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AN EXPERIMENTAL STUDY ON THE IMPACT OF WATER INJECTION ON THE PERFORMANCE AND EMISSIONS OF A NATURAL GAS – DIESEL PILOT ENGINE

Nic Tuma
Michigan Technological University, nctuma@mtu.edu

AN EXPERIMENTAL STUDY ON THE IMPACT OF WATER INJECTION ON THE PERFORMANCE AND EMISSIONS OF A NATURAL GAS – DIESEL PILOT ENGINE

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Nicolas C. Tuma

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Department of Mechanical Engineering – Engineering Mechanics

Thesis Co-Advisor:	*Dr.* Jeffery *D. Naber*
Thesis Co-Advisor:	*Dr. Jeremy Worm*
Committee Member:	*Dr. Brian Eggart*
Committee Member:	*Dr. Scott Miers*
Department Chair:	Dr. *William W. Predebon*

Table of Contents

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List of Abbreviations

NG – Natural Gas

DSR – Diesel Substitution Ratio

WFR – Water to Fuel Ratio

THC – Total Hydrocarbons

NOx – Nitric Oxide + Nitrogen Dioxide

CO – Carbon Monoxide

CO_2 – Carbon Dioxide

PM – Particulate Matter (Soot)

TWC – Three-Way Catalytic Converter

NI – National Instruments

DAQ – Data Acquisition

CAS – Combustion Analysis Software

DPF – Diesel Particulate Filter

EGR – Exhaust Gas Recirculation

SCR – Selective Catalytic Reduction

VGT – Variable Geometry Turbocharger

AFR – Air-to-Fuel Ratio

SOI – Start of Injection

EOI – End of Injection

°bTDC – Crank Angle Degrees Before Top Dead Center

CAD – Crank Angle Degrees

H_20 – Water

O_2 – Oxygen

CI – Compression Ignition

SI – Spark Ignition

ICE – Internal Combustion Engine

LHV – Lower Heating Value

CR – Compression Ratio

ID – Ignition Delay

BMEP – Brake Mean Effective Pressure

IMEP – Indicated Mean Effective Pressure

NMEP – Net Mean Effective Pressure

BTE – Break Thermal Efficiency

NG-DP – Natural Gas – Diesel Pilot

Abstract

Natural gas has been gaining popularity as an alternative fuel due to its high availability, low CO2 emissions, and low cost. In this experimental study, water injection's impact on medium and heavy-duty engine operation fueled by natural gas and pilot diesel injection for ignition was studied under stochiometric operation for use with a three-way catalytic converter to meet criteria emissions for off-road power generation. To retain high efficiencies, a high compression ratio of 17.3:1 was used. Maintaining stoichiometric operation with a high compression ratio leads to combustion knock, pre-ignition, and high NOx formation. Conventionally, cooled EGR can be used to reduce NOx, but results in increased soot and does not eliminate combustion knock and pre-ignition. As an alternative to EGR this work utilized port injected water to provide on demand charge cooling, successfully reducing both NOx and soot while enabling high-load operation. A combination of both high and low speeds and loads were tested to study the impact of water injection on the emissions and performance of the natural gas, diesel-pilot engine. Additionally, water injections impact on diesel only operation was tested to provide comparison metrics and aid in a better understanding of the mechanisms at work when injecting water in an internal combustion engine.

At full load, 16.8 bar BMEP, it was found that a water to fuel ratio of 0.5:1 was sufficient to enabling the knock free operation without significant increase in combustion duration or instability where operating at this load without water resulted in pre-ignition. Increasing the water to fuel ratio to 1:1 enabled a 21 bar BMEP load. At 12.5 bar BMEP, the NOx emission was reduced from 13.5 g/kwh to 7.2 g/kwh with a water to fuel mass ratio of 1.5:1. In addition to solving the high NOx and pre-ignition problem, a water to fuel ratio of 2.5:1 at 16.8 bar BMEP also decreased the soot content in the exhaust by a factor of 3.5 with only a small penalty in efficiency, decreasing break thermal efficiency from 41 to 40%.

1 Introduction

This chapter is intended to give a background to the natural gas-powered internal combustion engine (ICE), provide the motivation and overview of the project's goals and objectives, and outline the remainder of contents in this thesis.

1.1 Background

This project compares and evaluates the impact of WI on diesel-only and natural gas with diesel-pilot (NG-DP) engines. A background of the workings and existing technology for each of these topics is provided in the three sections below. Specific results and capabilities of these engines along with the motivation for the combination of a natural gas ICE with WI will be covered later in the literature review of this paper.

1.1.1 Diesel Operation

Although already well understood, as the first diesel engine was created in 1890 and have been the topic of countless research projects, the benefits and downfalls of diesel engines are outlined here for the purpose comparison to the focus of this project, a natural gas-diesel pilot engine.

Diesel engines are typically known for their high brake thermal efficiency (BTE), commonly ranging above 40% BTE. This fuel efficiency benefit stems from the high compression ratio (CR), unthrottled operation, high combustion efficiency, and globally lean charge within the cylinder [1]. Additionally, the high pressures experienced in compression ignition (CI) engines requires robust construction leading to highly durable engines that last longer than typical gasoline spark ignited (SI) engines. The lean burn and high combustion efficiency of diesel engines results in low HC and CO emissions although particulate matter and NOx are high and require an extensive aftertreatment system.

What is known as the NOx-Soot tradeoff is the main downfall of diesel engines where tuning to decrease NOx typically results in an increase in PM, and vice versa. As mentioned, an extensive after treatment system must be utilized to reduce these harmful emissions and meet environmental tailpipe out emission regulations. A three-way catalytic converter (TWC) cannot be used as a stoichiometric operation is required for the TWC to operate properly. Instead, a diesel particulate filter (DPF) and diesel oxidation catalyst (DOC) are used to collect and oxidize engine out PM. Next, NO and NO_2, collectively known as NO_x, are passed through a urea mist or diesel exhaust fluid (DEF) in the decomposition reactor before passing through the selective catalytic reduction (SCR) system where NO_x and urea are converted to N_2 and H_2O [1]. Although capable of reducing harmful emissions to "near zero", new regulations are becoming more stringent and require larger, heavier, and more expensive aftertreatment systems to meet these standards. Additionally, engine tuning can be done to reduce emissions, but result in a BTE decrease.

1.1.2 Natural Gas in Internal Combustion Engines

Generally, there are two types of natural gas engines, SI and dual fuel CI. This section will outline the basic operation for each of these engines where the specific benefits and downfalls will be covered in the literature review of this thesis.

Overall, natural gas is an inexpensive, readily available fuel holding a high hydrogen to carbon ration and high lower heating value (LHV) in comparison to diesel and gasoline.

SI natural gas engines are simple where natural gas is either direct injected (DI), port injected (PI), or carbureted and is ignited with a spark plug at the desired time. CI natural gas engines bring some additional complexity as natural gas has a low cetane number making CI extremely difficult. A high cetane fuel such as diesel is used instead to start the combustion. The use of two fuels gives this operating mode the name, dual fuel, however the naming convention grows more specific when diesel contribution quantity is explored. Diesel pilot and diesel micro-pilot engines are both dual fuel but are specified to have small diesel contributions at near 15% and less than 5% diesel contribution, respectively. The experimental study conducted in this project used a CI dual fuel engine, more specifically, a natural gas-diesel pilot (NG-DP) engine with 15% diesel contribution by energy.

1.1.3 Water Injection in Internal Combustion Engines

WI in ICEs has been used for many years as a method for controlling combustion. An overview of how and why WI is used will be covered in this section where specifics will be covered later in the literature review.

The high expansion ratio of water that takes place during the phase change from liquid to gas brought on the use of steam engines many years ago. Water is also a common flame suppressant due to its high specific heat capacity. The combination of these characteristics makes water an excellent combustion control mechanism where it has been direct injected, port injected, and emulsified within fuel in many different research projects over the years [3,4,5,6,7]. In general, water is used to reduce in-cylinder temperatures and slow combustion, mitigating knock, and decreasing NOx. Depending on injection timing, the conversion of water to steam can also be used to increase cylinder pressure, utilizing heat that would otherwise be expelled in the exhaust. The use of port injected water was utilized in this project to reduce knock, enable high load, and reduce engine out NOx emissions.

1.2 Project Goals and Objectives

The goals of this project are to observe the impact WI has on diesel only operation and diesel-pilot with natural gas operation. Additionally, the comparison of performance and emissions between these operating modes, with and without water, is studied. The motivation for these goals at each stage of the project are covered below.

- Diesel only and NG-DP operation are first studied to compare their performance and emissions and with the goal of obtaining baseline values for comparison with the addition of WI.
- Diesel operation with WI was studied next with the goal of maintaining high BTE and low IMEP COV while decreasing NOx.
- NG-DP with WI was the final experimental stage and is the main focus of this study. Motivated by current problems found in literature, the goals of this section

were to reduce knock, reach high load, and reduce NOx while targeting high BTE and maintaining combustion stability with a COV of IMEP below 3.0.

- The final goal of this project is to compare the combustion, performance, and emissions of these operating modes to aid in further engine design and operation for dual fuel engines with water injection.

In meeting the goals outlined above, project objectives were formed starting with a literature review on related topics, followed by engine and test cell setup. With goals for each study in mind, an experimental test plan was created and conducted, followed by analysis and comparison of results. Project objectives in meeting these goals are as follows:

1. Review relevant literature on natural gas fueled ICEs, WI, and comparable industry ICEs to obtain comparison values and develop objectives to pursue with the experimental data.
2. Create an experimental test plan based off the objectives determined from the literature review.
3. Plan, fabricate, and install engine and test cell modifications required to achieve the defined test objectives.
4. Install and setup engine and test cell instrumentation to monitor and collect data during the experimental tests.
5. Perform the experimental tests and collect data for analysis and comparison.
6. Quantify and summarize the results of experimental tests and compare to similar studies with results obtained in the literature review section.

1.3 Thesis Outline

This section provides an overview and the motivation for the remaining chapters found in this paper. The research outlined in figure 1 was conducted to provide a comparison between diesel only operation and diesel-pilot natural gas operation as well as creating a better understanding for the impact WI has on the two engine operating conditions.

Chapter 2: Literature Review

Motivation: Form a knowedge base on existing natural gas engines and water injection to create experimental ojbectives and values for comparison.

- **2.1**: Review of current performance and emissions of ICEs similar to this project's.
- **2.2**: Review on the impacts of water injection in other projects.
- **2.3**: Summary of findings and how they are used to create experimental objectives.

Chapter 3: Experimental Setup and Test Plan

Motivation: Prepare the engine, test cell, and experimentor for conducting experiments.

- **3.1**: Detailed overview of engine and test cell setup and modifications.
- **3.2**: Instrumentaion installation and specifications.
- **3.3**: Experimental test plan and data analysis methods.

Chapter 4: Experimental Results & Discussion

Motivation: Present and explain experimental results and provide a comaprison to similar engines.

- 4.1: Diesel with Water Injection
- 4.3: Natural Gas with Diesel Pilot + Water Injection
- 4.4: Summary of Results

Figure 1. Thesis Chapter Outline

2 Literature Review

This section will cover the literature that was reviewed for the motivation of conducting this research. In particular, the transportation and power generation industries' increasing interest in alternative fuels for emission reductions and efficiency increases. After determining the industries current position, motivation, and goals on this matter, natural gas engine performance and emission capabilities are covered to provide guidance in creating objectives and comparisons for this project's experimental data. An overview of WI in ICEs is then covered to gain insight on the impact it has on combustion, NOx emission reduction, and enabling of high load operation. Finally, a summary of the literature review and summary table of the reviewed results are provided to be used for comparison to this project's experimental data.

2.1 Motivation

CI Diesel engines, well known for their high fuel conversion efficiency and reliability, are extremely prominent in both the transportation and power generation industries [8]. Diesel engines, however, are also known for emitting high levels of NOx and PM [9]. PM particles have been identified as a carcinogen by the World Health Organization in 2012 along with other studies proving PM to be extremely detrimental to human health [10,11,12]. NOx is also a detrimental emission and is a major cause for both smog and acid rain [13]. For these reasons, both NOx and PM tail-pipe out emissions are being tightly regulated for Non-road CI engines at 0.04 and 0.02 g/kWh by the EPA Teir 4, emission standards, respectively [14].

Increasingly stringent emission standards imposed by the EPA are driving researchers and industry alike to pursue the development of alternative fuels [14]. Natural gas is an attractive alternative fuel for many reasons including its previously mentioned low cost, high availability around the world, and most importantly, its chemical properties. Natural gas engines are by no means, a new concept, and have been used vehicles since the 1920s, but in recent years are utilized mainly in power generation [15]. Natural gas is the main source of electric energy in the United States, where 34% of primary energy consumption in 2020 was generated with natural gas [16]. Only 4% of transportation in the United States is powered by natural gas however, which is dominated by petroleum products at 90% [16]. Composed of at least 80% methane (CH_4), the shortest hydrocarbon, natural gas has a high hydrogen to carbon ratio (HC ratio) of 4:1. Table 1 provides a comparison of the properties for these three fuels.

Table 1. Fuel Property Comparison

	Fuel Type		
Property	**Natural Gas**	**Diesel**	**Gasoline**
Lower Heating Value [MJ/kg]	45	42.8	44.0
Hydrogen to Carbon Ratio	~ 4:1	1.9:1	2.25:1
Stoichiometric Air to Fuel Ratio	14.5	14.4	14.6
Specific Heat - Vapor (Liquid) [kJ/kg*K]	2 (-)	1.7 (1.9)	1.7 (2.4)

The high HC ratio of natural gas means there is less carbon present to form CO_2 emission, a clear advantage of natural gas over conventional fuels. Additionally, its high LHV in comparison with diesel and gasoline means the fuel contains more energy per mass, meaning higher BTE can be achieved [1]. The fuel properties described are a major contributor to the interest in natural gas as a fuel. The increased availability and thus decreased price of natural gas due to developments in production technology also contribute to the industries interest [16]. There are also renewable forms of natural gas (RNG), which is collected from landfills, livestock, and wastewater treatment centers that would otherwise release the natural gas into the atmosphere [17]. Combusting RNG and effectively converting methane to CO_2, has a positive impact on the environment because methane traps 28 to 36 times more heat than CO_2 over a 100-year time scale [17]. The use and development of RNG is beyond the scope of this paper but is important to consider in evaluating feasibility of NG as an alternative fuel.

The increase in natural gas engine research and utilization has brought many new avenues for experimentation. Both SI and CI engines are common for natural gas operation, each holding their own benefits and downfalls. Experimentation within each of these operating modes includes variables such as spark timing, diesel pilot quantity, start of diesel injection, natural gas induction method, EGR use, diesel substitution ratio, air to fuel ratio (AFR), and in this projects case, WI. In the research reviewed, these variables and their impact on combustion, emissions, and performance were studied. The following sections outline the relevant findings and how they were used to guide this project's experimental objectives.

2.2 Natural Gas Engines

As mentioned above, natural gas engines are already prevalent in both the transportation and power generation industries. Both CI and SI NG engines exist, and each hold their own advantages and disadvantages.

2.2.1 Spark Ignited Natural Gas Engines

SI natural gas engines are currently more common due to their simplicity of conversion from gasoline operation. A natural gas induction method simply needs to be added to the engine where an already present throttle and spark plug can be used to meter and ignite the natural gas. Mono-fueled SI natural gas engines exist in light, medium, and heavy-duty applications and are typically operated at a stoichiometric mixture to achieve high combustion efficiency and utilize a three-way catalytic converter. The problem with SI engines though is their decreased efficiency experienced at part load from pumping losses experienced in throttled operation [18]. Additionally, SI engines suffer from decreased BTE compared to CI engines due to their lower knock limitations. Regardless of the higher knock resistance of natural gas compared to gasoline, the slower burn rate of natural gas increase chances for end gas ignition [19]. Both EGR and lean burn have been used to mitigate knock, lowering combustion temperatures, and therefore mitigating knock. These methods bring new problems however where lean burn disables the use of a TWC and can cause combustion instability and EGR reduces combustion efficiency and results in increased PM emission [20]. The efficiency disadvantages of SI NG operation as well as the large amount of research already conducted on them lead the focus of this project to CI NG engines.

2.2.2 Compression Ignition Natural Gas Engines

CI engines are commonly known for their high torque and high efficiency capabilities. This makes them an attractive option for use in developing a high efficiency medium or heavy-duty natural gas engine. A problem with this concept though comes from natural gas's high auto-ignition temperature, or low cetane number. This fact makes it extremely difficult to achieve reliable compression-ignition with natural gas. Where previously referenced as impossible by several researchers [15,18, 21], one paper by Naber et al has shown it is in fact possible with the correct modifications [22].

Instead of using SI or CI, natural gas can be ignited in one of two ways:

- <u>Hot surface assisted CI</u> – Natural gas is direct injected onto a hot surface such as a glow plug, igniting the mixture. Efficiency and implementation advantages are achieved with this method as storage of only one fuel is required and no additional, lower LHV fuel is required. This method, however, suffers from durability and maintenance issues due to the high temperature required to be maintained on the glow plug. Additionally, the requirement of direct injecting natural gas causes a locally rich region and thus higher PM emissions. The complicated ignition system package design and low combustion efficiency acquired in this method are reasons for less recent research interest in this method [8].
- <u>Dual fuel</u> – A high cetane fuel, typically diesel, is direct injected, to create an ignition source for the natural gas. This method allows NG to be introduced with DI, PI, or a carburetor (mixer). Additionally, different diesel injection strategies allow for a wide range of optimization. The engine can also be reverted to diesel-only operation with this method in the case natural gas is not available. Engines utilizing this ignition system benefit from the many variables available for optimization making it the focus of many recent research papers. [8]

With the goals of this project in mind, a dual fuel CI engine configuration was chosen as it is best suited in meeting low emissions, high efficiency, and medium load operation while operating at stoichiometric operation, utilizing already available engine infrastructure.

2.2.2.1 Combustion Characteristics

Diesel combustion is typically categorized into four stages: [1]

1. **Ignition delay** – time between start of injection and start of combustion measured at the initial rapid increase in heat release.
2. **Premix Combustion** – rapid heat release when atomized diesel reaches the flammability limit.
3. **Mixing Controlled Combustion –** occurs after the premix combustion has occurred and injected fuel is rapidly combusted.
4. **Late Combustion –** unburnt fuel and rich burn products are burned after fuel injection has stopped.

In a dual fuel operation however, the mixing-controlled combustion phase is replaced with the flame propagation of natural gas. The ignition delay and premixed combustion states still exist and are critical to the quality of the natural gas combustion. This is where a large amount of research stems from as injection timing, duration, and pressure can all be optimized to obtain high combustion efficiency.

The in-cylinder pressure rise in dual fuel operation has shown to be both lower and higher when compared to diesel only operation [23,24]. This can be attributed to the timing and quantity of the diesel pilot injection where earlier and larger injections increase pressure rise rate. The larger diesel pilot injection forms a larger flame kernel thus reaching and igniting more natural gas at once. The opposite effect happens when timing is retarded, and diesel injection quantity is decreased. [24]. The AFR also has a strong impact on pressure rise rate where lean burn natural gas engines suffer from lower burn charge temperature and thus lower pressure rise rates [18].

Ignition delay is an important metric and if extended too long can cause a low temperature, slow burning flame kernel due to over dilution of the injected diesel [9]

In dual fuel engines, ignition delay has been found to increase in comparison with diesel only operation [24,25]. The major reason of this ignition delay extension is mainly due to chemical factors where it has been shown that the free radicals of natural gas interfere with the diesel pilot injection, delaying the combustion of the diesel pilot [25]. The higher specific heat capacity of natural gas can also have some impact as the higher heat transfer to the fuel in comparison with air lowers the pre-combustion in-cylinder temperature. Lower in-cylinder temperature has been shown to increase the diesel pilot ignition delay [26]. A shorter ignition delay is optimal for producing high combustion efficiency meaning a larger and earlier diesel injection quantity are preferred when considering combustion efficiency alone.

The heat release rate is another important combustion metric to review as it is a good indicator of overall combustion quality. Papagiannakis et al showed the AHR and cylinder pressure trace collected for diesel only operation along with 52% and 85% natural gas [27]. Increasing natural gas contribution resulted in decreased peak heat release rate. The decreased heat release rate amplitude and increased duration as diesel quantity is decreased was attributed to the slower burning natural gas when operated lean. Combustion phasing was also shown to be later with increasing natural gas content. The retarded combustion phasing is due to the increased ignition delay previously discussed. The late burn results in high exhaust temperature, wasting fuel energy and causing lower fuel conversion efficiency due to the combustion "chasing" the piston down the cylinder [27].

To combat the increased ignition delay and slow heat release, stoichiometric operation is becoming more common in research and industry [2]. Stoichiometric natural gas operation results in a higher flame propagation rate of the natural gas, increasing the heat release rate and effectively decreasing the ignition delay [23]. Mohand et al demonstrated this effect where a stoichiometric dual fuel engine reached a higher peak heat release in comparison to its diesel baseline engine [23]. It should be mentioned that the diesel pilot ignition delay is not significantly impacted by the stoichiometric natural gas mixture, but instead the transition between the pre-mix combustion phase and flame propagation phase is decreased and only effectively decreases the ignition delay when measured on an apparent heat release curve.

Stoichiometric operation was also shown to “fix” the late and slow combustion of lean natural gas engines and decreased the ignition significantly in comparison with lean burn natural gas engines [23]. Stoichiometric operation however can bring rise to new problems including pre-ignition, NOx increase, and BTE decrease of which will be discussed in following sections.

2.2.2.2 Emissions

Being one of the main motivations for the research and development of natural gas engines, many studies have been conducted on both lean and stoichiometric engines with varying combustion strategies. The impact on emissions that researchers have found with these combustion strategies are covered in the following section.

NOx

Diesel engines are commonly known to produce high engine out NOx. This detrimental byproduct emission is a major contribution in the development of natural gas engines to replace diesel engines. The major mechanism to NOx formation is temperature, where it forms above 1800 K and increases exponentially as temperatures increase [28 29].

In a typical diesel combustion process, the initial injection starts rich around $\Phi=4$ where PM is produced [9]. As the diesel combusts and transitions into the diffusion burn stage, the combustion reaches a stoichiometric condition, and the high temperatures form large amounts of NOx [30,31,32]. In dual fuel operation, the initial rich, diesel pilot injection still exists but only contributes to a small amount of the total fuel. The natural gas mixture thus dominates the emission formation where a lean natural gas mixture results in low temperatures and low NOx, and vice versa for stoichiometric operation.

The variables mentioned above were studied to gain insight to each variables impact on NOx formation.

The first, more obvious combustion parameter is AFR due to its strong relationship with combustion temperature where lean mixtures correlate with low combustion temperatures [1]. Studies conducted on lean and stoichiometric dual fuel engines confirmed this where NOx was found to decrease with increased lambda [24,27,33,34]. These experiments concluded that lean dual fuel operation decreased NOx values by up to 53% in comparison with diesel only operation. In a stoichiometric operated dual fuel engine, NOx values increased in comparison with their diesel baseline due to the increased combustion temperature in comparison with diesels globally lean mixture [35].

Natural gas contribution was another key variable in determining NOx emissions. Studies found that increased natural gas (decreased diesel pilot) resulted in decreased NOx emissions. This can be attributed to the increased ignition delay resulted from a smaller diesel pilot. The increased ignition delay as mentioned in the previous section causes late burn and thus lower combustion temperatures, again, reducing NOx formation.

Consistent results on the formation of NOx regarding engine speed in both diesel and dual fuel engines were found where increased speed resulted in decreased NOx formation. This was attributed to less time for the NOx to form in the cylinder at the higher RPM.

In summary, lean burn dual fuel engines reported a significant decrease in NOx emission in comparison to their diesel baseline where stoichiometric dual fuel increased from their diesel baseline. Within these studies, increased diesel injection quantity, decreased engine speed, and advanced diesel SOI were also contributors to an increase in NOx.

Although NOx levels are already low in lean burn NG engines, a SCR is typically utilized and capable of reducing NOx emission by 95% [2]. Stoichiometric dual fuel engines can utilize a less costly and lightweight TWC where NOx, CO, and HC conversion can reach 95 % [35].

PM

Particulate matter or soot formation is more complicated than NOx where chemical, kinetic, and thermal mechanisms have significant effect on its formation. Soot forms where temperatures are above 773 K, where fuel is partially oxidized. Temperatures below this threshold typically result in HC emission [1]. Soot formation also increases when oxygen availability is low, and when vapor particles are present to enhance particle coagulation [1]. In diesel engines, most of the soot is burned off as the combustion continues due to mixing and high combustion temperatures. Regardless of the soot burn off, diesel PM is still high in diesel engines and requires expensive aftertreatment systems [9]. An obvious advantage of PI and carburetor dual fuel engines is the homogenous charge of natural gas and air. All studies on dual fuel engines found that dual fuel operation decreased particulate matter in comparison to their diesel baseline attributed to the homogenous charge and the high H/C ratio [23,24,27,36]. Additionally, all dual fuel engine studies concluded that increased diesel injection quantity resulted in increased PM. This is attributed to two factors. One, the obvious increase in locally rich fuel from the diesel injection causing increased soot formation due to the mixing mechanism described as well as the additional carbon present from lower HC ratio diesel fuel. Two, the higher combustion temperatures brought on by higher heat release rate discussed in previous sections.

In a study conducted on the AFR of a dual fuel engine, the expected result of increased soot was correlated with a decrease in lambda due to the decreased access to oxygen.

In summary, natural gas combustion results in lower soot in comparison with diesel only operation. Within dual fuel operation, decreased diesel substitution and factors causing decreased heat release rate result in lower soot emission.

CO

CO is commonly known to be poisonous to humans. Formation of this emission results from combustion with insufficient access to oxygen and low temperature combustion [1]. This means that both rich and lean conditions can cause CO formation. CO is formed in lean operation due to the decreased combustion temperature, where CO is formed at temperatures below 1450 K [1]. Fuel rich conditions on the other hand result in a lack of oxygen availability causing incomplete combustion and thus, CO formation.

Literature reviewed on lean burn dual fuel engines found that CO emission increased significantly in comparison with their diesel baseline tests [34,37,38]. This result was attributed to the homogeneous natural gas mixture not fully combusting due to trapped fuel mass in the crevice volume and quenching due to the gaseous mixture contacting the cylinder walls [8].

One project found that as the fuel mixture became rich and approached lambda 1, CO decreased from the higher combustion temperatures found in fuel rich operation.

Within dual fuel operation, diesel pilot quantity and timing proved to have significant effect on CO emission. As discussed in the combustion section of this review, combustion efficiency in dual fuel operation is increased and therefore CO decreased, with increased diesel pilot quantity, advanced SOI, and stoichiometric mixtures [33,34,38,39].

CO_2

CO_2 emission formation is directly related to a high combustion efficiency where ideal combustion or 100% combustion efficiency results in only H_2O and CO_2 emissions. CO and CO_2 hold an inverse relationship due to this fact where CO_2 replaces CO when combustion efficiency is high. Due to the high hydrogen to carbon ratio of natural gas in comparison to diesel fuel, CO_2 levels are expected to decrease in dual fuel operation.

Within dual fuel operation, combustion parameters that increase combustion efficiency cause an increase in CO2 emission. Opposite to CO formation, increased diesel injection quantity, advanced timing and stoichiometric mixtures cause increased CO2 emission.

HC

Reviewed literature determined that a direct trade-off relationship between HC and NOx existed. HC emission is the result of unburned fuel in the exhaust and indicates low combustion efficiency. Overall, HC emissions in dual fuel engines were much higher than their diesel counter parts by up to 100 times. This result is attributed to two factors.

1. **Valve overlap:** Present to assist with exhaust blowdown, intake and exhaust valves are open for around 70 CAD near TDC. In normal diesel operation, this is not a problem as the direct injected fuel is not introduced until both valves are closed. In a dual fuel engine however, a small amount of unburnt natural gas is transferred between the higher-pressure intake and lower pressure exhaust valve during the valve overlap period. The unburnt fuel entering the exhaust is a direct contributor to HC increase in dual fuel engines. During NG-DP operation the pressure delta between intake and exhaust was around 20 kPa where this could be reduced by resizing the turbocharger to provide additional backpressure in the exhaust.
2. **Crevice Volume and Flame Quenching:** An issue arising again from the transition from DI to PI. The homogenous natural gas/air charge throughout the entire cylinder volume means that the crevice volume contains fuel which cannot combust. Additionally, the fuel present at the cylinder walls experience quenching from the relatively cool cylinder walls. The unburnt fuel quantity can be decreased by increasing engine load, decreased lambda, increased diesel injection quantity, and advanced SOI. These four parameters cause higher combustion temperatures as mentioned and thus decrease the amount of unburnt fuel from crevice volume and quenching.

Within dual fuel operation lean burn was found to result in high HC emissions which was attributed to the low combustion efficiency [8]. This could also be attributed to the crevice volumes created from the in-cylinder pressure transducers used in research where the homogenous natural gas mixture fills the volume and fails to ignite.

2.2.2.3 Performance

Efficiency, load capability, and knock of dual fuel engines were evaluated and summarized below.

Lean dual fuel engines compared to their diesel baseline were found to have lower BTE due to the lower combustion efficiency and decreased compression ratio (in some cases) found in lean burn NG engines, as discussed in previous sections [33,38,39,40,41]. One study found that when load was increased and AFR decreased, a BTE increase of 3% in comparison with its diesel baseline was achieved [40]. Another study examined the dual fuel performance in a stoichiometric engine. They found that the efficiency was slightly less than that of its diesel baseline [31].

Knock and pre-ignition can become an issue in dual fuel operation because of the early presence of fuel in the cylinder compared to normal diesel operation where fuel is not present until the desired start of injection. Natural gas has a high methane number of at least 70, making it fairly resilient to knock although many projects have run into knock when attempting to reach high load [42]. It is common practice to decrease the compression ratio of the engine to mitigate the knock experienced at high load [20,21,43]. In other studies, intake temperature and pressure were increased in attempt to decrease ignition delay and achieve higher combustion efficiency [44]. This was successful but resulted in knock when used at high loads [44]. Another dual fuel engine increased diesel pilot quantity and advanced the pilot timing to improve the BTE which again, resulted in knock at high loads [39].

The reviewed literature has shown that there is a clear inverse relationship between BTE and knock at high load. As knock cannot pass a certain threshold without destroying an engine, BTE is sacrificed instead. In attempt to maintain a high BTE while mitigating knock, several research projects have used WI which will be covered in the following section.

2.3 Water Injection

WI, like natural gas operation, is by no means a new concept and has been used since WWII where it was used to enable high loads during airplane takeoff. Dryer et. al published a paper on the mechanisms of WI in 1952 covering the chemical and kinetic effects of WI [5]. Increased interest in recent years has shown that water can successfully be used to control combustion and has been shown to decrease NOx, mitigate knock, and increase load capability with a small decrease in BTE. Several recent papers involving different operating types were reviewed and summarized below.

A project investigating the impacts of WI on a heavy duty, turbocharged, stoichiometric, SI, NG engine was reviewed [3]. WI was found to decrease the burning velocity of natural gas, decreasing peak in-cylinder pressure, peak heat release rate, and combustion temperature. The combustion was both slowed down and retarded by WI. The combustion changes described resulted in decreased NOx and mitigated knock, but resulted in increased THC and CO, indicating decreased combustion efficiency. This project found that WI enabled earlier spark timing and was able to increase BTE from 27.8% to 28.2% with a water to fuel ratio of 0.35:1 [20].

A study was conducted on the impact of indirect steam injection and CR on a dual fuel, liquefied petroleum gas and diesel fuel, single cylinder, naturally aspirated, engine [45]. Researchers found that steam injection increased BSFC from ~490 g/kWh to ~575 g/kWh at a CR of 18:1. HC and CO emissions were measured Selim et al and found that increased steam resulted in an increase of these emissions. Selim did not measure NOx but referenced and inferred a NOx decrease with WI [6].

WI can be useful in diesel only operation as well, reducing the typically high NOx and reducing engine knock. Tauzia et. al experimented on a 2.0L, turbocharged DI diesel engine with port intake water injectors to compare the ability of water and EGR to reduce NOx and modify combustion [7]. A water to fuel mass ratio of 0.6:1 was able to obtain a 50% reduction in NOx, showing to be slightly more effective than EGR in terms of NOx reduction. The benefit of using WI instead of EGR is clear when comparing the PM emission resulting from WI and EGR. The PM in EGR operation measured significantly higher than that of similar concentration water injection [7].

Note that there was an increased lambda in the WI trendline which contributed to the reduced PM where a direct, same lambda comparison was not available. WI also resulted in a small decrease in efficiency, starting at 35% without water and dropping to 32% with a dilution ratio of 13% [7]. Similar to the previous projects covered, HC and CO were increased as water was increased.

Several examples of WI's ability to decrease NOx while maintaining high efficiency and low PM have been shown. Decreased combustion efficiency, indicated by the increased CO and HC emission, is the main contributor to the decreased in BTE. There is a clear thermal mechanism at work when water is injected as its high specific heat capacity allows significant heat transfer to the water without a significant increase in temperature. Another mechanism is present however when liquid water is vaporized, consuming a large amount of heat energy but expanding significantly in the process. This can result in increased cylinder pressures and increase BTE if injection timings are tuned [4]. Roberts et al studied the effect of water on soot formation and found that water is capable of decreasing soot levels, which was attributed to a chemical mechanism [46]. The chemical mechanism happens when injected water is present in the flame zone and is dissociated into O, H, H_2, and OH, increasing the oxygen and hydrogen available for combustion, therefore reducing soot formation. Roberts also found that the thermal impact of water was responsible for the increase in PM where decreased combustion temperature resulted in increased soot. The chemical mechanism described outweighed the thermal mechanism, resulting in a net decrease in soot. This however is not always the case as shown in the outlined experimental engine tests.

WI in both dual fuel and diesel only engines has shown to be beneficial in reducing NOx emission while maintaining high performance.

2.4 Summary

The goals of this project are to experimentally determine the impacts of WI on the combustion, performance, and emissions of a DP-NG engine and provide comparison to its diesel baseline as well as other comparable engines in industry. From the reviewed literature, this thesis focuses on a stoichiometric DP-NG engine with 15% diesel contribution by energy, utilizing a NG mixer and PI WI.

3 Experimental Setup & Test Plan

Prior to conducting experiments, both the test cell and the test engine, a Cummins 6.7L ISB, required many modifications and instrumentation. Specifications for the modifications and instrumentation installed are included in this chapter to first give an overview of the system and also allow for the possibility of repeating the experiments conducted. Additionally, the test plan motivated by the objectives outlined in the Introduction of this paper is included in this chapter. Finally, the process used in calculating parameters and analyzing data is outlined.

3.1 Instrumentation

This section provides a detailed overview of the instrumentation, modifications, and specifications of both the test cell and engine used in this project. A summary of the fuels used in this project is also given. Figure 2 shows the fully instrumented engine in the test cell.

Figure 2. Engine and Dyno In Test Cell

3.1.1 Test Cell Instrumentation

3.1.1.1 AVL Dynamometer

A 445 hp AVL308 AC dynamometer was used to measure the torque output of the engine. The dynamometer was controlled by setting the desired speed which was then maintained regardless of the load input by the engine.

3.1.1.2 Natural Gas Flow Measurement Cabinet

To operate the natural gas-powered engine, the test cell was first prepared to measure the supplied natural gas. Shutoff valves, plumbing, and flowmeters were enclosed in a cabinet to isolate and detect any NG leaks. The cabinet is shown in figure 3 with labeled components followed by an explanation of their specifics and functions.

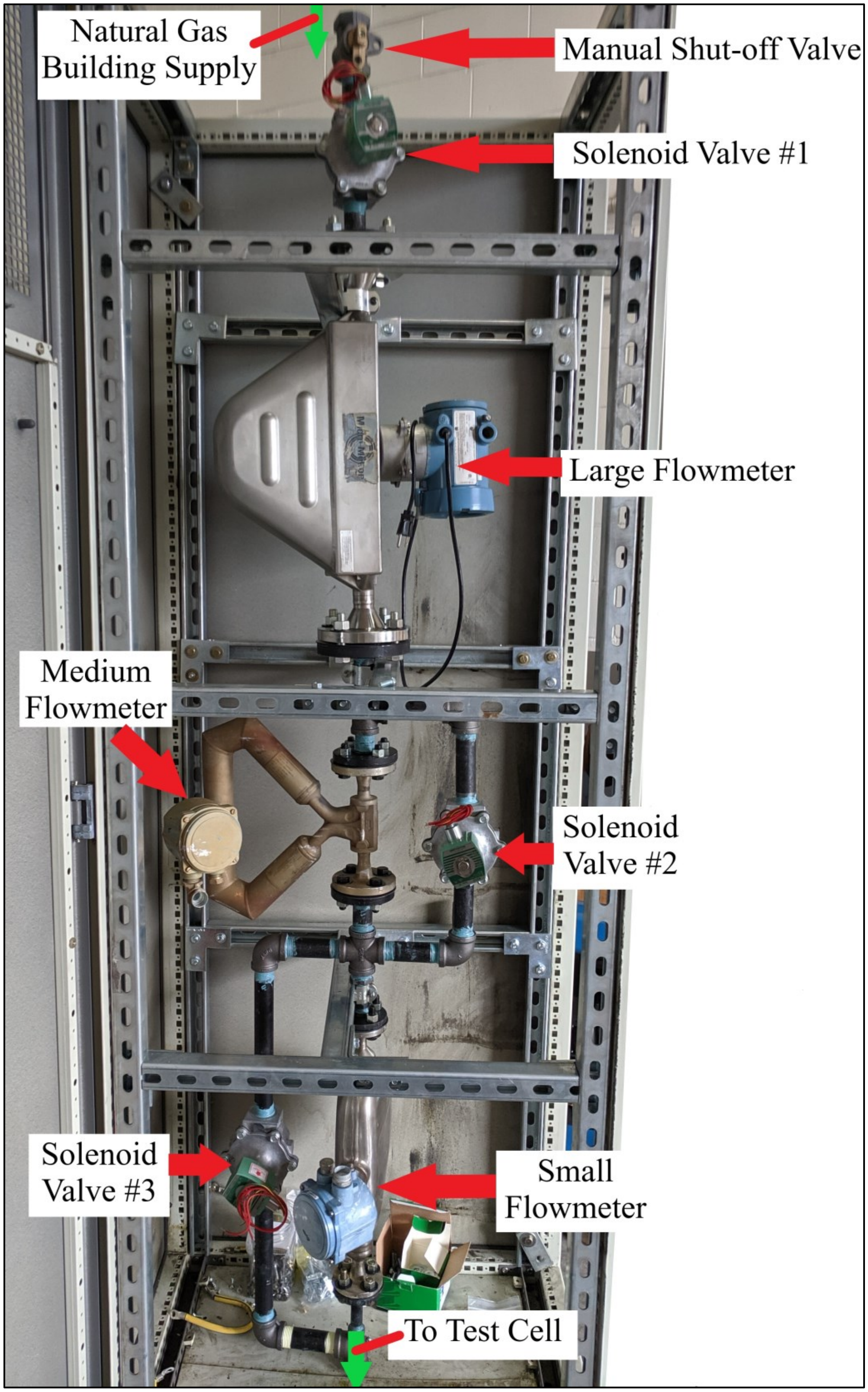


Figure 3. Natural Gas Flow Cabinet

NG Flowmeters:

The low, 2psig, NG supply pressure and high flowrate needed to reach high loads required three separate Coriolis meters to be utilized at different flow rates in attaining accurate measurements. These flowmeters are summarized in table 2.

Table 2 Natural Gas Cabinet: Flowmeter Specifications

Flowmeter	**# 1**	**# 2**	**# 3**
Flowmeter Model #	F100S209CCAAE	CMF050MA14N2BAE	CMF025M335NQBAE
Transmitter Model #	2700I12BBAEZZZ	2700R12BBAEZZZ	2700R12BBAEZZZ
Max Flowrate $\left[\frac{g}{s}\right]$	35.7	10.7	3.8
Nominal Flowrate $\left[\frac{g}{s}\right]$	23.7	7.4	2.5
Accuracy [%]	± 0.35 • flowrate	± 0.35 • flowrate	± 0.5 • flowrate
Repeatability [%]	0.175 • flowrate	0.175 • flowrate	± 0.25 • flowrate

Solenoid Valves:

Three 2-way, normally closed, 300 SS, 1” NPT,24V, red-hat solenoid valves were installed and wired to the NI Veristand DAQ to allow control from the test cell computer. The three valves were used as follows:

Table 3 Natural Gas Cabinet: Solenoid Valve Functions

Valve	**Closed**	**Open**
1	NG Supply to cabinet OFF	NG supply to cabinet ON
2	Direct flow through flowmeter #2 for medium flow operation	Open bypass #1 for high flow operation
3	Direct flow through flowmeter #3 for low flow operation	Open bypass #2 for medium and high flow operation

<u>Natural Gas Alarm (not shown in figure):</u>

A Macurco GD-2B combustible gas detector was installed at the top of the cabinet to detect any leaks present in the natural gas cabinet. This alarm is set to detect flammable vapors over 25% lower explosive limit (LEL), conforming to standard UL1484.

<u>Manual Shutoff Valve:</u>

A full port, 1" ball valve was installed to manually shutoff the natural gas supply to the cabinet when not in use.

3.1.1.3 Water Supply Control

A reverse osmosis filter, water pump, and water line were already existing in the test cell prior to this project, however the pump duty cycle controller required installation. The engine controller was used to send a digital PWM signal to a relay that switched the pump power supply on and off. A PID controller was implemented in the controller logic to allow the operator to set the desired water pressure. A schematic of the water system is outlined in figure 4.

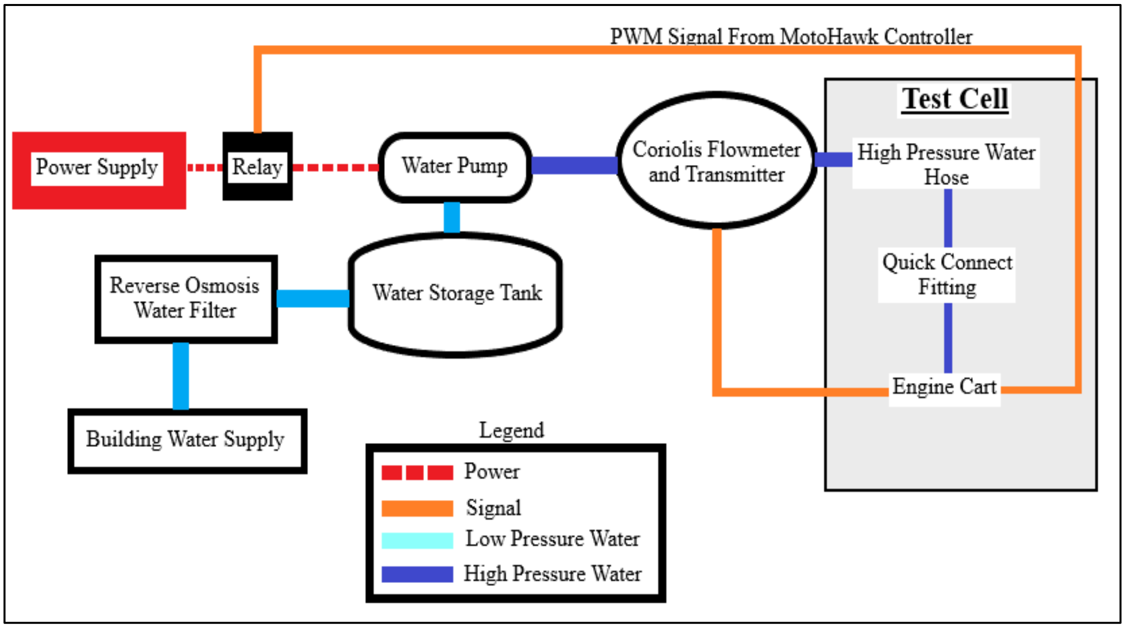
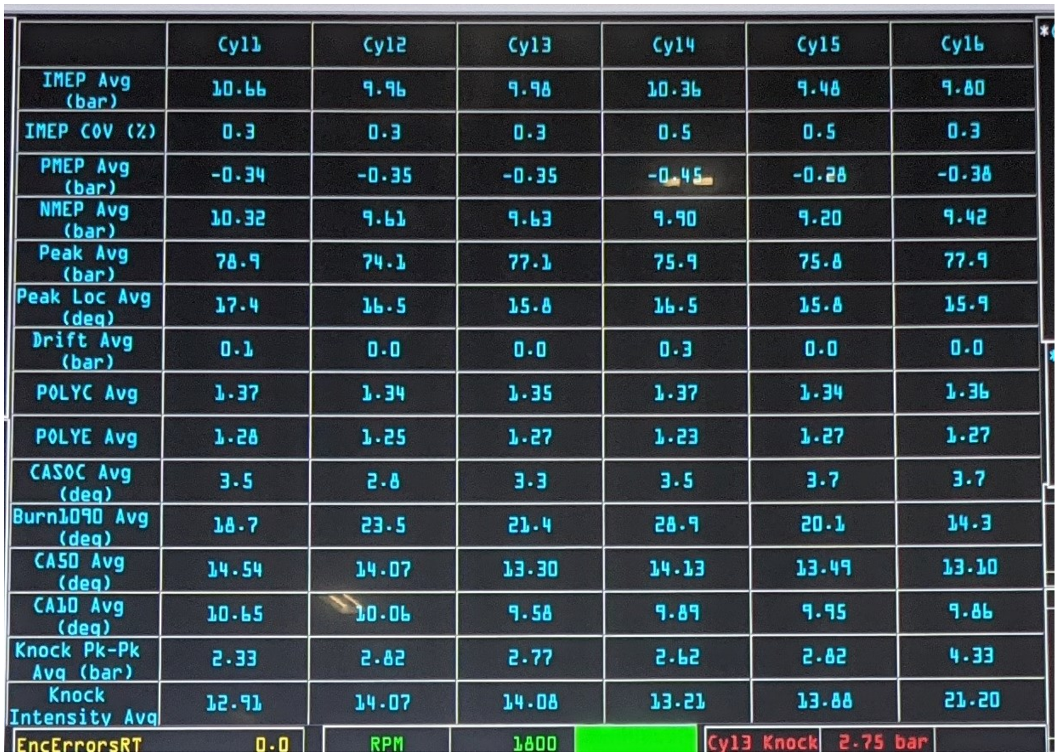


Figure 4. Water Supply System

Outlined in the schematic above, water supplied from the building is first filtered through reverse osmosis and stored inside a tank. The water pump then creates pressure in the high-pressure water system containing a micromotion Coriolis flowmeter to measure the water flow rate. The high-pressure line runs from the water cart to the test cell where a hose is used to connect it to the engine cart containing a common rail injection system.

3.1.1.4 Combustion Analysis Software

A Redline 2, Advanced Dynamometer and Powertrain-Combustion Analysis System block and software was used to collect and monitor a real-time pressure trace using the in-cylinder transducers, an optical encoder, and MAP pressure. The CAS then calculated and reported the desired combustion metrics shown in figures 5, 6, and 7.



	Cyl1	Cyl2	Cyl3	Cyl4	Cyl5	Cyl6
IMEP Avg (bar)	10.66	9.96	9.98	10.36	9.48	9.80
IMEP COV (%)	0.3	0.3	0.3	0.5	0.5	0.3
PMEP Avg (bar)	-0.34	-0.35	-0.35	-0.45	-0.28	-0.38
NMEP Avg (bar)	10.32	9.61	9.63	9.90	9.20	9.42
Peak Avg (bar)	78.9	74.1	77.1	75.9	75.8	77.9
Peak Loc Avg (deg)	17.4	16.5	15.8	16.5	15.8	15.9
Drift Avg (bar)	0.1	0.0	0.0	0.3	0.0	0.0
POLYC Avg	1.37	1.34	1.35	1.37	1.34	1.36
POLYE Avg	1.28	1.25	1.27	1.23	1.27	1.27
CASOC Avg (deg)	3.5	2.8	3.3	3.5	3.7	3.7
Burn1090 Avg (deg)	18.7	23.5	21.4	28.9	20.1	14.3
CA50 Avg (deg)	14.54	14.07	13.30	14.13	13.49	13.10
CA10 Avg (deg)	10.65	10.06	9.58	9.89	9.95	9.86
Knock Pk-Pk Avg (bar)	2.33	2.82	2.77	2.62	2.82	4.33
Knock Intensity Avg	12.91	14.07	14.08	13.21	13.88	21.20

Figure 5. CAS Display - Realtime Calculated Parameters

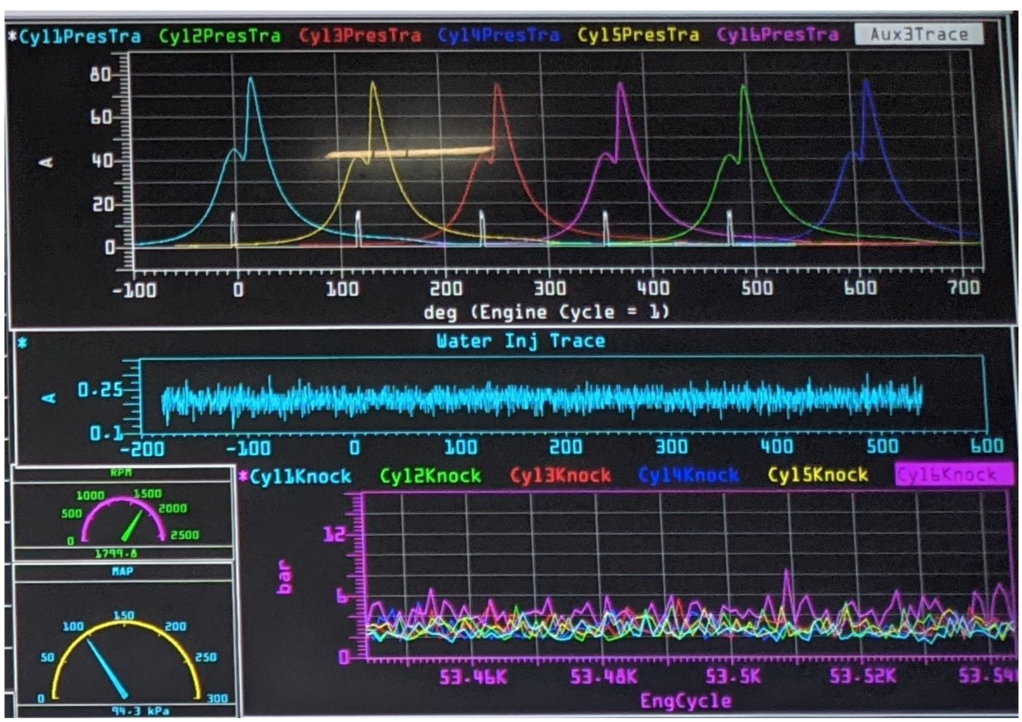


Figure 6. CAS Display - Realtime Pressure, Knock, WI, RPM, MAP

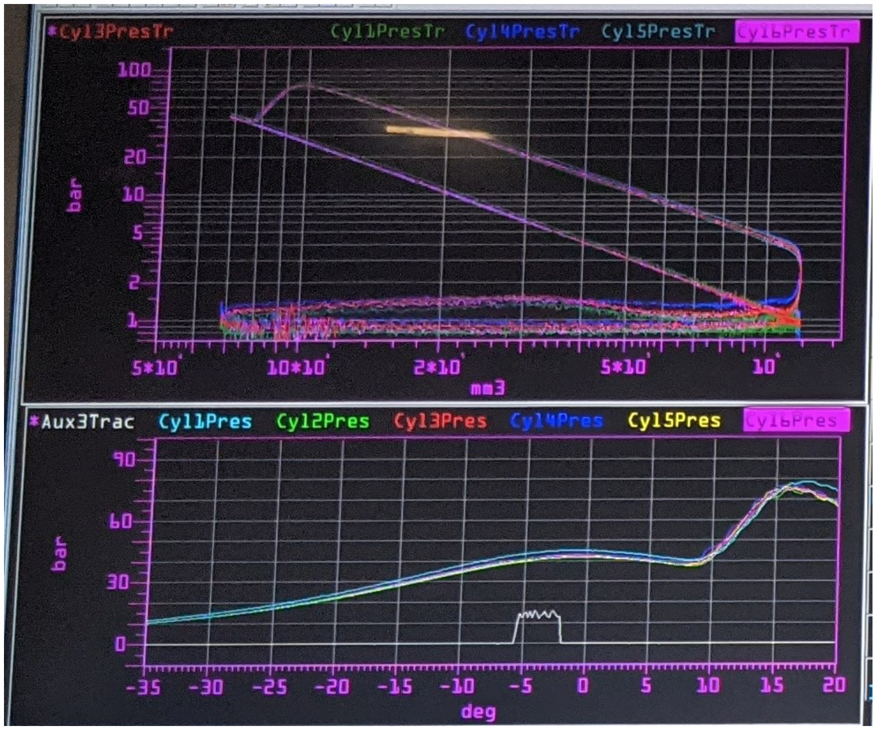


Figure 7. CAS Display - Realtime Pressure Trace and LOG PV Diagram

3.1.1.5 National Instruments DAQ

A National Instruments DAQ was used in conjunction with NI Veristand Software to monitor and control the test cell instrumentation as well as all engine thermocouples and piezo-resistive pressure transducers. This system is capable of sending and receiving both digital and analog signals making it useful for powering many test-cell and engine components. The NI Veristand user interface used for controlling the NI DAQ is shown in figure 8.

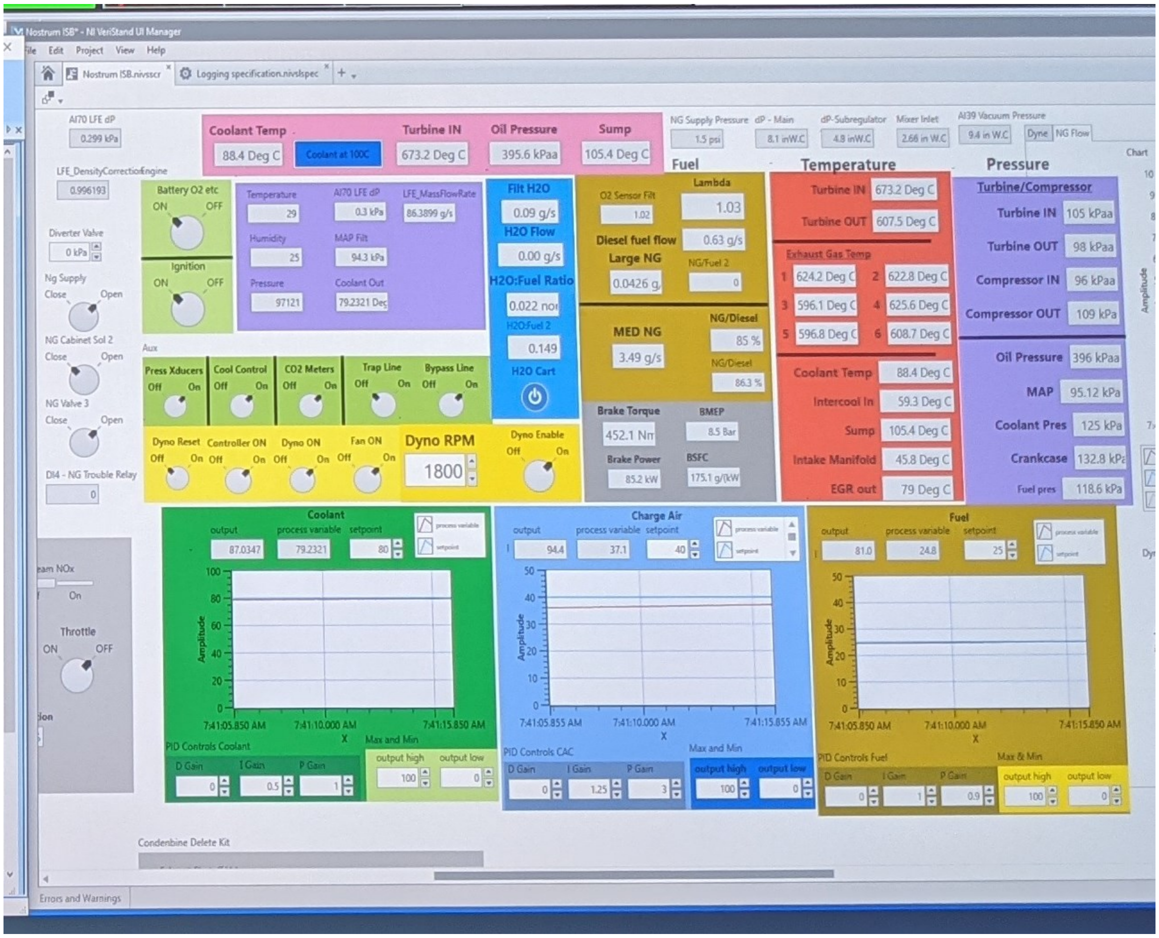


Figure 8. Veristand User Interface

The instrumentation controlled and monitored using the NI Veristand user interface are summarized in Appendix A of this thesis.

3.1.1.6 Gaslab CO_2 Sensor

A K33 ICB 10% CO_2 Sensor was installed on the engine and provided a serial output which was monitored and recorded using the software, GasLab. CO_2 content in the intake manifold was monitored using this sensor and used to calculate EGR percentage using equation 1.

Equation 1: $$\text{EGR \%} = \textit{Intake } CO_2\ \% / \textit{Exhaust } CO_2\ \%$$

3.1.1.7 AVL Smoke meter

An AVL, Filter Type, Smoke meter was used to measure the particulate matter concentration of the engine out exhaust. The Smoke meter operates by drawing a selected volume of exhaust sample through a piece of filter paper before measuring the resulting blackening of the sample. This does not measure particle size but gives an accurate reading of soot concentration (mg/m3) and filter smoke number (FSN) in the exhaust.

3.1.1.8 Horiba 5 Gas Analyzer

The engine out emissions were monitored and recorded using a 5-gas Horiba Motor Exhaust Gas Analyzer 1600D. Exhaust sample was collected and passed through a heated filter before running through a heated sample line to the machine, located outside of the engine test cell. The analyzer is capable of measuring CO_2, CO, THC, NO_x, and O_2 which were calibrated and spanned prior to each day of testing. The calibration gases used in this project and the machine's emission analysis method for each emission are outline in table 4.

Table 4. Span Gas Concentrations

Span Gas	Span Gas Concentration
CO_2	16%
CO	1%
THC	9900 ppm and 1600 ppm
NO_x	1609 ppm NO, 1005 ppm NO_2, and 1625 NO
O_2	4%

3.1.2 Engine Instrumentation

The baseline engine used in this project is a Cummins, 6.7L ISB that was modified and instrumented to operate as a diesel-pilot, NG engine with PI WI. Engine specifications are outlined in table 5.

Table 5. Test Engine Specifications

Parameter	Value	Units
Model	Cummins ISB6.7 CM2150	-
Cylinders	6	#
Bore & Stroke	107 x 124	mm
Connecting Rod Length	192	mm
Displacement	6.7	L
Compression Ratio	17.3 : 1	-
Aspiration	Turbocharged VGT + Charge Air Cooler + Throttle Valve	-
Diesel Pilot Injection System	Stock Bosch Injector - 8 holes	-
Natural Gas Fueling System	IMPCO 400VF3 Mixer Upstream of Compressor	-
Water Injection System	Nostrum 500cc Kinetic Water Injectors	-
Rated Power	191 (260)	kW (HP)
Rated Speed	2400	RPM
Peak Torque	895 (660)	Nm (ft·lb)
Peak Torque Speed	1600	RPM
Peak BMEP	16.8	bar
Minimum Throttling Pressure	70	kPa
Peak Cylinder Pressure	<150	bar

In preparing the engine for experimentation, the engine was placed on an engine cart and the following modifications and instrumentation were installed on the engine.

3.1.2.1 MotoHawk ECU

Complete control and monitoring of all engine controls, measurements, and modifications required the OEM ECU to be replaced with the ECU-5554-112-0902-C/F, MotoHawk ECU. A custom engine code was written and flashed on the MotoHawk ECU which was then operated using the MotoTune control interface.

The MotoTune interface allowed the operator to override engine actuators including pilot injection timing and quantity, diesel rail pressure, VGT position, WI quantity and timing, water rail pressure, NG vacuum solenoid opening, EGR valve position, and throttle position.

3.1.2.2 Water Injection System

The engine had previously been equipped with a custom billet aluminum intake manifold manufactured to mount six Nostrum Kinetic water injectors as shown in figure 9.

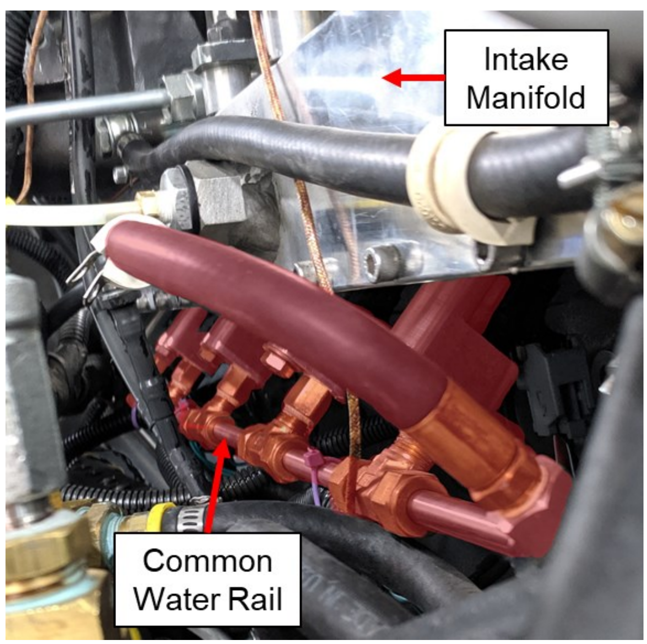


Figure 9. Water Injector Mounting

Each injector was pointed at the intake valve for its corresponding cylinder. The injectors were fed by a common rail which was fed by the cell water supply. Injector timing and duration were controlled by the MotoHawk ECU coded with custom MotoTune control logic.

3.1.2.3 Intake Air Cooler

A constant intake air temperature is necessary to achieve consistent and comparable results. An intake air cooler and controller were installed post-compressor. The intake air temperature was controlled using an electronically actuated gate valve and PID controller where the target temperature was set in NI Veristand which modulated the valve position to reach the desired intake air temperature.

3.1.2.4 Throttle

The natural gas mixer intake system installed on the engine required the installation of an air throttle. The throttle was installed downstream of the compressor and intake air charge cooler and was controlled via PWM input sent from the MotoHawk ECU.

3.1.2.5 EGR Valve Actuator

The stock EGR control valve actuator is difficult to control with the technology that was available. Instead, a linear actuator was used to adjust the position of the stock EGR valve. The actuator was controlled by a Pololu PWM control module which was commanded by the MotoHawk controller.

3.1.2.6 Natural Gas Intake System

The low-pressure natural gas supply used in this project required the use of a natural gas mixer as shown in figure 10.

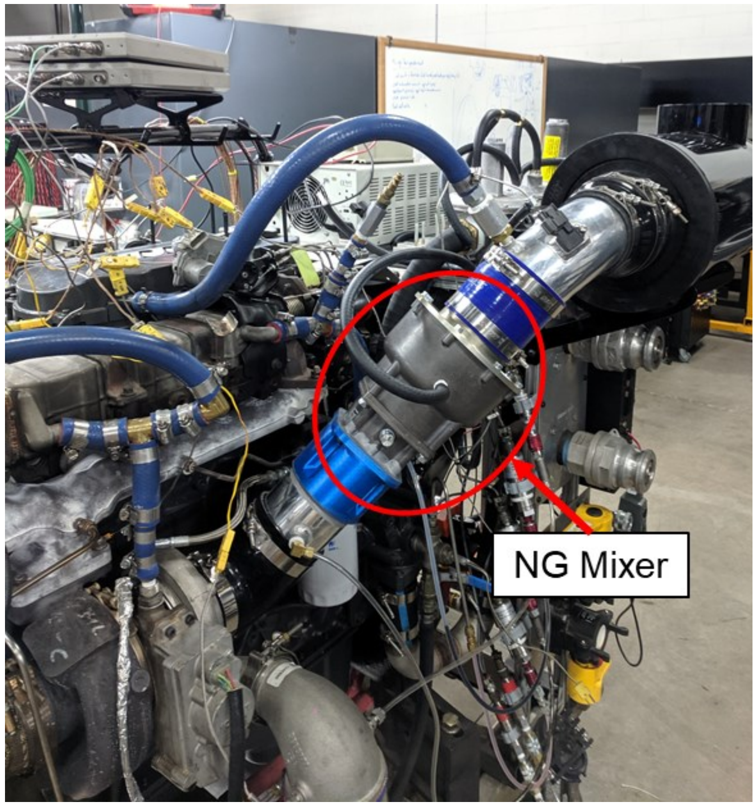


Figure 10. Natural Gas Mixer

The natural gas mixer, installed pre-compressor, is an IMPCO-400VF with AA2-69 air horn, AB1-66-3X downdraft air valve body. Two mixer valves were purchased for stoichiometric and lean operating modes. The NG mixer requires a NG intake pressure of 5 in w.c (1.2 kPa) and requires a main and sub-regulator to maintain consistent input pressure. The main regulator, a Maxitrol 325-9L, decreased the 1.5-2.0 psi NG supply pressure from the natural gas flow measurement cabinet down to 11 in w.c. The sub-regulator, a Maxitrol RV81, then decreased the pressure to the NG mixer's required 5 in w.c.

Real-time AFR control was required to maintain stoichiometric operation during experiments in this project where the NG mixer contained three methods for AFR control. The mixing valve, power screw, and NG intake pressure. The mixing valve must be installed prior to testing and the power screw was specified by IMPCO to only impact

AFR at high load. This left the NG intake pressure to be controlled which was done with the use of the two regulators and the vacuum system shown in figure 11.

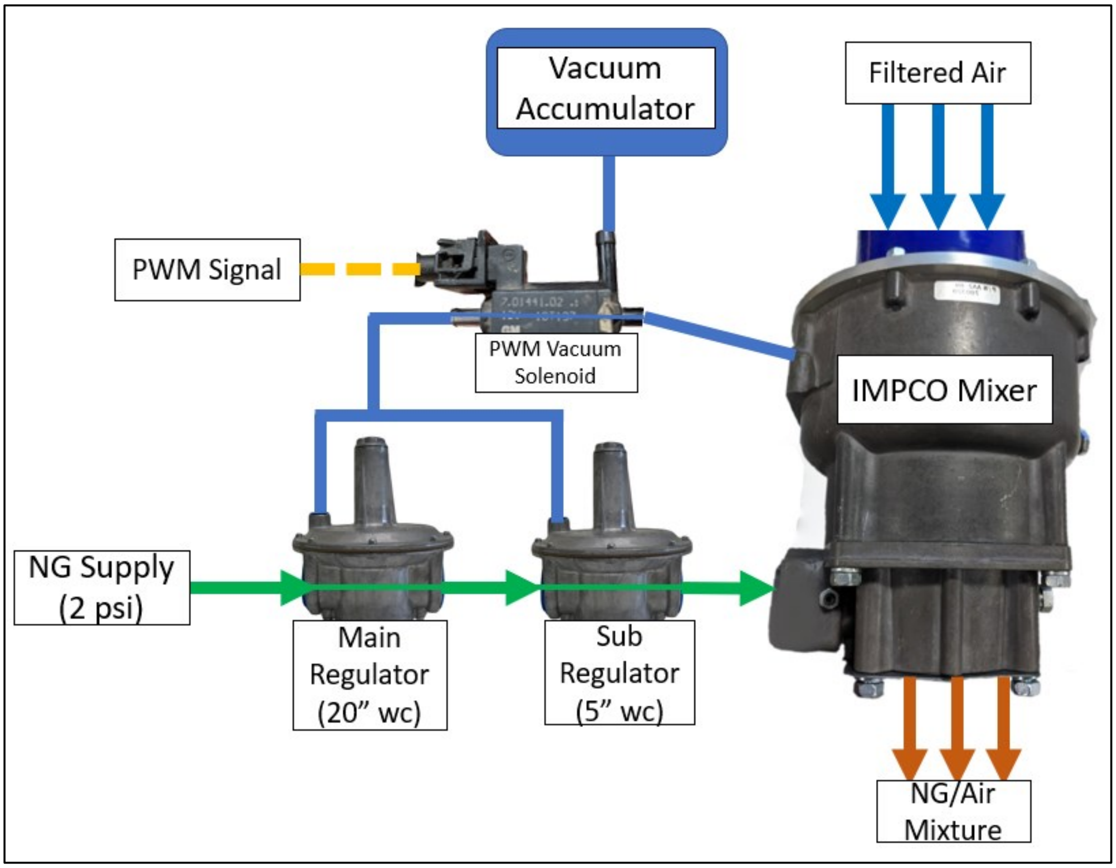


Figure 11. Natural Gas AFR Control

The vacuum system consisted of a vacuum pump, vacuum chamber, relief valve, 3-way PWM controlled solenoid valve, and vacuum lines connecting to the NG mixer inlet, vacuum chamber, and the atmospheric reference for both regulators. To control AFR, a constant vacuum was maintained in the vacuum chamber, where a constant leak was introduced with a needle valve to keep the pump at a constant duty cycle. At 100% duty cycle, the vacuum solenoid opened the regulator atmospheric references to the vacuum chamber, closing the regulators and shutting off natural gas flow. As the solenoid duty cycle was decreased the regulators increasingly referenced the NG mixer inlet, allowing more natural gas to flow, decreasing the AFR. In experiments where a leaner AFR was required, the power screw was adjusted, or a new NG valve was installed. This system allowed for simple AFR control, robust NG operation at low supply pressure, and sufficiently mixed air and fuel. Additional specifications on the regulators and NG mixer can be found in Appendix B.

3.1.2.7 In-Cylinder Pressure Transducers

To obtain an in-cylinder pressure trace, Kistler 6125A-C, piezo-electric transducers were installed at the top of each cylinder. The SN, model, and sensitivity of each transducer are summarized in table 6.

Table 6. In-Cylinder Pressure Transducer Specifications

Cylinder #	Brand and Model	SN	Sensitivity
1	Kistler 6125 CU20	3053773	30.92
2	Kistler 6125 B	1615110	15.90
3	Kistler 6125 B	1611793	15.74
4	Kistler 6125 B	1615109	15.01
5	Kistler 6125 A	645193	15.85
6	Kistler 6125 A	560292	19.90

Each transducer was installed inside a mounting sleeve requiring the cylinder head to be machined. The transducer mounting sleeve passed through the coolant passage in the cylinder head, above the exhaust for each cylinder, and was sealed with heat resistant sealant to ensure no coolant escaped into the cylinder. Additional information regarding the transducers and the mounting of the transducer sleeves is included in Appendix C of this thesis.

3.1.3 Fuel Properties

A summer blend, #2 ultra-low sulfur diesel (ULSD #2) was used for all experiments along with a diesel fuel detergent additive to prevent injector coking at low injection quantities. The natural gas was sourced from the Advanced Power System Laboratory's municipal supply from Semco Energy. Properties for the diesel, diesel additive, and natural gas are outlined in table 7 where additional information can be found in Appendix D of this thesis.

Table 7. Fuel Properties

Diesel (ULSD #2)	**Density (kg/m^3) @ 15°C, 1 atm**	**Lower Heating Value (MJ/kg)**	**AFR_s**	**H/C**	**Cetane Number**
	849	45.8	14.6	1.85	46-48
NG	**Density (kg/m^3) @ 20 °C, 1 atm**	**Lower Heating Value (MJ/kg)**	**AFR_s**	**H/C**	**Methane Number**
	0.727	47.5	16.30	3.80	83.0

The fuel additive used is made by Lubrizol and is the POWERZOL 9048 diesel fuel additive. The additive was mixed with the diesel fuel at a mixture ratio of 8mL of additive per 10 gallons of diesel fuel, as specified by the manufacturer.

3.2 Experimental Test Plan

This section first provides the specific test procedures that were followed to obtain consistent and accurate experimental data. Next, the experimental test plan is laid out, followed by the calculations and data analysis methods used in processing the collected data.

3.2.1 Test Conditions

Control variables were defined to obtain reliable and comparable experimental data and are outlined in table 8.

Table 8. Experimental Control Variables

Control Variable	Set Point	Unit
Intake Air Temperature	40	°C
Coolant Temperature	80	°C
Low Pressure Fuel Temperature	25	°C

After defining test constants, procedures were developed to maintain engine health and consistent testing methods.

3.2.1.1 Engine Warmup Procedure

Proper engine warmup is essential in maintain engine health and ensuring all engine conditions were stabilized before testing started.

Prior to starting, all engine sensors were checked to ensure they were reporting correctly. Next the engine was motored with the dynamometer at 750 RPM and the CAS monitoring was started, and pressure traces monitored. A small pilot and main diesel injection were then commanded and tuned to produce a load of approximately 100 Nm. The dynamometer speed was then adjusted to 1200 RPM and engine load adjusted to approximately 250 Nm, adjusting diesel injection timing and durations appropriately. Engine temperatures and pressures as well as calculated polytropic coefficients of expansion and compression and knock were monitored at this time to ensure safe and normal operation. Once the coolant temperature reached approximately 50°C, the dynamometer speed was gradually moved to 1800 RPM while injection timing and durations were adjusted to maintain load and operate without knock. Injection timing and durations, diesel rail pressure, and VGT position were then adjusted to reach a load of approximately 350 Nm. The engine was operated at this condition until the engine coolant reached 80°C and engine oil temperature reached approximately 90°C. Once these conditions were reached the engine was sufficiently warmed up and ready to begin testing.

3.2.1.2 Control Test Procedure

Prior to conducting tests, a control point was run in diesel only mode, motoring, and diesel pilot-NG mode. For the diesel and diesel pilot-NG control tests, control parameters were set and steady state conditions were reached before recording on all DAQ systems. After conducting the diesel only control test, fueling was shutoff and motoring data was collected with all DAQ systems except emissions. The control tests were observed first in real-time to ensure the engine load and polytropic matched previous control point. After each testing session, the collected control data was analyzed and documented to ensure engine health was maintained over time.

3.2.1.3 Experimental Test Procedure

Prior to each testing session, a test plan was first made and discussed to ensure all testing objectives were met. When sweeping a variable such as WI, the tests were randomized to negate any bias or hysteresis present. Additionally, tests were generally repeated 3 times to ensure no outliers existed within the collected data. If time was an issue, 2 repeats were conducted and analyzed to ensure no outliers existed within the collected data.

After warming up the engine and running the control point, the target test condition was then achieved by first matching the target engine speed, followed by engine load, and finally the remaining variables were met. Once the test condition was met, the engine was run at steady state for at least one minute while engine load, target parameters, and engine temperatures were observed. When all target conditions and temperatures were stabilized, all DAQ systems were set to record for 2-3 minutes. If all parameters remained stable while recording, the test point was deemed successful and the next test point was targeted. Otherwise, the test point was marked as invalid and repeated.

3.2.1.4 Engine Shutdown Procedure

Upon completing the planned experimental tests, the control points were ran and recorded before starting the engine shutdown procedure.

Engine load was first reduced to approximately 100 Nm by reducing fueling while engine speed was adjusted to 1200 RPM. This condition was ran for several minutes to ensure engine components such as the turbocharger have sufficient time to cool off. The fuel was then shutoff and the engine motored at 750 RPM for another minute before shutting off the dynamometer. With the engine and dynamometer off, all recorded data was then compiled and analyzed.

3.2.2 Experimental Test Plan

The goal of this project was to experimentally investigate the impact of WI on the combustion, performance, and emissions for both diesel-only operation and NG-DP operation. Power generation is the target use for this study therefore speeds and loads were chosen based off common genset input from their engine. The 12.5 bar BMEP, 1800 RPM load point represents 100% load for medium and heavy duty gensets. Several stages of experimental tests were conducted to achieve this goal and are summarized in table 9 followed by a detailed description for each test section.

Table 9. Experimental Test Plan Overview

Operating Mode	Engine Speed [RPM]	Engine Load [BMEP]	Independent Variables Swept
Diesel Baseline	1200	12.5	-
	1800	4.2	-
	1800	8.4	-
	1800	12.5	-
	1800	16.8	-
Diesel + Water	1200	8.4	WFR
	1800	4.2	WFR
	1800	12.5	WFR, SOI
Dual Fuel	1200	< 4.0	Diesel RP, Lambda, SOI
	1800	8.4	DSR, SOI, Diesel RP, Lambda, EGR
	1800	12.5	SOI, EGR, DSR, Lambda
	1800	16.8	Pre-Ignition
Dual Fuel + Water	1800	8.4	SOI, WFR, EGR
	1800	12.5	SOI, WFR, EGR
	1800	16.8	SOI, WFR

Upon reviewing the goals and objectives, to obtain data suitable for comparison, it is necessary to match speed and load between operating mode and to change as few variables as possible. Also, as this is a genset application, the engine should be operated in steady state conditions and data recorded only after all variables have stabilized. Specific operating procedures for each operating mode are outlined below.

3.2.2.1 Diesel-Only

Experimentation in this mode was conducted first, to obtain OEM operating parameters (fueling, MAP) and second, to obtain baseline diesel results for comparison to the other operating modes. The OEM operating controls were obtained by using the OEM engine controller, where a simulated throttle controller was used to change load while speed was held constant with the dynamometer. All engine instrumentation was used however injection timing and duration, MAP, diesel fuel flow, and diesel RP the focus of these tests as they were necessary to operate OEM conditions with the open MotoHawk ECU.

After obtaining the OEM operating parameters, the MotoHawk ECU and injector driver was installed and the same tests, outlined in table X, were conducted, and compared to the OEM baseline, ensuring the engine could be controlled sufficient to match the OEM operating conditions.

3.2.2.2 Diesel with Water Injection

With the baseline experimentation complete, experimentation on the impact of WI on diesel operation could be conducted. Three speed/load points from the baseline data were chosen to conduct testing as shown in table 9. During experimentation, the baseline condition was first matched before injecting water. Once the condition was matched and engine conditions stabilized, water was injected at five different WFR spanning from 0 to 2.5 in increments of 0.5. WI quantity was the only variable changed in these tests and all other controls were left unchanged by the operator. This was done to study the impact of WI without any live user tuning. Specifically, CA50 was retarded as WFR was increased and could have been combatted by advancing SOI. This was not done however as the goal of this section was to study the impact of WI on diesel operation alone without modifying other control parameters.

As mentioned, WFR was swept from 0 to 2.5 in increments of 0.5. A WI strategy was developed and discussed to allow for a consistent comparison between WFR as well as minimizing the chance for water to become trapped in the intake manifold. For all tests WI pressure was maintained at 500 kPa above MAP. The end of injection (EOI) for each water injector was set at 25 CAD before intake valve close (IVC) for each injectors corresponding cylinder. Controller logic was implemented allowing the operator to increase WI duration, the ECU would then calculate the water SOI change required to maintain EOI at 25 CAD before IVC. After any WI was conducted, the engine was run at high load for several minutes to evaporate and expel any excess water. Additionally, the oil was checked regularly, and a periodic oil change were conducted to ensure water was not accumulating in the oil. Throughout the duration of the project, no evidence of water in the crankcase was found, however trace amounts of "foamy" oil were found in the oil filter and in the valve cover. Pictures are provided in the results section of this thesis. The WI strategy outlined above was utilized for all WI experimentation as optimization of WI strategy was not within the scope of this project.

3.2.2.3 Natural Gas – Diesel Pilot

The NG system instrumentation was installed in order to begin conducting the NG-DP experimentation. This operating mode was performed to provide comparison to the diesel baseline conditions, verify past research results, and obtain NG-DP baseline for the water injection testing. Baseline NG-DP conditions are outlined in table 10.

Table 10. NG-DP Baseline Parameters

Parameter	Setpoint
DSR	85%
Diesel RP	1000 bar
Lambda	1.00
EGR	0%
Combustion Phasing	MBT

In maintain these parameters, engine operation was conducted as follows. The engine was first operated with one diesel injection before opening the NG supply. With the VGT set at 100% open, the throttle was used to restrict NG/air flow, and therefore control load. To reach an experimental test point, the diesel RP, EGR, and speed were controlled to their setpoint. Next, the throttle was used to reach the test load while DSR and lambda were maintained by controlling diesel injection duration and the NG vacuum controller, respectively. Finally, MBT was achieved by adjusting SOI. Once all parameter setpoints were achieved and engine conditions stabilized, data was collected.

After baseline conditions were established and data was collected at each speed/load point, the parameters outlined in the table were swept to gain insight to their impact on NG-DP combustion.

3.2.2.4 Natural Gas – Diesel Pilot with Water Injection

This final operating mode was conducted to study the impact of WI on NG-DP operation and compare it to diesel with WI. Maximizing BTE and minimizing emissions was the goal of this operating stage.

In this operating mode, different from the diesel WI mode, combustion phasing (CA50) from the NG-DP baseline was maintained by advancing SOI while WFR was increased. After comparing WFR to the baseline NG conditions, EGR and SOI were swept with WI to provide additional insight to the mechanisms behind our results.

3.2.3 Calculations & Data Analysis

Before continuing, operating a NG-DP engine with WI, requires two variables to be defined, water to fuel ratio (WFR) and diesel substitution ratio (DSR).

Water to fuel ratio is the ratio of water mass injected to total fuel mass and is defined in equation 2.

Equation 2: $$\boldsymbol{WFR = \dot{m}_{water}/(\dot{m}_{NG} + \dot{m}_{Diesel})}$$

Diesel substitution ratio is the ratio of natural gas energy to total fuel energy and is defined in equation 3.

Equation 3: $\boldsymbol{DSR = (\dot{m}_{NG} \times LHV_{NG}) / [(\dot{m}_{NG} \times LHV_{NG}) + (\dot{m}_{diesel} \times LHV_{diesel})]}$

Data analysis was conducted using MATLAB where all collected data was first complied into an excel spreadsheet before combustion metrics were re-calculated and plots were produced. In this project, CAS recorded cylinder pressure every 1/3 of a CAD for 300 engine cycles. Combustion figures in this paper display the average of these 300 cycles and all points in scatter plot figures use the average of the test repeats conducted.

Ignition delay in natural gas operation is calculated as the time between the hydraulic SOI (electronic SOI + 0.3 ms) and 5% MFB [49]. The hydraulic SOI is the time between the sending of the electronic open signal to the time the injector actual opens due to hydraulic lag. In this thesis, the SOI refers to the electronic SOI unless otherwise stated. Note that at 1800 rpm, the 0.3 ms delay results in a 3 degree delay in injection timing. The 5% MFB was determined by reviewing MFB and AHR curves and choosing a MFB suitable to measure SOC.

In section 4.2, NOx emissions for some of the tests were determined to have an offset. This was due to a low concentration span gas, 1005 ppm NO_2, being used after the 1625 ppm NO bottle was emptied. In order to correct the collected NOx data, the Horiba 5-gas analyzer was first calibrated with the 1005 ppm NO_2 gas then spanned with 1625 ppm NO and 5010 NO. Using these known concentrations, a linear fit for the measured NOx vs actual NOx was obtained. A transfer function was then created to correct the mismeasured NOx data to the linear fit from the 1625 ppm NO calibration. The test points measured with the bad calibration were ran again with a good calibration to obtain correct values. To verify the transfer function accurately corrected the data, the mismeasured NOx data was plugged in to ensure it matched the good calibration NOx measurements.

4 Experimental Results & Discussion

This chapter covers the results from experimentation broken into two sections, diesel with water injection and NG-DP with water injection. Each section provides an overview of the test covered and detailed discussion on the results of combustion, performance, emissions.

4.1 Diesel with Water Injection

Table 11 outlines the tests conducted in studying the impact of WI on regular diesel operation.

Table 11. Diesel with Water Injection Test Point Overview

Test Set Name	Torque [Nm]	BMEP [bar]	Engine Speed [RPM]	Independent Variable
A	672	12.5	1800	WFR
B	224	4.2	1800	WFR
C	448	8.4	1200	WFR

The combustion, emissions, and performance are described in the following sections for each data point outlined above.

4.1.1 Combustion

The combustion impact resulting from WI is reviewed and discussed below where each test point is separated into its own section.

4.1.1.1 Test Point A - 12.5 bar BMEP - 672 Nm @ 1800 RPM

WFR was swept from 0 to 2.5 by increments of 0.5 in diesel only operation and resulted in the AHR curves shown in figure 12.

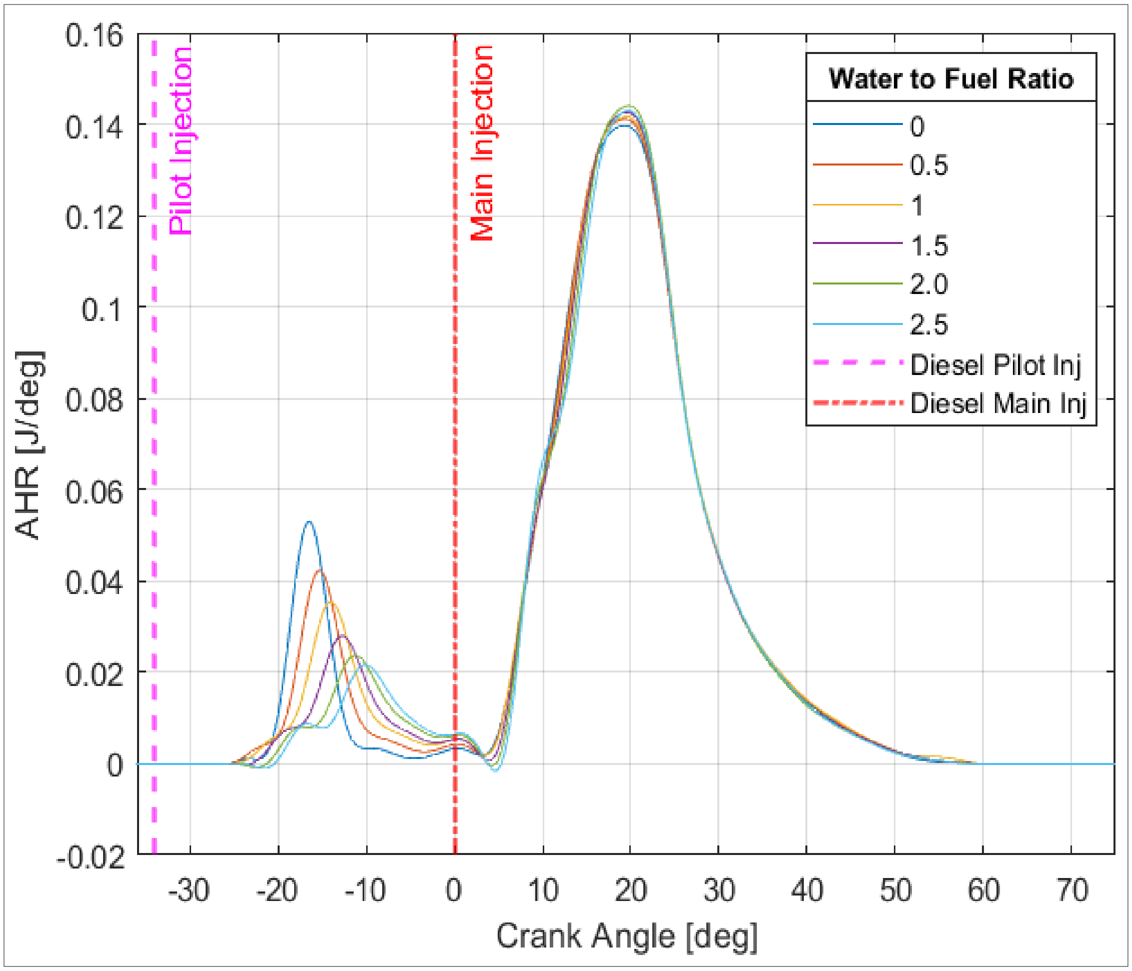


Figure 12. Apparent Heat Release - Test Point A

From the figure, it was concluded that regarding heat release rate, only the pilot injection is significantly impacted by the water, where the peak heat release rate of the pilot injection is decreased from 0.055 J/deg to 0.02 J/deg. The main injection incurs negligible change in both its timing and peak heat release rate. This finding suggests that the water is completely vaporized during the compression stroke, ignition delay, and pre-mixed combustion phase, resulting in little to no impact on the diffusion combustion phase.

Reviewing the MFB for the WFR sweep in figure 13 provides further example of the decreased pilot burn as WFR is increased from 0 to 2.5.

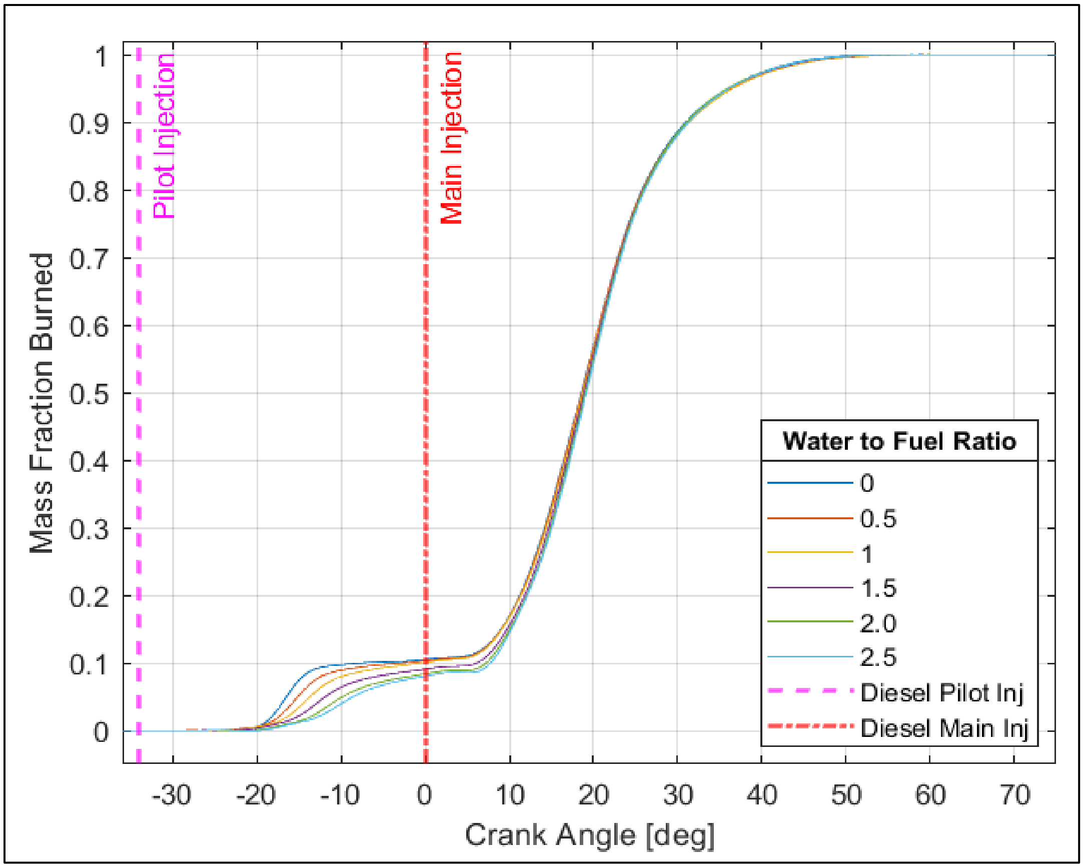


Figure 13. Mass Fraction Burned - Test Point A

The decreased peak heat release rate shown in figure 12 is visible in figure 13 as well, where the slope of the MFB decreases with increased WFR. The increased ignition delay becomes more obvious in the MFB plot. Looking at a MFB of 0.05, the start of combustion is significantly delayed from 0 to 2.5 WFR. Although ignition delay is increased, the main injection's MFB magnitude and timing are almost unchanged as additional water is injected. Previous studies experienced opposite results depending on their WI method where the diffusion combustion was impacted more than the pre-mixed combustion stage [7]. This finding indicates that WI has various impacts depending on the combustion system and WI method, giving room for development, and tuning to the engineers and operators.

The increased ignition delay and decreased peak heat release caused by WI resulted in a later CA50 but decreased the CA10-90 burn duration as shown in figures 14 and 15.

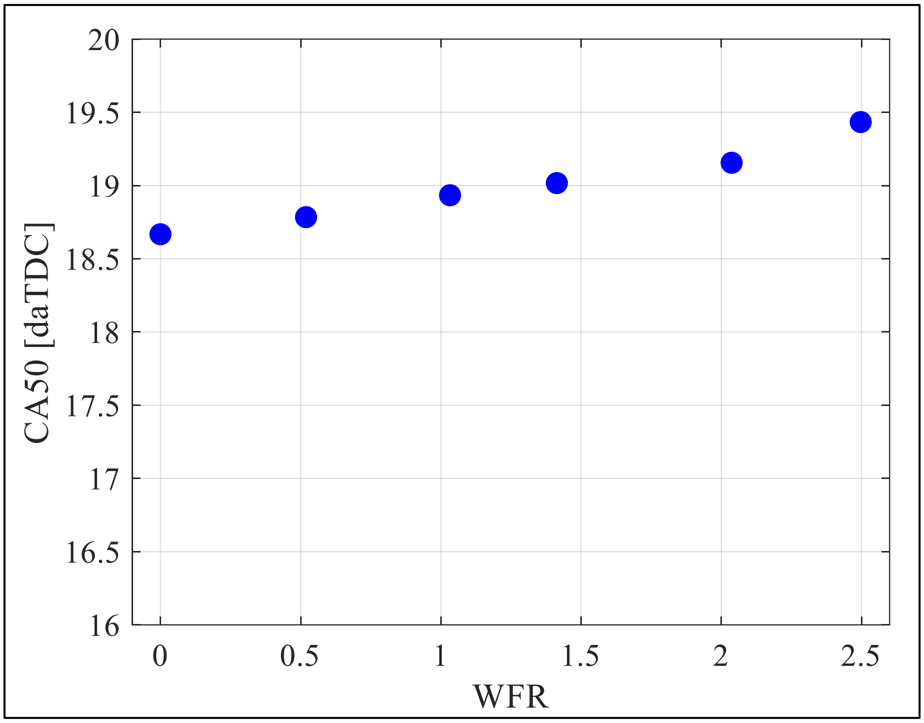


Figure 14. CA50 vs WFR - Test Point A

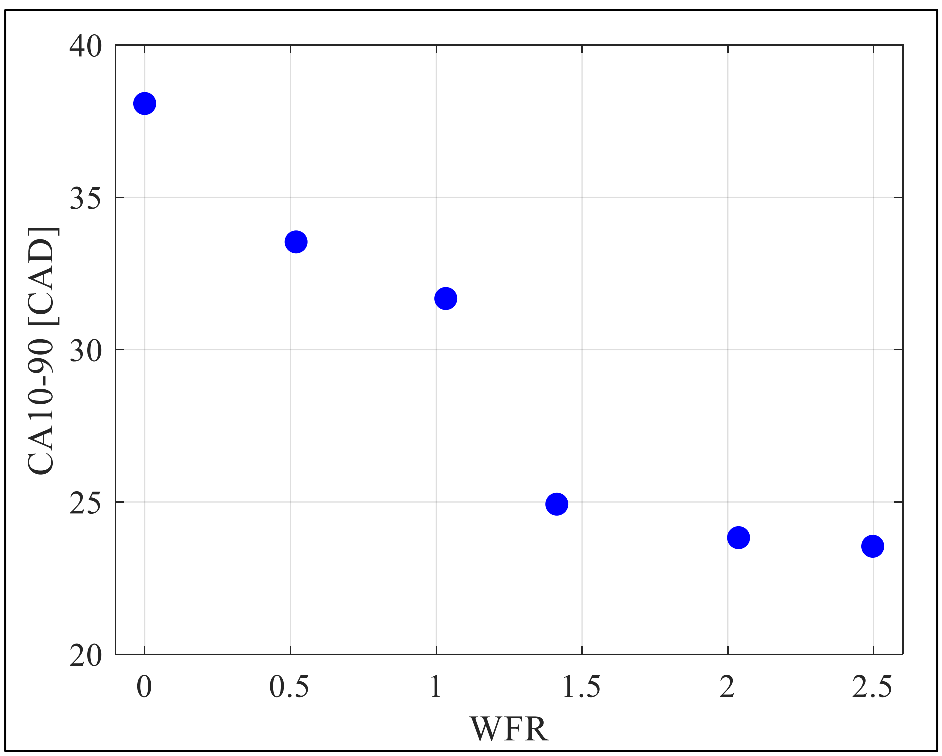


Figure 15. CA10-90 - Test Point A

The burn duration decrease is the result of the retarded SOC or 10% MFB and unaffected 90% MFB location due to the unaffected main injection. The combination of retarded CA50 and decreased CA10-90 is an indication of decreased combustion efficiency where a slower burn should result in a longer burn duration if combustion efficiency were maintained. Further verification of a decreased combustion efficiency with increasing WFR is provided in the emission result section.

The in-cylinder pressure trace resulting from the WFR sweep is shown in figure 16.

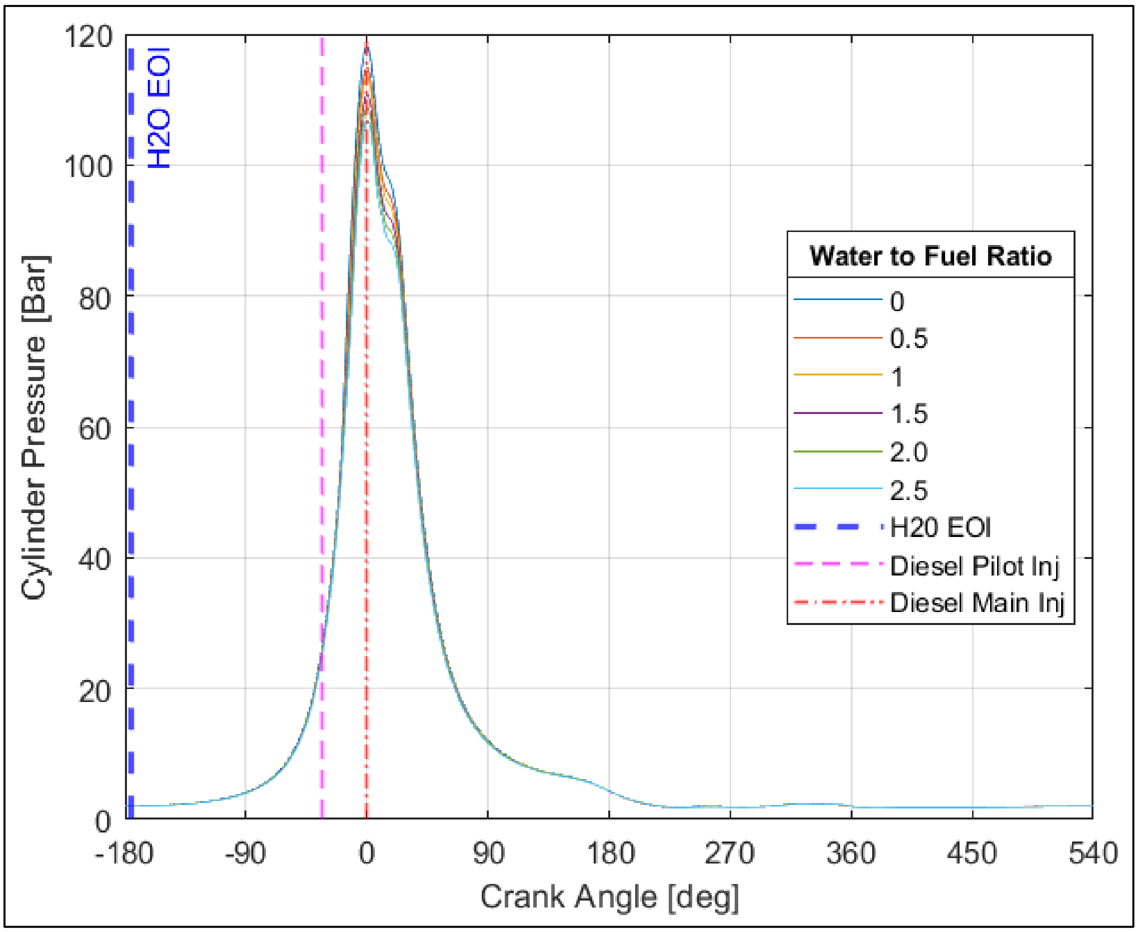


Figure 16. Cylinder Pressure Trace - Test Point A

WI decreased the peak cylinder pressure by a maximum of 10 bar despite the diffusion burn incurring little impact as reviewed above. The peak cylinder pressure decrease can be attributed to two factors. One, the decreased in cylinder temperature caused by heat transfer to the injected water which lowered the charge temperature and thus the in-cylinder pressure. Two, the discussed impact on the pilot burn of decreased heat release rate and lengthened ignition delay that both occurred before TDC and thus lowered the peak cylinder pressure.

Although a net decrease of in-cylinder pressure was recorded, there is the possibility of a pressure increase with further tuning, utilizing the expansion of steam as water is vaporized. This would require extensive tuning on this engine because of the limited freedom of injection timing that comes with using PI water where Wei et al. achieved this by using late injection, direct injected water in a SI gasoline engine [4]. IMEP is calculated using cylinder pressure and therefore was decreased with increased WFR while fuel flow remained constant resulting in a slight BTE decrease.

Another study by Tauzia et al. conducted on a DI diesel engine with WI had similar results at high loads where the diffusion burn was unaffected as WI was increased [7]. Tauzia utilized a similar method to inject water, where water was injected into the intake manifold and did not decrease the amount of air inducted by the cylinder. As the diffusion burn is dominated by the amount of available air and fuel which remained unchanged with additional water, the diffusion combustion ROHR experienced little to no impact as WFR was increased [7].

4.1.1.2 Test Point B - 4.2 bar BMEP - 224 Nm @ 1800 RPM

At low load the WI had less impact on combustion. Figure 17 shows the heat release for all six WFRs tested.

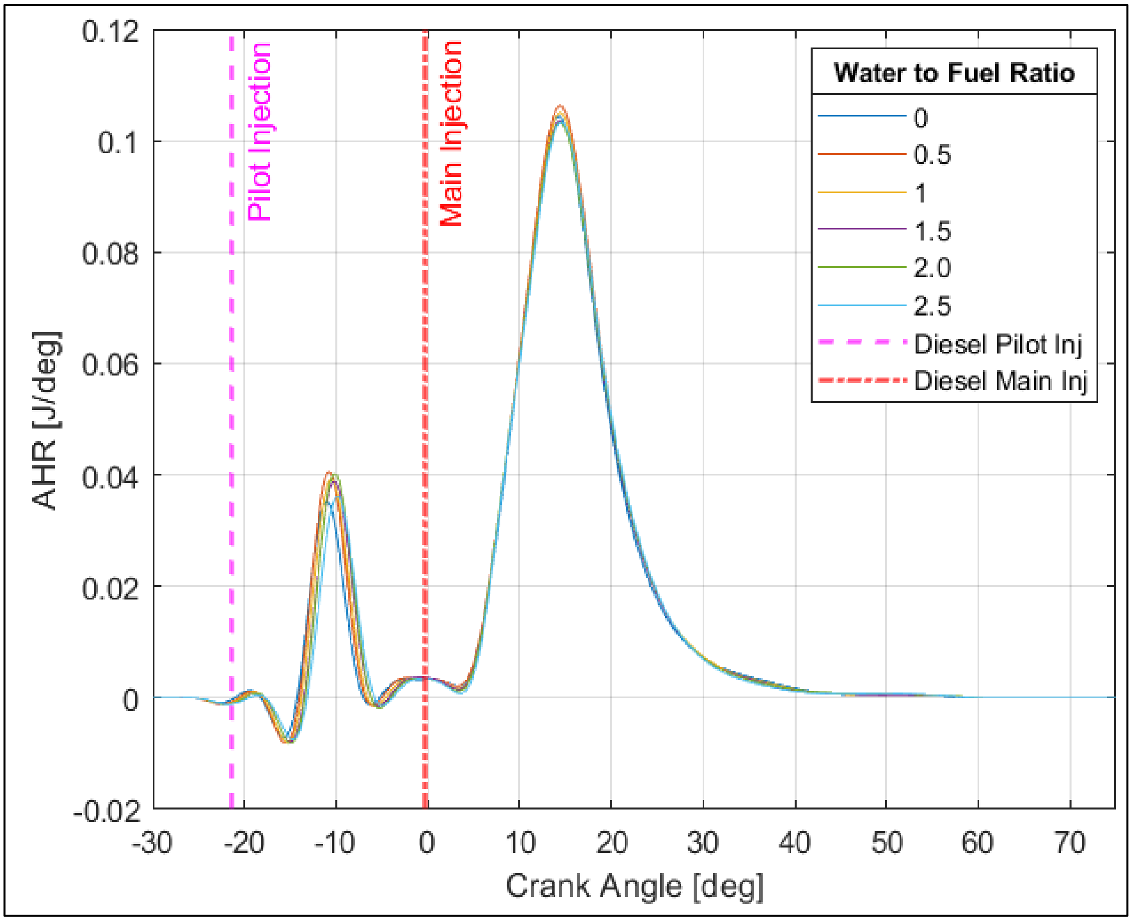


Figure 17. Apparent Heat Release – Test Point B

At this load, both the pilot and main burn are almost unaffected by the increased WI where only a small decrease in peak heat release rate is seen in the pilot injection. The combustion timing is virtually the same for all WFRs tested at this speed and load. The small impact on the ROHR at low loads in comparison to high load points can be attributed to injection timing and absolute water quantity injected. Although AFR was not controlled during diesel operation, AFR remained constant as WFR was increased, meaning air availability did not contribute to the change in AHR as WFR increased. Additionally, both polytropic coefficient of expansion and compression were unaffected with the increase in WFR. Due to WFR being calculated on a per mass basis, higher loads, needing more fuel, result in a higher absolute water quantity. During the compression stroke, this water is compressed, heated, and vaporized where a lower absolute water content will vaporize more completely than at high loads with high absolute water. This is the main factor attributed to decreased impact of WI on combustion at low load. In the high load case, the pilot is injected at 36° bTDC giving sufficient time for the water to mix with the injected fuel and decrease the laminar flame speed of the pilot injection. Additionally, the earlier combustion takes place when less water has been vaporized by the cylinder heat and compression. In the low load case, a later pilot at 22°bTDC gives less time for the injected diesel to mix with the water before reaching its combustion point as well as additional time for the water to vaporize as the piston compresses. The injected water therefore has little impact on the premixed and diffusion combustion at low loads. Additional work exploring the effect of SOI and pilot contribution could help explain this finding further. The decreased impact is further exemplified when reviewing figures for the combustion phasing and duration in figures 18 and 19.

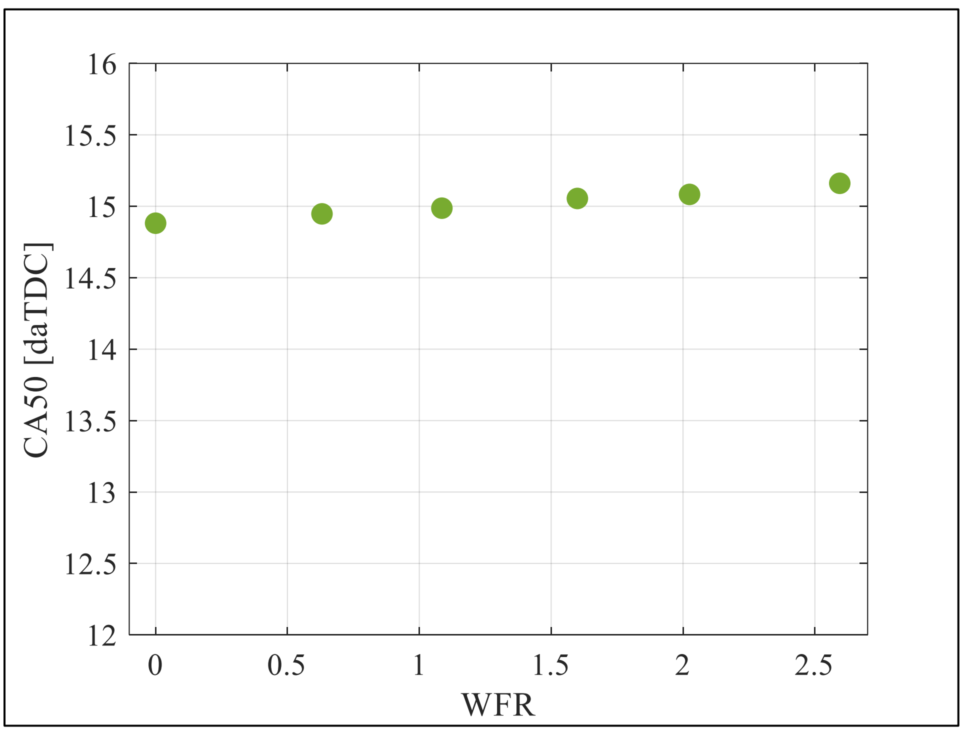


Figure 18. CA50 vs WFR - Test Point B

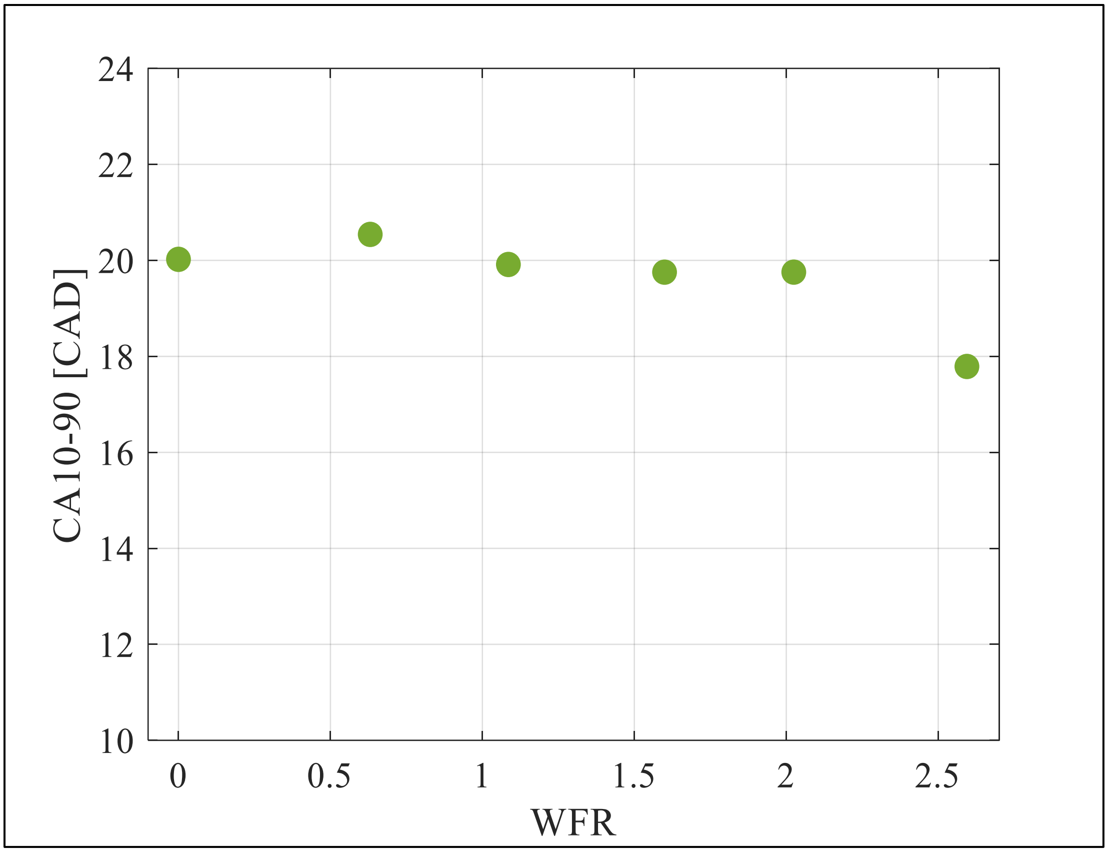


Figure 19. CA10-90 - Test Point B

As shown, CA50 was retarded by approximately 0.25 CAD at low load in comparison to the 1 CAD experienced at high load. Burn duration at low load was decreased by 2 CAD where the high load condition burn duration was decreased by 15 CAD from 0 to 2.5 WFR.

4.1.1.3 Test Point C – 8.4 bar BMEP – 448 Nm @ 1200 RPM

The medium load, low speed tests showed similar results to the low load testing where WI had negligible impact on combustion. The AHR is shown in figure 20 where a small decrease in peak heat release rate for the pilot injection is shown as water is increased.

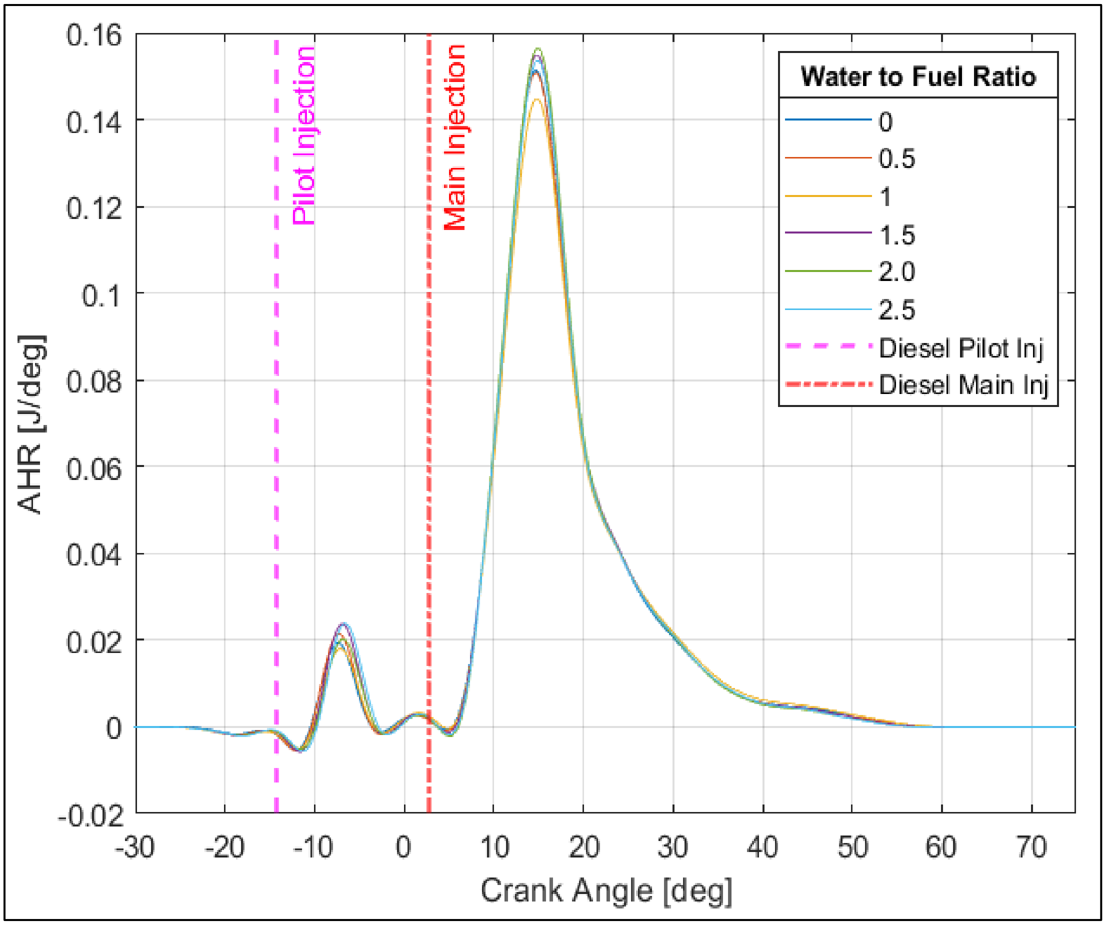


Figure 20. Apparent Heat Release - Test Point C

Plotting CA50 and CA10-90 in figures 21 and 22 further exemplifies the small impact WI had at this speed load point where both remained relatively constant.

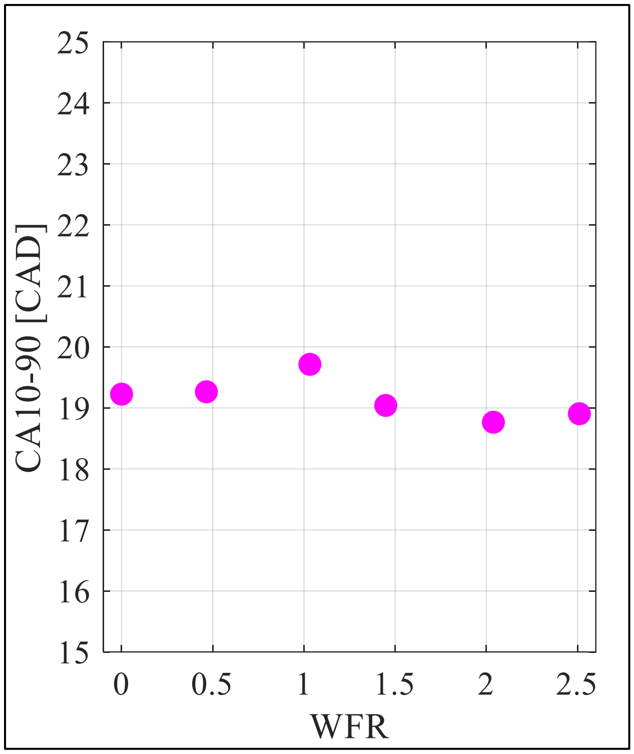


Figure 21. CA10-90 vs WFR - Test Set C

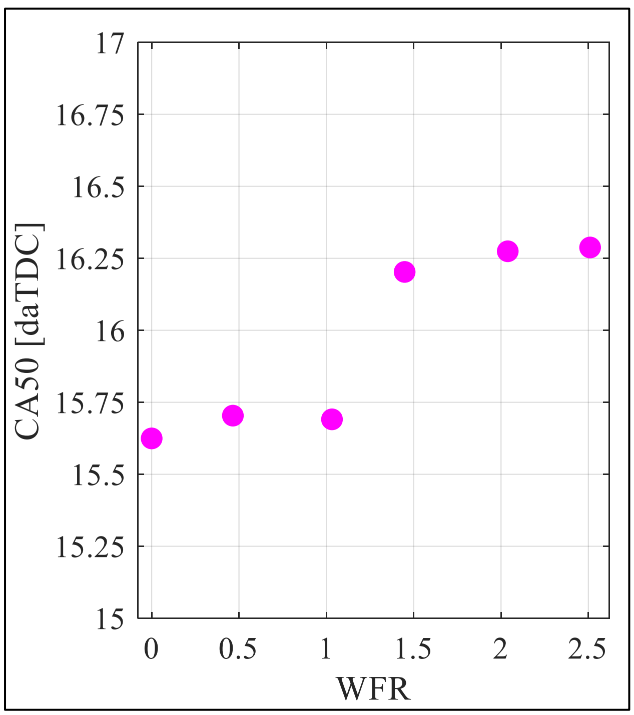


Figure 22. CA50 vs WFR - Test Set C

In addition to the mechanisms attributed to lower combustion impact from water at low loads, the low speed at this condition allowed for increased heat transfer to the injected water prior to TDC and thus vaporized more of the water before SOC.

4.1.2 Emissions

The emission results for test points A, B, and C are reviewed and discussed concurrently in this section.

4.1.2.1 Test Points A, B, and C – High, Medium, and Low Load

The NOx and soot emissions were recorded and displayed in figure 23 and 24 for all WFRs tested.

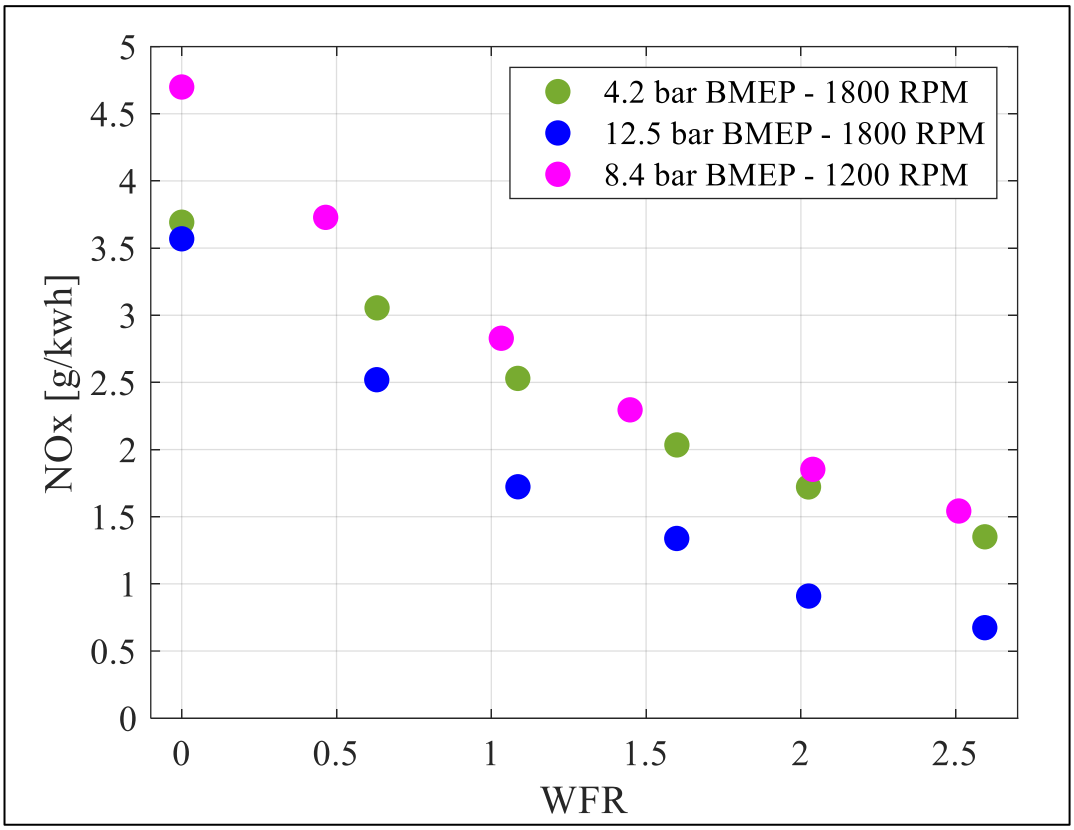


Figure 23. NOx vs WFR - Test Points A, B, and C

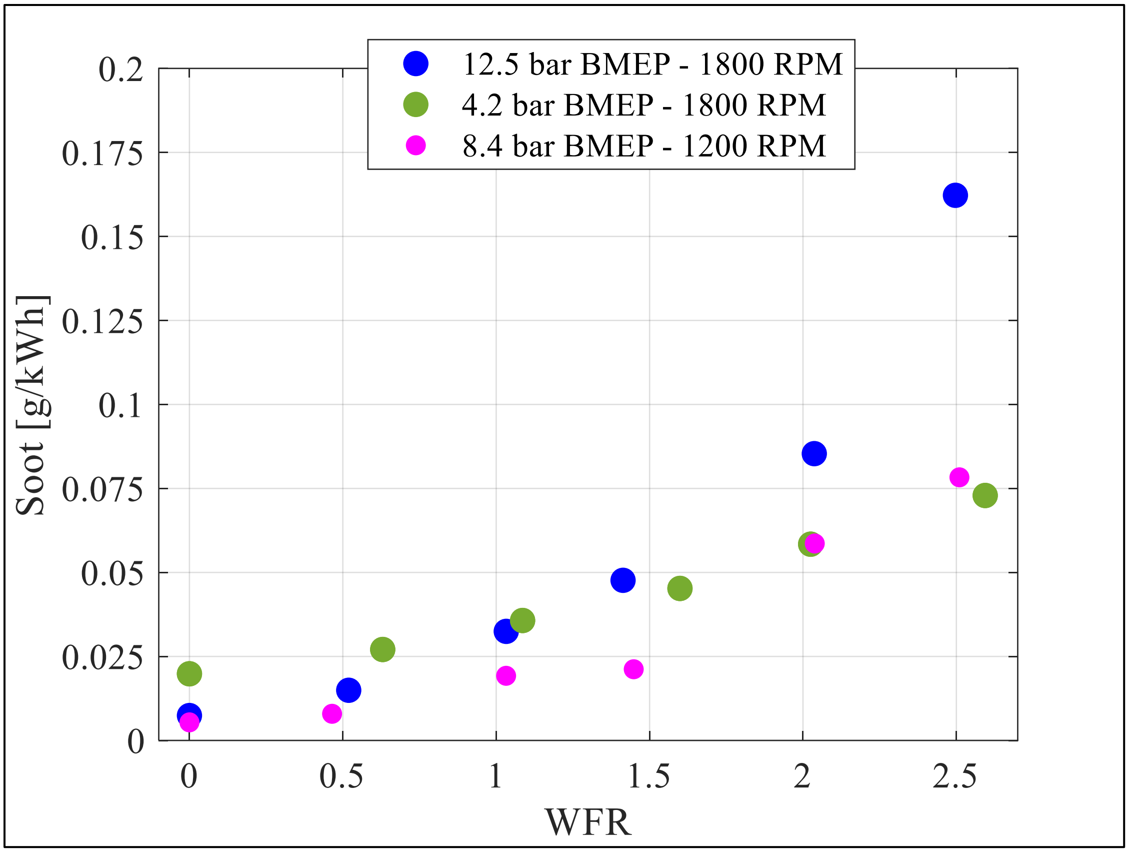


Figure 24. Soot vs. WFR - Test Points A, B, and C

There was an 83% decrease in NOx from 0 to 2.5 WFR at high load and 62.5% decrease at low load. This result provides validation to several other studies that conducted WI experiments on diesel engines [3,6,7]. The NOx-Soot tradeoff mentioned in the literature review section is a temperature dependent mechanism and is shown to be present with WI where soot is shown to increase by a factor of 18 from 0 to 2.5 WFR at high load.

The NOx reduction is due to the decreased in-cylinder temperature as WFR is increased. The soot increase is likely due to the reduction of soot oxidization. It is likely that less soot is formed however, due to the increased ignition delay allowing increased mixing of diesel and air before combustion. Additionally, decreased combustion temperatures resulted from the energy transferred to water in the process of vaporization, can shift the diesel combustion into low temperature combustion where less soot is formed. The measured increase in soot shows that soot oxidization in no-water conditions has a more significant effect than the mechanisms decreasing soot formation. This explanation is validated further when observing the CO trend outlined in figure 25.

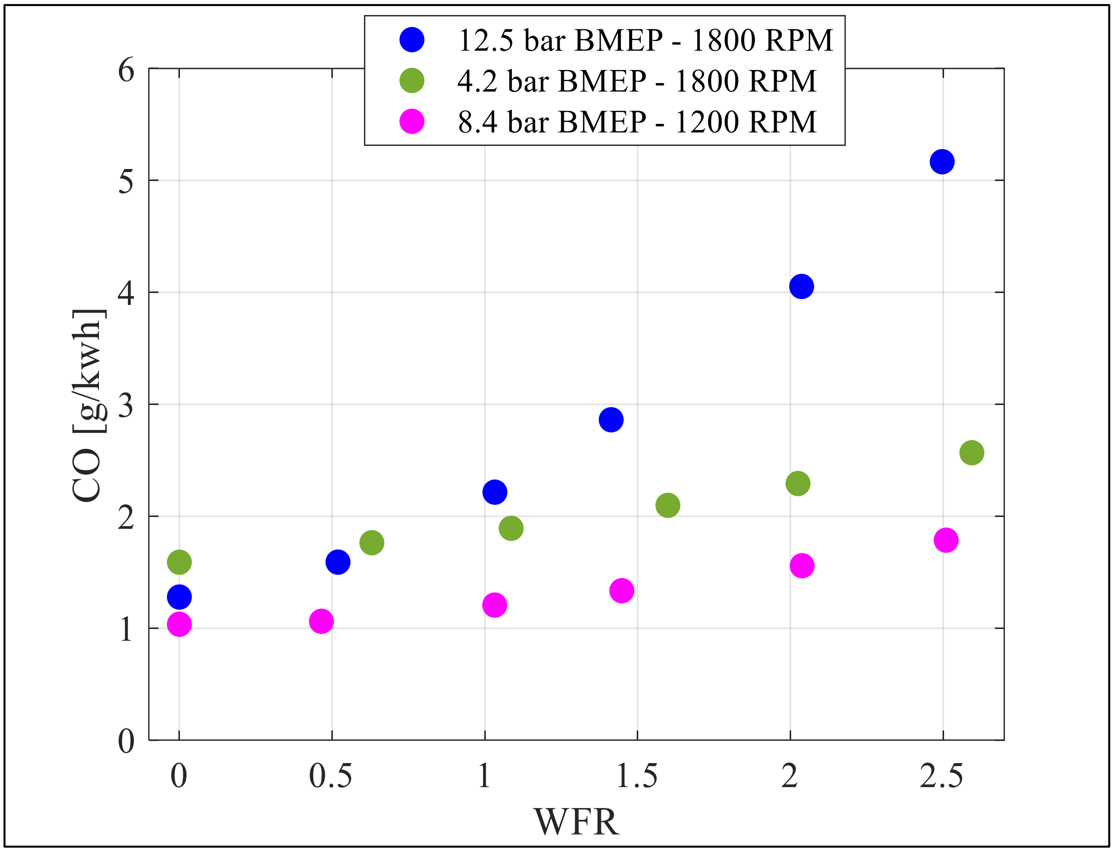


Figure 25. CO vs WFR - Test Points A, B, and C

The increase in CO indicates decreased combustion efficiency and low temperature combustion that is unable to completely burn all the hydrocarbons. Figure 26, showing the exhaust gas temperature as WI quantity is increased further verifies this explanation where low temperature end gas burns off less HC, CO, and PM.

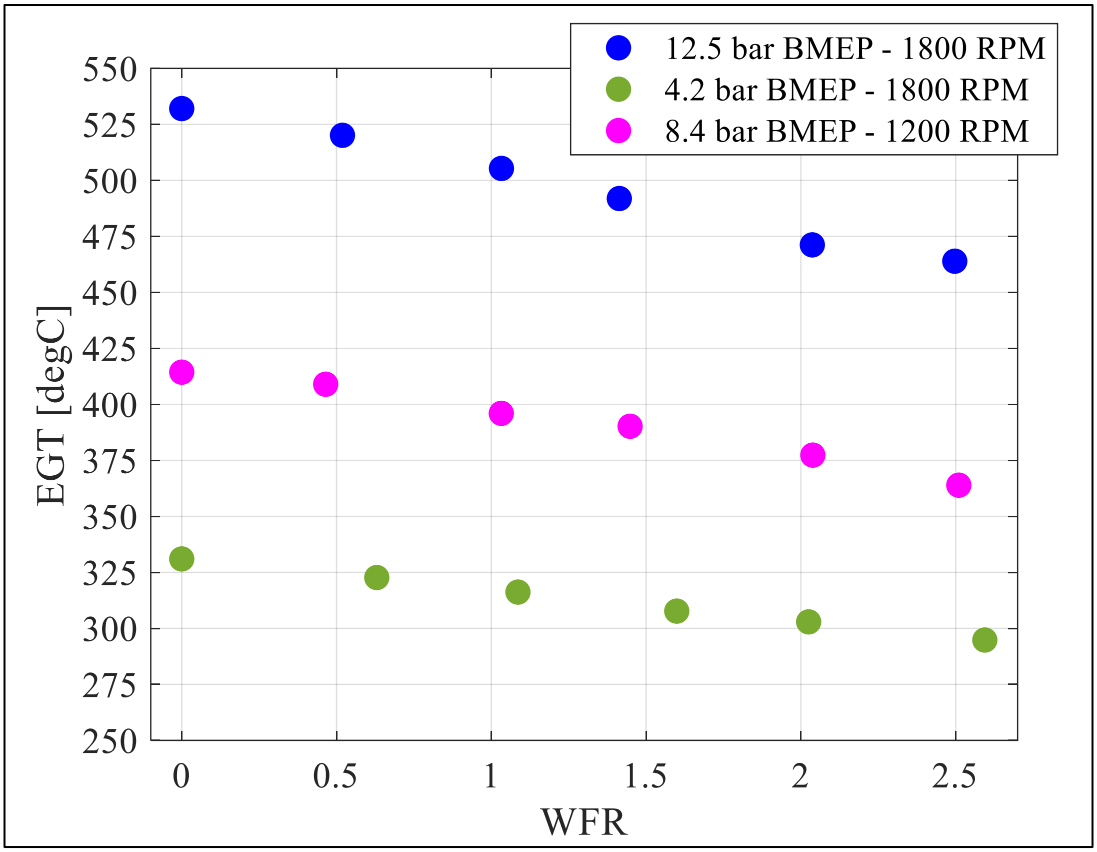


Figure 26. EGT vs WFR - Test Points A, B, and C

The smaller absolute WI quantity and its decreased impact on combustion is reflected in smaller EGT decrease at lower load.

4.1.3 Performance

As mentioned in the test plan section of this paper, the diesel pilot and main SOI were held constant while WFR was increased to isolate the impact WI has on combustion. As shown in figure 27 and 28, the BTE and BMEP were shown to decrease at high load and slightly increase at low load.

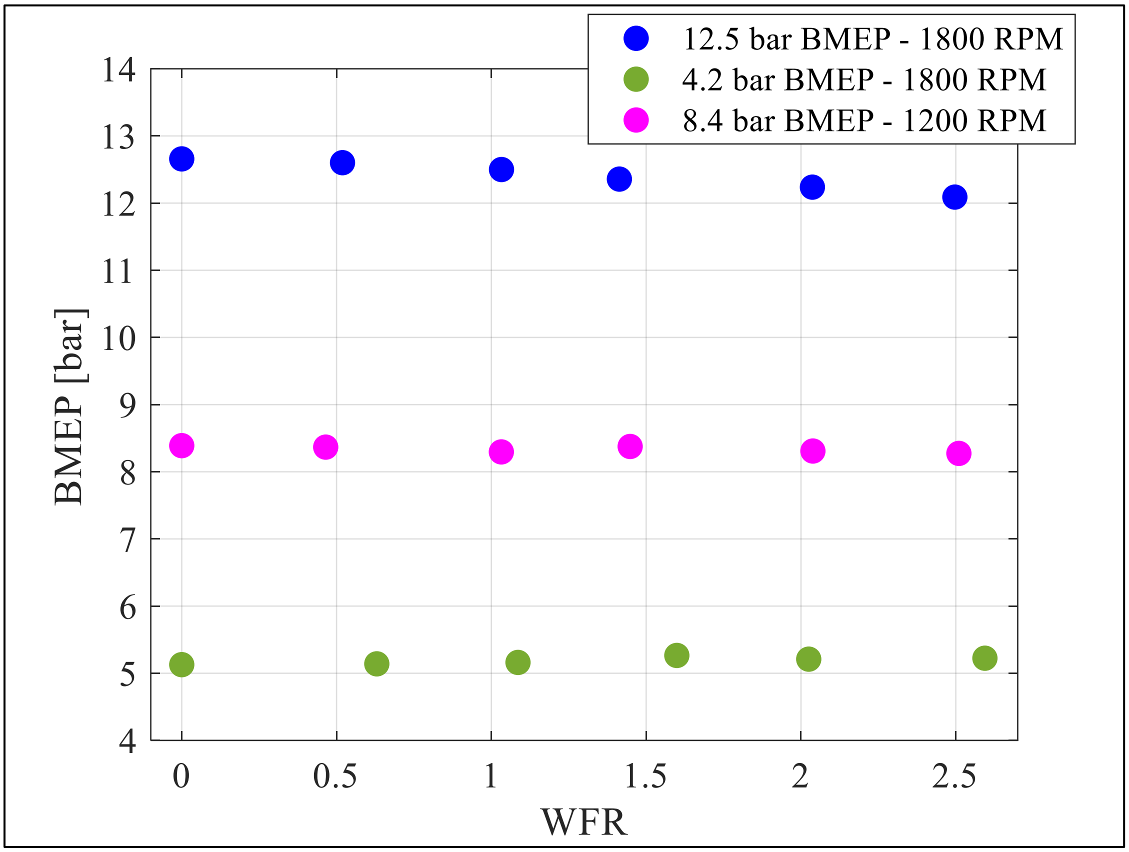


Figure 27. BMEP vs WFR - Test Points A, B, and C

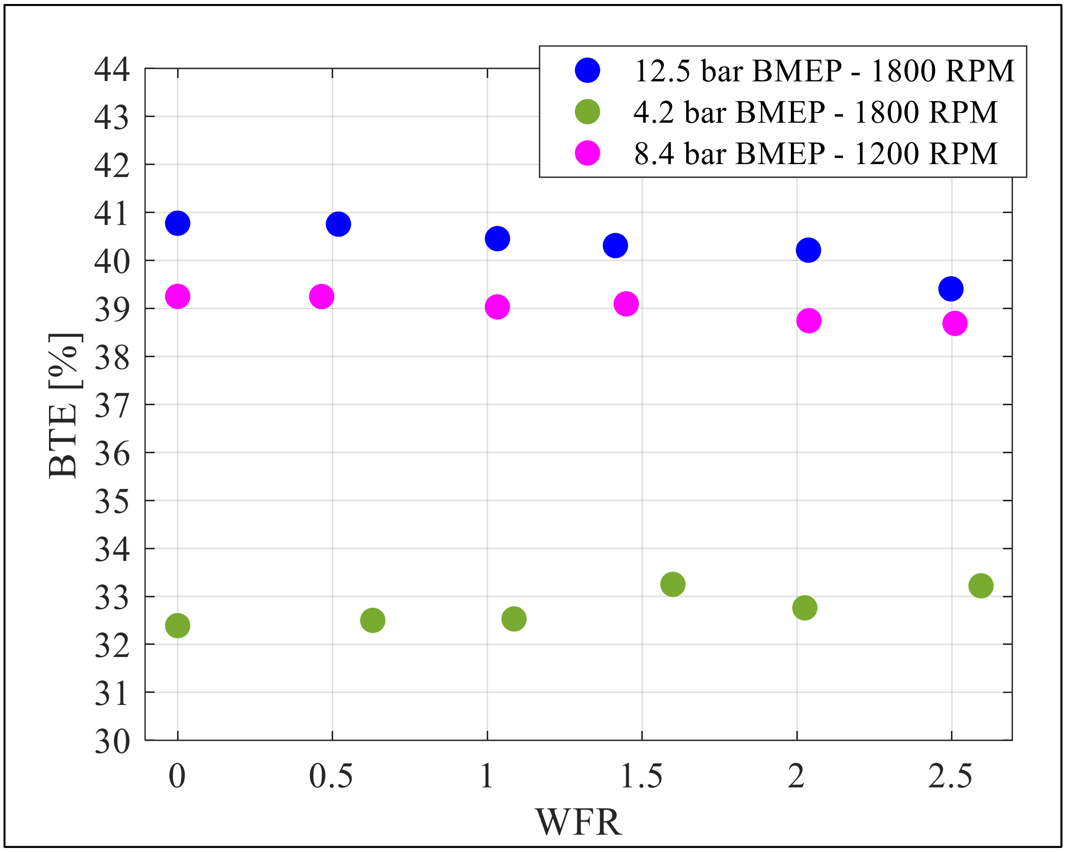


Figure 28. BTE vs WFR - Test Points A, B, and C

The decreased load can be attributed to the cooling effect of the WI, lowering the in-cylinder pressure, and thus decreasing BMEP while fuel flow is held constant. The increased ignition delay, causing a late CA50 as discussed, also contributes to the decreased BMEP as the combustion phasing is shifted away from MBT. Other works have advanced SOI while water was added and found that BTE could be maintained or increased however a combustion phasing study was outside of this projects scope and was not pursued [4].

The small combustion impact shown in the previous section at low loads is reflected in the BTE increase, where the addition of water increased pressure slightly due to the expansion of steam or addition of mass while fueling was maintained. As MAP was maintained while WFR was increased, the additional volumetric exhaust flowrate required the VGT to be opened, decreasing backpressure, and thus decreasing pumping losses, resulting in an increase in BMEP and BTE.

4.1.4 Diesel + Water Injection Summary

The impact of WI on diesel operation has been studied before with similar results to what was found in this research [3,4,5,7,19]. Water had the tendency to impact the pilot combustion only, having little to no impact on the ROHR for the main burn. Additionally, in-cylinder pressure was only decreased slightly from the cooling effect of WI where some of this cooling effect was offset by the expansion of the water when transitioning between a saturated liquid to vapor stage and even increasing pressure at low loads where the impact of the steam expansion overcame the impact of the cooling from the water.

As expected, NOx emissions were decreased significantly for all speed, load points. The decreased combustion temperatures responsible for reducing NOx also resulted in a decreased combustion efficiency and thus increased CO and PM emissions.

The BMEP and thus BTE of the engine were decreased slightly at high load as WI was increased and increased at low load. WI proved to decrease knock however, which could enable advanced SOI, increased IMAT, and VGT tuning in future work to combat the decreased power and combustion efficiency found at high load. Finally, COV of IMEP was below 3.0 for all tests conducted in this section showing that WI created no issues in terms of combustion stability.

4.2 Natural Gas-Diesel Pilot with Water Injection

While conducting the literature review, only two research papers were found that included experimental results for WI, dual fuel engines [3,6]. Lacking experimental precedent, this stage of experimentation was the focus of this work and thus the majority of tests were conducted in this operating mode. The test points reviewed in this section are outlined in table 12.

Table 12. Test Set Summary for Natural Gas - Diesel Pilot Experimentation

Test Set Name	Torque [Nm]	BMEP [bar]	Engine Speed [RPM]	Independent Variable
D	672	12.5	1800	WFR and SOI
E	896	16.8	1800	WFR and SOI
F	448	8.4	1800	SOI, DSR, RP, EGR, Injectors
G	672	12.5	1800	WFR
H	< 224	< 4.2	1200	RP, SOI, DSR, DSR
I	448	8.4	1800	Lambda

The experimental results are broken into sections below, focusing on combustion, emissions, and performance. Unlike the previous section, speed/load points are not separated and are compared side by side as needed.

4.2.1 Combustion

During diesel with WI experimentation, SOI was held constant while WFR was increased. For the NG-DP experiments, constant combustion phasing was desired although several tests were conducted to verify the findings of previous researchers, where WI was found to increase ignition delay and decrease peak heat release rate [3.6]. Figure 30 shows the apparent heat release for this verification where the impact of WI is studied at 12.5 bar BMEP and 1800 RPM, comparing constant combustion phasing (CA50) versus constant SOI. The CA50 and SOI are shown below in figure 29.

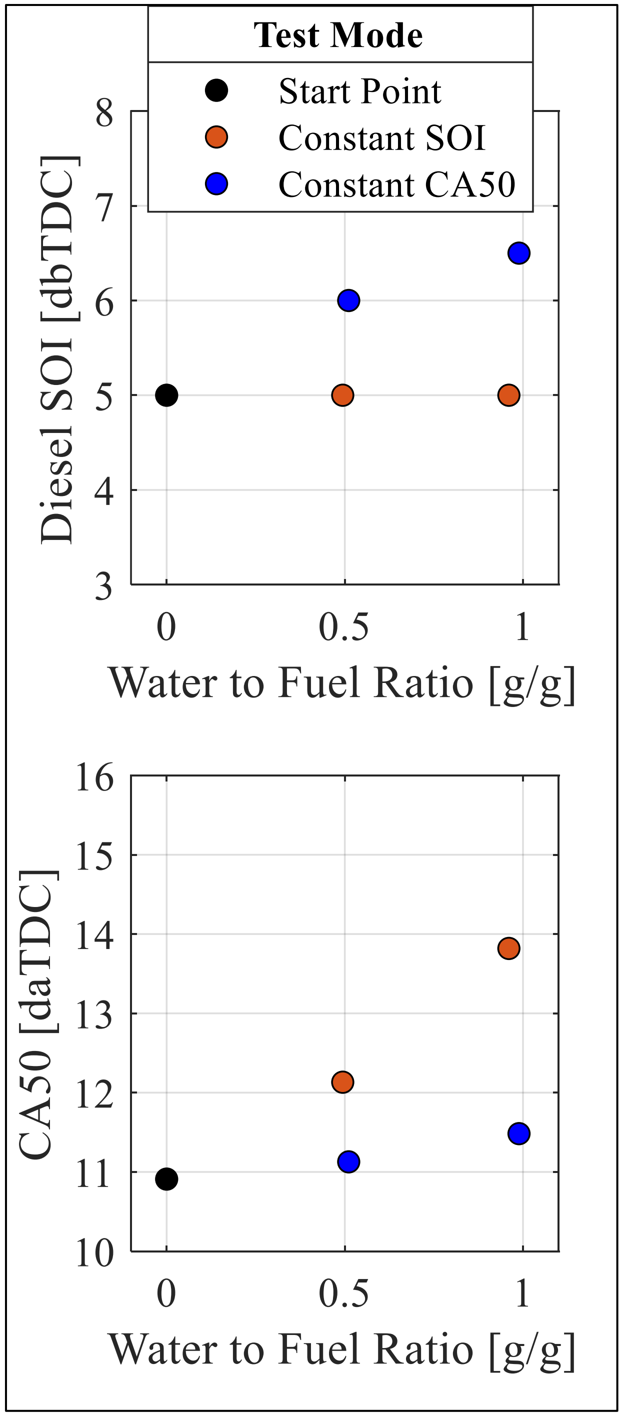


Figure 29. SOI and CA50 - Test Set D and G

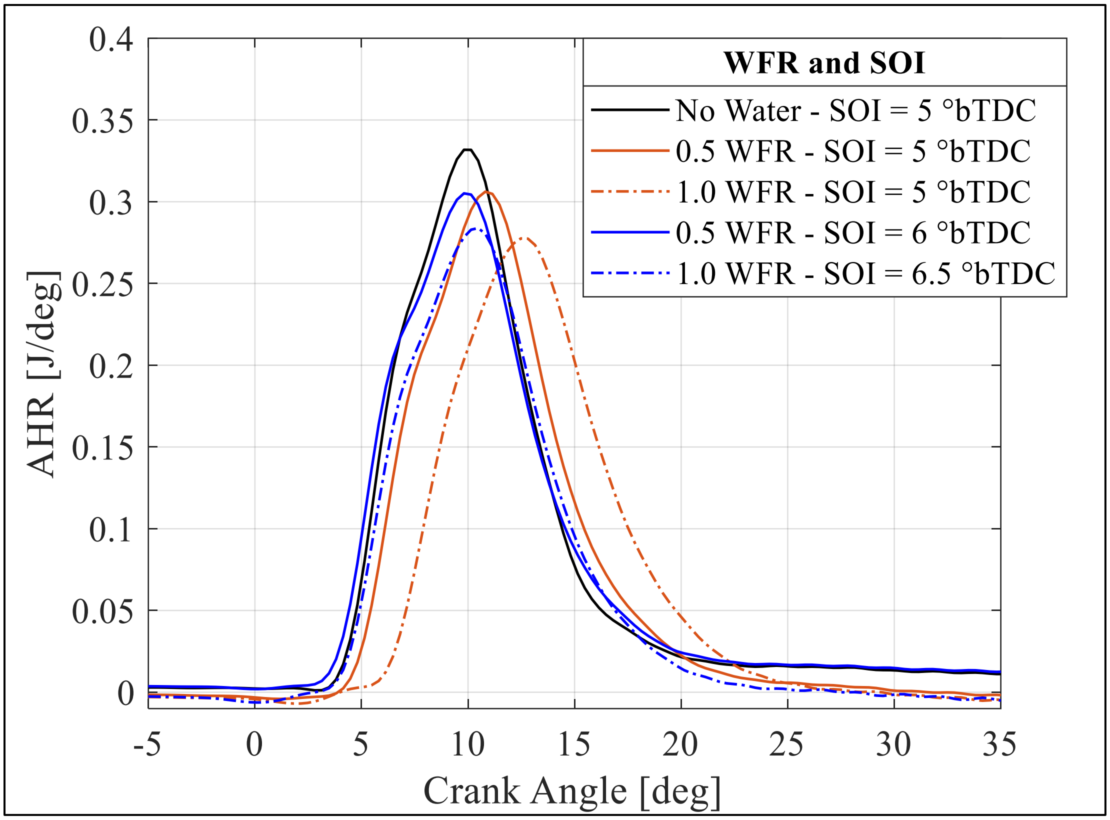


Figure 30. Apparent Heat Release - Test Set D and G

As WFR is increased and the diesel SOI is maintained at 5 °bTDC, the AHR is delayed and the peak is decreased. Advancing the diesel SOI to maintain CA50 while increasing water mitigates the retarded timing from water, but still results in a decreased peak AHR. This result verifies the findings summarized by Wei, where a decreased diesel pilot quantity in NG dual fuel engines resulted in late and decreased combustion of the NG. Furthermore, it was stated that the late phasing resulted from a decreased diesel pilot could be mitigated by advancing SOI, but the peak AHR could not [8]. Our study found that the pilot injection was impacted most by the WI in the diesel with WI section and that finding is reflected here as the natural gas peak heat release, shown in the literature review to be controlled by the pilot burn quality, is shown to decrease as WFR is increased, regardless of SOI timing.

Further verification is provided when reviewing the MFB shown in figure 31.

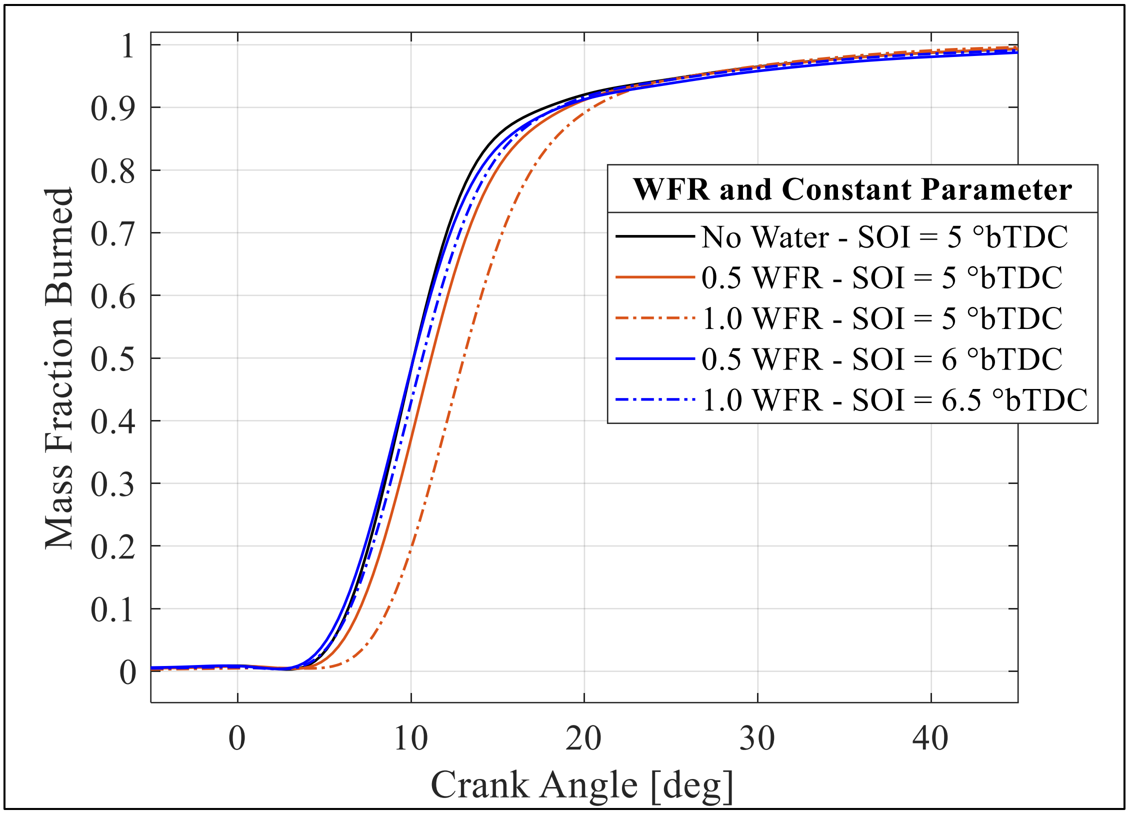


Figure 31. Mass Fraction Burned - Test Set D and G

Like the diesel with water testing, the combustion after the pre-mixed diesel combustion is mostly unaffected where the AHR rate and AHR duration for the no-water and water case are matched.

The decrease in peak AHR can mainly be attributed to the heat transfer to the injected water, where heat release from the fuel may occur, but is not calculated if transferred to the water. As mentioned in the diesel with water review section, the expansion of water into steam can mitigate the calculated heat loss if timing and quantity are optimal. Figure 32 shows the recorded CP trace for the combustion phasing study conducted.

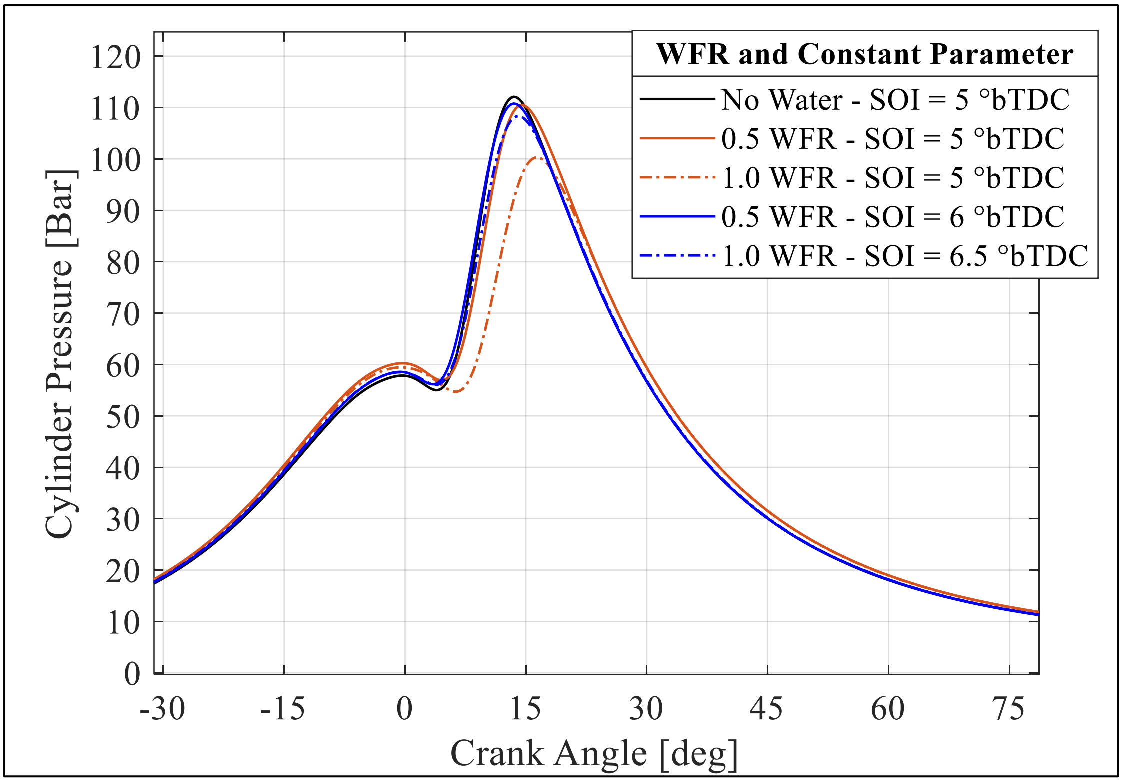


Figure 32. Cylinder Pressure Trace - Test Set D and G

The cylinder pressure trace makes the SOC after TDC obvious, shown near 5°aTDC where pressure increases rapidly. Comparing the 1.0 WFR tests, the advanced SOI proves to maintain high cylinder pressure were maintaining SOI results in a significant decrease in cylinder pressure, and thus load and BTE. In pursuance of higher engine performance, combustion phasing was maintained by advancing SOI while WFR was increased at loads of 12.5 and 16.8 bar BMEP at 1800 RPM. Figure 33 shows the SOI advance and resulting constant CA50 and CA10-90 for these tests.

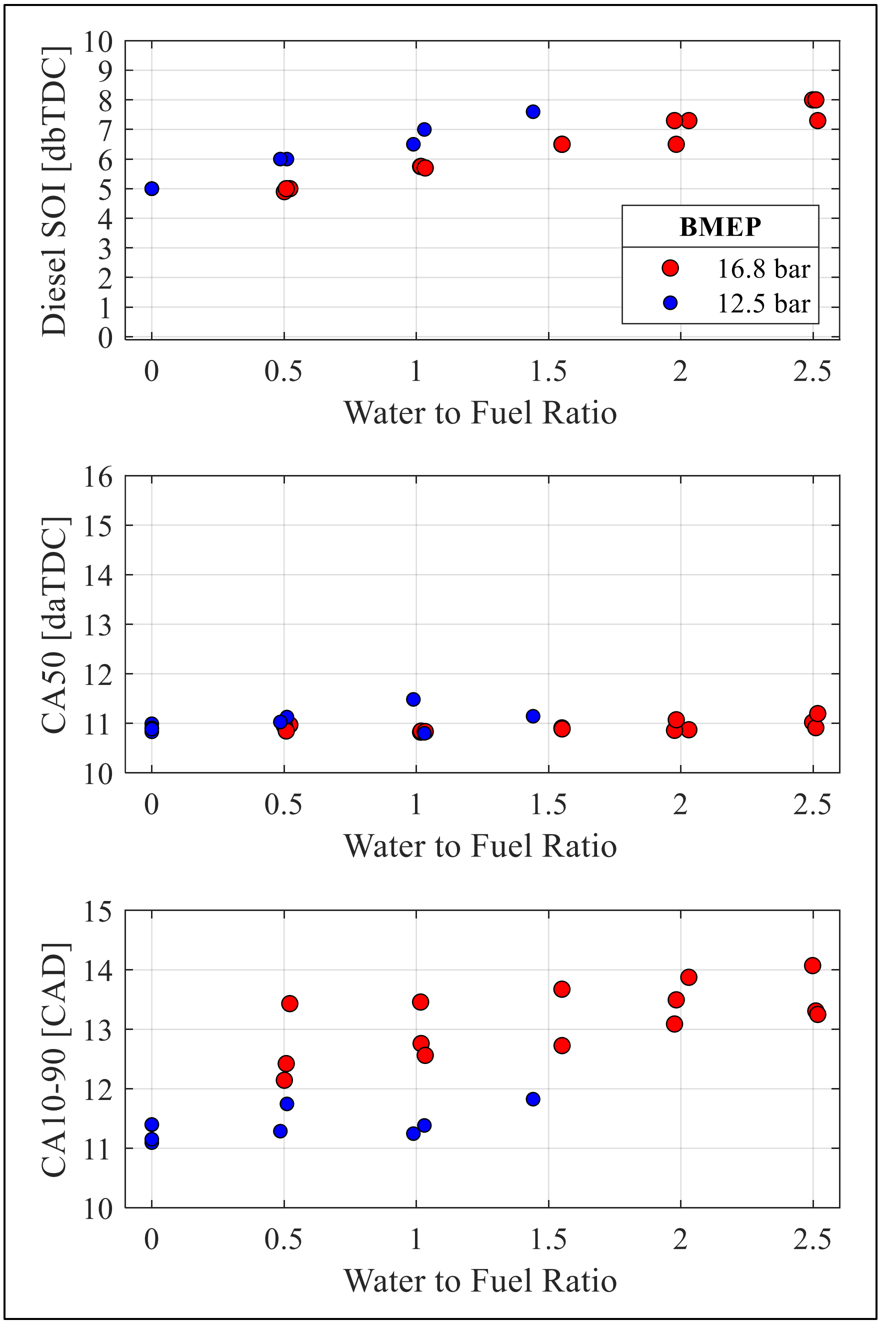


Figure 33. CA10-90, SOI, and CA50 vs WFR - Test Set D and E

The increased burn duration shown as WFR was increased is better visualized using the MFB plot in figure 34 where only the no-water and maximum water cases are shown for each load.

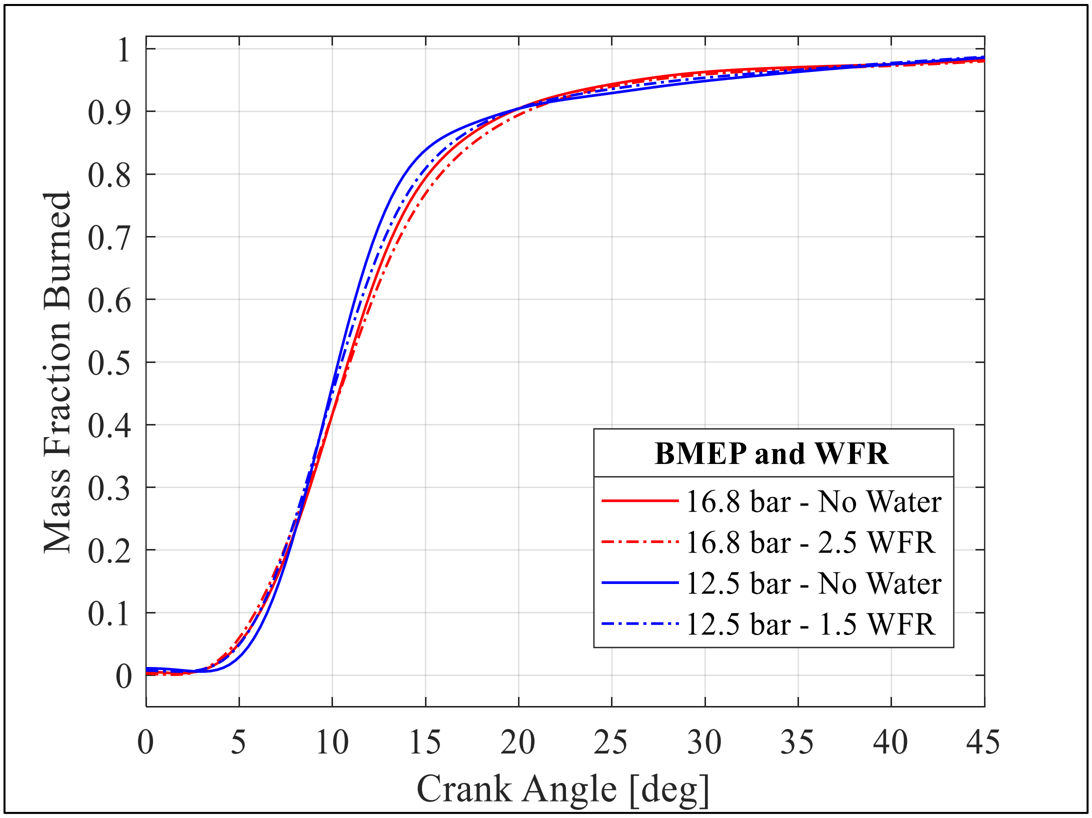


Figure 34. Mass Fraction Burned - Test Set D and E

The increase of WFR resulted in measured decrease in burn rate which can be attributed to a decreased laminar flame speed caused by lower in-cylinder temperatures from the injected water where the EGT trend is shown in figure 35.

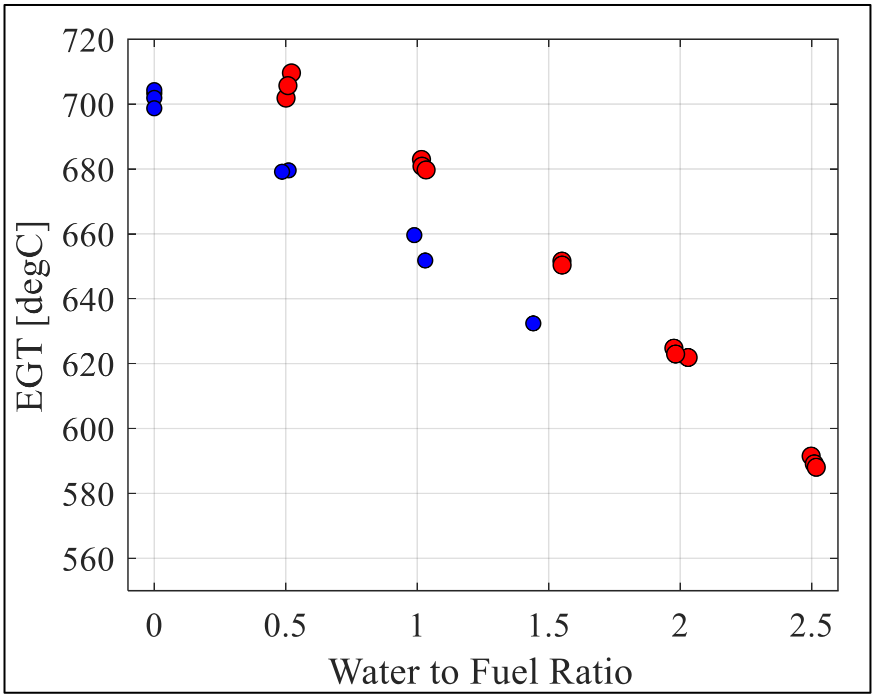


Figure 35. Exhaust Gas Temperature - Test Set D and E

The decreased in cylinder temperature as shown slows the combustion of natural gas in the cylinder, causing a longer duration and decrease in peak AHR, shown in figure 36. There are some benefits to this cooling however where in stoichiometric operation, EGT can reach temperatures capable of harming the turbocharger where the turbocharger present on this engine was limited at 750°C. Additionally, excess heat in the exhaust means there is excess energy in the exhaust gases that could have been used to increase engine power. Finally, a decreased cylinder temperature can and has been used to mitigate knock which will be covered in the following section.

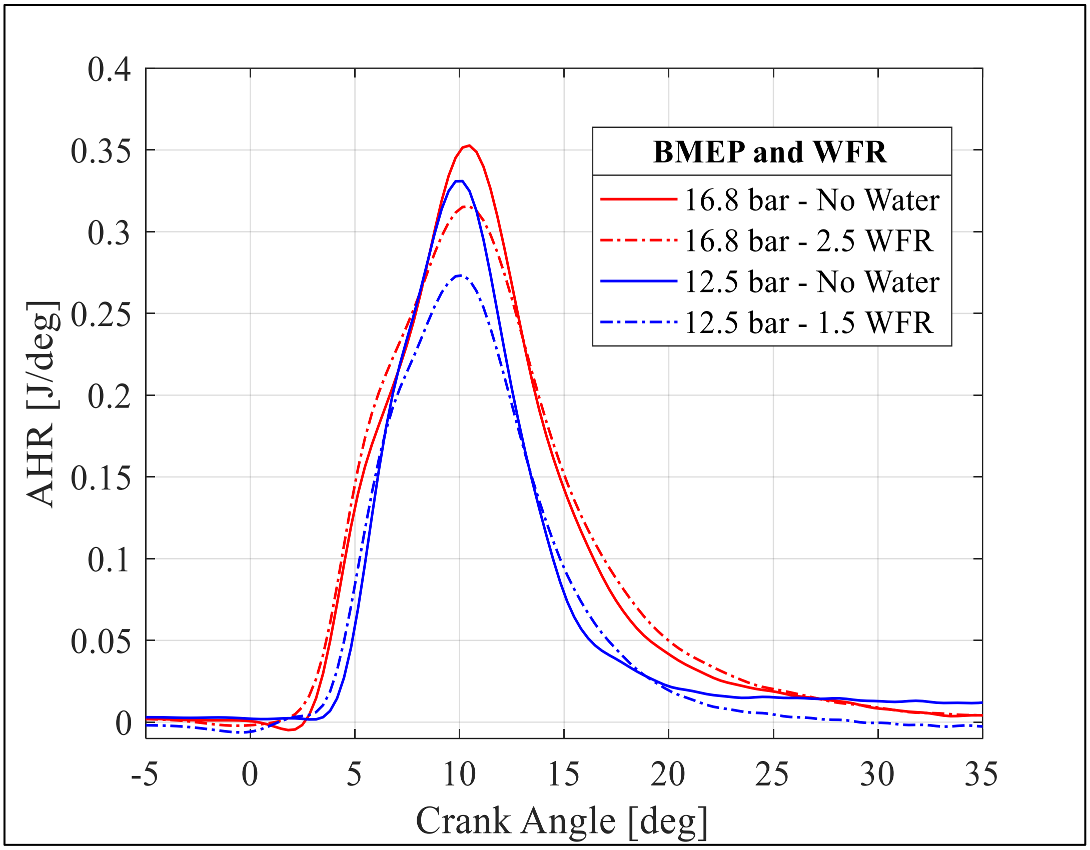


Figure 36. Apparent Heat Release - Test Set D and E

Although the combustion phasing was able to be matched to the no water conditions for both loads, the peak AHR with WI was still decreased. This decreased peak AHR is again, attributed to the decreased calculated AHR from the heat transfer to the injected water and decreased cylinder pressure as shown in figure 37.

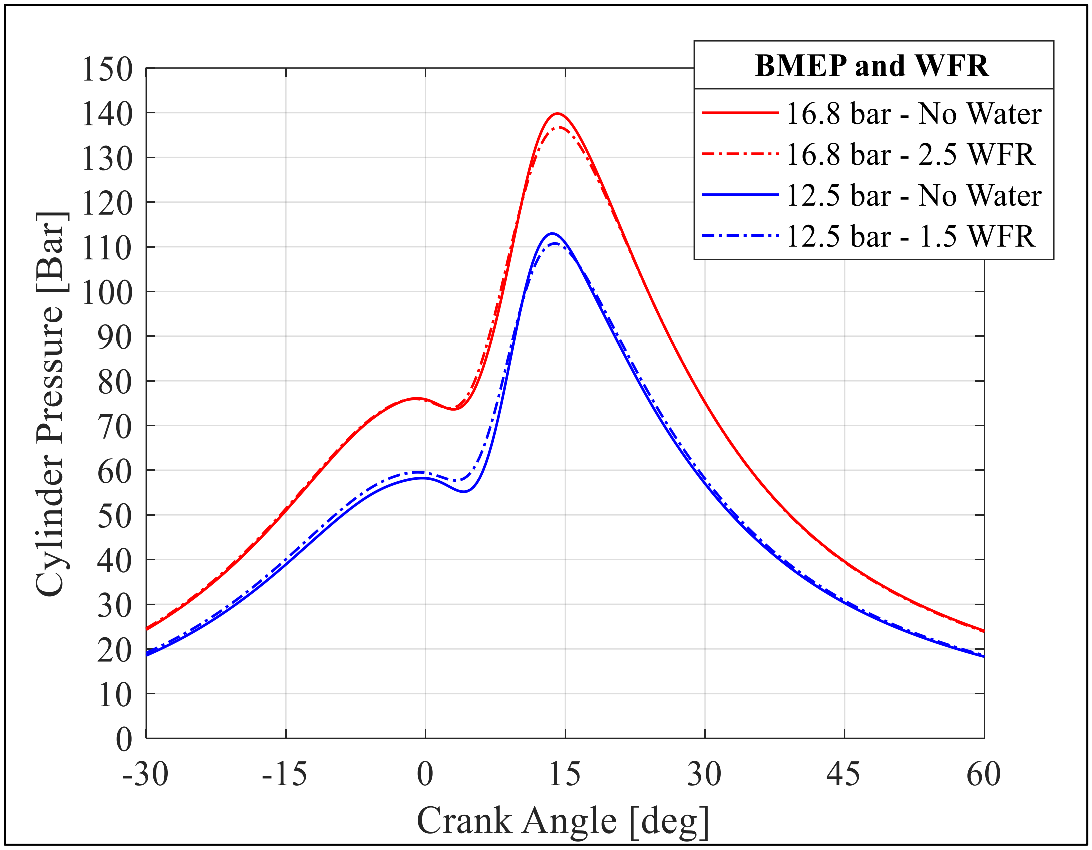


Figure 37. Cylinder Pressure Trace - Test Set D and E

The decrease in peak cylinder pressure with increased WI is similar to the high load diesel injection results. The fueling however was increased during the NG-DP with WI experiments to maintain the load while water was increased. Increasing fueling while maintaining load of course resulted in a decreased BTE as discussed in the following section.

WI's impact on DF-DP combustion is straightforward in that as WFR increases, ignition delay is increased while burn rate is decreased. Both of these trends are attributed to the cooling effect of water, where the high specific heat capacity of water causes high heat transfer to the injected water from the cylinder charge as the cylinder compresses and the fuel combusts. Slow and late combustion as a result of WI cause decreased load output, decreased combustion efficiency, and ultimately decreased BTE. Some of these combustion impacts can be mitigated however where the advance of the diesel pilot was shown to mitigate the increased ignition delay. Increased IMAT or diesel pilot contribution could be used to mitigate the slow combusting NG as shown in other projects, however that is beyond the scope of this experimental study [47].

4.2.2 Performance

During experimentation, as WI was increased, load was impacted, where fueling was then increased or decreased to reach the target BMEP while maintaining a stoichiometric AFR and 85% DSR. The increased or decreased fueling required to reach the same load had a direct impact on BTE.

The BTE resulting from the discussed combustion impact of WI is shown in figure 38 for both the constant SOI and constant CA50 tests.

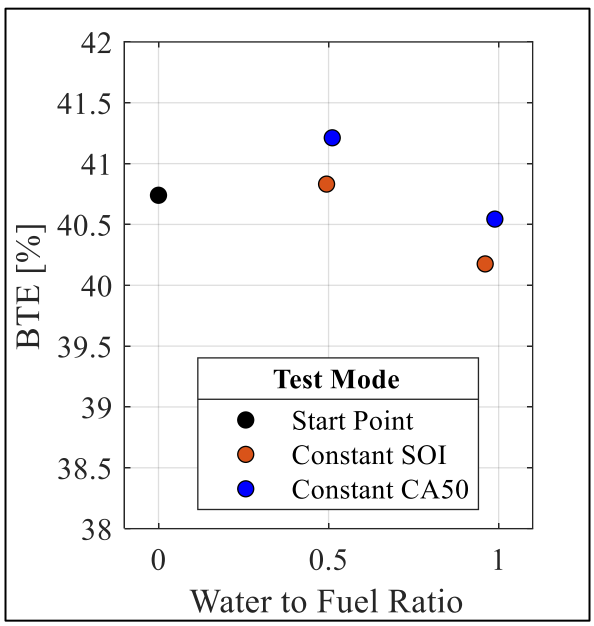


Figure 38. BTE - Test Set D & G

As shown, the BTE increases slightly for both the constant SOI and constant CA50 case at 0.5 WFR and begins to decrease at 1.0 WFR. This points to an optimal WFR near 0.5 where both NOx emission and efficiency are decreased. The decrease in BTE past 0.5 WFR signals that combustion efficiency is being decreased when excessive amounts of water are injected. Advancing SOI, maintaining CA50 at MBT, resulted in an increased

BTE in comparison with the constant SOI tests. As mentioned in the combustion section, additional tests were conducted while holding CA50 constant. The BTE for 16.8 and 12.5 bar BMEP with swept WFR is shown in figure 39. Polytropic coefficient of expansion and compression remained constant despite the increase in WFR.

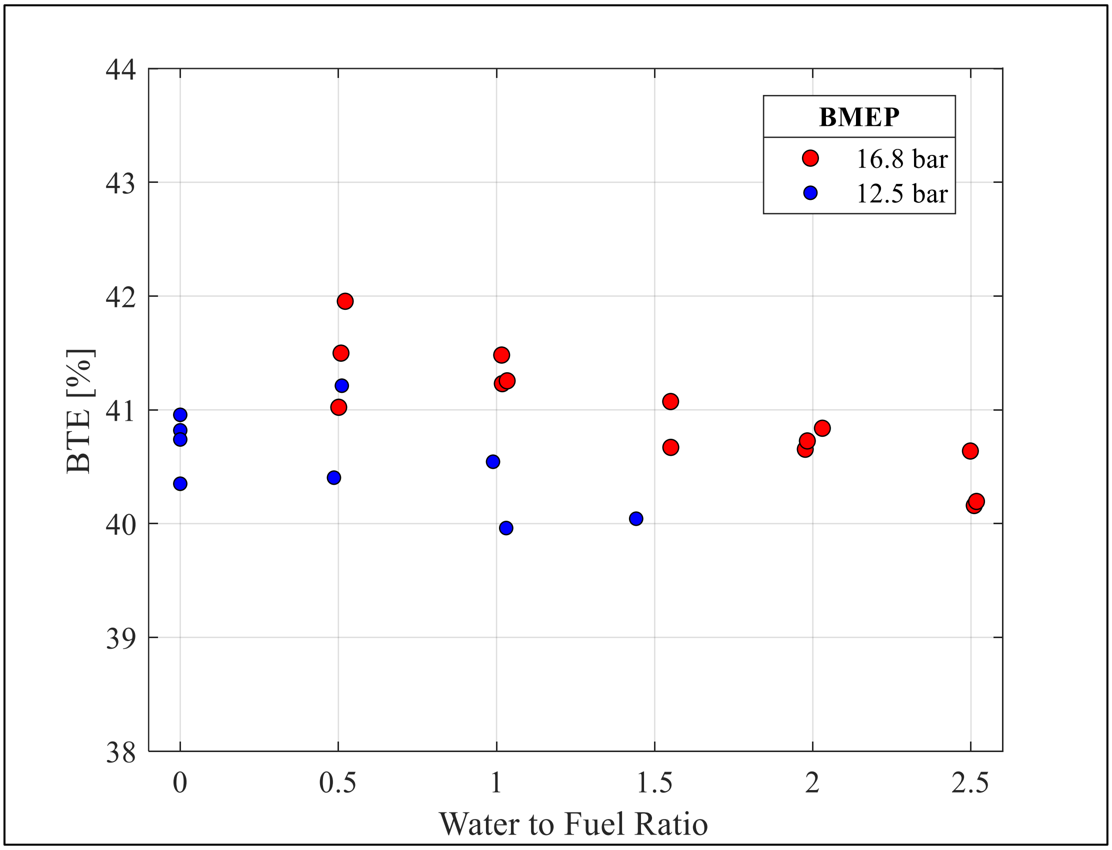


Figure 39. BTE - Test Set D & E

As shown, efficiency first increased from 0 to 0.5 WFR for the 12.5 bar BMEP load before decreasing at each increase in WFR. Note that there is no data point for 0 WFR at 16.8 bar because this condition could not be operated without experiencing pre-ignition.

There are two mechanisms impacting efficiency in opposite manners. As mentioned, the high specific heat of water allows for high heat transfer to the water, decreasing laminar flame speed, peak HRR, and combustion efficiency. Some benefit is gained through the heat transfer to water as the intermolecular bonds are broken and the saturated water turns to vapor, expanding and creating a rise in pressure [4]. The expansion of the steam increases cylinder pressure, similar to the function of a steam engine, and increases efficiency by utilizing heat energy that would normally be expelled in the exhaust. Both mechanisms have been observed in other projects where the net impact on efficiency depends on operating mode and WI timing and method [4].

When operating the engine at a load of 16.8 bar BMEP at 1800 RPM without water, the pre-ignition or "super-knock" event shown in figure 40 occurred. Note that the injection timing shown in figure 40 is the hydraulic SOI. This event is classified as a pre-ignition event due to the pressure rise occurring before the hydraulic diesel SOI. One cycle of this pre-ignition event was captured before the engine was promptly shut down to prevent damage. This pre-ignition event becomes evident when compared to the normal cycles with peak cylinder pressures of 120 bar where the PI event reaches at least 170 bar, as shown in the figure.

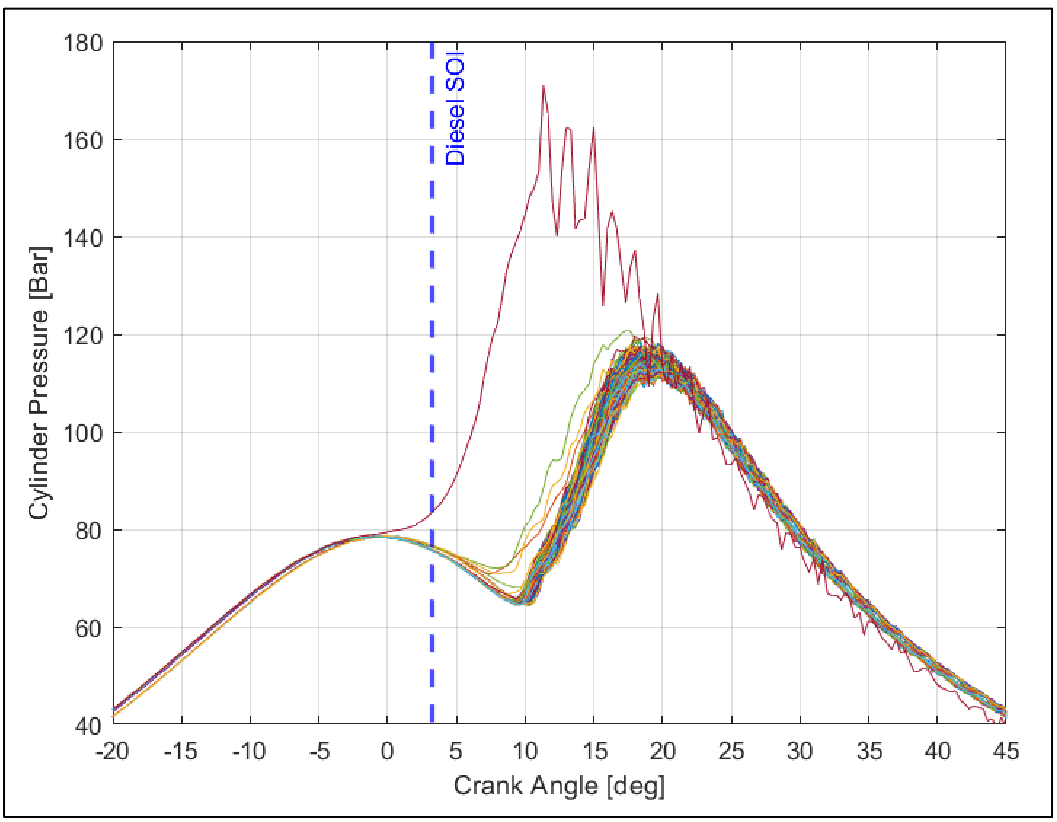


Figure 40. Cylinder Pressure Trace of Pre-Ignition Event - Test Set E

From the figure we can see that the normal combustion of the natural gas does not happen until about 7 CAD after diesel SOI. The pre-ignition event however started at TDC, giving evidence to ignition of natural gas prior to the diesel pilot ignition. This extreme pressure rise can cause catastrophic engine failure where each pre-ignition event increases cylinder temperature for the next cycle which then creates a higher likelihood of pre-ignition. This cycle of pre-ignition and cylinder temperature increase will continue until the engine is destroyed or in our case, fueling cut off and forced to a stop by the dynamometer.

With a WFR of 0.5, the 16.8 bar BMEP load was able to be run safely. Figure 41 shows the average of 300 cycles for the no-water test, excluding the pre-ignition event along with each WFR swept where no pre-ignition events were experienced for all levels of WI. Additionally, WI allowed combustion phasing to be advanced towards MBT due to mitigated knock from the cooling effect of the water, advancing SOI from 0 °bTDC at 0 WFR to 5 °bTDC at 0.5 WFR. Note the SOI shown in the figure is the hydraulic SOI which is retarded 3.2 CAD from the electronic SOI.

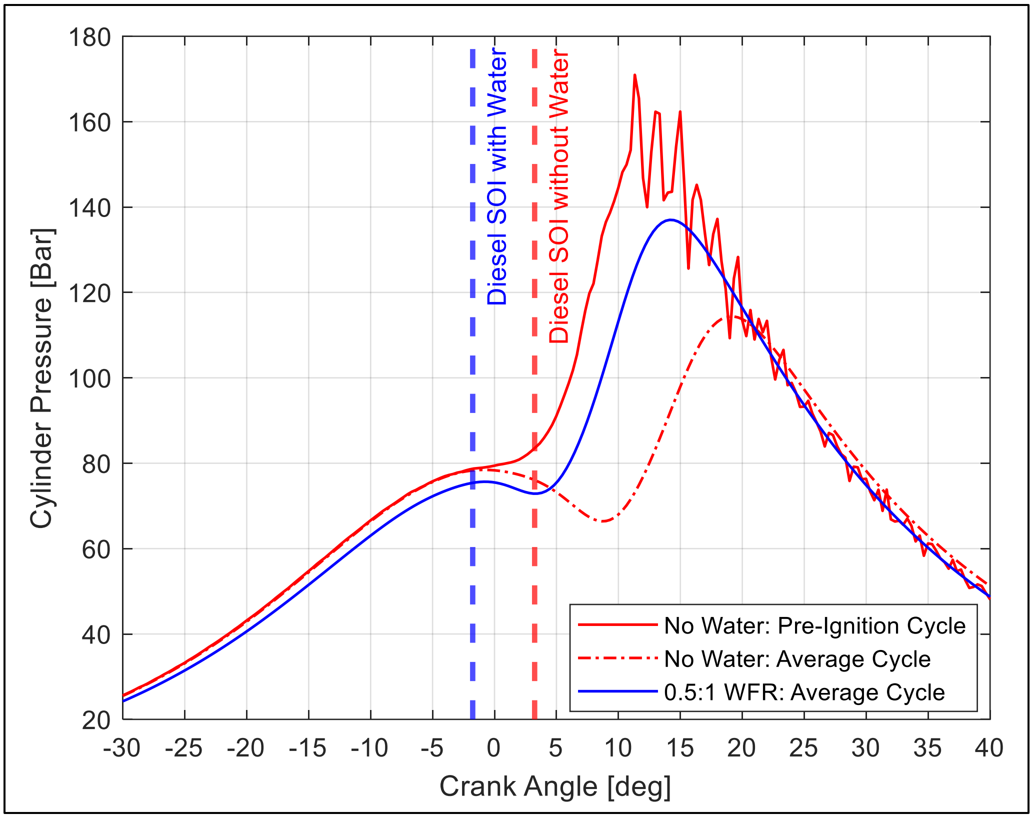


Figure 41. Cylinder Pressure - Pre-Ignition versus Normal Operation - Test Set E

In previous research conducted on natural gas dual fuel engines, high load could only be achieved safely by reducing CR, and thereby reducing BTE. In the tests summarized above, WI not only mitigated knock, but also increased BTE at certain WFRs. In addition to the 16.8 bar BMEP load achieved, a 21 bar BMEP load was achieved with a WFR of 1.0, although no data was recorded for this point. The explanation for knock mitigation by WI is obvious where the cooling effect of the water decreases the in-cylinder temperature, mitigating end-gas ignition and pre-ignition.

WI was shown to benefit both the BTE and knock of NG-DP operation, increasing BTE and decreasing knock/pre-ignition. Too much water however has a negative impact on BTE where the decrease in pressure from the cooling effect overcomes the increase in pressure from the expansion of water. The knock mitigation allows for further BTE increase as SOI advance is made possible without increasing knock to dangerous levels.

Finally, COV of IMEP remained below 3.0 for all tests conducted showing combustion stability was not an issue in results of WI in NG-DP operation.

4.2.3 Emissions

NOx, THC, CO, and PM were recorded using the AVL Smoke meter and Horiba 5-Gas Analyzer and were summarized and investigated to provide explanation for the impact of WI on emissions.

4.2.3.1 NOx

The reduction of combustion temperature is a well-known method for decreasing NOx emission whether it be through SOI retard, EGR, or IMAT decrease. WI is another method that has been used to decrease combustion temperature where the cooling effect discussed causes lower temperature combustion and thus decreased NOx formation. Figure 42 shows the NOx decrease at 12.5 bar and 16.8 bar BMEP as WFR was increased.

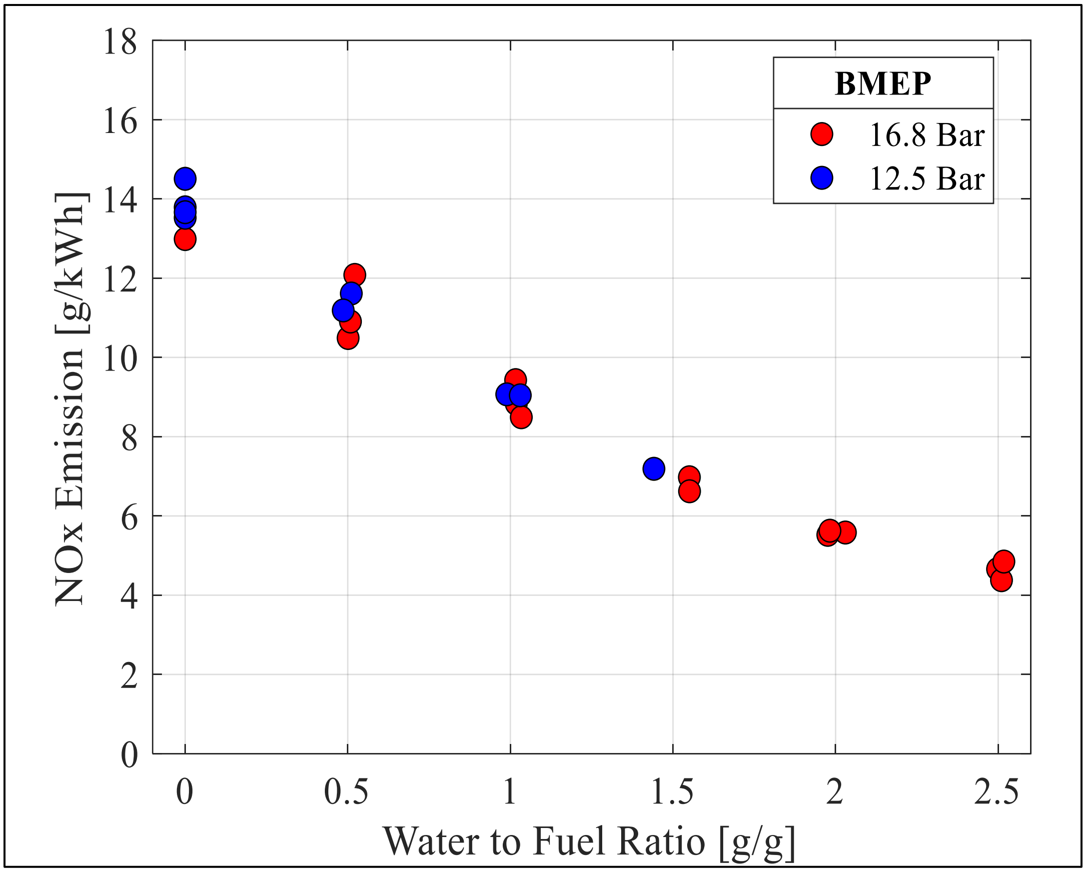


Figure 42. NOx vs WFR - Test Set E

As mentioned in previous sections, SOI was advanced with increasing WFR to match combustion phasing. Advancing SOI however generally increases in-cylinder temperatures where combustion occurs near TDC when cylinder pressure is high from compression alone, leading to increased NOx. This trend is shown in figure 43 where the diesel pilot SOI was swept from 6 °bTDC to 2 °aTDC at 12.5 bar BMEP and 1800 RPM.

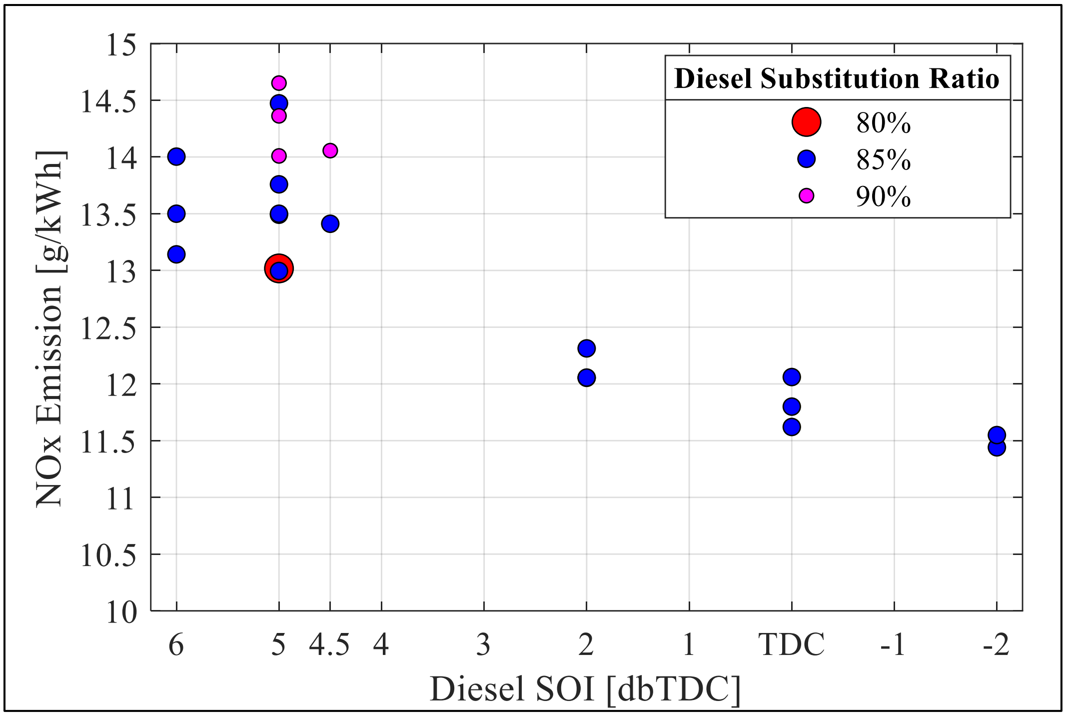


Figure 43. NOx vs SOI - Test Set E

As expected, NOx is decreased as SOI is retarded and vice versa with SOI timings of 5 and 6 °bTDC having the highest NOx emission. Notice also that an increase in DSR resulted in increased NOx formation and vice versa. This could be attributed to a more rich mixture of natural gas and air when diesel pilot quantity is decreased and AFR is maintained. Additionally, the high flame temperature of natural gas in comparison to diesel contributes to this rise in NOx as natural gas contribution is increased. As the NG and air mixture approach lambda 1, combustion temperature is increased, therefore increasing NOx formation.

Diesel SOI was also swept at various WFR for 12.5 and 16.8 bar BMEP loads. Figure 44 shows the NOx and BTE result for these tests.

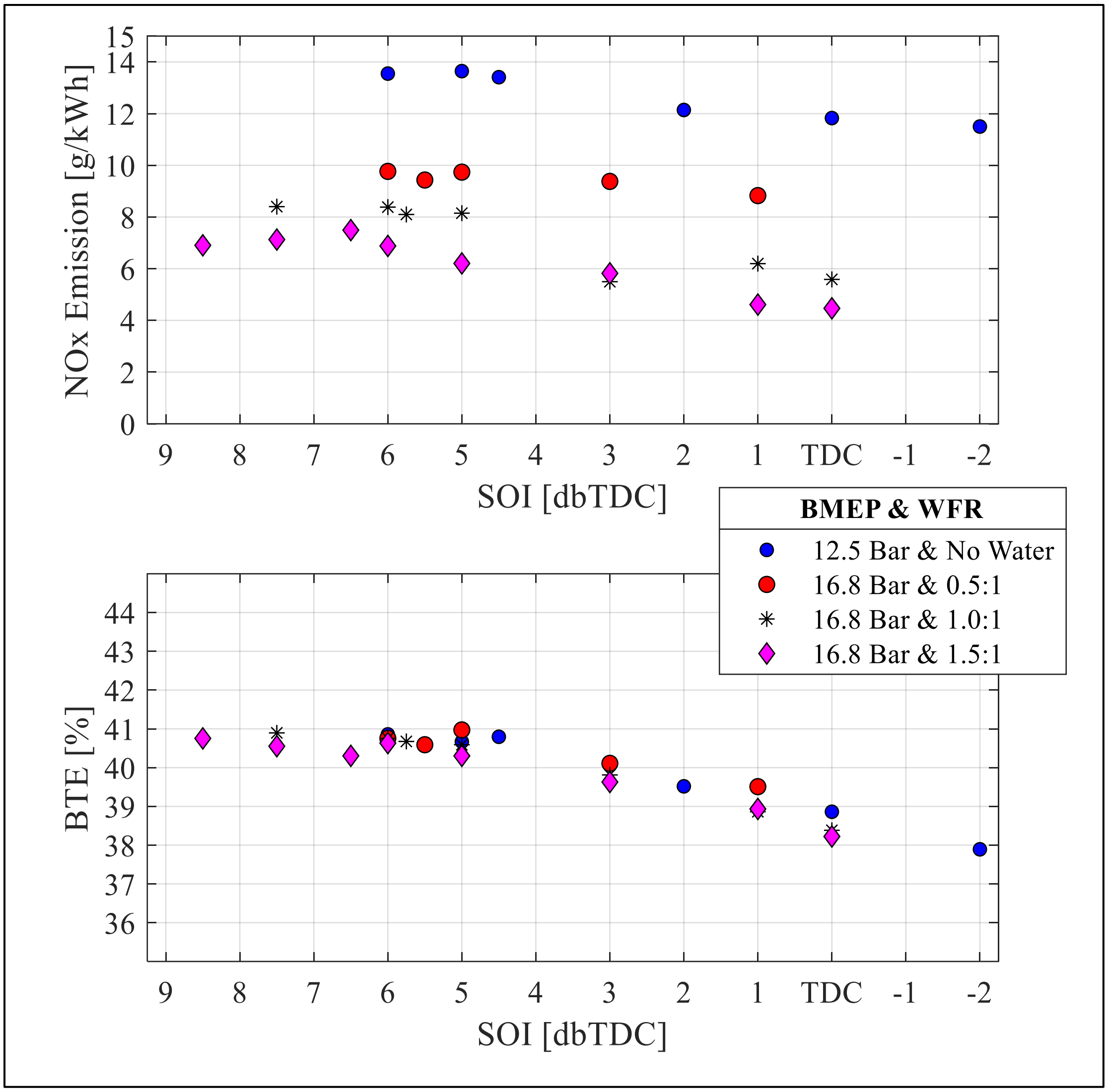


Figure 44. NOx and BTE vs SOI at Different WFRs - Test Set D & E

Advancing CA50 to reach MBT without WI results in combustion knock and an increase in NOx where WI enabled an SOI advance by mitigating combustion knock and simultaneously decreasing NOx.

4.2.3.2 Soot

Without a diesel particulate filter, particulate matter becomes a concern as it cannot be removed with a TWC. Running at stoichiometric conditions, soot levels reached 0.09 g/kWh where tier 4 EPA standards require 0.02 g/kwh [14]. Contrary to the soot increase experienced by Mohamed et al, with injected steam and this projects diesel with WI results, WI with NG-DP operation caused a decrease in PM emission [P]. Figure 45 shows this trend at 12.5 bar and 16.8 bar BMEP.

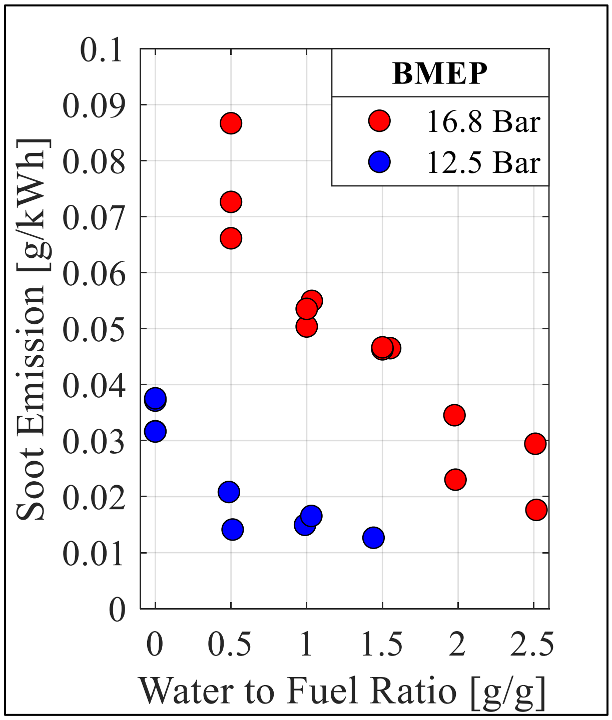


Figure 45. Particulate Matter vs WFR - Test Set D and E

This result of reduced soot with increased WI is particularly interesting in that it was expected that water would cause decreased oxygen access for the fuel, particles for soot to form on, and the cooled combustion would decrease soot burn off, all of which would increase soot emission. This finding gives an obvious advantage for WI over EGR as a NOx reduction method where EGR typically results in increased soot. The exact reason for the particulate emission decrease is yet unknown although additional tests were conducted to form several theories for this finding.

Soot formation is a complicated process to study as there are many factors that impact its formation and burn off during combustion. This fact makes it difficult to pinpoint the exact reason for the soot decrease, but by isolating known soot formation factors in combustion, we can theorize several explanations to guide further research.

The hydrogen to carbon ratio of diesel is 1.8:1 and for NG is 4:1. The larger HC ratio held by natural gas suggests that even at an 85% DSR, the diesel will contribute to soot emissions more than the natural gas due to the larger amount of carbon present. Additionally, when the natural gas enters the cylinder, it has already passed through the compressor and charge cooler, providing a homogeneous charge through sufficient mixing. The diesel pilot on the other hand is locally rich and poor mixing with the NG/air mixture could be the main driver behind the soot formation in this engine. To test this theory, SOI and DSR were independently swept.

Conventionally, as SOI is advanced, combustion has more time to take place, which in turn burns off soot. SOI was advanced as water was added to maintain a constant CA50, but this brought the possibility of the soot reduction being due to injection timing and not the WI. To show that SOI advance was not the main driver of PM decrease in NG-DP operation, SOI was swept at 12.5 and 16.8 bar BMEP for several different WFR.

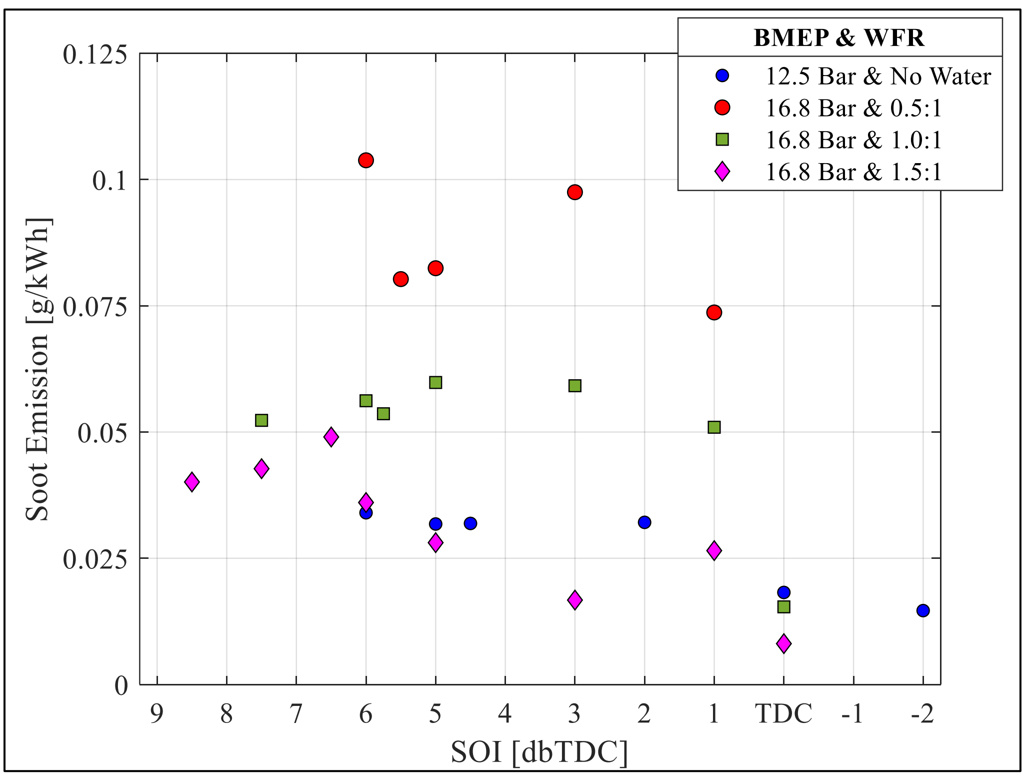


Figure 46. Soot vs SOI at Different WFRs - Test Set D & E

As SOI was advanced, little to no decrease in soot was measured where at 16.8 bar BMEP and 1.5 WFR, PM actually increased as timing was advanced. As WFR was increased from 0 to 1.5, soot decreased dramatically, further proving WI to be the main driver of PM decrease.

Although maintained constant for all WFR testing, and therefore not a culprit for the decrease in soot, DSR was swept to gain a better understanding of the diesel pilot quantities impact on PM emission. Figure 47 shows the DSR sweep at 8.4 bar BMEP and 12.5 BMEP. DSR was not swept at high loads to prevent injector tip damage and coking reported to take place at DSR great than 85% at high loads [18].

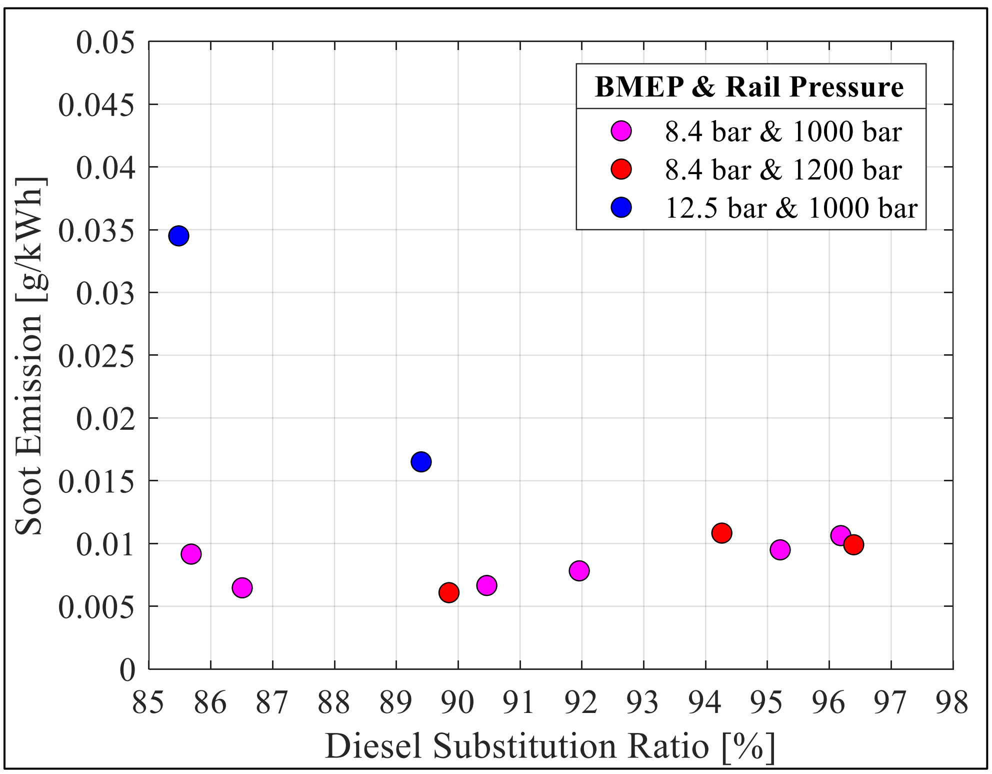


Figure 47. PM vs DSR at different loads - Test Set D & F

At both loads, PM was shown to decrease from 85% to 90% DSR. At DSR higher than 90% however, PM increased. The initial decrease in PM was expected, where previously mentioned, the diesel pilot injection is locally rich and causes high soot formation. The increase in PM at DSR greater than 90% however was unexpected but is attributed to the pilot quantity falling below the Bosch injector's minimum injection quantity, causing poor diesel atomization and therefore, increased soot formation. The diesel rail pressure was also studied in figure 47 and showed to have no significant impact on PM emission.

The investigation of the diesel pilot characteristics showed that pilot injection quantity, timing, and pressure have some impact over PM emission but is not the main driver for the PM decrease found with increasing WFR.

One common factor of PM formation is availability of oxygen. When fuel in the presence of combustion cannot access sufficient oxygen, PM can form. With increased mixing time or ignition delay, there is additional time for the diesel pilot to mix with the intake charge mixture. As shown in the figure below, WI increases ignition delay, allowing this additional mixing time and potentially reducing soot formation.

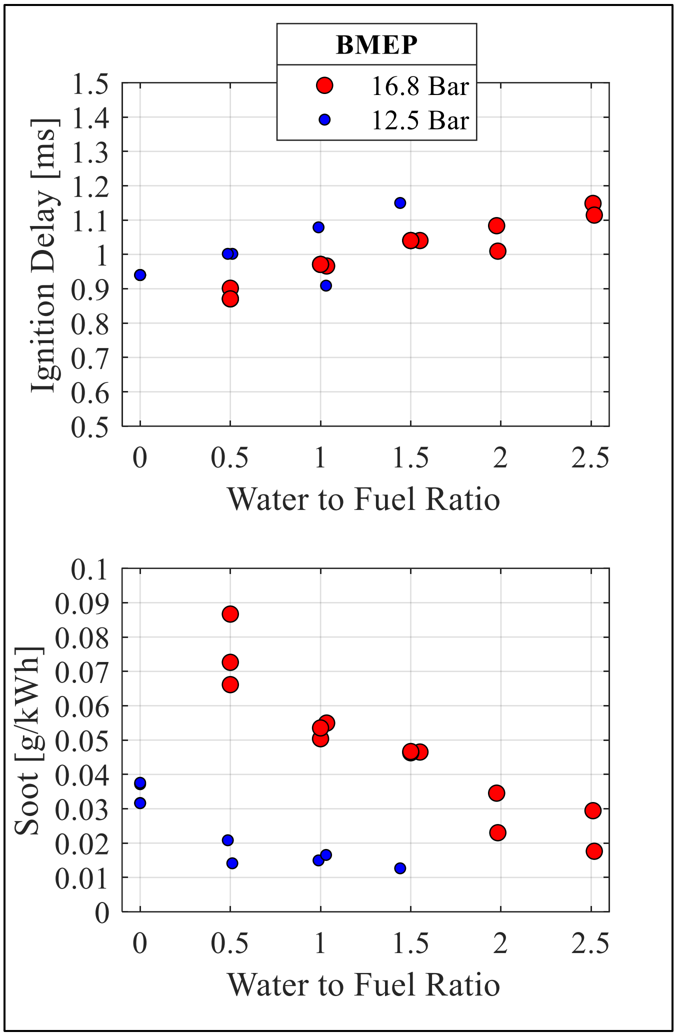


Figure 48. IGN Delay and PM vs WFR - Test Set D & E

There is an inverse relationship between ignition delay and PM emission whereas water causes ignition delay to increase and PM to decrease.

To further investigate the impact of oxygen availability on PM formation, Lambda was swept at 8.4 bar BMEP, shown in figure 49.

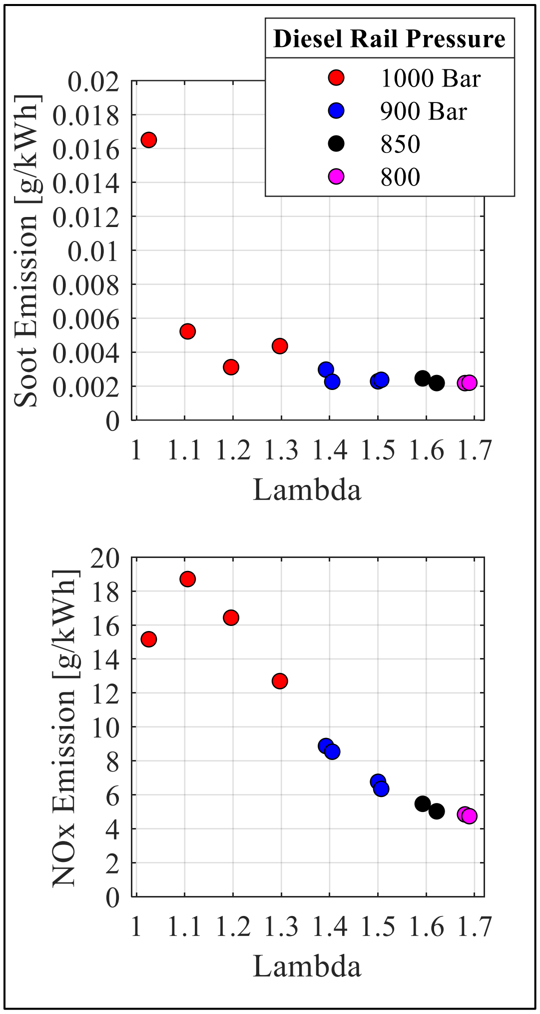


Figure 49. PM and NOx vs Lambda - Test Set I

The increase in lambda resulted in a decreased PM emission as expected. A large decrease from lambda 1.0 to 1.7 relative to the remainder of the lambda sweep was shown indicating that excess oxygen present in the cylinder charge significantly reduces PM formation. There is also a thermal mechanism to consider when evaluating PM formation. The NOx trend shown indicates that combustion temperature is increased from lambda 1 to 1.1. A high combustion temperature has two opposing impacts on PM emission, one increasing PM and the other decreasing. PM requires high temperatures to form but also is burnt off after forming when combustion temperatures are high. Regardless of the thermal mechanisms however, PM decreased significantly throughout the entire lambda sweep suggesting that oxygen availability has greater impact on PM emission.

Further investigation into combustion temperature and ignition delay provides further insight on the formation of PM. EGR was used to study these effects and was swept from 0% to 18% at 8.4 and 12.5 bar BMEP. The resulting ignition delay was plotted in figure 50, followed by the NOx and PM emission in figure 51.

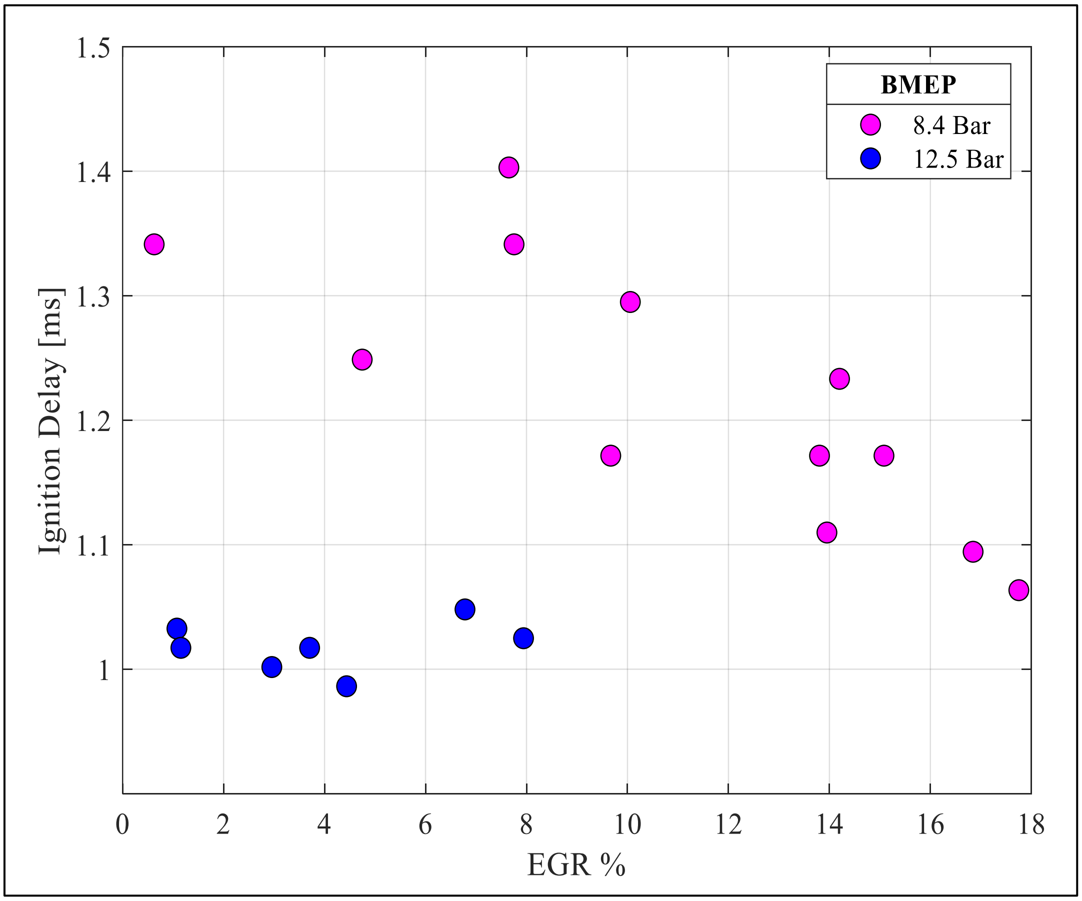


Figure 50. Ignition Delay vs EGR - Test Set D & F

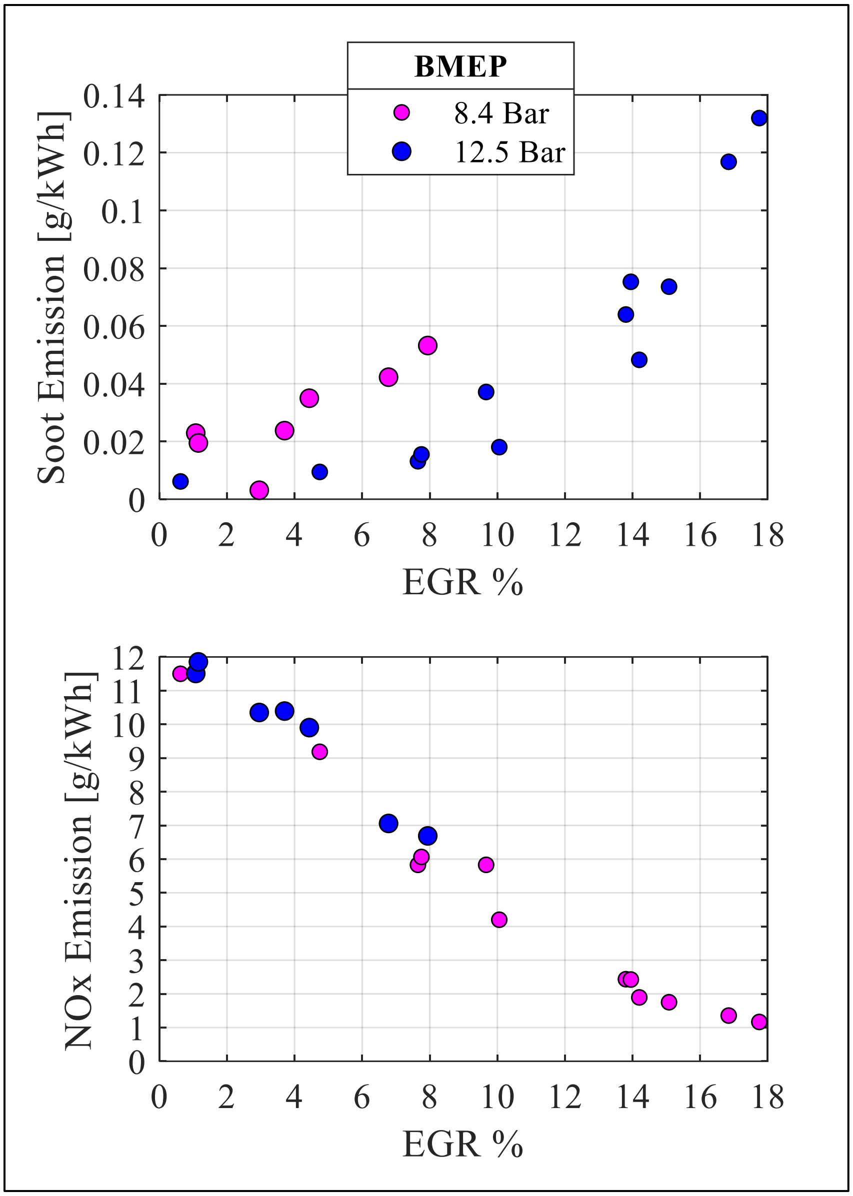


Figure 51. Soot and NOx vs EGR - Test Set D & F

At 8.4 bar BMEP, ignition delay decreased as EGR was increased. This is likely due to the increased intake manifold temperature from the EGR, causing higher cylinder temperatures and therefore igniting the diesel pilot earlier. At the 12.5 bar BMEP load however, the intake manifold temperature was maintained at 40 °C by increasing charge cooling to prevent knock. In result, the ignition delay at 12.5 bar BMEP remained relatively constant. The PM emission still increased with increasing EGR, regardless of ignition delay. This trend works against our theory that increased ignition delay decreases PM by allowing the diesel pilot to mix with the in-cylinder charge more.

The NOx and PM trends do however verify past research that utilized EGR to decrease NOx in dual fuel engines. EGR brings many other factors to consider when measuring PM and therefore the ignition delay theory cannot be ruled out. Further testing using diesel fuels with higher and lower cetane could be performed to study the impact of ignition delay more directly but was beyond the scope of this project.

In a study conducted on waters impact on PM formation in combustion, Roberts et. al summarized that there are two key mechanisms to soot formation, thermal and chemical [46]. He then went on to conclude that the chemical mechanism outweighs the thermal mechanism and stated that the additional of water to flame resulted in decreased PM because of its breakdown into O, H, H_2, and OH, increasing availability to oxygen and hydrogen radicals [46].

4.2.3.3 Hydrocarbons and Carbon Monoxide

The reduction of soot means the carbon contained in the PM must be emitted as another form unless measurement error is present. Increases in both CO and THC verify that the carbon was not “disappearing” with increased water. Figure 52 shows this increase in both emissions as WFR was increased.

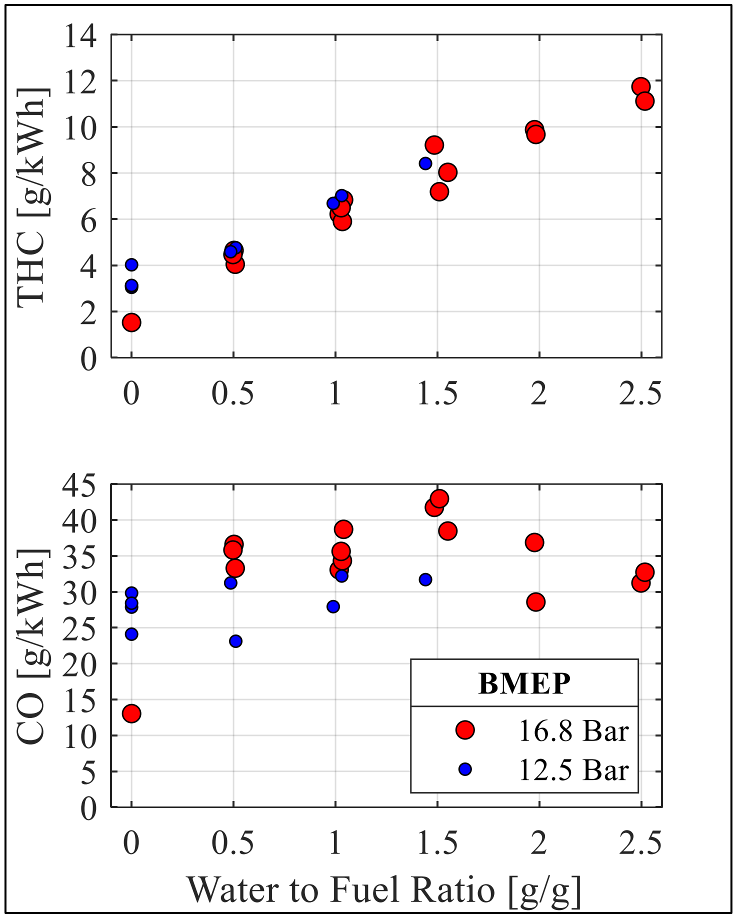


Figure 52. THC and CO vs WFR - Test Set D & E

The increase in CO & HC are both indications of decreased combustion efficiency which can be attributed to the cooling effect of water as mentioned in the combustion section. Both emissions can a be reduced significantly by the TWC meaning the switch from the non-treatable PM to the treatable CO and THC has a positive impact on tailpipe out emissions and can simplify and reduce the cost of aftertreatment systems.

4.3 Results Summary

Experimentation was conducted to investigate the impact of WI on both diesel and NG-DP operation. The significant findings are summarized below:

- In both diesel and NG-DP operation, WI increased ignition delay and decreased peak AHR causing late burn and decreased combustion efficiency. The increased ignition delay was successfully mitigated by advancing diesel pilot SOI, however the peak AHR could not be mitigated without increasing fueling.
- WI had very little impact on BTE for both diesel and NG-DP where across all WFRs swept, a maximum increase in BTE of ~1% and maximum decrease of ~2% was recorded.
- For stoichiometric NG-DP operation at 16.8 bar BMEP and 1800 rpm with an 85% DSR, a maximum efficiency of 42% was achieved with a WFR of 0.5.
- A WFR of 0.5 enabled safe NG-DP operation at 16.8 bar BMEP and 1800 rpm. When this operating point was run with a 0 WFR, pre-ignition occurred.
- Combustion knock was mitigated at all WFRs which allowed the SOI to be advanced, shifting combustion phasing to MBT.
- WI significantly reduced engine out NOx emission for both diesel and NG-DP operation.
    - From 0 to 2.5 WFR, an **81% reduction in NOx** was measured for diesel operation at 12.5 bar BMEP and 1800 RPM.
    - From 0 to 1.5 WFR, an **47% reduction in NOx** was measured for NG-DP operation at 12.5 bar BMEP and 1800 RPM.
    - From 0 to 2.5 WFR, an **63% reduction in NOx** was measured for NG-DP operation at 16.8 bar BMEP and 1800 RPM.
- WI increased engine out PM emission for diesel operation and decreased engine out PM emission for NG-DP operation. These conflicting results were attributed to the shift of the diesel diffusion burn into low temperature combustion as well as diesels locally rich mixture resulting from direct injection compared to the homogeneously mixed NG and air mixture. The soot decrease in NG-DP operation was attributed mainly to chemical effects where water breaks down into O, H, H_2, and OH in the presence of combustion temperatures however further investigation is required to confirm the reason for soot decrease.

- CO and HC both increased with increasing WFR (indicating decreased combustion efficiency) caused by lower in-cylinder temperatures from an effective heat loss to the water, which has a high specific heat capacity.
- COV of IMEP was maintained below 3.0 for all tests conducted showing WI does not cause significant combustion stability at these conditions.
    - During diesel operation, COV had a less than 0.1 change across all WFRs
    - COV remained less than 1 for all NG-DP data points with water injection

Table 13 summarizes the performance and engine out emissions of the four operating modes investigated in this paper and provides comparison to two similar NG engines.

Table 13. Results Summary and Comparison for All Operating Modes

Engine	Operating Mode	BMEP [bar]	Power [kW]	Speed [rpm]	BTE [%]	NOx [g/kWh]	PM [g/kWh]	CO [g/kWh]
Nostrum/MTU Cummins 6.7L ISB	Diesel	12.6	126	1800	40.8	3.6	0.007	1.3
	Diesel + 0.5 WFR	12.6	126	1800	40.8	2.5	0.015	1.6
	Diesel + 2.5 WFR	12.1	126	1800	39.4	0.7	0.162	5.2
	NG-DP	12.5	126	1800	40.7	13.9	0.035	27.6
	NG-DP + 1.5 WFR	12.5	126	1800	40.0	7.2	0.013	31.7
	NG-DP + 2.5 WFR	16.8	169	1800	40.3	4.6	0.024	32.0
	NG-DP + λ=1.7 + 90% DSR	12.5	126	1800	42.3	4.8	0.002	5.2
	NG-DP + 7% EGR + 90% DSR	12.5	126	1800	38.2	6.9	0.05	36.5
12L ISX SI Natural Gas[19]	NG + 17% EGR	7.9	145	1836	31.5	-	-	-
9.7L SI I6[3]	NG + 0.7 WFR + λ=1.0	3.9	50	1600	27.8	12.5	-	4.0
EPA Nonroad CI Tier 4 2014+		**-**	**75-130**	**1800**	**-**	**0.40**	**0.02**	**5.0**

Table 15 gives the calculated aftertreatment emissions based on the generalized TWC conversion efficiencies outlined in table 14 [35]. The tailpipe out emissions meeting the EPA Nonroad CI Tier 4 2014+ emission standards are highlighted in green and emissions not meeting the standard highlighted in red.

Table 14. TWC Conversion Efficiencies

Emission	Conversion Efficiency
NOx	95%
PM	5%
CO	95%

Table 15. Results Summary with TWC Emissions

Engine	Operating Mode	Power [kW]	Speed [rpm]	BTE [%]	NOx [g/kWh]	PM [g/kWh]	CO [g/kWh]
Nostrum/MTU Cummins 6.7L ISB	Diesel	126	1800	40.8	0.18	0.01	0.1
	Diesel + 0.5 WFR	126	1800	40.8	0.13	0.01	0.1
	Diesel + 2.5 WFR	126	1800	39.4	0.04	0.15	0.3
	NG-DP	126	1800	40.7	0.70	0.03	1.4
	NG-DP + 1.5 WFR	126	1800	40.0	0.36	0.01	1.6
	NG-DP + 2.5 WFR	169	1800	40.3	0.23	0.02	1.6
	NG-DP + λ=1.7 + 90% DSR	126	1800	42.3	0.24	0.002	0.3
	NG-DP + 7% EGR + 90% DSR	126	1800	38.2	0.35	0.05	1.8
12L ISX SI Natural Gas[19]	NG + 17% EGR	145	1836	31.5	-	-	-
9.7L SI I6[3]	NG + 0.7 WFR + λ=1.0	3.87 bar	1600	27.8	0.625	-	0.2
EPA Nonroad CI Tier 4 2014+		**75-130**	**1800**	**-**	**0.40**	**0.02**	**5.0**

The findings of this paper open several new avenues for experimentation. The following points are suggestions for conducting future research on a NG-DP engine:

- **PM Emission Investigation**

The reduction of PM with WI in NG-DP operation requires further investigation in determining the mechanism responsible for the decrease. The increased ignition delay and additional mixing time from WI is one suspected mechanism and could be isolated by testing with diesel fuels of different cetane numbers. Additionally, IMAT could be swept to study the impact in-cylinder temperature has on PM emission. Finally, more insight to the PM formation could be gained with the use of a PM particle size measurement device where this project only measured the concentration of PM in the exhaust. Finally, NG-DP studies could be conducted in a combustion vessel to use imaging in determining soot formation and burn off trends.

- **Combustion Chamber Design/Optimization**

As discussed in the literature review section, dual fuel operation typically results in high HC and CO from crevice volume and escaped unburnt fuel during the gas exchange process. A combustion chamber optimization including decreased crevice volume and intake/exhaust port geometry minimizing transfer from the intake to exhaust port during valve overlap. Additionally, the diesel piston crown could be optimized for NG-DP combustion.

- **Water injection Optimization**

There are many different WI parameters that could be investigated further. WI pressure and timing could be optimized as well as smaller increments in water quantity, testing at a WFR lower than 0.5.

- **Water Vaporization Study**

Determining the phase of the injected water throughout the combustion cycle using a program such as EES would be useful in analyzing the mechanisms at work effecting combustion and emissions. A similar study was conducted by Worm et. al for a water injected GDI engine [48].

- **MBT Exploration**

During the NG-DP testing stage, MBT was found by adjusting the diesel SOI to maximize efficiency for the target load while fueling was adjusted to maintain that load. In future testing, true MBT should be determined by sweeping SOI to find maximum load with constant fueling.

- **Water Impact on Three-Way Catalytic Converter**

There are possible decreases in TWC conversion efficiency due to excessive water vapor in the exhaust. Tests should be conducted with a TWC converter and emission measurements taken before and after to determine if conversion efficiency is impacted. EGT was decreased with WI but not low enough to impact the efficiency of the TWC. Impacts made from water vapor in the exhaust would be chemical.

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A Veristand Monitors and Controls

As mentioned, NI Veristand was used to both monitor and control engine and test cell instrumentation. The controlled and monitored instrumentation are outline in tables 16 and 17 below.

Table 16. NI Veristand Monitored Instrumentation

Measurement	Unit	Sensor
Turbine Inlet Temperature	°C	k-type thermocouple
Turbine Outlet Temperature	°C	k-type thermocouple
EGT 1 – 6	°C	k-type thermocouple
Engine Coolant Temperature	°C	k-type thermocouple
Intercooler Temperature Inlet	°C	k-type thermocouple
Oil Sump Temperature	°C	k-type thermocouple
Intake Manifold Temperature	°C	OEM Sensor
EGR Outlet Temperature	°C	k-type thermocouple
Intercooler Temperature Outlet	°C	k-type thermocouple
Coolant Inlet Temperature	°C	k-type thermocouple
Diesel Fuel Inlet Temperature	°C	k-type thermocouple
Turbine Inlet Pressure	kPaa	piezo-resistive pressure transducer
Turbine Outlet Pressure	kPaa	piezo-resistive pressure transducer
Compressor Inlet Pressure	kPaa	piezo-resistive pressure transducer
Compressor Outlet Pressure	kPaa	piezo-resistive pressure transducer
Measurement	**Unit**	**Sensor**

Oil Filter Pressure	kPaa	piezo-resistive pressure transducer
Manifold Absolute Pressure	kPaa	OEM Sensor
Crankcase Pressure	kPaa	piezo-resistive pressure transducer
Diesel Fuel Inlet Pressure	kPaa	piezo-resistive pressure transducer
NG Supply Pressure	Psi	piezo-resistive pressure transducer
Main Regulator Delta Pressure	Inches w.c	ProSense Air Differential pressure transmitter
Sub-Regulator Delta Pressure	Inches w.c	ProSense Air Differential pressure transmitter
NG Mixer Inlet Pressure	Inches w.c	ProSense Air Differential pressure transmitter
Vacuum Chamber Pressure	Inches w.c	ProSense Air Differential pressure transmitter
Ambient Pressure	kPaa	piezo-resistive pressure transducer

Table 17. NI Veristand Controlled Instrumentation

Control	Function
Cell Power	Enable Powe to Cell
Engine Power	Enable Power to Engine
Dyno Reset	Reset Dynamometer
Dyno Controller	Enable Dynamometer Controller
Dyno Unit	Turn on power to the dynamometer
Dyno Fan	Turn on dynamometer fan
Dyno RPM	Set desired dynamometer RPM

Dyno Enable	Enable the dynamometer to spin
H20 Cart Power	Turn on power to H2O cart pump
NG Valve 1	Open/Close NG Valve 1
NG Valve 2	Open/Close NG Valve 2
NG Valve 3	Open/Close NG Valve 3
Coolant Inlet Temperature	Adjust Desired Coolant Inlet Temperature
Charge Air Temperature	Adjust Desired Charge Air Temperature
Diesel Fuel Inlet Temperature	Adjust Desired Diesel Fuel Inlet Temperature

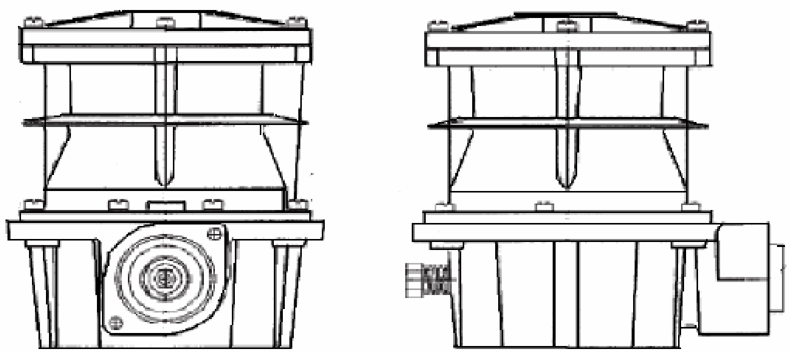
B NG Mixer Specifications

Specifications for the 400VF NG Mixer utilized in this project were provided by IMPCO in their “Master Parts Catalog” are shown below.

IMPCO

SECTION 1: CARBURETORS & MIXERS
400VF SERIES

MODEL 400VF STANDARD CARBURETOR



TECHNICAL DESCRIPTION

The 400VF Series Mixer is a concentric flow type air valve carburetor. A precision gas metering valve is attached to the air valve. The air valve and gas valve motion is linear in relation to the volumetric flow. The air fuel ratio is controlled by the adjustable power valve, the shape of the gas metering valve and the pressure from the gas regulator. The gas valve and matching jet is sized for the gas to be used. There are several jet sizes to adapt to various fuels from propane to landfill gas. Gas valves are shaped to provide A/F versus flow characteristics defined by each customer.

TECHNICAL SPECIFICATIONS

Fuel Type	
Propane	50-50 Propane Butane
Natural Gas	85% + Methane
Digester Gas	60-72% Methane
Landfill Gas	45-60% Methane
Gas Inlet Normal	5"H2O over Air Inlet (Idle Setting)
Air Inlet Maximum	517 kPa (75 PSI)
Back Fire Peak	1379 kPa (200 PSI)
Air Flow Capacity	500 CFM @ 2" HgΔP (T2-7 @ WOT)
Max Operating Temp	121° C (250° F)
Max Soakback	160° C (320° F)

Figure 53. IMPCO Mixer Specifications

C In-Cylinder Pressure Transducer Mounting

The mounting sleeve designed and manufactured to mount the Kistler transducers safely inside the cylinder head is shown in the technical drawing in figure 54.

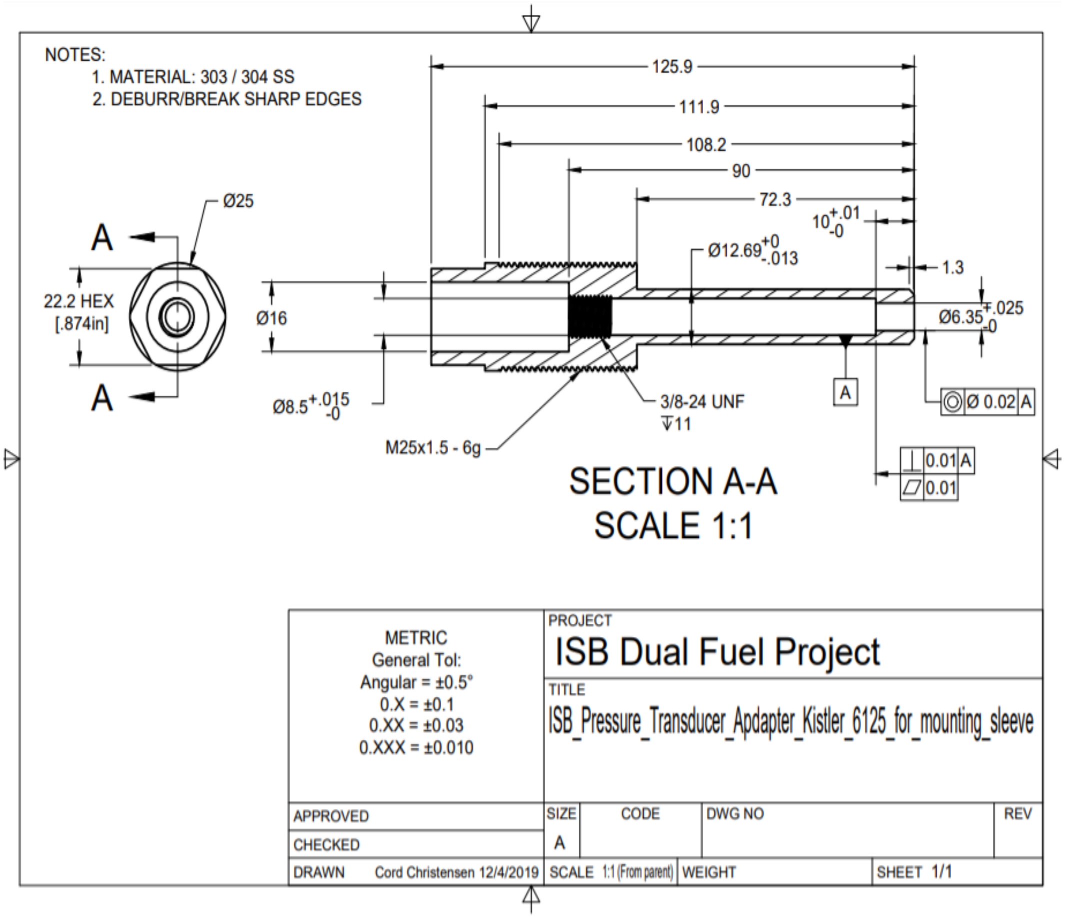


Figure 54. Transducer Adapter Sleeve Drawing

D Fuel Additive Specifications

Specifications for the diesel additive used to prevent injector coking are provided in figures 55 and 56.

Lubrizol

Product	Type
POWERZOL™ 9048	Aftermarket Diesel Fuel Additive

Designed for regular use in diesel fuel to prevent and reduce fuel injection system deposits.

Application

Use 2.6 oz per 100 gal (80 mL per 400L) of diesel fuel.

Recommended for use at: 200 ppm v/v

for regular use in diesel engines to prevent and gradually clean up fuel injector system deposits.

Physical Characteristics

	Minimum	Target	Maximum
FLASH POINT, C, PMCC		65	
LBS PER U.S. GAL @ 15.6 C		7.29	
LBS PER IMP GAL @ 15.6 C		8.75	
POUR POINT, C		-54	
SPECIFIC GRAVITY @ 15.6 C	0.854	0.874	0.894
VISCOSITY @ 40 C, CST		81	

Benefits

POWERZOL™ 9048 is a diesel fuel additive formulated with the revolutionary new Zer**0** Series diesel deposit control additive.

- Effective on nozzle deposits and internal injector deposits (IDID)
- Keeps fuel system clean
- Prevents power loss due to fouled injectors
- Helps restore engine performance with regular use

Figure 55. Diesel Additive Specification Sheet #1

POWERZOL™ 9048 Unloading, storage and blending instructions

General handling instructions - In general, The Lubrizol Corporation recommends, as a minimum, the use of neoprene or nitrile rubber gloves and safety glasses or chemical splash goggles. The Material Safety Data Sheet should be consulted for specific information and for information on health and safety when handling this product.

Fire and explosion hazard data			
	Flash Point (method)	Classification	
	65°C PMCC	Combustible	
Temperature recommendations			
Unloading	Pumping Temperature	Ambient°C	°F
	Maximum temperature*	50°C	122°F
Storage Maximum temperature for long-term storage		45°C	113°F
Blending Maximum base oil temperature for mechanical or in-line mixing		50°C	122°F
Equipment recommendations			
Type of Pump	Positive Displacement		
Type of transfer line	Ball Launched, Insulated,Steam Traced Using 107°C/225°F Steam Max.		
Transfer line size	2-3inch/5-8 cm.		
Heat source			
Type	Steam 107°C/225°F Max.		
Storage tank	Suction Heater Recommended in Cold Climated Only		
Viscosity data	**cSt**	**SUS**	
at 0°C,32°F	762	3517	
at 25°C,77°F	164	759	
at 40°C, 104°F	81	376	
Notes			
Pour Point	-54°C,-65°F		
Low Flash Point-Use caution when handling this material			

Figure 56. Diesel Additive Specification Sheet #2

E Water in Oil Investigation

Preceding all test sessions, the oil was checked to ensure excessive water was making its way into the crankcase as this has been a common issue in previous WI studies [48]. Checking the oil with the dipstick and draining the oil pan after approximately 100 hours of WI runtime showed no signs of water in the oil. Removal of the valve cover and oil filter however showed trace signs of "creamy" oil as shown in the figures below.

Although no oil was found in the crankcase it is clear that water was present in the oil which could cause engine wear issues over time. This finding requires further investigation into what WFR cause water in oil, why no oil is being found in the crankcase, and how much water is actually making its way into the oil. Figures 57, 58 and 59 show the "creamy" oil found.

Figure 57. Oil Filter Investigation

Figure 58. Oil Fill Cap with Creamy Oil

Figure 59. Closeup of Creamy Oil in Oil Filter