	262.9	284.7	292.7	298.8

As seen from Table 35 and verified by the pressure and HRR curves at the specified operating condition, in-cylinder temperature increases as CN decreases. This trend can also be seen in Figure 67. Figure 70 shows radar plots of NO_x, HC and PM emissions as well as BTE of each fuel at the specified operating condition. As seen in the figure and verified by in-cylinder temperatures, NO_x emissions are higher for the low CN fuel. HC emissions are also highest for the lower CN fuel but PM emissions are lowest due to the increased time available for mixing.

Figure 70: Radar Plots of NO_x, HC, BTE and PM of Each Fuel at Specified Operating Condition

4.7 Fuel Characteristic Effects at Optimized Control Strategies

To examine the effect of fuel properties at the optimized control strategy, in-cylinder combustion analyses were used to develop trends while examining emission concentrations from each fuel. Table 36 shows the optimized control strategies for each fuel as determined from the previous sections.

Table 36: Optimized Control Strategy of Each Fuel

		Fuel A	Fuel B	Fuel C	Fuel D
EGR	%	55	50	40-45	45-50
Rail Pressure	bar	800	1200	1200	1200
Fuel Split	%	50	30	50	50
Pilot Injection	°BTDC	40	40	30	35
Main Injection	°BTDC	0	0	0	0

As seen from the table, optimum EGR rates can mainly be correlated to the CN of the fuel. Fuel A has the highest EGR rate at 55% which consequently also has the highest CN. The higher the CN, the more EGR that can be introduced. A higher CN fuel has a shorter ignition delay, while more EGR increases the ignition delay, verified by the pressure and HRR curves generated in the previous sections. Conversely, the lowest optimal EGR rates correspond to Fuel C with the lowest CN. Figure 71 shows the in-cylinder pressure at the optimized control strategy of each fuel.

Figure 71: In-cylinder Pressure versus Crank Angle at Optimized Control Strategy of Each Fuel

From Figure 71, Fuel A appears to have a significantly lower peak pressure than the other fuels. The decrease in cylinder pressure may be a direct result of the high EGR rate since Fuel A has the highest CN. As seen in the previous sections, in-cylinder pressure drastically

decreases due to the higher heat capacity and corresponding lower oxygen concentration at high EGR rates. The lower cylinder temperatures of Fuel A also resulted in the lowest peak HRR and more retarded combustion compared to the other fuels as seen in Figure 72. Due to the decreased cylinder temperatures associated with Fuel A, NO_x was significantly lower compared to the other fuels as seen in Table 37.

Table 37: Emissions Data at Optimized Control Strategy

	HC (ppm)	NO_x (ppm)	HC (g/kW-hr)	NO_x (g/kW-hr)	BSFC (g/kW-hr)	Soot Based PM (mg/kW-hr)	Gravimetric PM (mg/kW-hr)
Fuel A	1911	13.6	5.22	0.118	261.4	N/A	319.2
Fuel B	791.6	73.2	3.35	1.05	312.6	59	232.7
Fuel C	904.3	104.1	3.40	1.33	268.7	14.3	214.2
Fuel D	729.5	60.3	2.94	0.837	303.4	83.42	379.7

Figure 72: Gross HRR versus Crank Angle at Optimized Control Strategy of Each Fuel

As seen in the figure, combustion becomes significantly retarded with the lower CN fuel (Fuel A) as opposed to Fuels B and D which have a premixed combustion event at 10°BTDC compared to TDC for Fuels A and C. Due to the higher EGR rate for Fuel A, combustion is

phased with Fuel C (10-15% less EGR) as seen in the HRR curve. Although Fuel C has the highest peak HRR during the main injection, it also has the highest peak HRR at premixed combustion. As previously mentioned, an increase in peak HRR can be correlated to higher cylinder temperatures as verified by the NO_x plot in Figure 77. Although NO_x is reduced with the higher CN fuels, PM is also consequently higher. The increase in PM with higher CN fuels may be attributed to the decrease in ignition delay as seen in Figure 73. As ignition delay decreases, the time available for mixing becomes limited, as a well mixed, homogenous burn is required for low PM emissions.

Figure 73: Mass Fraction Burned versus Crank Angle at Optimum Control Strategy of Each Fuel

Figure 73 illustrates the effect of the CN on the ignition delay of each fuel. Fuel C, with the lowest CN, shows a much more retarded combustion starting at approximately 8°BTDC compared to Fuels B and D. The increased ignition delay with high amounts of EGR can again be seen in the MFB curve, as the start of combustion for Fuel A is nearly identical to that of Fuel C. The effects of parameters such aromatics, sulfur content and volatility on combustion were not evident, as the CN was the contributing factor for trends discovered through the

combustion analysis. The effect of CN on combustion can also be seen in the PRR and burn duration, refer to Figure 74.

Figure 74: CA 10-90% Burn Duration and Max PRR at Optimum Control Strategy of Each Fuel

As seen from Figure 74, the maximum PRR for Fuel C is comparable to Fuel B which has a 5-10% higher PRR rate at 3.5 bar/deg, while the highest CN fuel (Fuel A) has the lowest maximum PRR at 2 bar/deg. The 10-90% burn duration is shortest for the lower CN fuel, a good indicator of better mixing during the increased ignition delay. Figure 75 shows the BTE and BSFC at the optimized control strategy for each fuel.

Figure 75: BTE and BSFC at Optimum Control Strategy of Each Fuel

Fuels A and C have the highest BTE at nearly 32% and 31.5% respectively. The significant increase in BTE can be contributed to the increase in ignition delay due to high EGR rates for Fuel A and low CN of Fuel C. As depicted in Figure 72 and Figure 73, the HRR and MFB curves for Fuels A and C are very similar. Both fuels have a premixed combustion event occurring around TDC while the diffusion combustion events occur 5-15°ATDC. Fuels A and C also have the lowest BSFC for the optimized control strategy at 260 g/kW-hr and 270 g/kW-hr respectively. Table 38 shows the calculated in-cylinder combustion parameters at the optimized control strategy for each fuel.

Table 38: Calculated In-cylinder Combustion Parameters at Optimized Control Strategy of Each Fuel

		Fuel A	Fuel B	Fuel C	Fuel D
Heat Released	kJ	0.606	0.611	0.670	0.620
Maximum Heat Release Rate	kJ/deg	0.029	0.048	0.052	0.039
Maximum Pressure	kPa	5965	7529	7221	7697
Location of Max Pressure	deg ATDC	2.75	2.0	11.5	2.0
Maximum Pressure Rise	bar/deg	2.06	3.63	3.64	4.23
Location of Maximum Pressure Rise	deg ATDC	-4.25	-7.75	-2.5	-8.0
Maximum Average Bulk Mixture Temperature	K	1254	1471	1616	1452
Location of Maximum Temperature	deg ATDC	21.50	21.0	20.5	20.0
Location of 10% Mass Fraction Burned	deg ATDC	-1.75	-7.25	-2.25	-7.0
Location of 50% Mass Fraction Burned	deg ATDC	14.0	11.5	13.0	10.75
Location of 90% Mass Fraction Burned	deg ATDC	33.75	23.75	23.25	24.25

As depicted in Table 38 and verified by Figure 72 , the heat released for Fuel A is slightly lower than for Fuels B-D. The lower heat released for Fuel A may be a direct result of the high EGR rate as previously explained. Similarly, maximum in-cylinder temperature is approximately 200 K less than the next lowest temperatures of Fuels B and D which can also be correlated to the NO_x plot in Figure 77 and maximum average bulk mixture temperature in Figure 76, as NO_x is mainly a function of in-cylinder temperature. The dependency of in-cylinder temperature on NO_x emissions can also be verified by Fuel C.

Figure 76: Maximum Average Bulk Mixture Temperature from Each Fuel

Figure 76 shows the maximum average bulk mixture in-cylinder temperature at the optimized strategy from each fuel. As seen in the figure, maximum temperature is greatest from Fuel C which can also be correlated to the higher NO_x values shown in Figure 77. The higher cylinder temperatures from Fuel C may be attributed to the high HRR before and after TDC.

Figure 77: Radar Plots of NO_x, HC, BTE and PM at Optimized Control Strategy of Each Fuel

5 Conclusions and Recommendations

5.1 Conclusions

Advanced combustion strategies were examined by implementing a GM 1.9L diesel engine with an open source controller capable of controlling operating parameters such as EGR fraction, fuel quantity, injection timing and rail pressure. Four fuels were tested on the engine at 2100 RPM and 53 N-m in order to examine the effects of EGR and fuel characteristics on advanced combustion strategies.

In the data presented here, it was discovered that the amount of EGR can drastically influence the allowable range of injection timings by increasing ignition delay and lowering combustion temperatures. Due to the lower cylinder temperatures, NO_x emissions were reduced nearly 80% for the highest CN fuel, however HC and PM emissions increased 60%. At large amounts of EGR, especially greater than 55%, pilot injection events could be further advanced to allow additional mixing time until wall wetting became the contributing factor for HC and PM emissions.

In order to examine the effect of fuel characteristics on both the emissions and the combustion process, a single operating strategy was selected. It was discovered that the CN of the fuel was the contributing factor for phasing combustion and allowing additional mixing time. The low CN fuel showed a 33% reduction in PM emissions over the high CN fuel due to the reduction in fuel rich zones from the additional mixing time. NO_x and HC emissions of the low CN fuel were however increased 85% compared to the higher CN fuel at the optimum operating condition for the discussed parameter sweeps.

Upon completion of the data analyses and comparing the range of fuels both through emissions and combustion data, it was determined that a low CN fuel shows favorable characteristics for low PM emissions but may require an aftertreatment device such as a catalyst to suppress the elevated NO_x and HC emissions. A high CN fuel however provides favorable characteristics for low NO_x emissions, but may require an aftertreatment device such as a DPF to suppress the elevated PM emissions.

5.2 Recommendations

The following are some recommendations to further aid fundamental research in the area of advanced combustion relative to this document.

- Further testing at a second, much lower speed and load set point
- Use of a 16-valve engine as opposed to 8-valve for enhanced intake charge mixing to align with recently published advanced combustion research studies
- Obtain soot based PM data for all tests
- Further exploration of a single injection strategy
- Further exploration of main injection strategies at each fuel
- Use of CFD to establish intake flow dynamics of the test engine

6 References

1. United States. Presidential Memorandum Regarding Fuel Efficiency Standards. Washington DC, 2010. Web. 12 Oct 2010.
2. Juttu, S., Thipse, S., Marathe N.V. and Babu M.K., "Homogeneous Charge Compression Ignition (HCCI): A New Concept Near Zero NO_x and Particulate Matter (PM) from Diesel Engine Combustion". SAE Paper No. 2007-26-020, 2007.
3. Ebert, Jessica, "The Ideal Engine". Biodiesel Magazine. 2010
<http://www.biodieselmagazine.com/article.jsp?article_id=1753>.
4. Kawano, D., Suzuki, J., Ishii, H., Goto, Y., Odaka, M., Murata, Y., Kusaka, J. and Daisho, Y., "Ignition and Combustion Control of Diesel HCCI". SAE Paper No. 2005-01-2132, 2005.
5. Andreae, M. and Cheng, W., "On HCCI Engine Knock". SAE Paper No. 2007-01-1858, 2007.
6. Neely, G., Sasaki, S., Huang, Y., Leet, J. and Steward, D., "New Diesel Emission Control Strategy to Meet US Tier 2 Emissions Regulations". SAE Paper No. 2005-01-1091, 2005.
7. Christensen, M. and Johansson, B., "The Effect of Combustion Chamber Geometry on HCCI Operation". SAE Paper No. 2002-01-0425, 2002.
8. Szybist, J. and Bunting, B., "Cetane Number and Engine Speed Effects on Diesel HCCI Performance and Emissions". SAE Paper No. 2005-01-3723, 2005.
9. Taylor, J., Sluder, S. and Wright, K., "Fuels for Advanced Combustion Engines (FACE)". National Renewable Energy Laboratory, Diesel Engine Efficiency and Emissions Research (DEER) Conference, August 17, 2007.
10. Kukwon, C., Han, M., Sluder, S. and Wagner, R., "Experimental Investigation of the Effects of Fuel Characteristics on High Efficiency Clean Combustion in a Light-Duty Diesel Engine". SAE Paper No. 2009-01-2669, 2009.
11. Vanegas, A., Won, H. and Peters, N., "Influence of the Nozzle Spray Angle on Pollutant Formation and Combustion Efficiency for a PCCI Diesel Engine". SAE Paper No. 2009-01-1445, 2009.

12. Parks, J., Prikhodko, V., Barrone, T., Lewis, S. and Storey, J., "Emissions from Premixed Charge Compression Ignition (PCCI) Combustion and Affect on Emission Control Devices". The North American Catalysis Society, 2006.
13. "Standard Test Method for Calculated Cetane Index by Four Variable Equation". American Society for Testing and Materials, West Conshohocken, PA, ASTM-D4737-10, 2010.
14. "Standard Test Method for Determination of Ignition Delay and Derived Cetane Number (DCN) of Diesel Fuel Oils by Combustion in a Constant Volume Chamber". American Society for Testing and Materials, West Conshohocken, PA, ASTM-D6890-10a, 2010.
15. Yates, A., Viljoen, C. and Metcalf, O., "An Accurate Determination of the Cetane Number Value of GTL Fuel". SAE Paper No. 2007-01-0026, 2007.
16. "Standard Specification for Diesel Fuel Oils". American Society for Testing and Materials, West Conshohocken, PA, ASTM-D7170-09, 2010.
17. United States Environmental Protection Agency Office of Transportation and Air Quality- National Vehicle and Fuel Emissions Laboratory, "Fluorescent Indicator Adsorption Analysis of Petroleum Fuels Method". NVFEL Reference No. 124, Ann Arbor, MI, 2002.
18. McCarthy, C., Slodowske, W., Sienicki, E. and Jass, R., "Diesel Fuel Property Effects on Exhaust Emissions from a Heavy Duty Diesel Engine that Meets 1994 Emissions Requirments". SAE Paper No. 922267, 1992.
19. Ullman, T., Spreen, K. and Mason, R., "Effects of Fuel Aromatics, Cetane Number, and Cetane Improver on Emissions from a 1991 Prototype Heavy-Duty Diesel Engine". SAE Paper No. 902171, 1990.
20. Kwon, Y., Mann, N., Rickeard, D., Haugland, R., Ulvund, K., Kvinge, F. and Wilson, G., "Fuel Effects on Diesel Emissions – a New Understanding". SAE Paper No. 2001-01-3522, 2001.
21. Baranescu, R., "Influence of Fuel Sulfur on Diesel Particulate Emissions". SAE Paper No. 881174, 1988.
22. Speight, J., The Desulfurization of Heavy Oils and Residua. New York, NY, Marcel Dekker Inc., 1981.

23. Rose, K., Samaras, Z., Jansen, L., Clark, R., Elliot, N., Fontaras, G., Zemroch, P., Hall, D., Almena, M., Higham, C. and Kalogirou, M., "Impact of Biodiesel Blends on Fuel Consumption and Emissions in Euro 4 Compliant Vehicles". SAE Paper No. 2010-01-1484, 2010.
24. Hills, F. and Schleyerbach, C., "Diesel Fuel Properties and Engine Performance". SAE Paper No. 770316, 1977.
25. Kouremenos, D., Hountalas, D. and Kouremenos, A., "Experimental Investigation of the Effect of Fuel Composition on the Formation of Pollutants in Direct Injection Diesel Engines". SAE Paper No. 1999-01-0189, 1999.
26. Ochoterena, R., Larsson, M., Andersson, S. and Denbratt, I., "Optical Studies of Spray Development and Combustion Characterisation of Oxygenated and Fischer-Tropsch FuelsGTL Fuels". SAE Paper No. 2008-01-1393, 2008.
27. Pulkrabek, W.W., Engineering Fundamentals of the Internal Combustion Engine. Second Edition, Pearson Prentice-Hall, Upper Saddle River, NJ, 2004.
28. Heywood, J.B., Internal Combustion Engine Fundamentals. McGraw Hill, New York, NY, 1988.
29. Nuszowski, J., "The Effects of Fuel Additives on Diesel Engine Emissions during Steady State and Transient Operation". Doctor of Philosophy Thesis, West Virginia University, 2008.
30. Hasegawa, R. and Yanagihara, H., "HCCI Combustion in DI Diesel Engine". SAE Paper No. 2003-01-0745, 2003.
31. Dec, J., "A Conceptual Model of DI Diesel Combustion Based on Laser-Sheet Imaging". SAE Paper No. 970873, 1997.
32. Dec, J. and Canaan, R., "PLIF Imaging of NO Formation in a DI Diesel Engine". SAE Paper No. 980147, 1998.
33. Aceves, S., Flowers, D., Martinez-Frias, J., Smith, J., Dibble, R., Au, M. and Girard, J., "HCCI Combustion: Analysis and Experiments". SAE Paper No. 2001-01-2077, 2001.
34. Daniels, C., "The Comparison of Mass Fraction Burned Obtained from the Cylinder Pressure Signal and Spark Plug Ion Signal". SAE Paper No. 980140, 1998.

35. Mobley, C., "Non-Intrusive In-cylinder Pressure Measurement of Internal Combustion Engines". SAE Paper No. 1999-01-0544, 1999.
36. Zhu, J. and Lee, K., "Effects of Exhaust Gas Recirculation on Particulate Morphology for a Light-Duty Diesel Engine". SAE Journal of Fuels and Lubricants. Vol. 114, No. 2, pp. 99-105, 2005.
37. Sjobberg, M. and Dec, J., "Influence of EGR Quality and Unmixedness on the High Load Limits of HCCI Engines". SAE Paper No. 2009-01-0666, 2009.
38. Murari R., "Effect of Exhaust Gas Recirculation on Combustion and Odorous Emissions in Direct Injection Diesel Engines". SAE Paper No. 2008-01-2482, 2008.
39. Lundqvist, U., Smedler, G. and Stalhammar, P., "A Comparison Between Different EGR Systems for HD Diesel Engines and Their Effect on Performance, Fuel Consumption and Emissions". SAE Paper No. 2000-01-0226, 2000.
40. "CAFE Overview". Web. 20 Nov 2010. <<http://www.nhtsa.dot.gov/CARS/rules/CAFE/overview.htm>.>
41. Kreso, A., Johnson, J., Gratz, L., Bagley, S. and Leddy, D., "A Study of the Effects of Exhaust Gas Recirculation on Heavy-Duty Diesel Engine Emissions". SAE Paper No. 981422, 1998.
42. Maiboom, A., Tauzia, X., Shah, S. and Hetet, J-F., "Experimental Study of an LP EGR System on an Automotive Diesel Engine, compared to HP EGR with respect to NO_x and PM Emissions and Specific Fuel Consumption". SAE Paper No. 2009-24-0138, 2009.
43. Wrap Off-Road Diesel Retrofit Guidance Document, "Low Pressure Exhaust Gas Recirculation". Emissions Advantage, Vol. 2 Sec. VII. 2005.
44. Walt Moore, "Living With Cooled-EGR Engines". Trade Association Business Publications. pp 68-69. Dec. 2003.
45. Yang, F. and Minggao, O., "Experimental Research on EGR in a Diesel Engine Equipped with Common Rail Injection System". SAE Paper No. 2003-01-0351, 2003.
46. Haiyong Peng, Yi Cui, Lei Shi and Kangyao Deng, Effects of EGR on Combustion Process of DI Engine During Cold Start. Higher Education Press and Springer-Verlag, 2008.

47. Dennis, A.J., Garner, C.P. and Taylor, D., "The Effect of EGR on Diesel Engine Wear". SAE Journal of Engines. Vol. 108, pp. 7, 1999.
48. Szybist, J., Kirby, S. and Boehman, A., "NO_x Emissions of Alternative Diesel Fuels: A Comparative Analysis of Biodiesel and FT Diesel". Energy & Fuels, Vol. 19, No. 4, pp. 1492, 2005.
49. Code of Federal Regulations, Title 40 CFR Part 1065 & 86, Washington, D.C., Office of Federal Register National Archives and Records Administration, 2007.
50. "Gas Phase Measurements," Dieselnet, September 2008,
<http://www.dieselnet.com/tech/measure_gas.html#tech>
51. Alpha Omega Instruments, "Oxygen Sensor Types". Alpha Omega Instruments, 2006.
<http://www.aoi-corp.com/additional_information/oxygen_sensor_types/>.
52. AVL, "AVL Micro Soot Sensor". Impressum, 2010. < <https://www.avl.com/micro-soot-sensor>>.
53. Yu, R., Bai, X.S., Lehtiniemi, H., Ahmed, S.S., Mauss, F., Richter, M., Alden, M., Hildingsson, L., Johansson, B. and Hultqvist, A., "Effect of Turbulence and Initial Temperature Inhomogeneity on HCCI". SAE Paper No. 2006-01-3318, 2006.

7 Appendix

Table 39: Test Fuel Properties

Fuel	Cetane Number	Specific Gravity	Flash Point (° C)	Sulfur (ppm)	T 90 Distillation Temperature (° C)	Aromatics (Vol %)	Net Heat of Combustion (MJ/kg)	Net Heat of Combustion (MJ/m ³)
A	56	0.8095	58.9	1.7	311	35*	43.32	35001
B	44	0.8496	70	8.4	307	28.8	42.86	36338
C	30.3	0.8081	58.7	1.2	274	23.5	42.80	34520
D	53.7	0.8411	74.4	2	340	20.4	42.80	35924
* Maximum Allowed

Figure 78: HDDE EGR Cooler

Figure 79: Engine Retrofitted with HDDE EGR Cooler and Heating Element

Table 40: In-cylinder Pressure Sensor Manufacturer Specifications

Type 6056A Piezoelectric Crystal Pressure Transducer		
Measuring Range	bar	0-250
Overload	bar	300
Sensitivity	pC/bar	-20
Natural Frequency	kHz	160
Shock Resistance	g	2000
Capacitance	pF	5

Figure 80: LabVIEW Screenshot

7-1

Table 41: Single Injection Emissions Data for Fuel A

EGR	HC (ppm)	NO _x (ppm)	HC (g/kW-hr)	NO _x (g/kW-hr)	BSFC (g/kW-hr)	BMEP	Gravimetric PM (mg/kW-hr)
45%	1948	287.0	7.883	3.884	325.0	3.34	338.3
50%	2099	457.2	8.789	6.420	335.2	3.03	621.0
65%	4075	18.66	10.701	0.158	244.6	3.66	654.3
70%	7757	7.544	16.415	0.048	229.6	3.75	739.9