THESIS

INVESTIGATION OF SUSTITUTION LIMITS AND EMISSIONS OF AN IN-LINE SIX CYLINDER DIESEL DERIVED DUAL FUEL ENGINE

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ABSTRACT

INVESTIGATION OF SUSTITUTION LIMITS AND EMISSIONS OF AN IN-LINE SIX CYLINDER DIESEL DERIVED DUAL FUEL ENGINE

New drilling techniques have expanded the availability and decreased the cost of oil and natural gas. This has resulted in an increase in drilling activity. Historically, power for drilling and hydraulic fracturing operations is supplied by diesel engines. The power demand requires large quantities of diesel fuel to be trucked to the well site. In an effort to reduce operating costs and disturbances to neighboring communities, natural gas is being utilized in these processes. One approach for achieving this is using natural gas from the local pipeline and routing it to the engine intake to convert the engine to a dual fuel engine. Some of the natural gas is thereby substituted for diesel in the combustion process reducing diesel consumption. Natural gas costs average around one sixth of the cost on a diesel gallon equivalent and the pipeline reduces the number of truck-loads of diesel fuel. However, there are limits to the amount of natural gas that can be substituted for diesel.

This work examines the different substitution limits that exist for natural gas-diesel dual fuel engines across their load range. A John Deere 6.8L Tier II diesel engine was converted to a dual fuel engine with a commercially available dual fuel retrofit kit. The dual fuel kit was originally commissioned as is typical of field operation based on the dual fuel kit supplier's previous commissioning experience. After the baseline commissioning, the dual fuel combustion process was examined with combustion pressure measurements and exhaust gas sampling. The testing showed engine knock caused by end gas auto-ignition limits substitution at high loads. High substitutions at low loads can cause governor instability but this limit exists beyond where

the emissions levels become unacceptable. Two methods were also investigated for their effectiveness on extending the substitution limits. These were adjusting the diesel injection timing and air manifold temperature. Retarding the injection timing was able to increase substitution levels by approximately 4% at full load while avoiding engine knock. Lowering the air manifold temperature was more successful, increasing the substitution levels by around 10% compared to the baseline commissioning. Advancing the timing at low loads was not able to achieve any significant benefits in allowable substitution. Preheating the incoming charge was able to increase substitution levels by 20% at 25% load.

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1. Introduction

1.1 Compression Ignition Engines and Emissions

Internal combustion engines still hold a high level of importance in power generation in transportation and stationary forms. The most common types are gasoline and diesel engines but interest in alternative fuels has increased. The interest is fostered by availability and environmental reasons associated with fuels developed from crude oil. Natural gas is becoming one of the most commonly used alternative fuels because of its availability and relative cleanliness. Dedicated spark ignited engines which operate exclusively on natural gas are being produced by original equipment manufacturers (OEM's). Another option to utilize natural gas is to use it in conjunction with diesel as a dual fuel engine.

Diesel engines are the most widely utilized engines because of their ability to achieve the highest efficiencies. Increasing the compression ratio has been shown to increase power and improve efficiency [1]. Spark ignited engines typically have compression ratios of 8-12 while diesel engines range from 12 to as high as 24. However, engine instability at high compression ratios can decrease efficiency as the compression ratio is increased too high. In gasoline engines, the limitation to compression ratio is abnormal combustion often referred to as knock because of audible sound caused by the event. In spark ignited engines knock is typically caused by the fuel-air mixture in front of the flame spontaneously igniting before the flame reaches the region. This form of knock is termed end gas auto-ignition. This causes erratic pressure waves that can cause damage to the engine. Other abnormal combustion events such as pre-ignition or surface ignition can cause end gas auto-ignition. The direct injection techniques used on compression ignition engines control the energy release because of the time required for fuel and air to mix. Diffusion rates slow combustion process allowing compression engines to have higher compression ratios than gasoline engines while avoiding knock.

Another limiting factor for internal combustion engines design is emissions. Emissions from combustion are known to cause health problems as well as environmental changes. The limits on emissions have been steadily reduced in the US and around the world. The US standards for diesel engine emissions are set by the Tier system. Currently most new diesel engines sold in the US are Tier 4. Tier 4 standards for non-road compression ignition engines are outlined in Table 1 [2]. The products of combustion for ideal are carbon dioxide and water. Nitrogen, which makes up a large portion of air, would not react. The general equation for combustion of hydrocarbon fuels is given as [3]:

$$C_xH_y + a(O_2 + 3.76N_2) \rightarrow xCO_2 + \frac{y}{2}H_2O + 3.76 * aN_2 \tag{1}$$

However, chemical kinetics, which are exponentially affected by temperature, play a large role in the process. This leads to other species in the exhaust such as NO, NO_2 (collectively NOx), CO, OH, H2, O2 and other species.

Table 1: Tier 4 Emissions Standards in g/kW-hr for non-road compression ignition engines.

Maximum engine power	Application	PM	NO_x	NMHC	NO_x + NMHC	CO
kW <19	All	20.40			7.5	36.6
19 ≤kW <56	All	0.03			4.7	45.0
56 ≤kW <130	All	0.02	0.40	0.19		5.0
130 ≤kW ≤560	All	0.02	0.40	0.19		3.5
	Generator sets	0.03	0.67	0.19		3.5
kW >560	All except generator sets	0.04	3.5	0.19		3.5

Particulate matter (PM) consists of solid carbon particles, or soot, with large hydrocarbon molecules absorbed on the surface. It can cause respiratory illness by obstructing air passages and introducing toxins from the hydrocarbons absorbed by the particles [4]. Acetylene and polycyclic aromatic hydrocarbons are widely accepted as the molecules formed before particulates [1]. From these molecules, nucleation can occur in fuel rich regions. Soot formation is theorized to begin with a carbon to oxygen ratio greater than one, which can be shown to relate to an equivalence ratio of 3 [1]. Direct injection of diesel creates locally rich regions suitable for soot formation. Particles sizes can increase from both growth and coagulation processes as hydrocarbons deposit on the soot nuclei and multiple nuclei agglomerate. This process is countered by oxidation reactions that can produce gaseous combustion products.

Diesel engines are notorious for producing high levels of oxides of nitrogen. These are different variations of oxygen and nitrogen, with the two major components being NO and NO_2. These emissions are collectively referred to as NO_x. NO is converted to NO_2 in the atmosphere, which reduces visibility by forming smog and contributing to acid rain [4]. NOx also harms the lungs by forming acid and attaching to hemoglobin so the blood cannot carry oxygen throughout the body [4]. Most fuel have low nitrogen content so the accepted production of NOx is from atmospheric nitrogen [1]. The chemical reactions responsible for NOx formation were presented by Zeldovich. The Zeldovich NOx mechanism is outline in Equations 2-4.

$$O + N_2 \Leftrightarrow NO + N \tag{2}$$

$$N + O_2 \Leftrightarrow NO + O \tag{3}$$

$$N + OH \Leftrightarrow NO + H \tag{4}$$

The activation energy for these reactions are high, which makes NOx formation exponentially dependent on temperature [1]. The reaction rates also dependent on oxygen concentrations. This is typically related to the equivalence ratio. The equivalence ratio relates the fuel to air ratio to the fuel to air ratio of stoichiometric combustion. Lean equivalence ratios mean there is excess fuel, while rich equivalence ratios mean there is excess fuel compared to stoichiometric combustion. It has been shown that NOx formation is greatest at slightly lean conditions and decreases as the equivalence ratio becomes either leaner or richer [1].

Carbon monoxide affects the lungs in a similar manor to NOx. It also deprives the body of oxygen by attaching to hemoglobin, which transports oxygen throughout the body [4]. Carbon monoxide is produced in the intermediate steps of normal hydrocarbon combustion. A general HC combustion process is presented by Heywood as

$$RH \rightarrow R \rightarrow RO_2 \rightarrow RHCO \rightarrow RCO \rightarrow CO \rightarrow CO_2 \tag{5}$$

where r is a hydrocarbon radical [1]. Heywood also notes the oxidation from CO to CO_2 is a slower reaction than the earlier stages. CO emissions are typically associated with rich burning engines since there is more fuel than oxygen for stoichiometric combustion. This results in some of the hydrocarbon fuel unused, stopping reactions at the CO stage. Diesel engines operate lean overall so CO emissions from diesel engines are usually low.

Complete stoichiometric combustion theoretically only produces H_20 and CO_2. All of the hydrocarbons should be disassociated to the equilibrium exhaust products. Hydrocarbons in the exhaust are thus due to incomplete combustion. Hydrocarbons are pollutant emissions regulated due to their reactivity in the atmosphere to create smog [1]. Methane does not react to form smog, which is why the regulated exhaust emission class presented in Table 1 is non-methane hydrocarbons (NMHC). Incomplete combustion can be caused by flame quenching at different locations in the combustion chamber. The cylinder walls are lower temperatures than the combustion process so a flame approaches the wall, it acts as a heat sink significantly slowing the combustion reactions [1]. Unburned hydrocarbons can also be exhausted due to crevice volumes or hydrocarbons absorbed into the oil on the cylinder wall. Diesel engines use direct injection of the fuel so less of the burning occurs near cylinder walls and crevice volumes. The unburned hydrocarbons from diesel engines are therefore determined by the cylinder conditions. The fuel air mixture can become too rich or too lean for combustion. During the injection process the fuel and air become stratified, to different extents. Some regions can overmix becoming too lean and other regions that mix slow are too rich for combustion [1].

Both methane and CO_2 are not regulated as pollutant emissions. However, both of these exhaust products are known greenhouse gases. Growing concerns over environmental impacts from greenhouse gases is likely to introduce regulations on these in the future. The only way to reduce the amount of carbon dioxide produced is to increase the fuel efficiency so more work is achieved from the same amount of fuel used or to burn fuels with a higher hydrogen to carbon ratio (H/C). The H/C ratio is given by y/x from equation (1) and increasing the ratio will decrease the percentage of carbon dioxide present in the exhaust.

Recent advances in technologies for diesel engines have enabled them to meet the EPA standards [5] [6] [7] [8] [9]. Many of the solutions are expensive, especially after treatment solutions. New techniques to reduce emissions without the use of after treatment are still being studied. Examples of methods for reducing emissions are electronic injectors with a multi-shot injection, exhaust gas recirculation, particulate filters, oxidation catalysts and selective catalytic reduction devices (SCR).

1.2 Natural Gas Resources

Fossil fuel resources are nonrenewable so their use in internal combustion engines will eventually be phased out. However, the 2015 Annual Energy Outlook predicts the energy consumed will remain largely from petroleum and natural gas until 2040 [10] so the use of fossil fuel will remain prominent for years to come as shown in Figure 1. One of the changes predicted through 2040 is the amount of natural gas consumed for producing energy is predicted to increase and therefore reduce some of the petroleum usage. Figure 1 shows a slight increase in renewable energy sources and the slight increase in natural gas. Although fossil fuels will be phased out it is not predicted to be a fast transition. Natural gas is a fuel used during the transition during the phase out to reduce global warming effects [11]. Natural gas is still a fossil fuel but it has a larger H/C ratio than diesel or gasoline, which will reduce the carbon dioxide emissions when it is burned. The study by Howarth examined the practicality of natural gas as a transitional fuel based on its emissions during extraction [11]. Methane has a significantly higher global warming potential at 25 times that of carbon dioxide over 100 years [12] so if too much methane slip occurs during exploration, storage, transmission or use of the resource the benefit of using natural gas over diesel or gasoline is lost.

The motivation for Howarth's study is the increase in natural gas production from unconventional sources has been proposed to have higher emissions than conventional wells. The use of hydraulic fracturing and horizontal wells to access these unconventional sources is one reason why production is increasing. The United States is transitioning from a net importer of natural gas to a net exporter of natural gas [10], which is lowering the cost of natural gas. The increase in natural gas production is because of non-conventional sources including tight gas, shale gas and coal bed methane [13]. A schematic of oil and gas resources is shown in Figure 2. Tight gases are generally found in dense sandstone deposits while shale gas is located in sedimentary rock. Both compositions of rock have a low permeability which was why they were previously thought to be resources which were inaccessible.

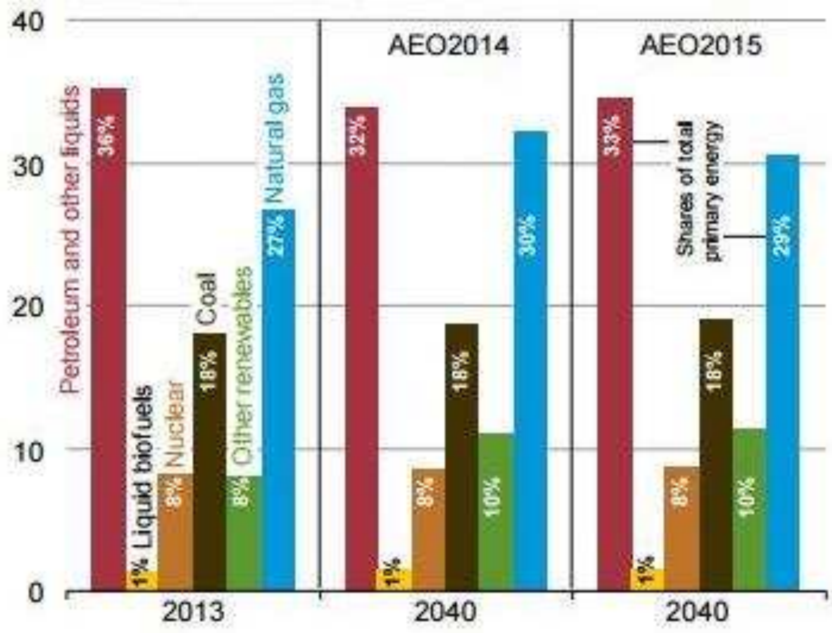

Figure 1: Primary Energy Consumption by fuel predicted by the annual energy outlook (quadrillion Btu) [10]

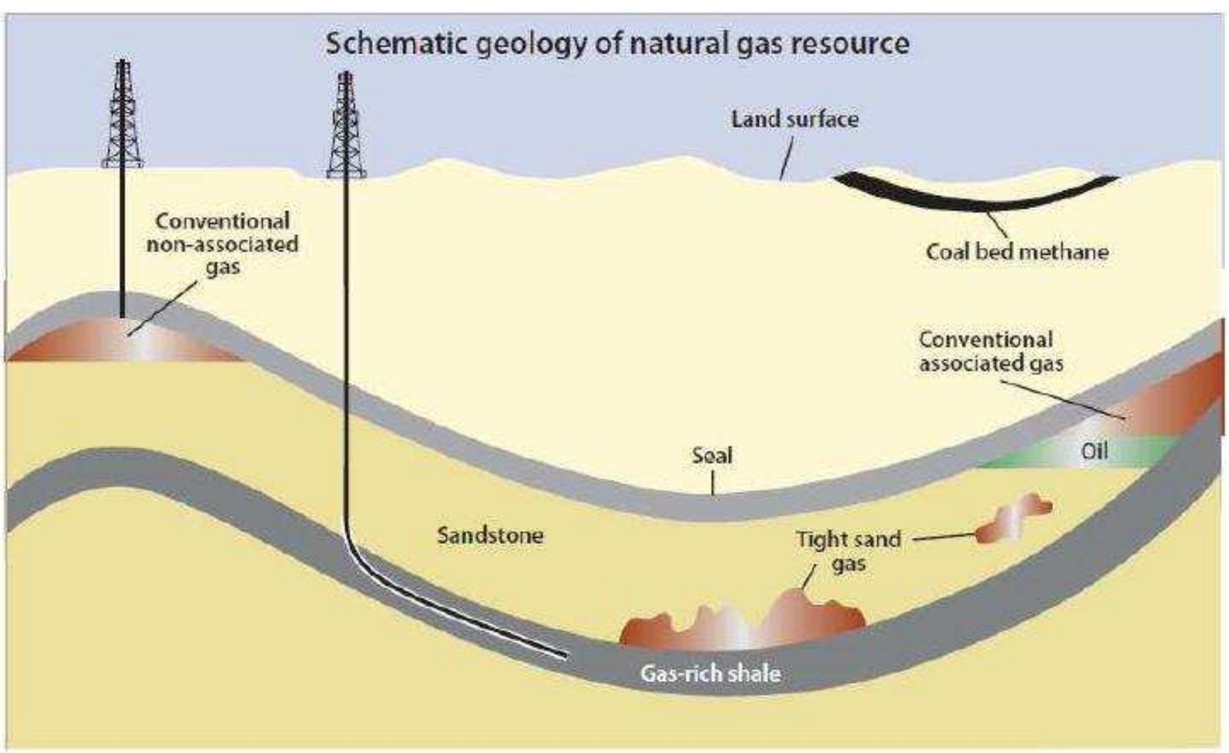

Figure 2: Types of natural gas resources based on location [13]

Creating fractures in the rock is required to provide permeability so the gas can be brought to the surface through the well. Natural fractures have been sources of shale gas for years but hydraulic fracturing, which causes artificial fractures, allows for more successful wells.

1.3 Dual Fuel Applications

The increased production of natural gas has increased the availability of the fuel in the United States and is reducing the cost. This has led to increased uses for the resource beyond its typical use for heating. This interest in expanding the uses of natural gas based on availability and the cost are some of the motivations behind using natural gas in internal combustion engines. Original equipment manufacturers have developed natural gas dedicated spark ignited engines. The natural gas is typically mixed with the air stream before the intake manifold and then ignited by spark plugs with a similar design to gasoline engines. Some larger spark ignited natural gas engines use direct injection or port injection. Another option for using natural gas is to retrofit diesel engines with a dual fuel kit. On these engines, the diesel engine is modified to have natural gas fumigated into the air stream. The natural gas enters the cylinder with the air and is ignited by the diesel injection.

One application that is utilizing this type of dual fuel combustion is hydraulic fracturing at natural gas well sites. The power demand to create a hydraulically fractured horizontal well is larger than a conventional petroleum or natural gas well. A large amount of power is required to drill the wells, pump the fracking fluid and provide electrical power to the control stations [14]. The well sites are not typically connected to the utility grid so this power is usually supplied by diesel engines. However, diesel fuel can be expensive posing high operating costs.

Natural gas has a similar energy density by mass and is lower cost for an equivalent amount of energy [15] [16]. Even with the recent decline in the price of diesel, it is still a 6:1

cost ratio over natural gas. The history of the diesel to natural gas cost ratio is shown in Figure 3. It is six times more expensive to operate the engine on diesel, as it would be to run on natural gas. The engine cannot be run entirely on natural gas but high substitution rates can be achieved which still provide significant cost reductions. The sites typically have a natural gas pipeline in the vicinity. Depending on the distance to the pipeline and pipeline ownership, nearby natural gas may be routed to the site for use in duel fuel engines. The amount of diesel consumed is less when operating as a dual fuel engine which means less diesel has to be hauled to the site by tanker trucks. This results in less surface traffic of these trucks that disrupt the neighboring communities surrounding the well. If natural gas from a nearby pipeline is not available, liquefied natural gas can be transported to the site. This is typically more expensive than local natural gas, but still results in a savings over diesel. The diesel to LNG cost ratio varies between 1 to 4 for a diesel gallon equivalent with an average ratio of 2.76.

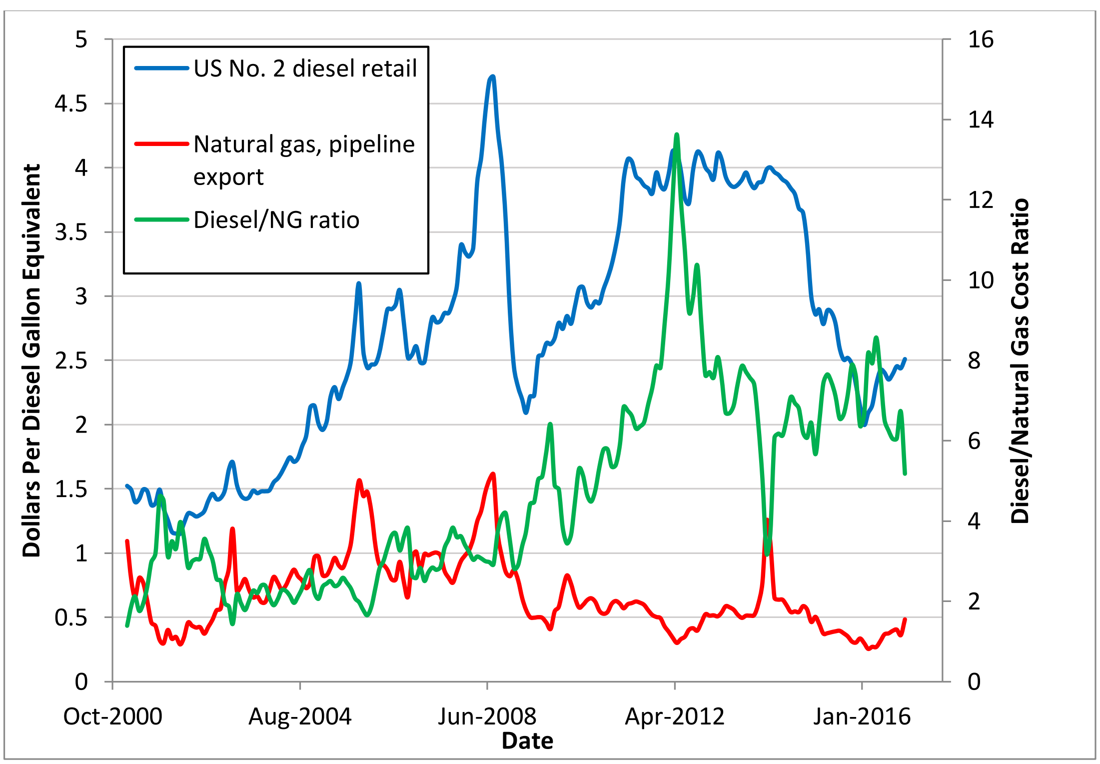

Figure 3: Prices of natural gas and diesel and the associated cost ratio [15] [16]

Other dual fuel applications are heavy duty mining trucks and agricultural machinery, over-the-road trucks and buses. Mobile applications of dual fuel engines have other challenges associated with storing the fuel. Since diesel is a liquid, it has a high density and is easily storable in small volumes for transportation applications. Because natural gas has a low density It is important to be able to store enough mass for sustained runtimes. Because natural gas has a low density, a large volume or a gas compression system is required. Liquefied natural gas (LNG) tanker ships have benefited from dual fuel engines. Natural gas can be compressed and stored at -160^{o}C to be transported across the ocean to its destination to be regasified and introduced into pipelines [17]. Much of the natural gas boils off while in transit because cryogenic temperatures are required to maintain natural gas in the liquid phase. This boiled off gas can be recaptured and returned to the cryogenic tanks or it can be routed to power the dual fuel engines, which drive propellers and electric generators.

1.4 Dual Fuel Operation

Operating an engine in dual fuel mode takes advantage of the differing ignition characteristics of the two fuels. Diesel, which the engine is originally designed to run on, has a high cetane number [1]. This is favorable for compression ignition since the fuel is intended to auto-ignite as it enters the combustion chamber. Natural gas is very difficult to auto-ignite, having a high octane number over 100 or a methane number typically in the range of 80 to 90 [18]. Methane is the most difficult fuel to ignite and is the primary component of natural gas. Smaller quantities of heavier hydrocarbons make up most of the remaining composition of natural gas. The natural gas can be fumigated into the air intake before a mixer or injected near the intake port. The lean pre-mixed air fuel mixture is compressed to high temperature and pressure and when the piston nears top dead center diesel is injected. The diesel spray creates jets that ignite as they enter the combustion chamber and form a diffusion flame. Once the flame is initiated, it is able to transition to a pre-mixed flame and propagate through the natural gas-air mixture. The schematic in Figure 4 shows the four strokes of a dual fuel engine.

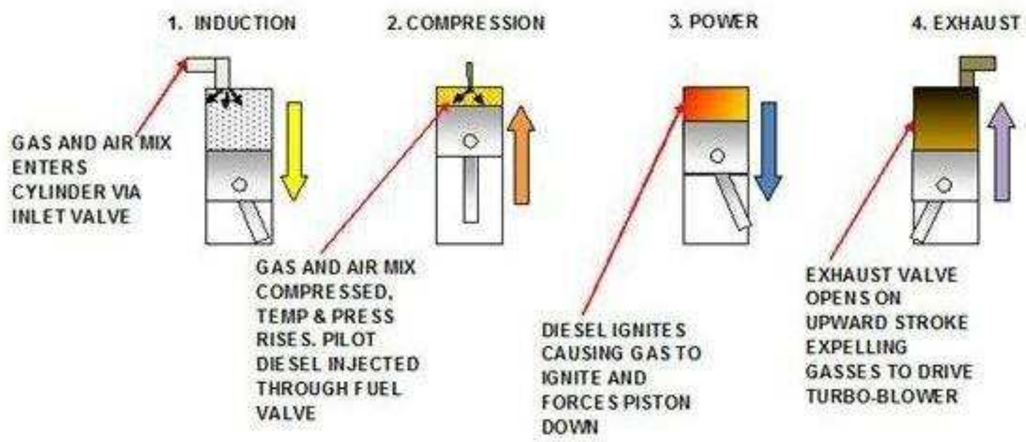

Figure 4: Schematic of the four strokes of dual fuel combustion [19]

When retrofit kits are installed most of the engine is left in its original condition. No changes are made to the Engine Control Unit (ECU) or to the compression ratio. This results in some conditions that are not ideal. Natural gas becomes the primary fuel of the engine; however, it has a slow flame speed and may reduce the overall combustion rate [20]. The injection timing, controlled by the ECU, remains unchanged and may result in a non-optimal center or combustion timing. Also, there is pre-mixed fuel being compressed at high compression ratios. This increases the potential for abnormal combustion events and knock. Natural gas is favorable compared to other fuels for this use though because it has a high methane number. Consequently, high substitution levels can be realized before knocking occurs.

1.5 Specific Aims of Research

The price of natural gas and its relative cleanliness makes it an attractive fuel compared to gasoline or diesel. Dual fuel operation is being increasingly employed to reduce operating costs in which utilization of natural gas in diesel engines reduces the amount of diesel burned. This avoids the cost of purchasing a new natural gas specific engine and the reduction in diesel fuel consumption results in cost savings. The engine cannot run on 100% natural gas though because natural gas requires an ignition source other than compression and lower compression ratios for safe operation. The limit of the amount of natural gas supplied to the engine is called the substitution limit. The substitution limit is determined by either high emissions levels or engine instability. The purpose of this research is to clearly understand the factors restricting a further increase in the substitution rates.

Aim 1: To identify the mechanism creating engine instability at the substitution limit of dual fuel operation.

Aim 2: Evaluate the effectiveness of different methods for increasing natural gas utilization before onset of the identified limits.

The primary goal of the research is to understand the limitations to the combustion of a natural gas and diesel mixture. Upon installation of a dual fuel kit a lookup table is created for given speed and loads which determines the amount of natural gas that can be added at a given operating point. A typical substitution profile is shown in Figure 5. At low loads, the substitution is generally limited by emissions levels and at high loads it is limited by an engine instability resulting in engine vibrations. It is desirable for the substitution to be as high as possible at every load to use more low cost natural gas and reduce diesel usage.

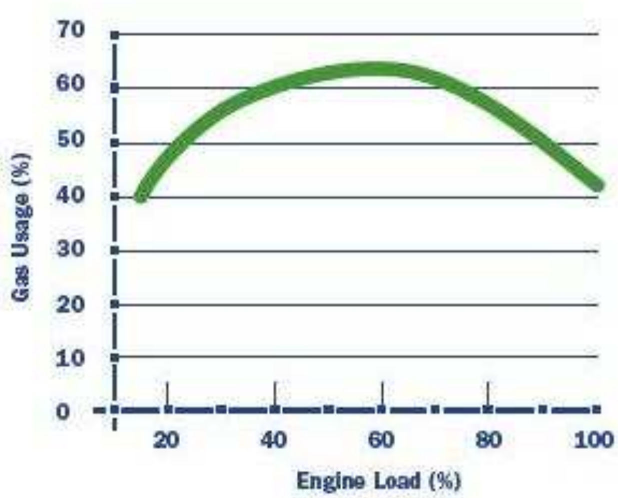

Figure 5: The typical substitution rate of gas for diesel for an Eden Innovations OptiBlend Kit [21]

Some engine manufacturers have developed dual fuel engines while other companies produce aftermarket products to retrofit diesel engines to benefit from the reduced costs of natural gas. A better understanding of the natural gas substitution limit may enable new designs to further increase the substitution. An increase in substitution will decrease drilling companies' operating costs. However, the engine must still meet the emissions limits to reduce the environmental impact and avoid costly fines due to non-compliance.

2. Literature Review

The principle of dual fuel combustion is that a flame spreads through natural gas after ignition of the diesel fuel. Natural gas is the main fuel source of dual fuel engines but it is burned in conditions designed for diesel operation. Having premixed fuel in engines with high compression ratios poses problems though. Premixed fuel in high pressure and temperature environments can cause engine knock. Incomplete combustion can become a problem if the conditions are not optimal. The purpose of the current study is to clearly define the substitution limitations across the entire operating range. Previous studies on dual fuel combustion have examined the typical engine parameters and their effects on engine performance and emissions. Dual fuel combustion has primarily been studied in single cylinder research engines. Studies have been performed to understand the effects of these different designs. Operating differences that have been studied for their effects on combustion and emissions are the equivalence ratio, injection timing and compression ratio.

2.1 Knock and Pre-ignition

At higher engine loads the general understanding is that knock causes the engine to be unstable. To better understand knock it is beneficial to understand dual fuel combustion phases shown in Figure 6 [22]. The combustion of dual fuel engines is described in 5 segments by Nwafor [22]. A to B is the period of ignition delay of the pilot diesel fuel. BC shows the combustion of the pilot fuel that becomes premixed with air during the ignition delay. C to D is the delay of combustion of natural gas. D to E is the combustion of the main fuel source and then diffusion combustion for the remainder of the curve from E to F. Karim explains the dual fuel combustion by separate phases of the heat release rate [23]. The initial heat release is from premixed combustion of diesel fuel shown by phase I in Figure 7. This corresponds to A-B in Figure 6. Then phase II is the heat release from natural gas combustion. This phase provides the highest peak of heat release rates but is a short duration. Phase III is described at diffusion combustion from the diesel fuel.

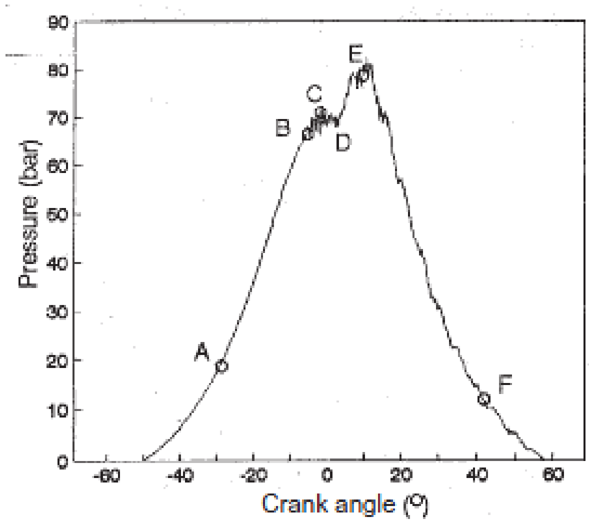

Figure 6: Theoretical dual fuel combustion phases based on cylinder pressure [22]

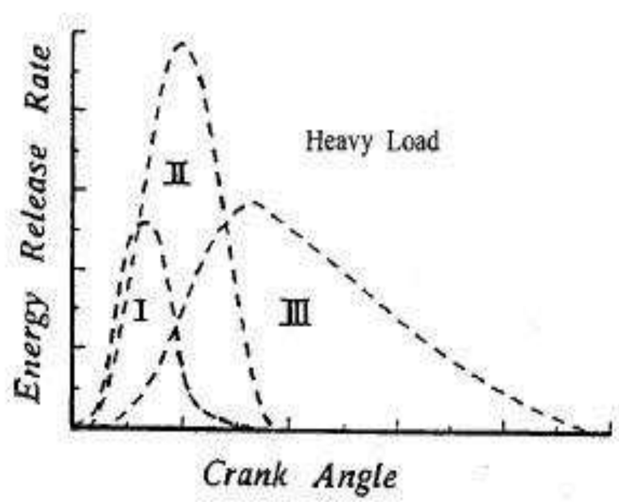

Figure 7: Theoretical dual fuel combustion phases based on heat release rates [23]

Nwafor observed different types of knock in dual fuel combustion [22]. First, there is diesel knock. Diesel combustion is an inherently noisy combustion because the process is started by auto-ignition. Dual fuel engines can also experience spark knock where the end gas auto-ignites before the flame front. This occurs in phase II, the premixed gaseous fuel combustion. A third type of knock was observed and called erratic knock. Erratic knock was observed with increasing substitution levels and created erratic engine operation and high pressure fluctuations. Karim observed an end gas knock when smaller pilot quantities are injected. If higher pilot quantities are injected, the natural gas can rapidly combust immediately following the diesel ignition [23]. Karim explains there are two effective ways to reduce the possibility of knock; decreasing the compression ratio and decreasing the intake temperature. Decreasing the compression ratio is generally not considered because a higher compression ratio leads to a higher efficiency of the engine. Reducing the intake temperature can be achieved by reducing the jacket water temperature or cooling the charge with a larger intercooler.

Abd Alla et al performed dual fuel research on a Ricardo E6 single cylinder engine. The engine has a bore of 75.2 mm and a stroke of 111.1 mm. The cylinder head has a Ricardo Comet MK V swirl combustion chamber and a compression ratio of 21.07. Their results showed a decrease in the engine torque when knocking began as the pilot fuel flow rate was increased [24]. Propane and methane were both used as the gaseous fuel which fumigated into the chamber before diesel injection. The knocking torque for methane was higher than propane at every pilot fuel flow rate. Another experiment performed by Abd Alla et al. showed that advancing the injection timing lead to lower engine torques when knock would begin [25]. The testing was performed at 1000 rpm with injection timings of 26-30 degrees before top dead center.

2.1 Equivalence Ratio and Injection Quantity

The equivalence ratio, given in Equation (6) by Φ, is a common parameter used to express the amount of air in the combustion chamber compared to fuel. The equivalence ratio, presented in Equation (6), is defined as the ratio of the fuel to air ratio divided the fuel to air ratio of a stoichiometric combustion process. It is an important engine parameter that changes the combustion significantly. An equivalence ratio of one signifies there is just the right amount of air and fuel for stoichiometric combustion. An equivalence ratio greater than one indicates a fuel rich process that has excess fuel compared to stoichiometric combustion. An equivalence ratio less than one is a lean mixture with excess air compared to stoichiometric conditions.

$$\phi = \frac{F/A}{F/A_{stoic}} \tag{6}$$

The equivalence ratio for dual fuel engines can be expressed as the ratio for just the premixed natural gas or the total equivalence ratio based on both fuel sources. The substitution in dual fuel engines compares the amount of fuel from natural gas to diesel. In most literature it is termed the substitution rate and defined as follows

$$substitution\ rate = \frac{\dot{m}_{NG}}{\dot{m}_{NG}+\dot{m}_{diesel}} \tag{7}$$

Between these two parameters the amount of each fuel and air can be determined for any operating condition. Changing these ratios has a large effect on the combustion process. Changing the equivalence ratio and the injection quantity effectively are changing the substitution.

Fredrik Konigsson studied limitations on dual fuel combustion on a single cylinder engine with a bore of 127mm and a stroke of 154mm. The engine is a 2 L Scania engine with a diesel common rail and natural gas injectors in the intake runners. The results were compared to a compatible SI engine. The spark ignited engine had an increase in the coefficient of variation (COV) of IMEP as the fuel mixture was leaner but the COV of IMEP for dual fuel operation remains below 2% even with 98% methane on an energy basis [26]. Although the combustion process is repeatable the emissions are analyzed to determine the effectiveness of combustion. It was found that at stoichiometric conditions the CO monoxide emissions are at a maximum. They become a minimum at slightly lean conditions and then increase as the combustion mixture is burned at leaner conditions [26]. Hydrocarbon emissions were at a minimum at stoichiometric conditions and increased with leaner mixtures. Konigsson explained at stoichiometric conditions there are local regions that are fuel rich and unable to fully oxidize leaving CO monoxide unreacted. As the mixture becomes leaner the combustion temperatures are reduced therefore exhausted higher levels of CO and HCs. NOx emissions were found to have a trend opposite to that of CO. At slightly lean mixtures the NOx emissions are at a maximum and decrease as the mixtures become leaner because of the reduction of the combustion temperature.

Carlucci et al performed testing on a 2 L 4- cylinder Fiat 154 D1.000 engine. The engine was turbocharged and supplied natural gas at ambient pressure. The engine was operated at 1400 rpm at 32Nm, 2000 rpm at 40 Nm, and 2000 rpm at 80 Nm. Tests were performed with natural gas adjusted and the pilot fuel tuned to reach a specific power set point and by fixing the pilot injection duration and adjusting the natural gas flow. For each of these operating conditions the NOx concentration increased with increased natural gas mass ratio supplied [27]. This trend has the steepest slope at high loads and speed. The THC emissions also increase with natural gas mass ratio. At high speed and high loads, the THC levels increase until a mass ratio of 40%. Above this, the THC emissions start to decrease slowly. As the natural gas usage is increased the particulate matter levels decrease.

Papagiannakis and Hountalas performed a dual fuel engine experiment with a single cylinder engine and varied the natural gas percentage [28]. A Lister LV1 four stroke diesel engine was tested at 2000 rpm at 40, 60, and 80% load with various substitution levels. The bore is 85.73mm and the stroke is 82.55mm. The compression ratio is 17.6:1. Their results on the naturally aspirated engine showed a decrease in the peak pressure for each load while increasing the mass ratio of natural gas. The addition of natural gas also increases the ignition delay period and slightly increases the burn duration. The brake specific fuel consumption worsened as the substitution was increased. The effect was minimal at 80% load but significant at 40 and 60% load. The emissions concentrations showed similar trends to literature, the soot opacity decreased with more natural gas. The NO emissions also saw a reduction with increased substitution. The effect was most noticeable at 80% load. The reduction was minimal at 40% load though.

The HC emissions were shown to increase with substitution. For each load tested, the increase in HC emissions is about the same until a mass ratio of about 50% natural gas to total mass of fuel. Above mass ratios of 50%, the low load test showed the most increase in HC exhausted followed by the 60% load test. The 80% load test maintained the same level of HC above a mass ratio of 50%. The CO emissions show an interesting trend. Initially as the mass ratio increases the CO emissions rise. At 80% load there is a peak around 50% after which the CO concentrations decrease. 60% percent load shows the trend at slower rate. Above a mass ratio of 50%, the CO levels do not decrease but stay the same. At 40% load, the CO emissions rise at a slower rate initially. The increase in emissions is greater at higher natural gas mass ratios with its maximum at a natural gas ratio of 85%.

An experiment by Karim showed a similar trend in the CO emissions to the trend observed by Papagiannakis and Hountalas. As the total equivalence ratio increases the CO emissions increase until a maximum around 0.45 [23]. After the maximum the concentrations decrease at a nearly the same rate as the increase at lower equivalence ratios. Increasing the mass of diesel injected increases the CO at the same total equivalence ratio. Karim explains the CO emissions are primarily from the methane fuel surrounding the diesel jets so increasing the area of the jets increases the amount of methane converted to CO. For this reason, the HC exhaust concentrations have a similar profile but the peak is at a gaseous fuel equivalence ratio of approximately 0.27 as shown in Figure 8.

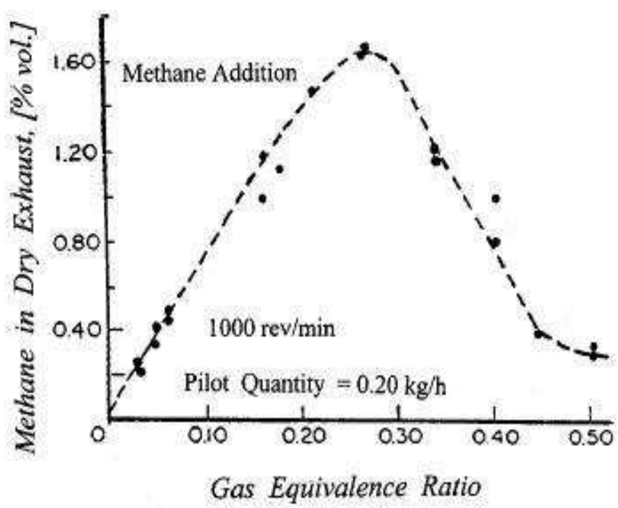

Figure 8: Methane concentrations in exhuast versus gas equivalence ratio [23].

When the methane emissions are examined on a percent of the fuel added to the cylinder, the amount of fuel unconverted is shown to decrease as more methane is added. For a given amount of methane fuel addition, increasing the amount of diesel injected lowers the amount of fuel left unconverted. The unconverted methane may be above 85% with little methane addition and small diesel injection quantities. The increase of methane increases the gas fuel equivalence ratio promoting better burning until a limiting conversion rate is achieved. A separate paper published by Badr, Karim and Liu discuss this in terms of the flame spread limits [29]. The paper discusses two engines each with a bore and stroke of 105mm and 152.5mm respectively. The compression ratios for the two engines are 14.2 and 14.7. The authors proposed four different operating regions and three limiting equivalence ratios as shown by the schematic in Figure 9.

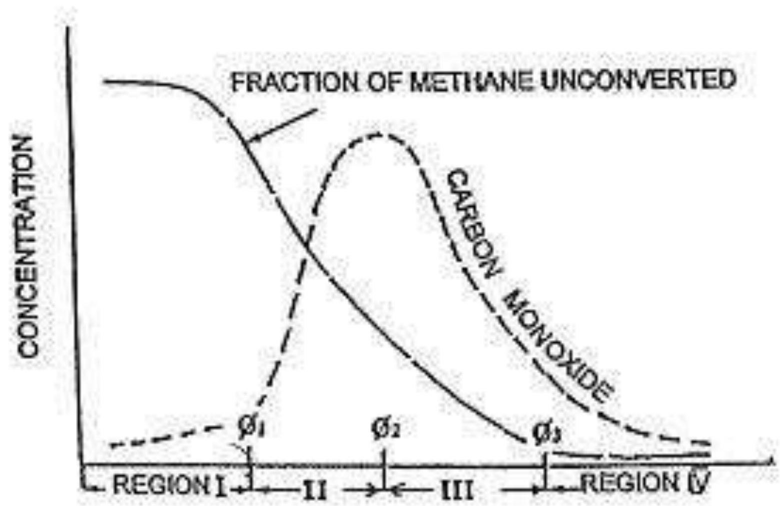

Figure 9: HC and CO Emissions versus equivalence ratio [29]

The limiting equivalence ratios Φ_1, Φ_2 and Φ_3 are described as possibly being the start of local oxidation, flame initiation and flame spreading respectively. The value Φ_3 is used as a way to determine flame spread limits at light throughout their testing. That is, the equivalence ratio is required to be greater than Φ_3 to successfully propagate a flame through the premixed fuel air mixture and reduce the emissions to acceptable levels.

2.2 Injection Timing

The injection timing is just as important as injection quantity on dual fuel engines. Adjusting the phasing of combustion affects the peak pressure, temperature, ignition delay and combustion duration. The study by Carlucci, introduced above, examined the effect of injection timing while adjusting the equivalence ratio. At various engine speeds and loads the injection timing is set to 19, 23 and 27 ^{o}btdc. The results show advancing the injection timing earlier before top dead center creates a longer ignition delay after every speed and load tested [27]. At low speed and low load, advancing the timing reduces NO emissions. Advancing the timing increases NO formation at high loads though, especially with high mass ratios of natural gas. The results show the HC emissions are slightly decreases from earlier injection timing. The effect on HC is smaller than the effect on NO. The CO emissions have a linear relationship with HC. At low loads they both increase together and at high loads the both decrease together. The soot opacity is largely unaffected by the injection timing changes at high loads. Retarding the timing at low loads did show a significant decrease in the soot formation though.

Abd Alla et al performed research on the injection timing of a dual fuel engine on the same Ricardo E6 engine as introduced above. Their testing also shows advancing the injection timing decreases the HC emissions [25]. The reduction is greater at higher total equivalence ratios. CO emissions were shown to have the same trend as the HC emissions. Advancing the timing has negative effects on the NOx emissions just as shown by Carlucci. The advanced timing tests have the highest thermal efficiencies and best conversion rate of HC and CO but the highest NO formation.

Krishnan et al performed a broader study of the effect of injection timing. The study was performed on a Caterpillar 3401 single cylinder engine with simulated turbocharging from pressurized natural gas port injection. The bore and stroke are 13.7 and 16.5 cm, respectively. The compression ratio is 14.5:1. The engine is operated at 42 kW and 1700 rpm. The study by Abd Alla et al. only varied the injection timing from 25 to 30 degrees before top dead center while Krishnan et al. varied the injection timing from 15 to 60 degrees before top dead center. An injection timing of 15 ^{o}btdc. has the lowest specific HC and NOx emissions but very low fuel conversion efficiencies [30]. The authors propose early injection timing as a method to reduce

NOx emissions while increasing the fuel conversion efficiency. Their results show an increase in HC emissions until 25 ^{o}btdc. The HC emissions then decrease until an injection timing of 45 ^{o}btdc, after which the emissions increase again. NOx emissions increase until 35 ^{o}btdc then steadily reduce with further advanced injection timings. The lowest NOx emissions are observed at 60 ^{o}btdc. The highest fuel conversion efficiency is shown at an injection timing of 45 ^{o}btdc. The diesel fuel is no longer confined to the areas of jet penetration but spreads out thoroughly into the natural gas air mixture with early injection timings. This makes more, small regions with diesel fuel, which creates many smaller ignition centers. Because the ignition centers are more distributed the equivalence ratio of the ignition centers are leaner. This is proposed to cause reduced local temperatures and a significant reduction in NOx. The distributed ignition centers provide better burning rates of the natural gas and air mixtures.

Krishnan also published another paper on the effect of the intake temperature on dual fuel combustion. The engine was operated at 50 and 25% load while operating with intake temperatures of 75, 95 and 105^{o}C. At 50% load the SOI is 60^{o}btdc and at 25% load the SOI is 55^{o}btdc. The results at 50% load show a reduction of HC emissions and a large increase in NOx emissions with increased temperatures [31]. The fuel conversion efficiency is the lowest at 75^{o}C and the COV of IMEP is above 10%. Increasing the temperature increases the fuel conversion efficiency by 7% and decreases the COV of IMEP to 3.8%. At 25% load, the COV of IMEP is above 26% when the intake temperature is 75^{o}C and is reduced to 6.6% when the intake temperature is 105^{o}C. The results show a reduction in HC emissions but a significant increase in NOx emissions. From 75 to 95^{o}C, NOx emissions increase from 66.7 to 78.0 mg/bkWh. Increasing the intake temperature to 105^{o}C raises NOx levels to 236.4 mg/bkWh. The combustion becomes more consistent at the expense of more NOx exhausted.

2.3 Dual Fuel Performance Compared to Diesel Only Operation

The papers presented previously discuss what affects emissions from dual fuel engines but it is important to compare dual fuel emissions to diesel only operation. Lounici et al performed dual fuel research on a Lister Petter TS1 single cylinder diesel engine. The output power is 4.5kW at 1500 rpm. The engine has a bore and stroke of 95.5 and 88.94 mm, respectively, and a compression ratio of 18. At low loads, diesel only operation produces higher peak pressures but at 60% load and above dual fuel operation produces higher peak pressures at 1500 and 2000 rpm [32]. At higher loads, the total bsfc is better for dual fuel operation but worse at low loads. For every operating condition tested, soot levels are reduced for dual fuel operation. NOx emissions are lower than diesel only operation except at 80% load. At 1500, 1800, 2000 and 2200 rpm dual fuel operation increase NOx emissions compared to diesel only. HC emissions are shown to be significantly higher than diesel only operation at every operating condition. CO emissions for dual fuel are higher at low loads but lower than diesel only at 80 and 100% load.

Papagiannakis and Hountalas also compare their results to diesel only operation at a wide variety of operating conditions. Their results show the peak pressure of dual fuel operation is lower than diesel for all test conditions at 1500 and 2500 rpm [33]. Their results also vary from Lounici et al in that test data shows worse total bsfc for dual fuel operation at all loads and speeds. Their emissions results show dual fuel creates less NOx than diesel only for all conditions tested. The difference is greater at higher engine loads and speeds. CO monoxide emissions are shown to be far greater than diesel only operation. HC emissions levels are also significantly higher than diesel only operation. Lounici showed HC emissions had a parabolic shape with the maximum levels around 50% load. Papagiannakis and Hountalas show the maximum HC emissions are lowest load and decrease as engine load is increased. Their results show the HC emissions are the same independent of engine speed. At 1500 rpm and 2500 rpm, the dual fuel HC emissions are nearly the same.

2.4 Dual Fuel Combustion Modeling

Recently two dissertations on dual fuel engine research have been conducted at Colorado State University. Andrew Hockett's dissertation was concerned with the modeling of dual fuel engines and comparisons to experimental results. Experimental results show advancing the SOI timing shortens the combustion duration and increases the peak pressures. The initial peak of HRR from the spray ignition is increased as the injection is advanced. A simulation in presented in the work shows a premixed ignition around the diesel spray, which transitions to a diffusion flame [34]. The simulation also shows a premixed natural gas flame spreading from the jet. Simulations also show given mass of combustion species at given times. The n-heptane and methane species concentrations also clearly show two distinct flames. As the natural gas flame propagates through the cylinder oxygen is consumed. However, the simulation shows there is leftover oxygen on the burned side of the propagating flame. This oxygen is able to react with the diesel diffusion flame. Hockett also notices HO_2 and CH_2O in end gas regions. This is described as auto-ignition reactions starting before the propagating natural gas flame consumes the fuel. The simulation shows NO formation is greatest in the diesel flame and little attributed to the premixed flame. The explanation for this is that stoichiometric non-premixed flames burn at higher temperatures than lean premixed flames. The simulation only accounted for thermal NOx production so the highest temperature regions display the highest NO formation.

Simulations were also performed to understand dual fuel knock. The simulation shows the auto-ignition centers are between adjacent sprays. As premixed flames propagate, they converge forming a small pocket of unburned natural gas [34]. This was shown to be closer to the center of the combustion chamber than at the cylinder walls. The cylinder walls provide cooling to the fuel mixture so the fuel further away from the walls receive less cooling. These are the areas that auto-ignite causing erratic pressure waves and knock. This is noted to be both similar and different than knock in spark ignited engines. Both are end gas auto-ignition events but in spark ignited engines there is one spherical flame propagating out towards the cylinder walls. The end gas region is thus at the cylinder walls rather than located closer to the middle of the combustion chamber in dual fuel knock.

Wan Nurdiyana Wan Mansor completed her Ph.D. at CSU studying dual fuel emissions. The study conducted also worked on dual fuel modeling but had more focus on experimental results than Hockett's. Wan Mansor observed the COV of IMEP of dual fuel operation was significantly higher than diesel operation especially at low loads [35]. At higher loads the fuel consumption and efficiency were improved by running the engine as a dual fuel. At low, loads the efficiency and bsfc were worse. Dual fuel operation was shown to have lower peak pressures at all loads except full load.

The emissions of dual fuel operation were compared to diesel only operation. At every load set point, dual fuel operation reduced NOx emissions comparatively to diesel only. The emissions of particulate matter were reduced at all loads except full load. The CO emissions were significantly increased at low loads but approached the levels from normal diesel operation at high loads. THC emissions are also far greater than diesel only operation. The work shows the emissions from dual fuel operation exceeds Tier II limits for all categories except particulate matter.

The Converge simulations performed by Wan Mansor examine pollutant emission formation. The simulation shows CO formation primarily occurs where the equivalence ratio is high [35]. THC emissions are shown to originate from regions with lean equivalence ratios. Since non-premixed flames burn at stoichiometric conditions, the lean equivalence ratios are associated with the natural gas-air mixture. The simulation demonstrates significantly more unburned HC at 12% load compared to 75% load because the natural gas air mixture is leaner. NOx emissions are shown to form in the high temperature regions around the diesel jet. Simulations show the PM formation is primarily due to regions where the equivalence ratio is greater than one. The present work is expanding on the work performed by Wan Mansor focusing on the substitution limits of natural gas for diesel. The same engine for the experimental portion is used but has been commissioned on a different test skid with some different modifications.

3. Design of Experiment

3.1 Engine and Generator Control

For this experiment, a John Deere 6068 Tier II diesel engine was commissioned as a generator set. The engine specifications are provided in Table 2 and the overall system including controllers and the dual fuel kit is outline in a schematic in Figure 10. The specifications are listed in Table 2. The Marathon Electric generator syncs and exports electrical power to the local utilities grid. Woodward Micronet, Easygen and MFR13 controllers monitor generator and grid conditions. The Micronet is the master controller of the system. The Easygen and MFR13 are slaves to the Micronet for providing generator and engine safety. The Micronet communicates to the slave controllers and to the John Deere ECU. The dual fuel kit was procured from Eden Innovations. The dual fuel kit is represented in the top left corner of Figure 10. The PLC does not communicate with the other controllers in the system. It has separate current and potential transformers (CTs and PTs) that are read by a power meter. The natural gas throttle position is determined solely on the power level.

The generator frequency and line phasing have to be synchronized to the local utility grid in order to export power. To achieve this, the throttle of the engine was controlled externally. The engine ECU performs the logic to control the amount of fuel injected based on the cam position and engine speed sensors. Fuel injection quantity is adjusted by the amount of time the injector nozzle is open. A portion of the wiring harness branches to a John Deere control box. On the control box the throttle can be selected between three options, internal analog, external analog and external pulse width modulation (PWM). The external analog setting is used to accept a throttle signal from the master controller for this testing. The engine ECU single range is 0-5V analog voltage signal for the throttle input. The master controller uses an analog output field termination module (FTM) to output a 0-20mA current signal for the throttle signal. The current signal is wired in a closed current loop configuration as necessary for current to flow. A large impedance inside the terminals on the control box ensures all of the current flows around the current loop back to the return current terminal at the FTM. The precision resistor is wired into the current loop creating the appropriate voltage difference. The ECU has leads on either side of the resistor and therefore processing the voltage drop as control signal. Using ohm's law, 20mA flowing through the current loop times 250 ohms equals a voltage drop across the resistor of 5V. Similarly, 4mA times 250 ohms equals a voltage drop of 1V across the resistor.

Table 2: John Deere engine specifications

Engine Model	6068HF 475
Engine type	In-line, 4-cycle
Number of cylinders	6
Displacement	6.8L
Rated Speed	2400 rpm
Rated Power	205 kW
Aspiration	Turbocharged and coolant to air aftercooled
Compression ratio	17:1
Bore X stroke	106 X 127 mm
Connecting Rod length	203 mm
Inlet valve opening	156.75 degrees before TDC
Number of injector nozzle holes	6

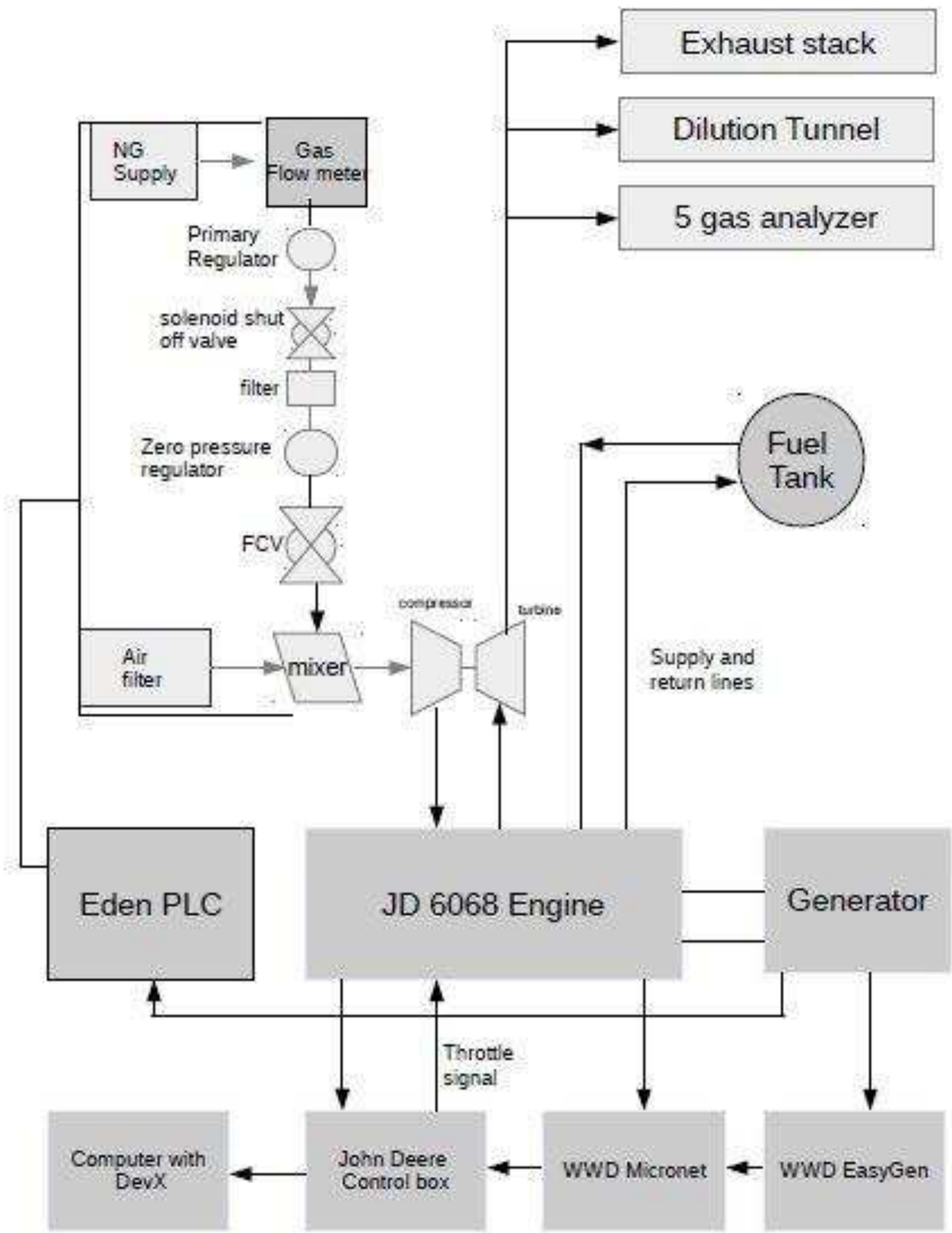

Figure 10: Schematic of the controls for the engine with the dual fuel kit

Once the engine is operating at 1800 rpm (60Hz generator output frequency), the engine can be synced to the grid. When commanded to start synchronization the Easygen controller starts to use a speed bias. The controller senses the frequency on the grid and the phasing of the three AC lines. It also measures the phasing of the voltages on each line of the generator. The Easygen controller determines if the engine needs to be sped up or slowed by comparing the sine wave of the voltage signal of the generator to the corresponding line voltage signal of the utility. A speed bias signal is calculated by the difference in zero crossing locations of the voltage signals. This speed bias signal commands a change of approximately 1% increase or decrease in throttle signal transmitted to the ECU. If the generator line frequency is leading the utilities the speed bias commands a decrease in engine speed. If the generator line frequency is lagging the utilities the speed bias signals for the engine to increase speed.

The speed bias signal is sent from the Easygen controller as a 4-20mA signal to an analog input FTM of the Micronet controller. The Micronet compares the bias signal to 12mA, a neutral offset. If the signal is greater than 12mA the speed of the engine is to be increased. If the signal is less than 12 mA the speed is to be decreased. The current offset determines the magnitude of the speed changed. The master controller calculates the speed change desired and converts this to a change in the throttle signal. Depending on if the speed is to be increased or decreased, the controller will output more or less current supplied to the John Deere control box.

The voltage output from the generator on each line also needs to match the utilities for synchronization to occur. The generator shaft has a permanent magnet generator that supplies power to a Marathon 2000E voltage regulator. Because the magnets are permanent, a specific speed will always output a specific voltage. This voltage is around 180Vac at 1800 rpm. This voltage cannot be changed to match the voltage on the grid so a voltage regulator is required to adjust the voltage on the main stator coils of the generator. When the voltage regulator has power it will supply an excitation voltage to the stator coils of the generator. The generator output voltage is proportional to the excitation voltage. The Easygen senses the voltages on the grid and on the generator and sends a voltage bias signal to the voltage regulator to increase or decrease the excitation voltage until the voltage matches, about 480V.

When the generator phasing of all three lines, the frequency and voltage match the grid within a small percentage the syncing controller sends a signal to close the generator circuit breaker. The signal from the controller triggers a relay in the switchgear. The signal powers the coil of a mechanical relay. When the relay becomes switched, a 120Vac signal is passed through to a stored energy operator that closes the breaker. The circuit breaker has a shunt trip, which is a safety that can open the breaker if the engine falls out of sync. Either the Easygen or the Micronet can trigger the shunt trip.

3.2 Dual Fuel Retrofit Kit

The natural gas was administered to the engine by an aftermarket retrofit dual fuel kit. The lab receives natural gas from the city pipeline at 26psi. A primary regulator drops the pressure to less than 5psi before entering the gas train. The gas train includes a manual shutoff valve, a solenoid operated shutoff valve, a filter, a zero pressure regulator and a mixer. The mixer is shown in Figure 11. The zero-pressure regulator is referenced to the mixer at the brass fitting on top of the mixer so that the natural gas flowing through the gas train is at the same pressure as the air coming from the air filter. Air flows from the air cleaner into the mouth of the mixer, which contains a venturi. The venturi reduces the air pressure and allows natural gas to be drawn through the port on the side of the mixer housing similar, to a carburetor. The amount of natural gas added is controlled by the flow rate of air and the control valve mounted directly to the side port of the mixer. The throttle control valve and mixer are shown in Figure 11.

The programmable logic controller (PLC) determines the position of the control valve. The dual fuel kit includes current transformers and potential transformers installed on the generator to measure power. The position of the control valve on the mixer is dependent on generator power. An engine vibration sensor is included in the Eden Dual fuel kit. The vibration sensor is used to determine the maximum natural gas substitution. The maximum substitution is determined for each of the five load break points. A lookup table of fuel control valve position and load values is created and programmed into the PLC. When the engine is operated at loads other than the five breakpoints the throttle position is linearly interpolated between the surrounding commanded positions.

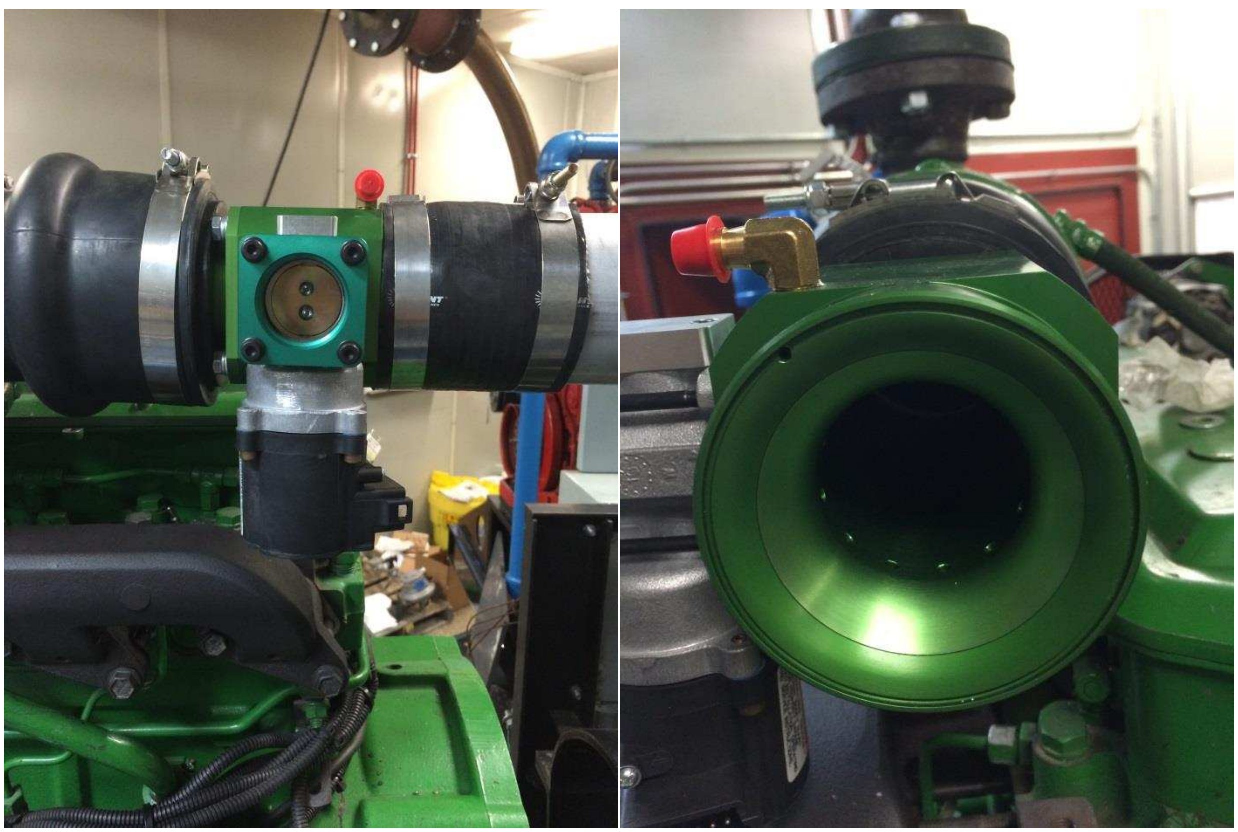

Figure 11: View of mixer for controlling natural gas addition to air stream.

3.3 Instrumentation and Operating Conditions

Various sensors were mounted around the engine to monitor operating conditions. Most of the sensors are thermocouples and pressure sensors. The specifications of the main experimental instrumentation are presented in Table 3. All of the thermocouples used are K type thermocouples. These monitored jacket water temperature, fuel line temperature, intake manifold temperature, chilling loop temperature and oil sump temperature. These values are below 100^{o}C so the measurement uncertainty is ±2.2^{o}C. Type K thermocouples have an error of 2.2^{o}C or .75% of reading, whichever is greater. Below 100^{o}C, 2.2 is the larger uncertainty. Type K thermocouples are also used for monitoring exhaust gas temperatures. These values are high enough that .75% of reading is the larger value of uncertainty.

Table 3: Main instrumentation specifications

Parameter	Measurement Device	Measurement Error
Jacket water temperature	Omega K type thermocouple	2.2^{o}C
Exhaust gas temperature	Omega K type thermocouple	0.75% of reading
In-cylinder pressure	Kistler 6056A	≤ 0.3 %FSO
Crank angle location	BEI Rotary encoder 924-01002-9390	0.25^{o} resolution
Natural Gas Flow	Omega FMA1845A-ST-CH4	<1.5% of full scale
Diesel Flow	DevX	NA

The natural gas mixer is located just upstream of the engine turbocharger as shown in the schematic in Figure 10. A laminar flow element flowmeter is installed just before the gas train, measuring the amount of fuel drawn into the air stream. The Omega flow meter is calibrated for natural gas flow with a range of 0-500 slpm. The fuel and air mixture are compressed in the turbocharger. At high loads, the compression from the turbocharger significantly increased the temperature of the charge before the intake manifold. A Frozenboost Intercooler, model INT000220, was installed in the air intake system to cool the charge temperatures. The intercooler is an air to water heat exchanger and shown in Figure 12. The coolant is supplied from the chilling loop, which circulates to the radiator fans dissipating heat to ambient conditions. A mechanical thermostat controls the flow rate of coolant through the intercooler. The thermostat is a Johnson Controls V47AB-3 thermostat. The sensing probe is located in the air intake. For most of the dual fuel testing, the thermostat was set to control the intake air temperature to 60^{o}C. At low loads, the compression from the turbocharger did not increase the temperature above the control set point so the thermostat closes leaving the intake manifold temperature the same temperature as at the exit of the turbocharger. This is around 55^{o}C at 25% load and 40^{o}C at 10% load. The locations of the sensors are shown in Figure 13. The chilling loop also flows through a parallel loop to the intercooler that has a shell and tube heat exchanger to cool the engine jacket water. The chilling loop temperature varies between 15-30^{o}C depending on the ambient temperature. The chilling loop temperature is measured on the inlet side of the intercooler in the red pipe just below the bottom of the Figure 12. The jacket water of the engine is maintained around 80^{o}C by in engine thermostats and the shell and tube heat exchanger. Oil flows through a plate type heat exchanger with the jacket water. This regulates the oil temperatures to around 80^{o}C also.

The diesel fuel is filtered and supplied to the high pressure pump. The fuel temperature is measured just before the inlet of the pump. The engine utilizes a common rail configuration and electronic injectors. The amount of fuel injected is determined from the John Deere ECU and

Figure 12: Side view of engine showing intercooler system and exhaust gas thermocouples.

throttle command from the Micronet. The injectors are controlled by a solenoid and a pressure differential on the injector needle. The diesel fuel flow is measured in DevX, the John Deere ECU human interface program, from the current duration for the solenoid and the rail pressure.

The injector factory calibration provides a diesel fuel flow in liters per hour based on duration and rail pressure. DevX is also used to log the start of injection timing and rail pressure. Combustion pressures are measured with four in-cylinder pressure sensors. Kistler 6056A piezoelectric pressure sensors are mounted in glow plug adapters. Charge amplifiers are used to increase the output voltage for better sensing. The pressure is located to crank angle with a BEI rotary encoder that has a resolution of a quarter of a degree. The pressure measurements are displayed in real time by a LabVIEW program developed at the EECL. Another LabVIEW program is used to post process the pressure data to determine the heat release rate, mass fraction burned, IMEP and coefficient of variation of each parameter. Pressure data is collected for 500 or 1000 cycles at each test condition.

The engine is equipped with eight exhaust thermocouples. Figure 14 shows the location of the instrumentation, including the exhaust thermocouples. There is one located in the exhaust port for each cylinder and two in the collector of the manifold just before the turbocharger. One of the thermocouples in the collector sent information to the Micronet, with a safety shutdown. The other thermocouple in the collector sends information to the dual fuel kit PLC. Increased substitution levels typically increase the exhaust gas temperatures because the premixed fuel

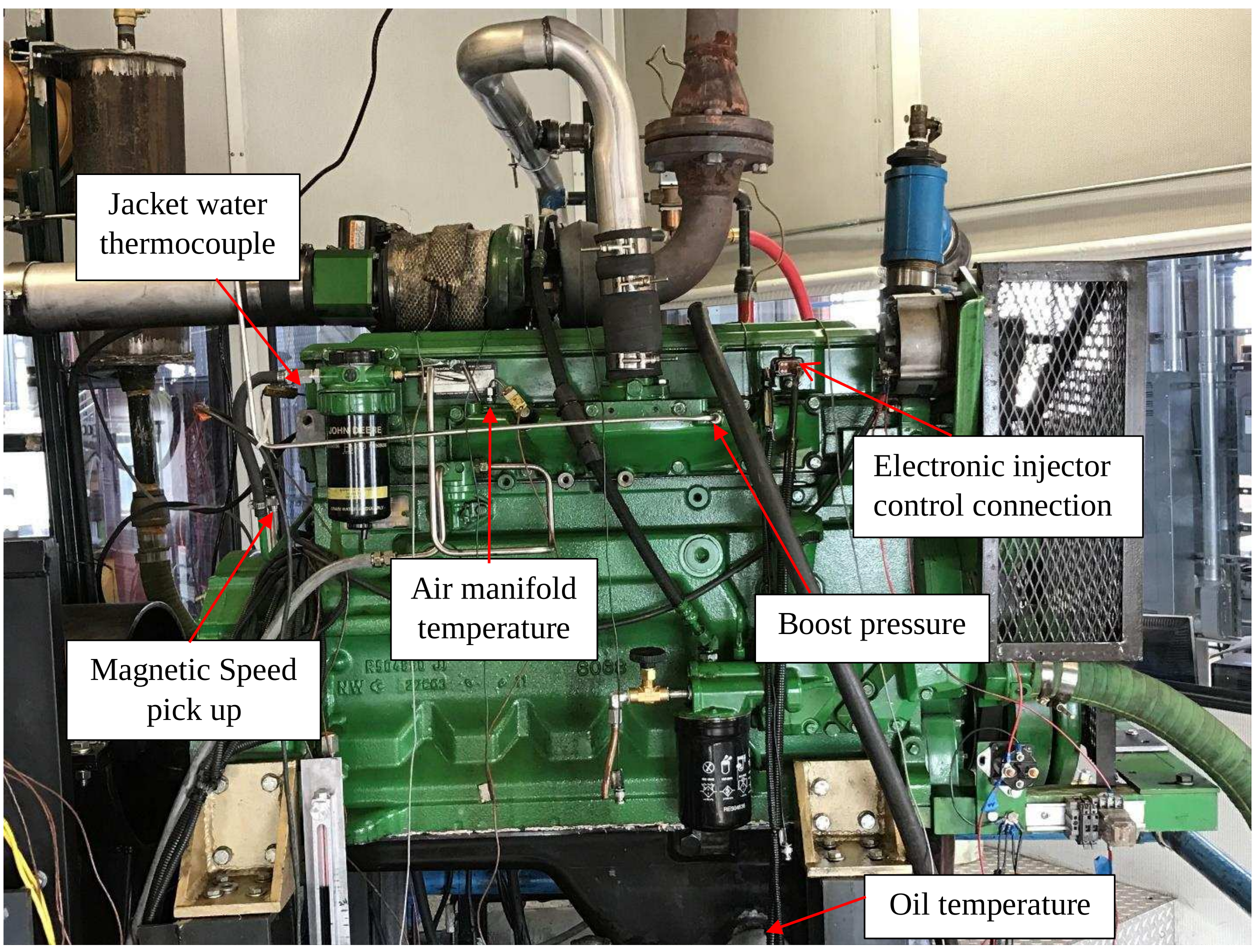

Figure 13: Sensor locations on engine

burns later. The PLC has a safety to decrease substitution levels if exhaust gas temperatures become too high. In Figure 14, the thermocouple on the left in the collector is the one connected to the Micronet and the one on the right is connected to the PLC.

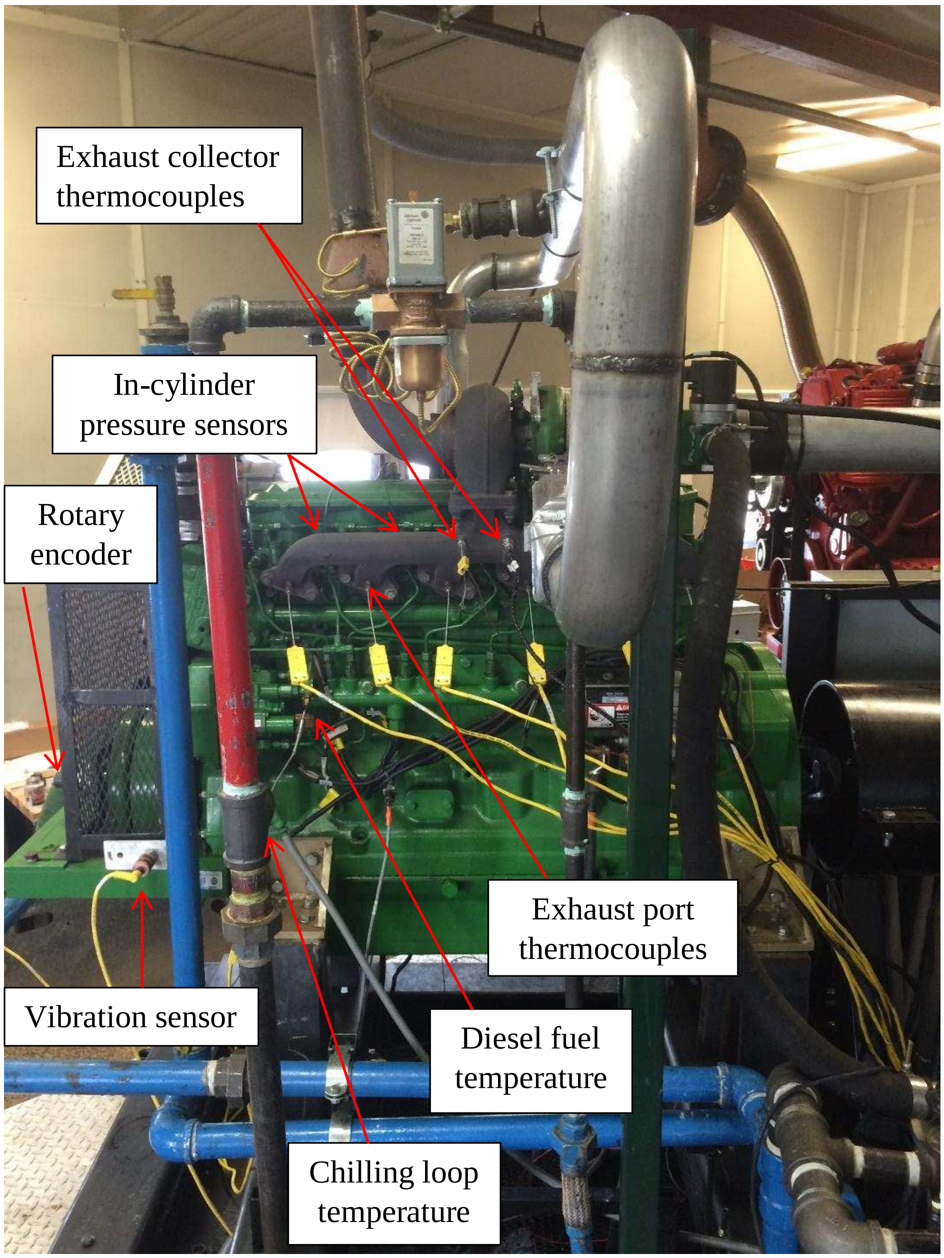

Figure 14: Sensor locations on opposite side of engine

The brake power of the engine is measured by the electrical power output of the generator. The apparent electrical power is the sum of voltage times the current on each line in kilovolt amps[kVA]. The real power is the apparent power times the power factor in kilowatts[kW]. The power factor is equal to cosine of the angle delay between the voltage and current of a single phase. The generator was operated with a power factor of 0.9 with the current lagging the voltage. The generator output 140kW of electrical power at full load. Generator efficiency curves are used to convert the electrical power to brake power of the engine. The generator is a Marathon Electric generator model 433RSL4021 three phase generator. While operating at full load with a power factor of 0.9, the generator efficiency is 94%. This corresponds to 148kW of brake power at full load. Accounting for the generator efficiency, the brake power at full load is 148 kW. The load break points are 148, 114, 77, 40 and 15 kW of break power at 100, 75, 50, 25 and 10% load respectively. The engine is operated at 1800 rpm and 60Hz generator output frequency to match the 60Hz electrical grid frequency.

3.4 Emissions

The exhaust gas is sampled via an averaging probe and an isokinetic probe directed parallel to the flow of gas located downstream of the turbocharger. The collected gas is supplied to exhaust gas analyzers including a Rosemount 5-gas analyzer and Fourier Transform Infrared(FTIR) spectrometer (averaging probe) and a dilution tunnel (isokinetic probe) through heated sample lines. The Rosemount 5-gas analyzer is capable of measuring CO, CO_2, THC, NO and NO_2 (collectively NOx). The capabilities of the 5-gas analyzer are shown in Table 4.

The ranges of each analyzer were set as follows for the presented emissions data: CO 0-1000 ppm, CO_2 0-12%, THC 0-5000 ppm, NOx 0-500 ppm and O_2 0-18%. This corresponds to 5ppm, 0.6%, 50 ppm, 2.5 ppm and .18% uncertainty for CO, CO_2, THC, NOx and O_2

Table 4: Specifications of the 5-Gas analyzer

	Device	*Measurement Technology*	*Minimum Concentration Range*	*Maximum Concentration Range*	*Linearity*
CO	Ultramat 6	IR	0 – 10.0 ppm	0 – 10000 ppm	< 0.5% of full-scale value
CO_2	Ultramat 6	IR	0 – 5.0 ppm	0 – 30 %	< 0.5% of full-scale value
THC	NGA 2000, FID	FID	0 – 1.0 ppm	0 – 10000 ppm	< +/- 1% of full scale
NO_x	NO_x MAT 600	Chemiluminescence	0 – 1.0 ppm	0 – 3000 ppm	< 0.5% of full-scale value
O_2	NGA 2000, PMD	Paramagnetic	0 – 1.0 ppm	0 – 100 %	+/- 1% of full scale

respectively. The carbon monoxide and carbon dioxide concentrations are measured by infrared spectroscopy. Infrared light is emitted into a chamber of the exhaust gas.

The wavelength which is emitted is varied over a range by a monochromator. When the wavelength of the radiation matches the energy of the bonds, the radiation is absorbed. When the infrared light is absorbed, less is transmitted to the detector which is recorded. A plot of light absorbed versus wavelength from the sample is compared to samples of known species and concentrations to determine the level of concentrations in the exhaust gas sample. A higher concentration of carbon monoxide or dioxide in the air would cause increased absorbance of the infrared light. A Fourier Transform Infrared (FTIR) spectrometer employs the same principles of radiating light and measuring the absorbance. The difference from infrared spectroscopy is the use of a mirror to alter the wavelengths of the radiated light. The absorbance is recorded versus mirror position but is converted to be plotted against the wavelength by a Fourier transform of the data. Then the plots are compared to reference samples similarly to the infrared process. An advantage of the FTIR spectrometer is its ability able to detect many more species including hydrocarbons and formaldehyde. An FTIR analyzer is also used to determine more species in the exhaust stream.

Total hydrocarbon emissions are measured by flame ionization detection (FID). The exhaust gas to be sampled is mixed with hydrogen fuel and air which is burned in a small chamber. The burner serves a secondary purpose as a cathode while an electrode is located above the burner, which is the collector. Burning of the sample creates ions which are attracted by the electrode. When the ions contact the collector current is enabled to flow and be measured. With increased concentrations of hydrocarbons more ions are produced resulting in higher current rates. The ions created are proportional to the amount of carbon present which is why a fuel without carbon is used for continuous burning of the flame. This method does not distinguish where the carbon atom was derived from which is why the concentration is labeled as the total hydrocarbon emissions. Heated tubes are used to transport the samples from the exhaust stream to the analyzer to ensure the sample remains in the gas phase and does not deposit along the tube.

Chemi-luminescence detection is similar to FID because a reaction in a controlled chamber is used to create a product, which is detectable electronically. However, instead of ions being produced, light is emitted from the reaction and is measured by a photodetector. The reaction of ozone with NO produces NO_2 and O_2. NO_2 is in an exited state which emits a small amount of light. The light absorbed by the detector is proportional to the flow rate of NO_2 which is directly related to the NO concentration of the exhaust gas. To determine the amount of NO_2

produced by the engine, it has to be first converted to NO then undergo the reaction with ozone to form the excited state of NO_2.

Oxygen is measured by using its paramagnetic properties which means it can be attracted by magnetic fields. It can also be measured by its thermal conductivity or through chemical means. In the paramagnetic technique two spheres are filled with nitrogen on a balance. The exhaust gas is allowed to enter the chamber containing the nitrogen spheres. Then a magnetic field is imparted causing the oxygen in the exhaust sample around the balance to be attracted by the magnetic field imparting a torque on the spheres. A coil with feedback is used to measure the torque required to keep the suspensions assembly in place. The sphere is held in place by a shining a light of a mirror located in the center between the two spheres of nitrogen. The light is transmitted to a photodetector when it is in the zeroed position. The photodetector interacts with the coil as the feedback to keep the balance in place. The amount of oxygen present will increase the magnetic pull and increase the torque required from the coil.

The dilution tunnel in the lab is used for sampling particulate matter. After the exhaust travels through the exhaust manifold into the turbine it enters the exhaust piping routed to the exhaust stack. In the exhaust line, a probe is positioned in the center of the tube with it opening perpendicular to the flow as in a pitot tube. The probe is located 4 feet from any bends and routes the exhaust gases to the dilution tunnel via heated flexible tubing to ensure the particles do not deposit on the tubing before reaching the dilution tunnel. Clean air is pulled through a high efficiency particulate air filter before mixing with the exhaust gas. The clean air flow and exhaust flow are controlled by valves to control the dilution rate of the exhaust sample. The flow rates are measured for the exhaust by a venturi meter and the clean air by a turbine meter. The air and exhaust mixture are allowed to equilibrate in a residence chamber before being pulled through a filter assembly designed to remove all particles larger than 10 microns. The filter is weighed before and after the test with the difference being the amount of particulate matter collected over a 20-minute sampling time.

The baseline commissioning conditions were tested up to a total of six times at each load. The error bars presented in the figures are determined from the average and standard deviation over these trials. Multiple tests were not able to be performed for each of the substitution limit and extending substitution limit tests due to time constraints. The error bars for these plots are assumed to have the same uncertainty as the baseline commissioning under similar conditions.

4. Baseline Commissioning

An Eden Innovations Dual Fuel kit was procured for this work and installed on the 6.8 L Tier II John Deere diesel engine. Eden Innovations has adopted the term “substitution level”, which is the amount of diesel reduction, to communicate effectively with customers. That is, the substitution level corresponds to the amount of diesel reduction realized as natural gas is added via the dual fuel kit. This terminology will be used throughout this work and is defined as follows:

$$substitution\ \% = \left[\frac{\dot{m}_{diesel\ only} - \dot{m}_{diesel\ in\ dual\ fuel\ mode}}{\dot{m}_{diesel\ only}}\right]100 \tag{8}$$

The diesel flow at each load break point was measured first when the engine was running on diesel only. The engine was then brought to full load of 140kW electrical power. To determine the substitution level at a specific load point, the natural gas throttle position is started at 0% and incrementally opened manually waiting to achieve steady state operation every few percentage points of opening. The throttle has 100 discrete locations from 0-100% of opening. That is, the minimal adjustment is 1% of opening. While adjusting the throttle control valve for the natural gas, the diesel flow rate was monitored through DevX. This process continues up to the substitution limit before returning natural gas flow to an acceptable level. A physical limit sets the substitution for 100% load due to audible knock but the rest of the set points are not as directly limited. Figure 15 shows the baseline substitution levels as well as comparisons between diesel flow rates of dual fuel and diesel only operation.

Substitution at 75% load is primarily determined by governor control, such that the engine operates smoothly as load changes through this range. The low load test points are determined based on the minimum diesel flow. The value of diesel consumption at 1800 rpm with no load is arbitrarily designated as the minimum flow for cooling of the electronic injectors. With an increase in load, the substitution is set so the diesel flow is slightly above the previous load up to 50%. From a controls perspective this is required for smooth transitions between load settings. The controller directly controls the diesel flow to increase or decrease the power output while the natural gas throttle is reactionary.

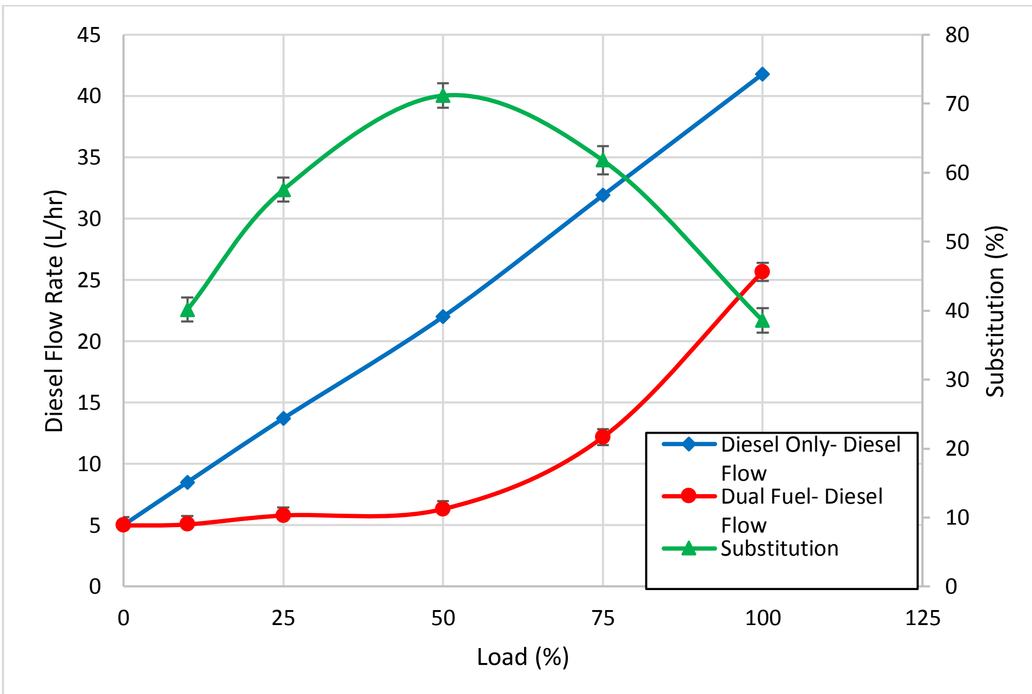

Figure 15: Diesel flow rates and substitution levels under baseline original commissioning conditions.

4.1 Combustion Analysis

After the initial commissioning, a full test was performed to evaluate combustion, performance and emissions characteristics. The in-cylinder pressure traces showed that dual fuel operation significantly raises peak pressure over diesel only operation. This is especially the case at 100% load as shown in Figure 16. This is the opposite result as what was found by [28]. Diesel and natural gas have different burning characteristics in internal combustion engines.

Diesel engines use electronic injectors that promote diffusion controlled burning while natural gas is a pre-mixed flame typically ignited by a spark plug. Dual fuel operation is a combination of both types of combustion processes. Dual fuel can be implemented in a couple of ways. Some large bore natural gas engines can use diesel as an ignition source as a replacement to a spark plug. The diesel injection supplied is under 2% of the energy supplied to the cylinder. In this case, the diesel is considered a pilot injection or micro-pilot.

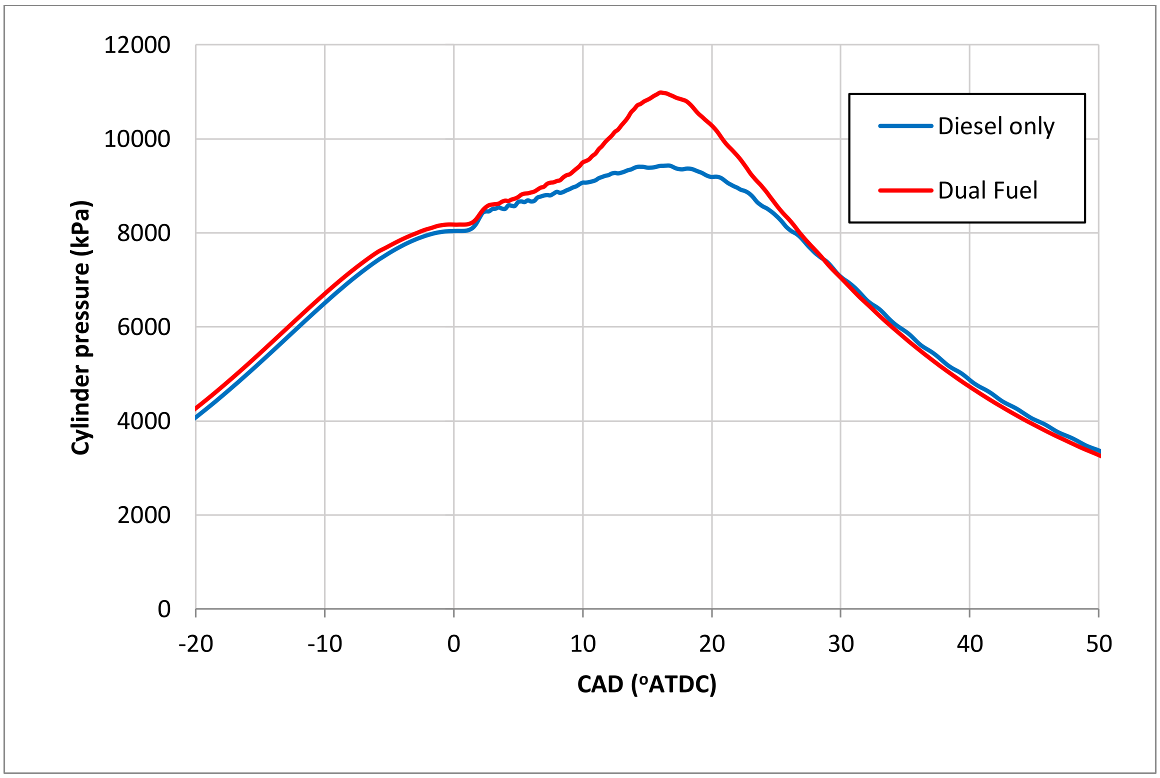

Figure 16: Cylinder pressure traces for diesel and dual fuel under baseline conditions at 100% load

On engines ignited by a micro-pilot, a propagating flame through the pre-mixed natural gas and air produces the majority of the energy. This is not representative of the method of burning observed in this work. As seen in Figure 17, somewhere between 30 and 65% of the energy is from combustion of diesel fuel. A proposed mechanism for this type of dual fuel combustion is the idea of multiple flames. There are simultaneous diffusion flames around the diesel jets and propagating pre-mixed flames. The increased pressure rise rate under dual fuel conditions is evidence of a propagating flame. The proposed idea of multiple flames implies an increased flame surface area allowing more reactions as evidenced by increased heat release rates.

The initial heat release rate profiles are nearly identical regardless of the operating mode. The ignition begins the same as normal diesel combustion. Some of the diesel jet evaporates until there is a pre-mixed zone surrounding the jet at an air-to-fuel ratio within the flammability limits. The pressure and temperature increase with compression to a point where auto-ignition of diesel can occur. This is consistent with the initial heat release rate being identical for both cases in Figure 18. After the initial energy release, the two processes differ significantly. Diesel operation has a decrease in heat release before ramping back up and maintaining a constant heat release rate for approximately 10 degrees of crank angle rotation. Dual fuel combustion does not have a decrease in the heat release rate and increases to a higher peak but does not maintain that level of energy release.

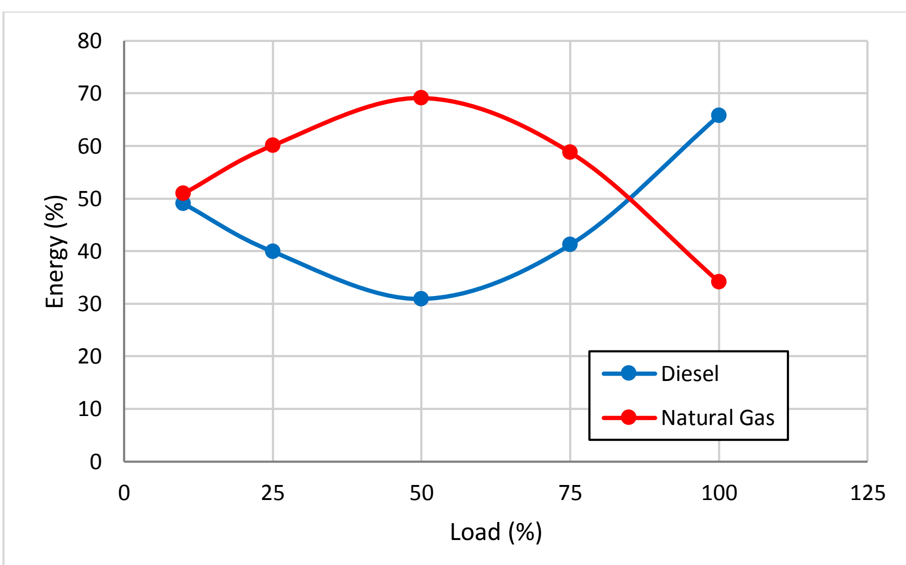

Figure 17: Percentage of energy supplied by diesel and by natural gas.

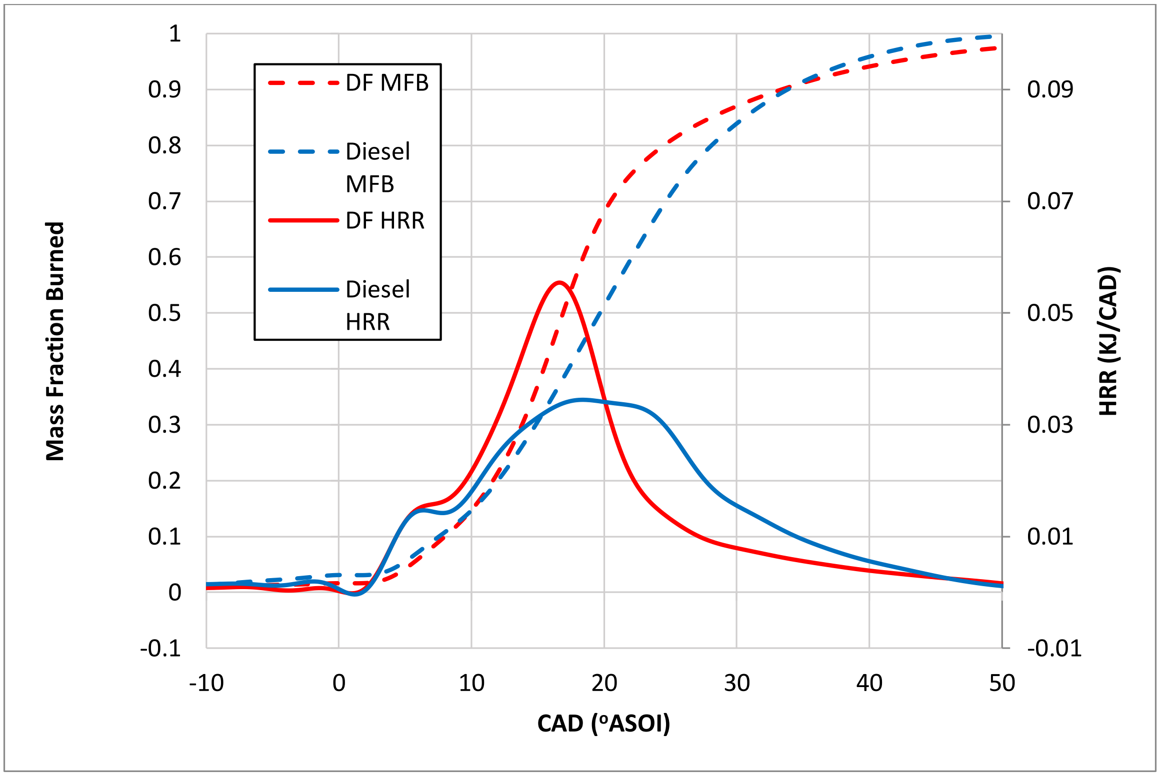

Figure 18: Burn curve and heat release rate for dual fuel and diesel only operation at 100% load. At 100% load, each condition had a start of injection timing of 3 degrees before top dead center.

The diesel fuel consumption is reduced to a rate similar to 50% load. A theoretical plot of the heat release rate being split into separate natural gas and diesel contributions is presented in Figure 19. The diesel heat release trace is assumed to be the same as the 50% load heat release case because the diesel consumption is reduced by shortening the injection duration. The portion of heat release associated with natural gas is assumed to be the difference between the heat release of the dual fuel case at 100%, which is measured, and the heat release rate of diesel only operation at 50% load. Natural gas starts to add energy to the process after the diesel reaches its first peak. It may start to add energy just before the first peak and increase the height of the first peak. This is consistent with the dual fuel case mass fraction burn curve leading the diesel only profile. The slow final portion of the dual fuel mass fraction burn curve is likely due to unburned hydrocarbons undergoing late reactions. The 10% burn and 90% burn locations match closely despite the differences in between 10 and 90% burn. Diesel burns at a slower rate in the middle portion of combustion. This is likely due to the diesel fuel beginning in the liquid phase. The liquid fuel evaporates before it is able to combust compared with natural gas already being in the gaseous phase and pre-mixed with air.

At 75% load, the pressure traces are similar to 100% load in that the dual fuel peak pressure is greatly increased. A difference between 100% and 75% load is caused by the start of injection timing. The ECU determines the start of injection timing based on diesel flow and speed. The engine is running as a generator at a constant speed of 1800 rpm but as the substitution level is increased the diesel flow is reduced. In general, a reduction of diesel flow advances the injection timing. At 100% load, the SOI timings are the same for dual fuel and

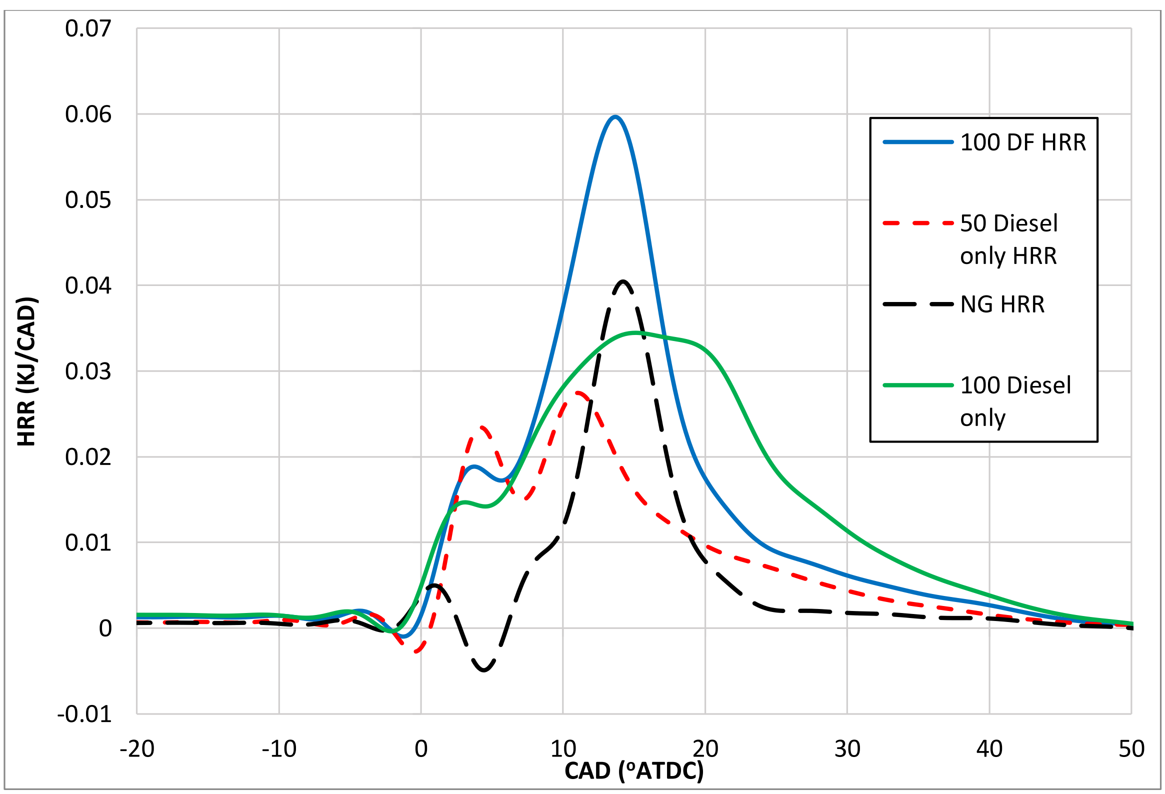

Figure 19: A theoretical heat release profile split into diesel and natural gas contributions

diesel only operation but at 75% load the dual fuel SOI is advanced resulting in the start of combustion occurring closer to top dead center compared to diesel operation. This is shown by the pressure trace from dual fuel operation shown in Figure 20. The conditions at 75% load enable a flame to propagate through the pre-mixed fuel.

The start of injection (SOI) timing for diesel only operation was 2.95 ^{o}btdc while for dual fuel operation the SOI was 5.6 ^{o}btdc. The heat release rate and burn curves are plotted versus crank angle degrees after SOI in Figure 21. The initial heat release occurs around the same crank angle degrees after SOI but with dual fuel reaching a higher initial peak. Similar to 100% load, the maximum peak of the heat release rate is higher than the diesel only max peak but then falls at a faster rate. Even with the slower late reactions, the dual fuel combustion reaches 90% mass fraction burned before diesel.

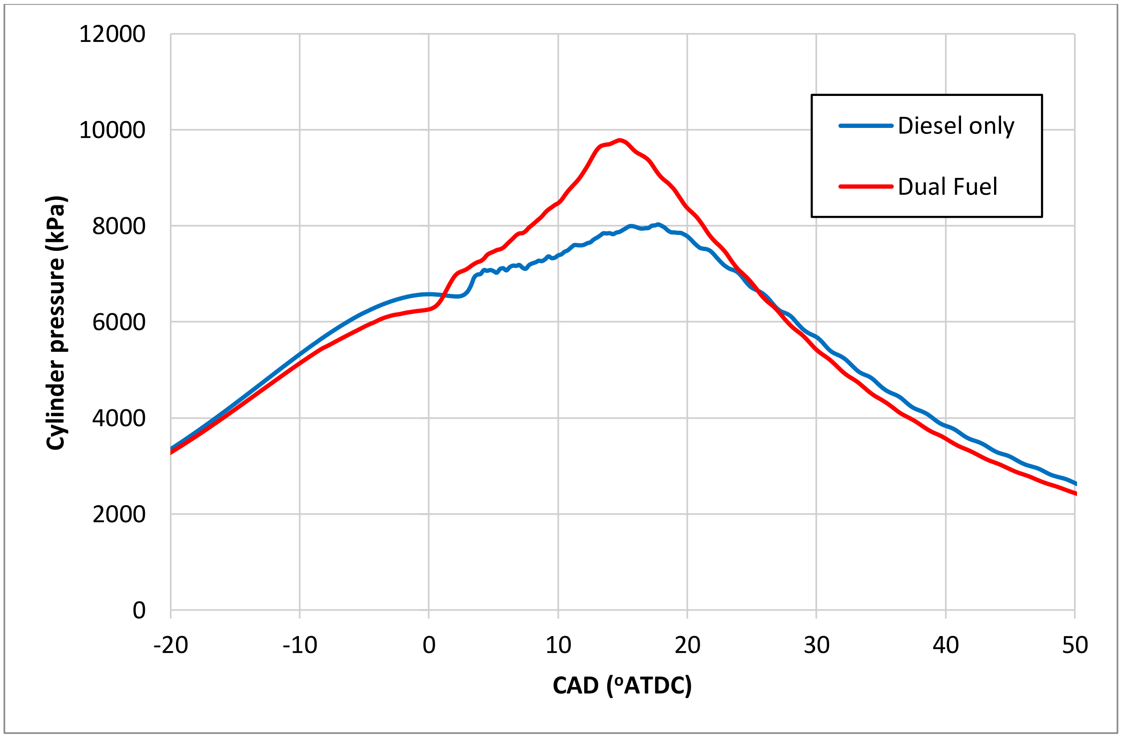

Figure 20: In-cylinder pressure traces of dual fuel and diesel only operation at 75% load. Without changes to the ECU the diesel start of injection timing for the dual fuel mode was advanced by a few degrees.

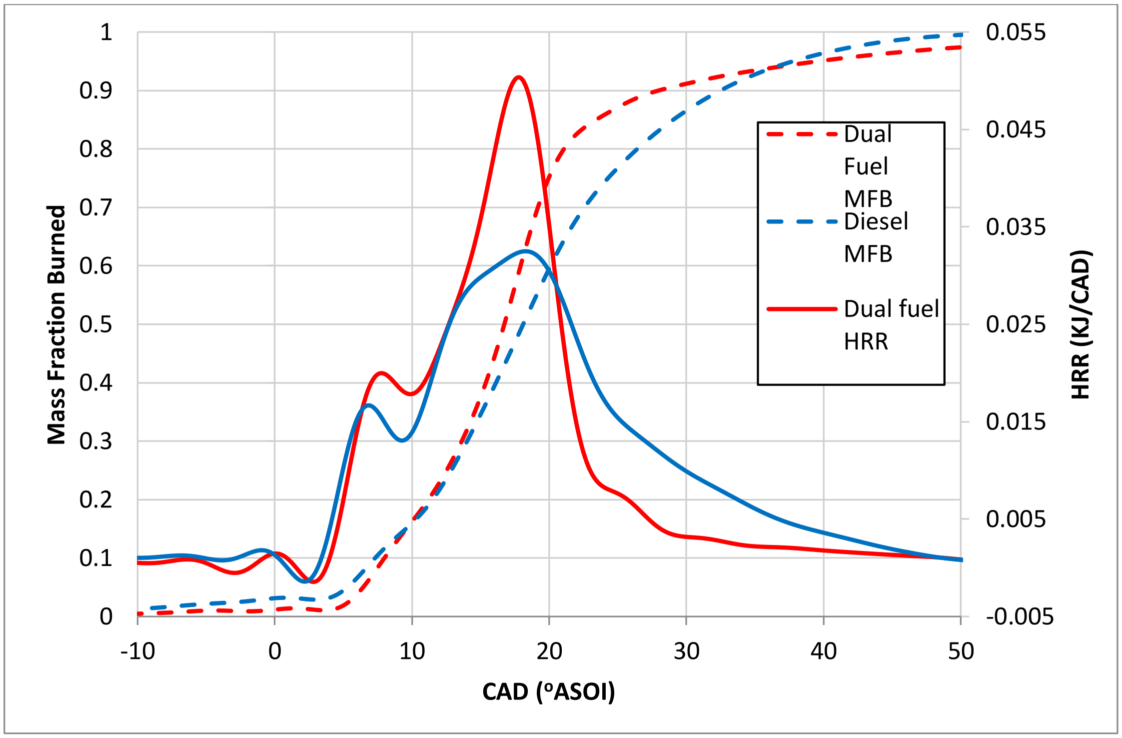

Figure 21: Mass fraction burned and heat release rate curves for dual fuel and diesel only operation at 75% load plotted versus degrees after SOI.

Collectively, 75 and 100% power outputs are termed high load set points that demonstrate similar behavior. The dual fuel heat release trace have good agreement with the theoretical heat release proposed by Hockett and Konigsson [34] [26]. It is proposed that at both of these conditions a flame is actually propagating through the natural gas. The ignition delay is primarily dependent on the diesel vaporization. The heat release trace shows characteristics of pre-mixed combustion. The rate of pre-mixed combustion is dependent on combustion chemistry while diffusion combustion rate is primarily dependent on the dynamics of transport and evaporation. After ignition, the dual fuel heat release has a pre-mixed combustion shape of a single peak, which quickly rises and falls. This is a major difference from diesel only combustion. Even when the energy release is reaching the final stages of combustion it is still more sustained when only diesel is burning.

A transition in the characteristics of combustion occurs at 50% load. Despite having the highest substitution of natural gas for diesel, 50% load has a pressure trace that resembles the diesel pressure much closer than the higher load tests. The pressure traces are presented in Figure 22. The peak pressure is rounded rather than a sharp peak and only a small percent higher than the diesel only peak pressure. The substitution causes earlier SOI timing. As a result, the initial pressure rise occurs closer to top dead center in a manor also observed at 75% load. The difference between 75 and 50% load is that for the 50% load case the pressure rises at the same rate as the diesel only conditions. The lower combustion temperatures generated at 50% load slow the reaction rates.

The heat release rates and mass fraction burned curves are plotted versus crank angle degrees after SOI in Figure 23. For high loads the initial release of energy is nearly identical in timing. However, at 50% load the dual fuel case energy follows the diesel energy release but lags behind a few crank angle degrees. This delay is likely due to the decreased rail pressure during dual fuel operation. A higher rail pressure promotes better atomization of the liquid fuel allowing faster evaporation. At 50% load the rail pressure is 72 MPa for dual fuel and 103MPa for diesel only. The largest difference between the two heat release profiles is in the second peak. The second peak is higher for diesel while the peak for dual fuel maintains a fairly constant heat release rate for approximately 10 crank angle degrees. A possible reason for the second peak of the heat release occurring later is the lower combustion temperatures causing a longer ignition delay of the natural gas. The burn curve of diesel leads the burn curve of dual fuel for the entire duration of the combustion event.

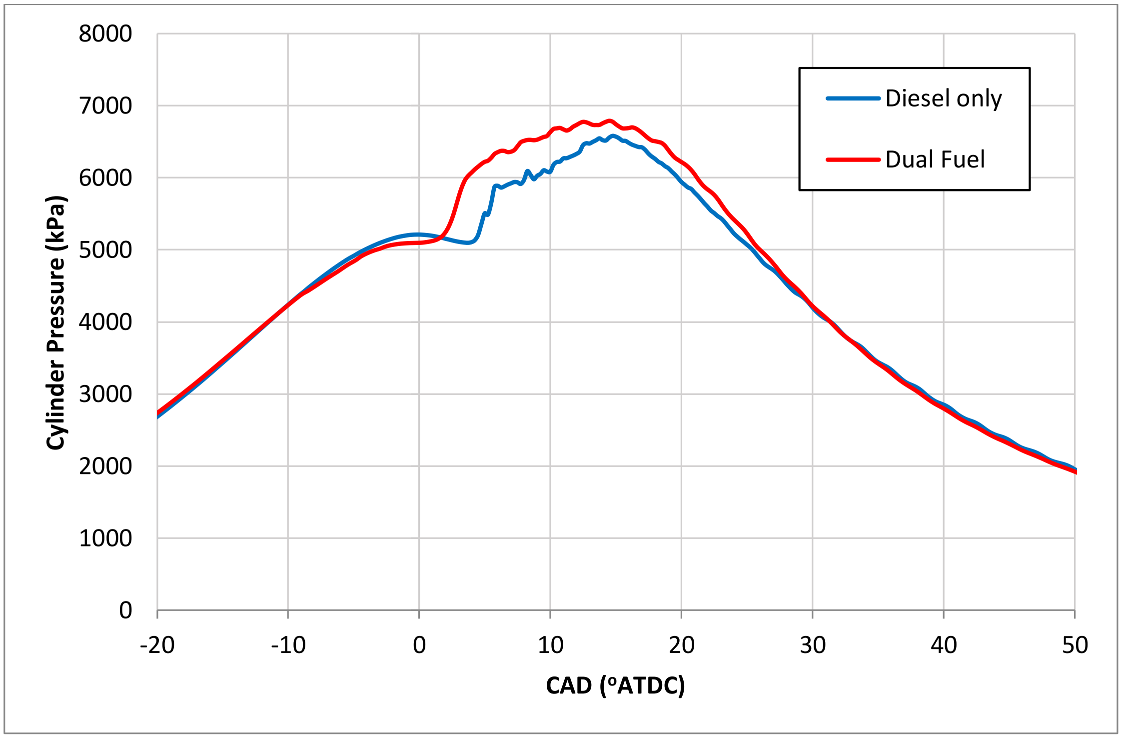

Figure 22: Average in-cylinder pressure traces for dual fuel and diesel only at 50% load.

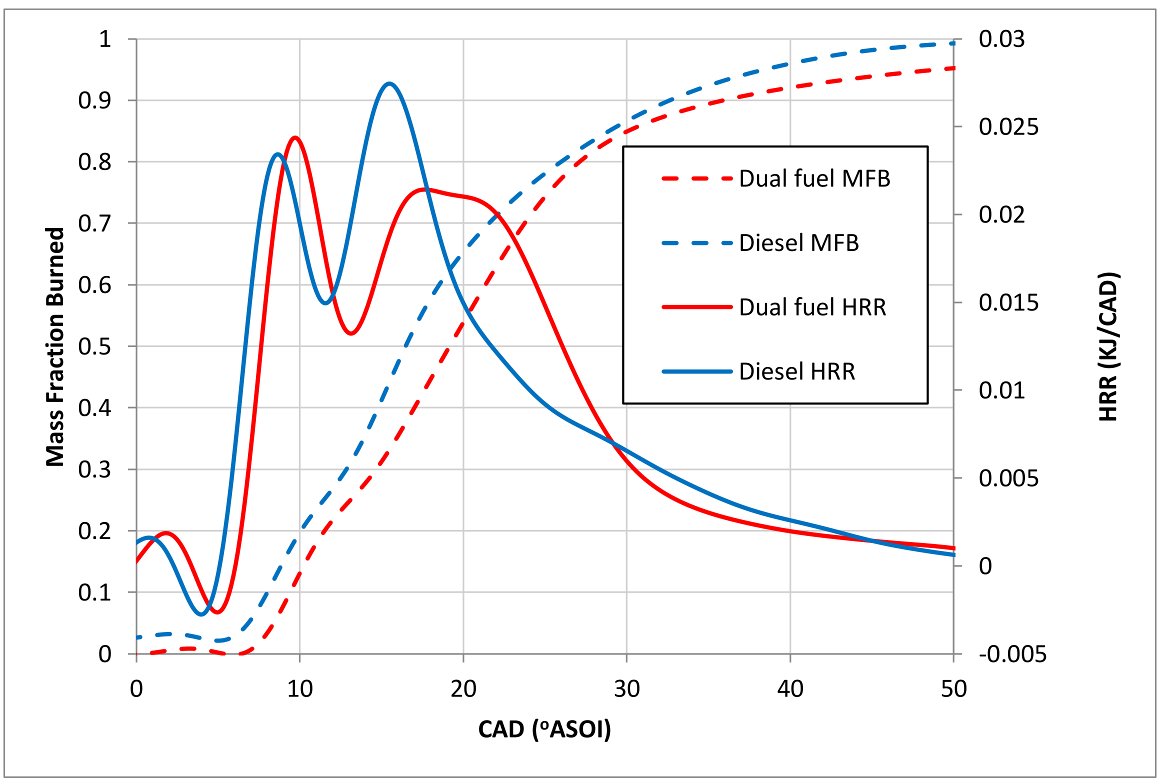

Figure 23: Heat release rates and mass fraction burned curves for 50% load for dual fuel and diesel only.

Both of the low load cases display similar characteristics. The pressure traces, heat release rates and mass fraction burned curves for 25 and 10% load are presented in Figures 24-27. Dual fuel operation adds natural gas before the turbocharger. The addition of natural gas lowers the ratio of specific heats of the air and fuel mixture. The motoring pressures for dual fuel operation should be lower because of this. However, the dual fuel and diesel baseline tests were performed on different days. An exhaust leak or intake leak changed the boost levels for the two tests. The dual fuel tests were performed first with normal boost levels, which is why the dual fuel cases have higher motoring pressures in the low load cases presented.

Apart from different pressures during the compression stroke, the pressure traces for diesel only and dual fuel operation at low loads closely resemble each other. There is an ignition delay where the pressure decreases from the maximum from motoring then rises sharply to the

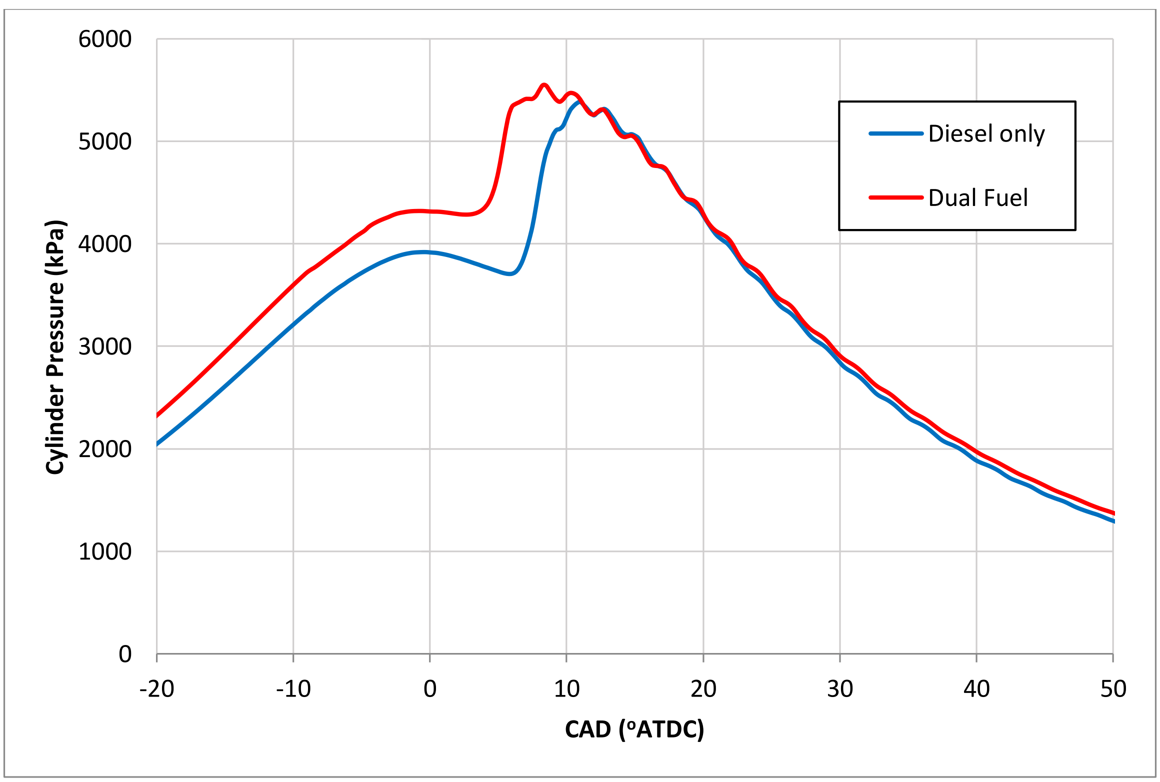

Figure 24: 25% load average in-cylinder pressure traces fod dual fuel and diesel only operation.

peak. The heat release profiles are also similar in shape. The longer ignition delays in the diesel cases are likely due to the lower cylinder pressures. A longer ignition delay typically results in a higher peak of energy released during the pre-mixed combustion phase as there was more time for preliminary reactions to form radicals. Each case has some energy release after the main peak of the heat release rate.

The mass fraction burned data are summarized in Figures 28 and 29. The SOI timing for diesel and dual fuel is summarized in Figure 30. For most of the conditions tested, dual fuel operation creates conditions with an earlier injection timing than when only diesel is being burned. This causes the 10% mass fraction burned location to be earlier for dual fuel. Generally, the 0-10% burn duration is longer for dual fuel operation though. At low loads the burn durations are longer for dual fuel operation. At 50% load, the burn locations are closely matched. At high loads the combustion process is faster.

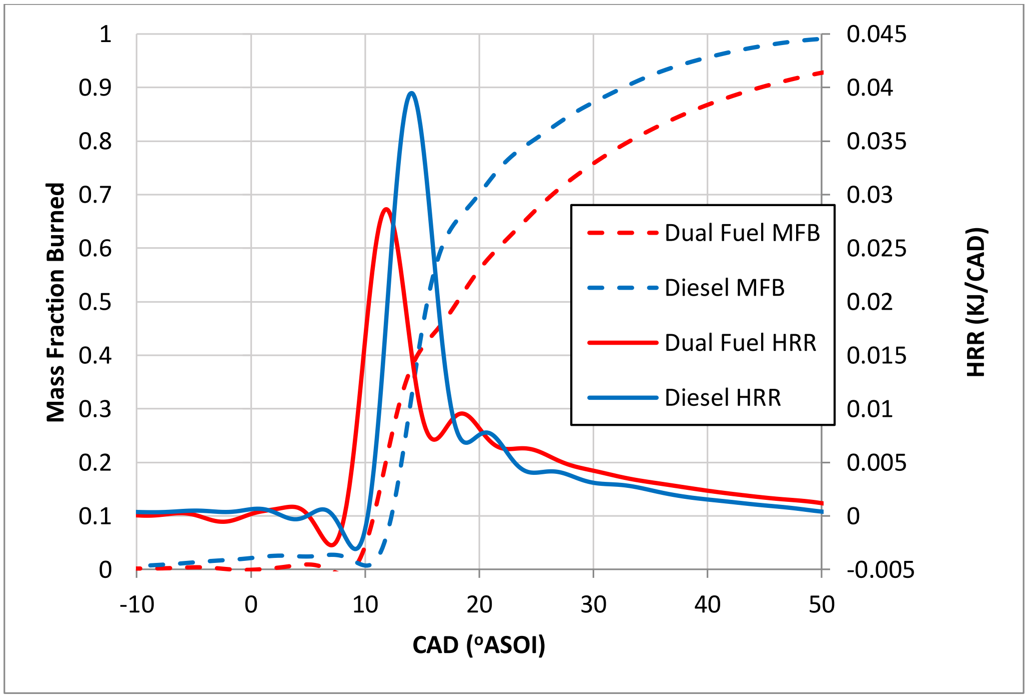

Figure 25: 25% load average heat release rates and mass fraction burned curves for dual fuel and diesel only operation.

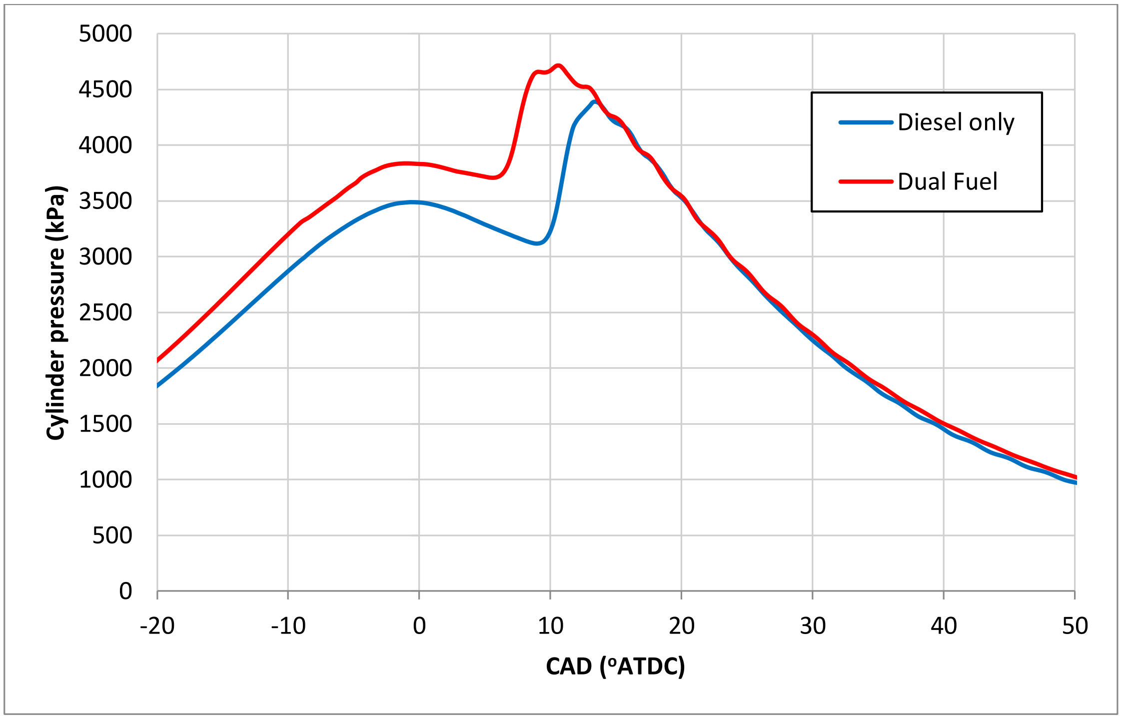

Figure 26: 10% load average in-cylinder pressure traces under for dual fuel and diesel only operation.

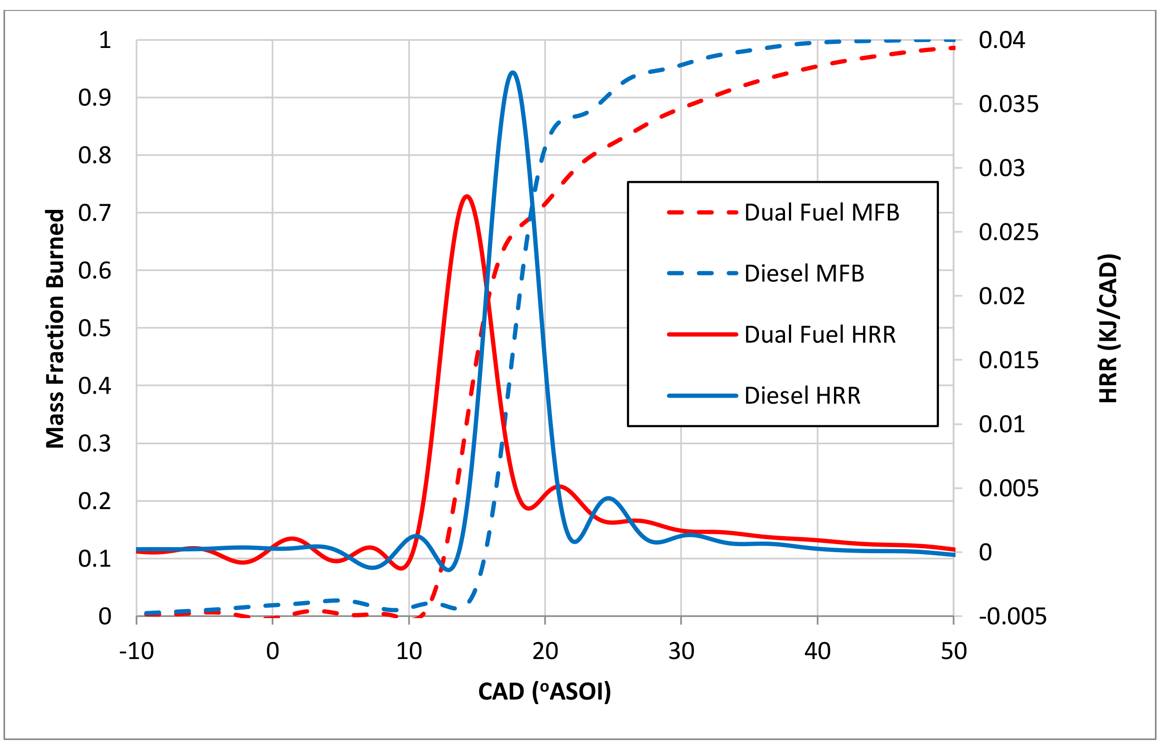

Figure 27: 10% load heat release rates and mass fraction burned curves for dual fuel and diesel only operation.

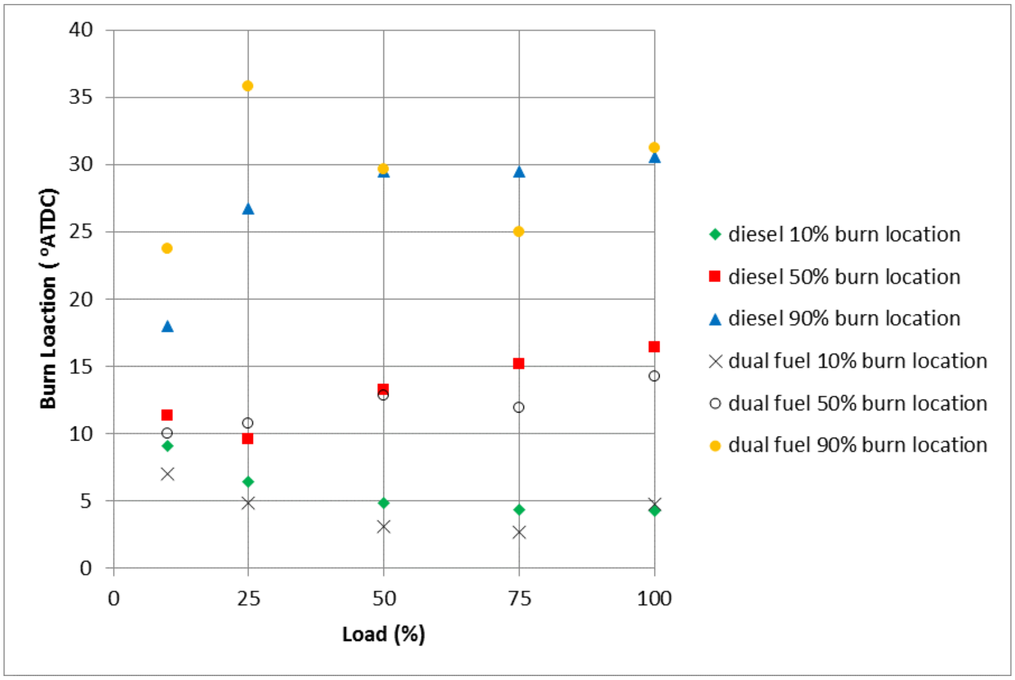

Figure 28:Burn locations for 10, 50 and 90% mass fraction burned for dual fuel and diesel only operation.

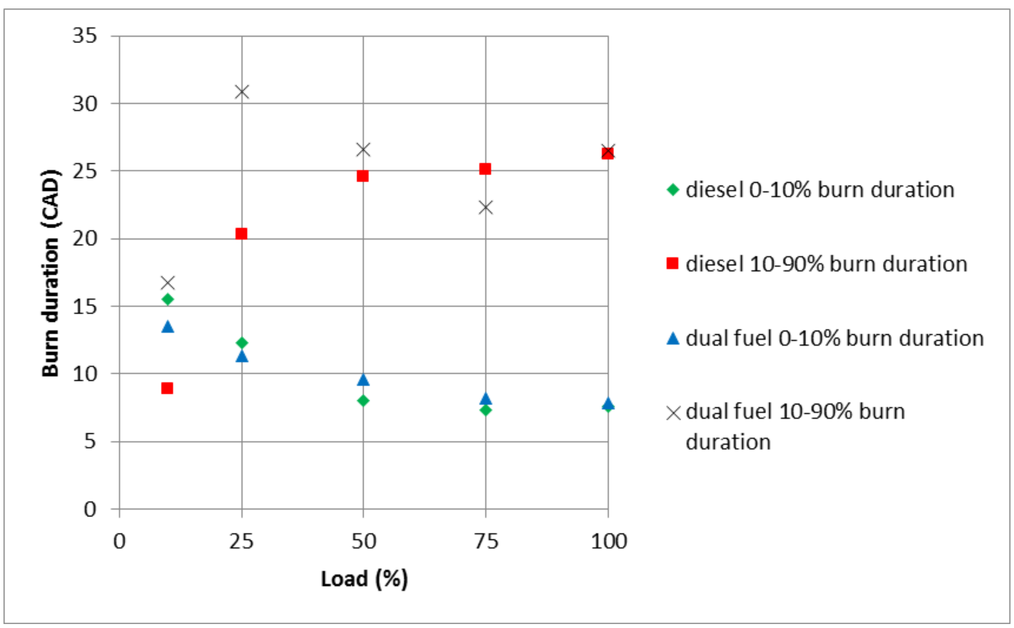

Figure 29: Burn durations for 0-10 and 10-90% mass fraction burned for dual fuel and diesel only operation.

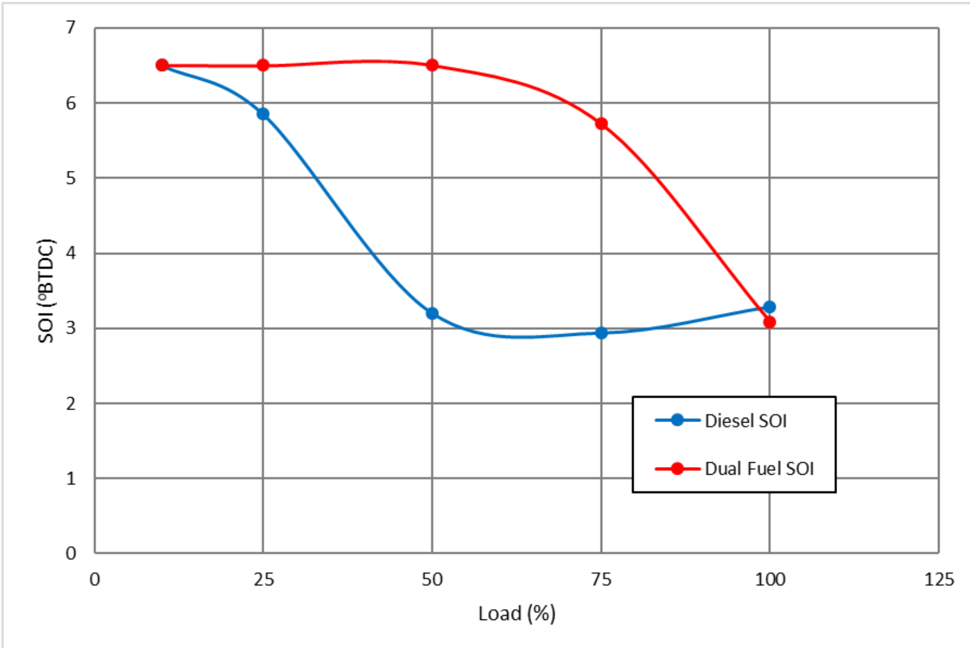

Figure 30: SOI timing of dual fuel and diesel only operation with stock ECU settings.

The dual fuel combustion events observed can be categorized in three categories. The three categories are based on the quality of natural utilization not substitution. The categories are defined as follows: 1) flame propagation through natural gas and large majority of gas used, 2) flame propagation through natural gas but flame quenching and 3) no flame propagation through natural gas. This can also be generally split between high loads, 50% load and low loads. At high loads the pressure and heat release profiles show signs of pre-mixed flame propagation. As discussed previously, it is believed there are diffusion flames around the diesel jets and pre-mixed flames simultaneously burning. The high load cases have significantly higher peak pressures because of the increased flame surface area. Dual fuel operation results in higher peak pressures than diesel only operation. The transition point is at 50% load. The substitution level is a maximum at 50% load but the peak pressure is marginally higher than diesel only operation. Under these conditions a pre-mixed flame is likely still propagating but at slower rates and may be quenching before all of the pre-mixed fuel is utilized.

Lower pressures and temperatures compared to the high loads slow the chemical kinetics and could be a cause of the peak pressures not increasing relative to diesel. Later in this section the emissions will be analyzed, which will provide evidence supporting pre-mixed flame propagation. The emissions of THC at 50% load increase compared to high loads, which suggests late quenching leaving some of the pre-mixed fuel unutilized. At low loads it is unlikely a flame propagates completely through the pre-mixed fuel and air. A possible method for the dual fuel operation at low loads is primarily diesel is being burned and only the natural gas immediately surrounding the jet undergoes oxidation reactions. The peak pressures are similar for either mode of operation. As air is being mixed around the diesel jet, natural gas is also transported with it and introduced to the diffusion flame front. The swirl levels thus become important at low loads to utilize the pre-mixed fuel. It is also possible that some flame propagation occurs through the natural gas at low loads but this is less likely based on the very low equivalence ratios of the pre-mixed fuel as shown in Figure 31.

For all load levels dual fuel combustion is consistent. Figure 32 shows the peak pressures for each load and the coefficient of peak pressure. The coefficient of variation of the peak pressure is in the range of 1.8- 2.3%. The diesel fuel properties control the ignition delay so even at low loads and low equivalence ratios there is a low likelihood of misfiring. For each region, the natural gas burns similarly each cycle. At high loads the pre-mixed flame is propagating consistently each cycle. The low COV shows combustion does not transition from flame propagation to flame quenching. The low load cases indicate there is no alternating between flame quenching and flame propagation, which is common near the lean limit for natural gas spark ignition engines. Any oxidation of premixed natural gas occurs consistently from cycle-to-cycle. At the lowest loads, the COV of peak pressure for dual fuel is even better than diesel only.

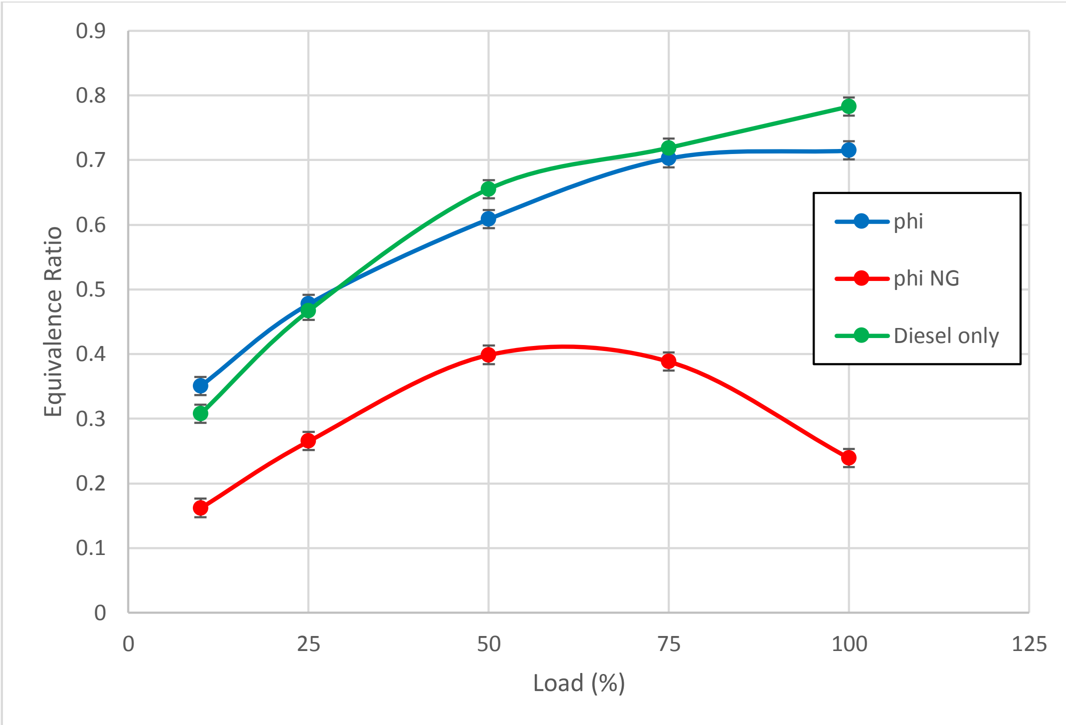

Figure 31: Overall and natural gas equivalence ratios of the baseline commissioning.

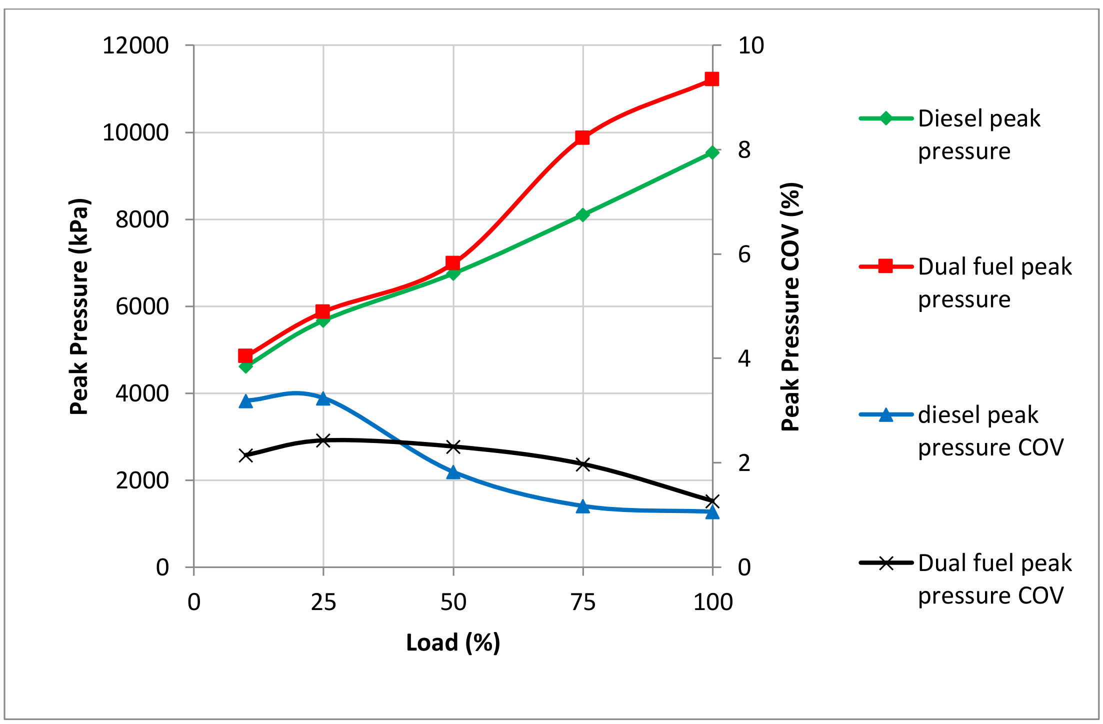

Figure 32: Peak pressure and COV of peak pressure of the baseline commissioning.

The location of peak pressure and COV of location of peak pressure for dual fuel and diesel only operation are presented in Figure 33. The location peak pressure of dual fuel operation leads dual fuel only operation on a degree atdc basis. This is most likely due to earlier start of injection. The heat release rate is also faster at high loading conditions causing earlier peak pressures. The location of peak pressure COV at 50% increases over diesel only. A possible reason for this is the flame quenching of the premixed natural gas flame. At 10 and 25% load it is unlikely a flame can propagate through the lean natural gas and air mixture. At 50% load the natural gas is near the flammability limit enabling flame propagation on some cycles but being quenched during the next cycle.

indicated mean effective pressure (IMEP) is determined from the pressure volume work not including friction losses. The calculated IMEP from in-cylinder pressure data is presented in Figure 33. Although the pressure traces discussed previously appear to produce different work per cycle values for the same load, for each load point the engine produced the same IMEP regardless in dual fuel or diesel only mode.

The coefficient of variation of the IMEP is low for dual fuel. Figure 34 shows it only significantly differs from diesel only operation at 25% load. At 25% load the COV is under 3% signifying repeatable power produced from the combustion event. A COV of 10% is typically considered a high value for misfiring or unstable combustion is occurring. Some inconsistency in combustion is observed at 50 and 25% loads compared to diesel only. At 50% load the peak pressure location varies more than any other load while 25% load has the highest COV of IMEP. It is proposed flame quenching is beginning at 50% load. Variance of flame quenching could cause the location of peak pressure to change cycle to cycle.

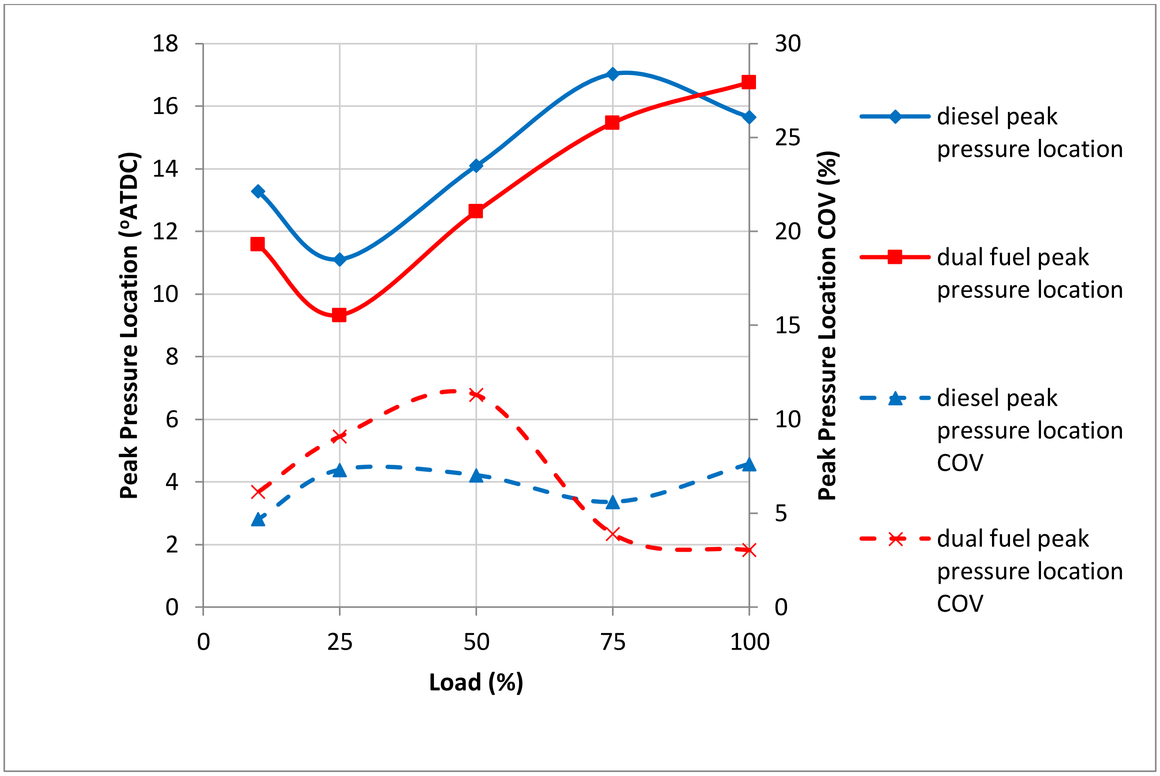

Figure 33: Peak Pressure location and COV of peak pressure location

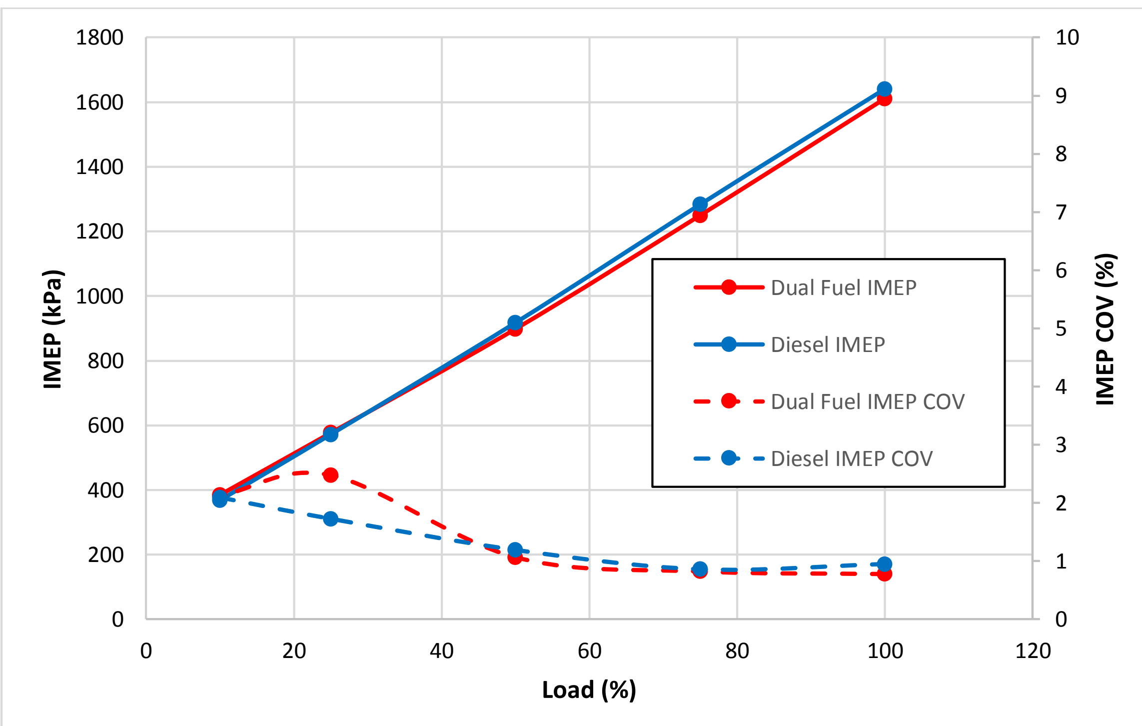

Figure 34: IMEP and COV of IMEP for diesel and dual fuel operation.

4.2 Pollutant Emissions

Combustion performance analysis indicates dual fuel is a suitable alternative to diesel only operation for field uses. The combustion is consistent and repeatable in producing power even at high substitution levels. The substitution levels achieved will provide significant cost reductions for fueling engines. Dual fuel operation can even lower the brake specific fuel consumption at high loads. Brake specific fuel consumption (BSFC) and brake efficiency are presented in Figures 35 and 36, respectively. The brake specific fuel consumption is the total fuel consumed including both diesel and natural gas on a mass basis normalized by power output. At high loads when pre-mixed flame propagation is occurring the brake specific fuel consumption is less than diesel only operation. However, poor utilization of natural gas at low loads worsens the fuel consumption. Natural gas and diesel have similar lower heating values so it is reasonable for the dual fuel mode to have reduced fuel consumption and higher efficiencies.

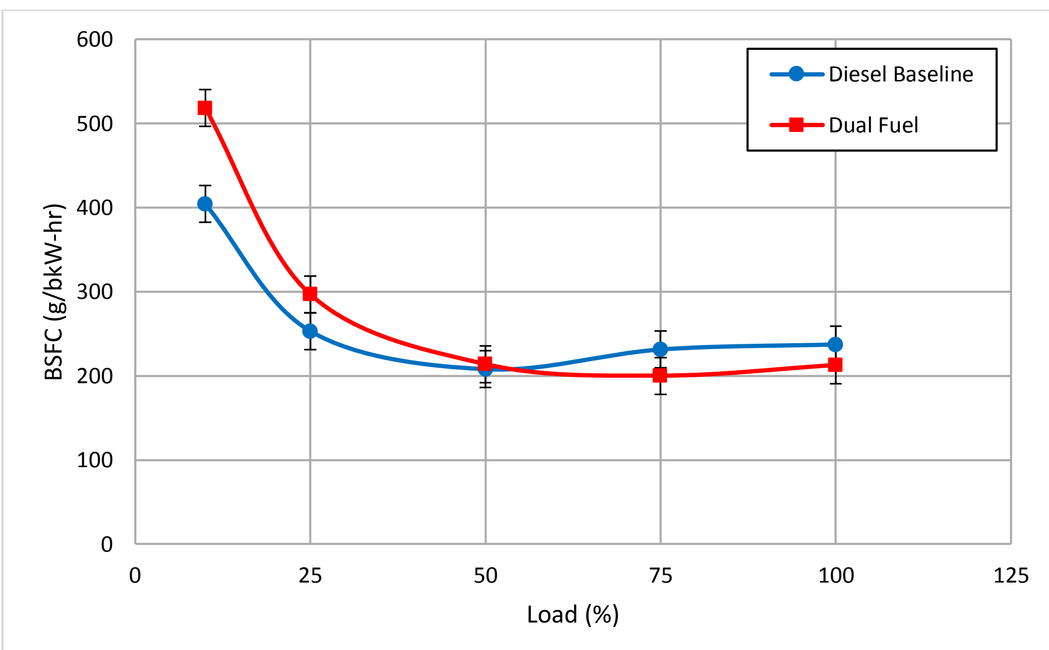

Figure 35: Brake specific fuel consumption for baseline commissining and diesel only operation..

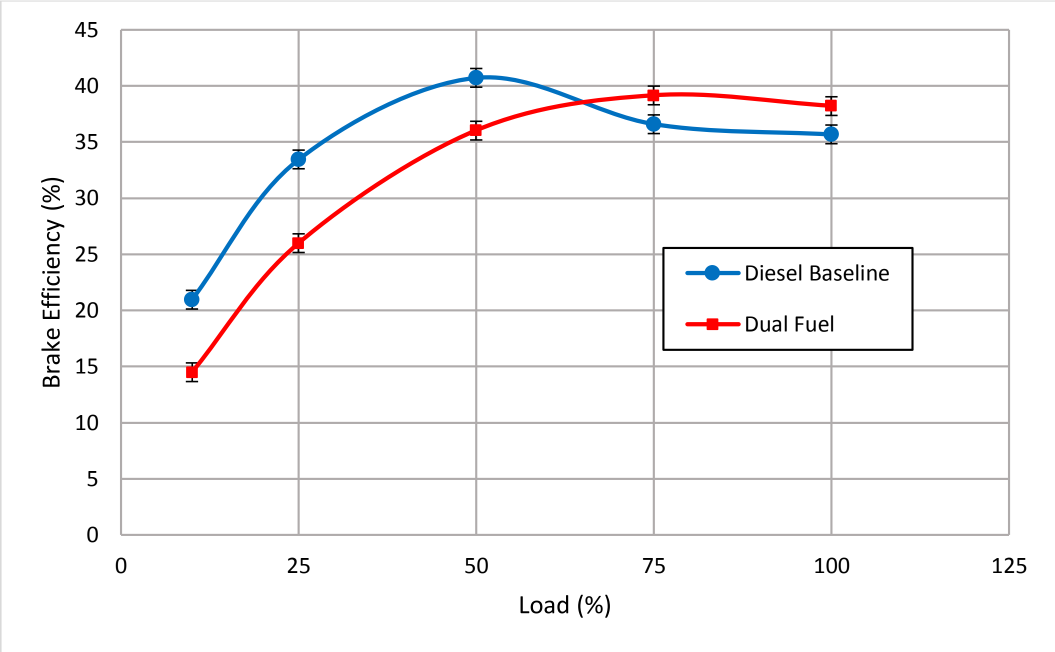

Figure 36: Brake efficiency for baseline commissioning and diesel only operation.

The most substantial weakness of operating in dual fuel mode is the increase in pollutant emissions levels, especially unburned hydrocarbons. Hydrocarbon (HC) emissions for diesel only and dual fuel are shown in Figure 37. Hydrocarbon emissions from diesel only operation are small across the load map. However, dual fuel operation emits large quantities of hydrocarbons, primarily at low load conditions. A calculation was performed examine the natural gas supplied that was not being burned. The HC levels of diesel only operation are subtracted from the HC levels of dual fuel operation. This was converted from a mole fraction to a mass flow rate. The incoming natural gas was converted into a mass flow rate of hydrocarbons by normalizing the constituents for inert gases present in the fuel gas such as small amounts of carbon dioxide and nitrogen. The result is the hydrocarbons emitted in the exhaust as a percentage of natural gas fuel supplied. The results are plotted in Figure 38.

At high loads only 2% of fuel hydrocarbons are not burned. This is similar to HC emissions from spark ignited engines [36]. The hydrocarbon emissions at high loads are likely due to typical hydrocarbon emission mechanisms from premixed spark ignition engines such as crevice volumes and absorption by oil on the cylinder walls. At 50% load most of the natural gas is burned but the amount of natural gas not used is starting to increase. Again, this is evidence that a flame is propagating but experiencing late quenching. The hydrocarbon emissions rise sharply as load is reduced from 50%. At 10% load the unburned natural gas is nearly 20% of hydrocarbons supplied by the fuel gas. There are two possible reasons for the amount of unburned natural gas. The first possible reason is that the natural gas is too lean to propagate a flame. Only the natural gas that is directly surrounding the diesel fuel spray burns. As in normal diesel combustion, which is dependent on mixing to introduce more air to the diffusion flame front, dual fuel combustion is dependent on mixing to introduce air and natural gas to the diffusion flame front. The rail pressure is decreased from 71 MPa at 25% load to 67 MPa at 10% load. The penetration of the diesel spray jet is reduced because of the lower rail pressure. The second possible reason is the natural gas flame is propagating but is being quenched quickly leaving large quantities of natural gas unreacted. The first reason appears more likely since combustion instability (e.g. COV of peak pressure and COV of IMEP) does not significantly increase at low load.

The carbon monoxide emissions follow the same trend as the hydrocarbons for dual fuel combustion. Figure 39 shows at every load they are higher than diesel only operation but extremely high levels at low loads. CO emissions are a result of incomplete combustion. CO is a required product before the formation of CO_2. Overall, the engine is running lean so there is oxygen available for the final oxidation reaction. The high levels of CO are because of time considerations not availability.

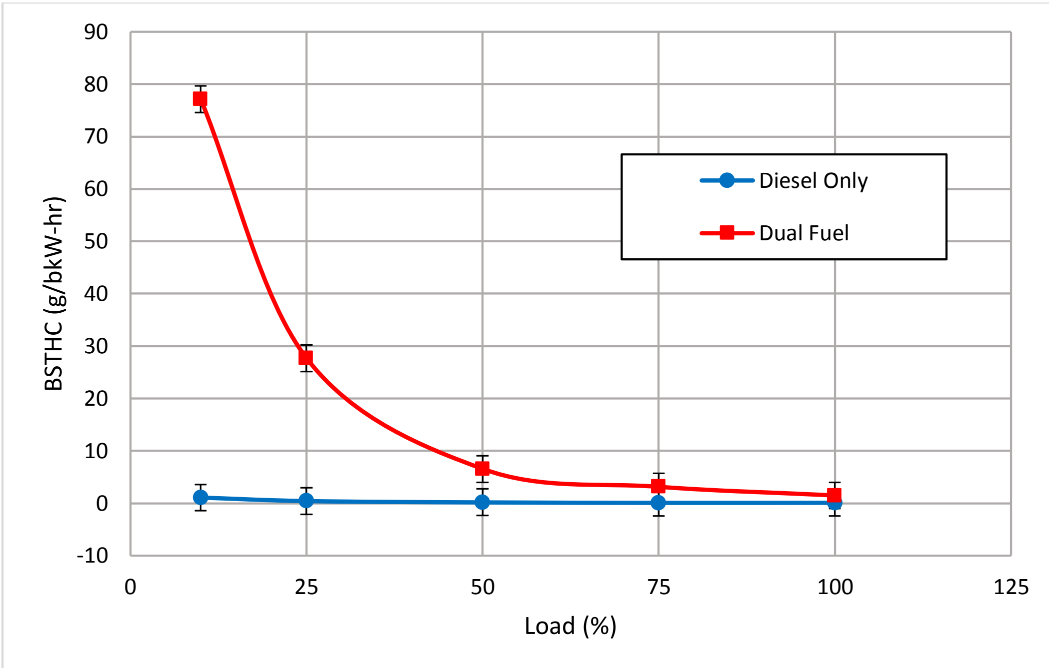

Figure 37: Brake specific total hysrocarbon emissions versus load for diesel only and dual fuel operation.

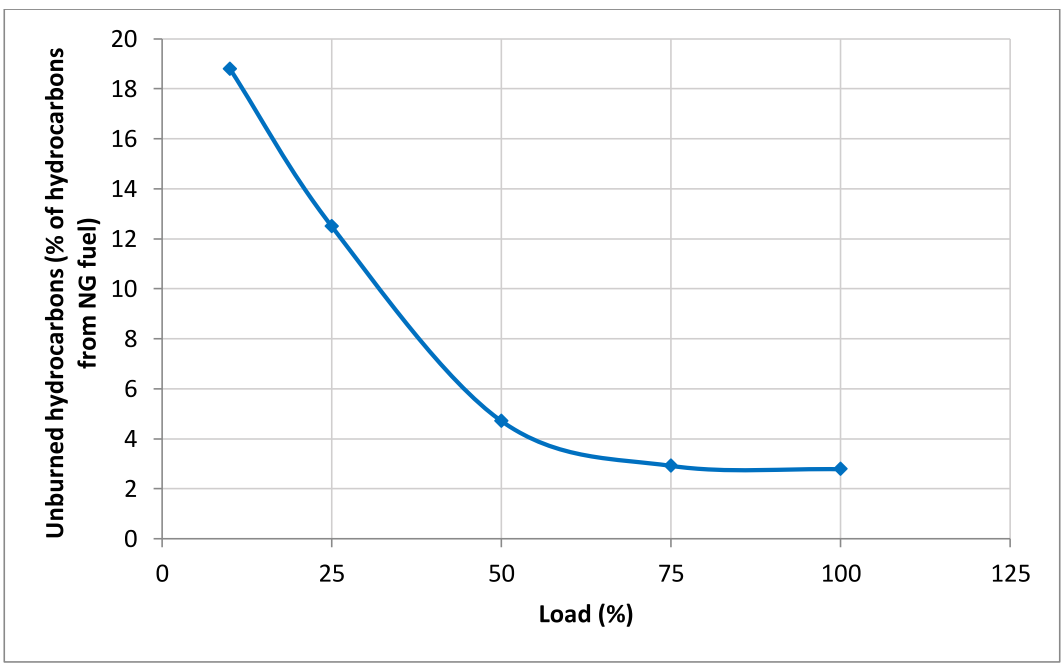

Figure 38: Unburned hydrocarbon emissions as a percentage of supplied hydrocarbons from gas fuel

Previous literature is mixed on the effect of dual fuel combustion on NOx emissions. Some studies show NOx emissions are reduced compared to diesel [23] [28] while others show

NOx are increased [27] [34] [25]. At low loads, the data collected from this experiment show reduced NOx emissions but at 50% load and above the emissions levels are nearly the same as diesel only operation. The NOx emissions levels are presented in Figure 40. NOx emissions are exponentially dependent on temperature. The combustion pressures and temperatures are high under dual fuel operation resulting in similar NOx emissions. The overall conditions are also lean and favor NOx formation since there is excess oxygen available to react with Nitrogen.

NOx levels are reduced to some degree at low loads but the most substantial reduction in emissions from dual fuel operation is from particulate matter. The particulate matter emissions are presented in Figure 41. It is widely accepted that particulate matter formation occurs in fuel rich zones with a long resonance time. Diesel sprays zones are fuel rich zones that allow hydrocarbons to rearrange forming large carbon rings and progressing to soot.

The primary way for reducing diesel flow rates is shortening the diesel injection time, admitting less fuel to the cylinder. This reduces the volume of the diesel spray jets where most of the particulate matter forms. If this were used on a higher Tier engine that used a diesel particulate filter (DPF), dual fuel mode would be a benefit. DPF's require a regeneration to unclog the filter so as not to cause too much back pressure in the exhaust manifold. There are different methods for this but they typically involve using some of the fuel to burn the rest of the particulate matter. Using fuel for regeneration has adverse effects on the fuel economy. Dual fuel operation would need to regenerate the DPF less often so the fuel economy would be improved.

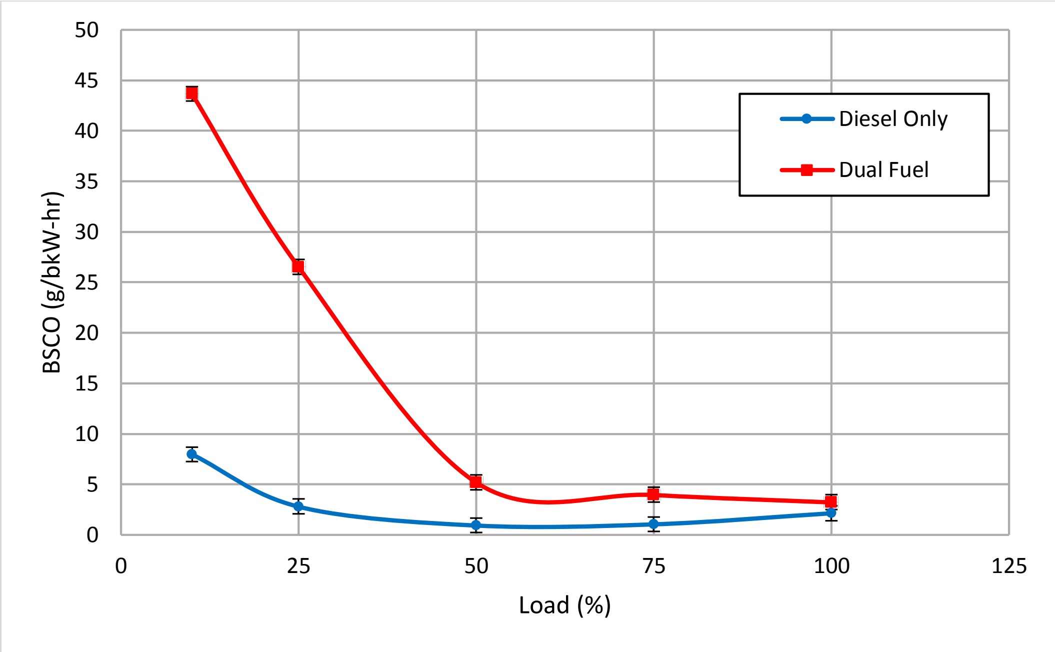

Figure 39: BSCO emissions for dual fuel and diesel only across the load range

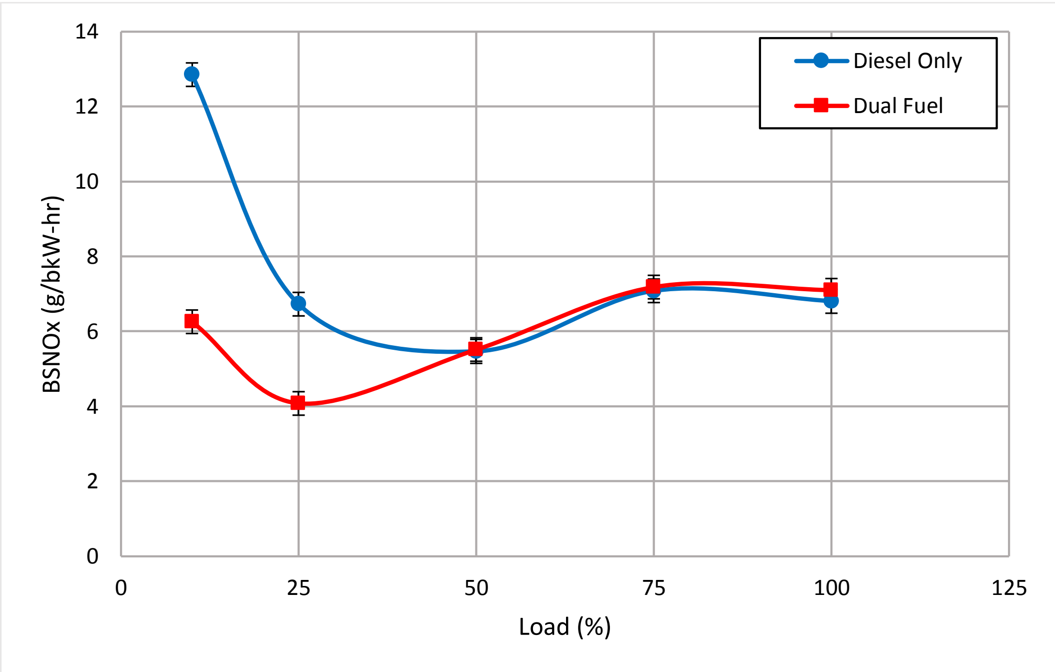

Figure 40: Brake specific NOx emissions for dual fuel and diesel baseline operation over the operating range

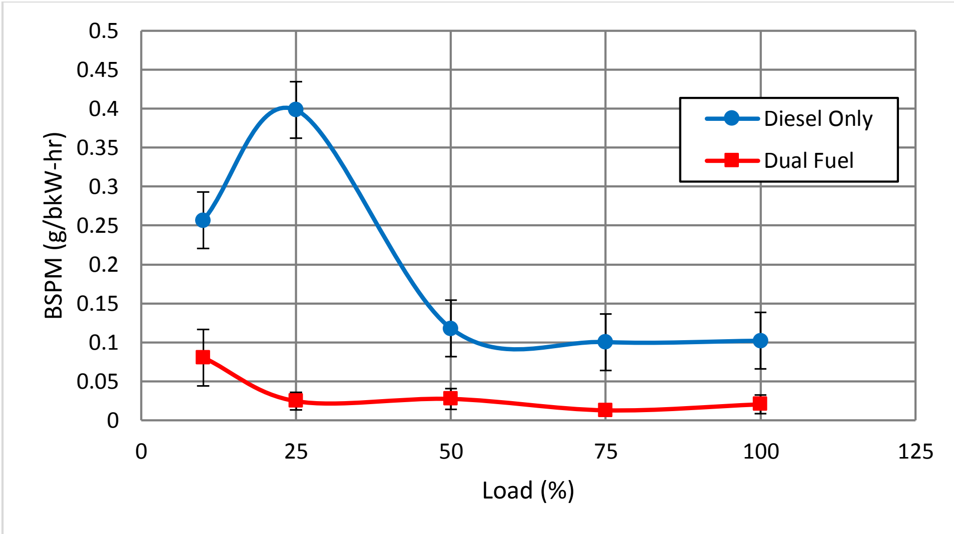

Figure 41: Brake specific particulate matter emissions for dual fuel and diesel only operation.

Arguably, the most important emissions assessment is comparison to the applicable EPA standards. For a dual fuel retrofit the engine is still required to meet the diesel engine permit level. In this case the engine is a Tier II model so the dual fuel mode must meet Tier II emissions levels, evaluated based on an ISO 8178 test cycle. The standard sets weights for each load based on the percentage of the time a stationary engine is operated at that load [37] [38]. The intermediate loads carry the most weight because the engine is operated between 25 – 75% the majority of the time. The ISO weighted emissions are presented in Figure 42. While particulate matter is within the standards carbon monoxide and non-methane hydrocarbons + NOx emissions are not able to meet the requirements. Testing was performed at an elevation of 5000 feet. At sea level the engine would meet Tier II emissions in diesel only mode for all categories. The NOx emissions for diesel only operation exceed the limit by a small amount because of lower ambient pressure at elevation. The NMHC+NOx levels are further out of compliance for dual fuel mode mainly due to hydrocarbon emissions. To operate in dual fuel mode with the baseline NG substitution curve and be able to meet Tier II limits an oxidation catalyst would need to be added to reduce the CO and NMHC levels.

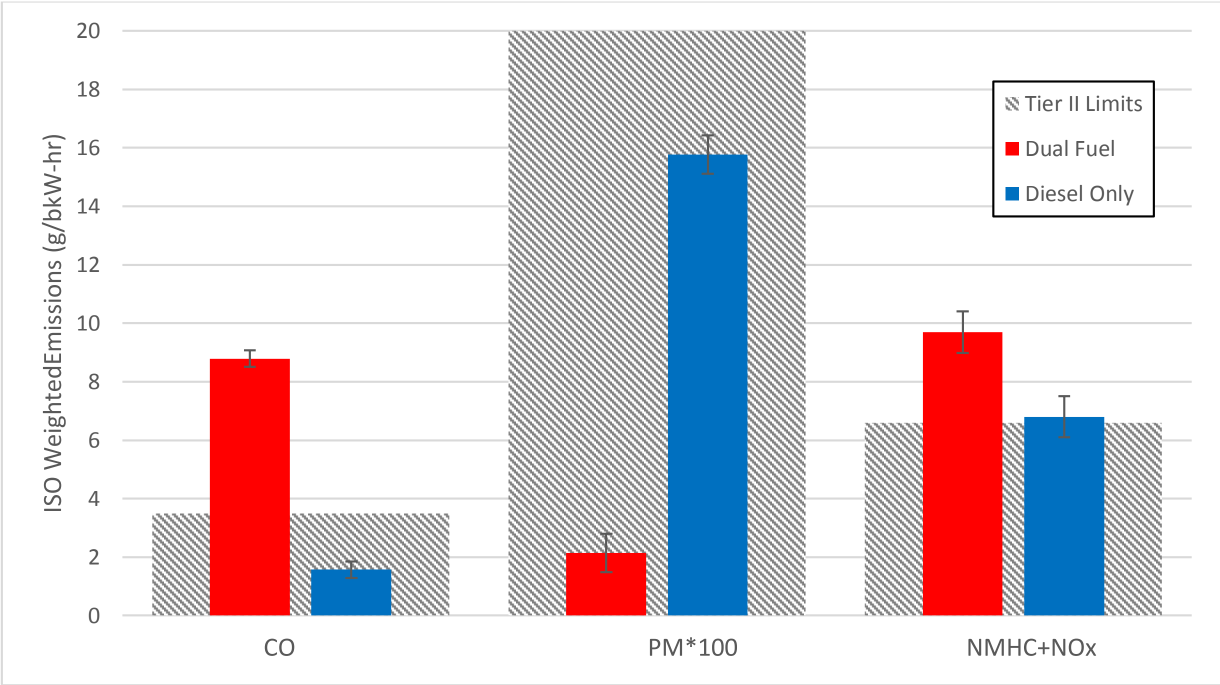

Figure 42: ISO 8178 weighted emissions for baseline commissioning and diesel only operation.

5. Substitution Limits

The baseline commissioning was configured based on previous experience of the dual fuel kit supplier, but may not be the physical limitations of the engine. The engine was tested without changes to the ECU to determine the maximum substitution and the mechanisms responsible for limiting natural gas utilization. This testing was carried out with combustion pressure and emissions measurements. At 100% load maximum substitution matches the baseline commissioning. Substitution was limited by engine knock, which was confirmed with combustion pressure measurements. However, Figure 43 shows three of the five break points in the baseline substitution map are higher than allowed based on limiting mechanisms. The rest of this chapter will analyze the limiting mechanism at each load in depth.

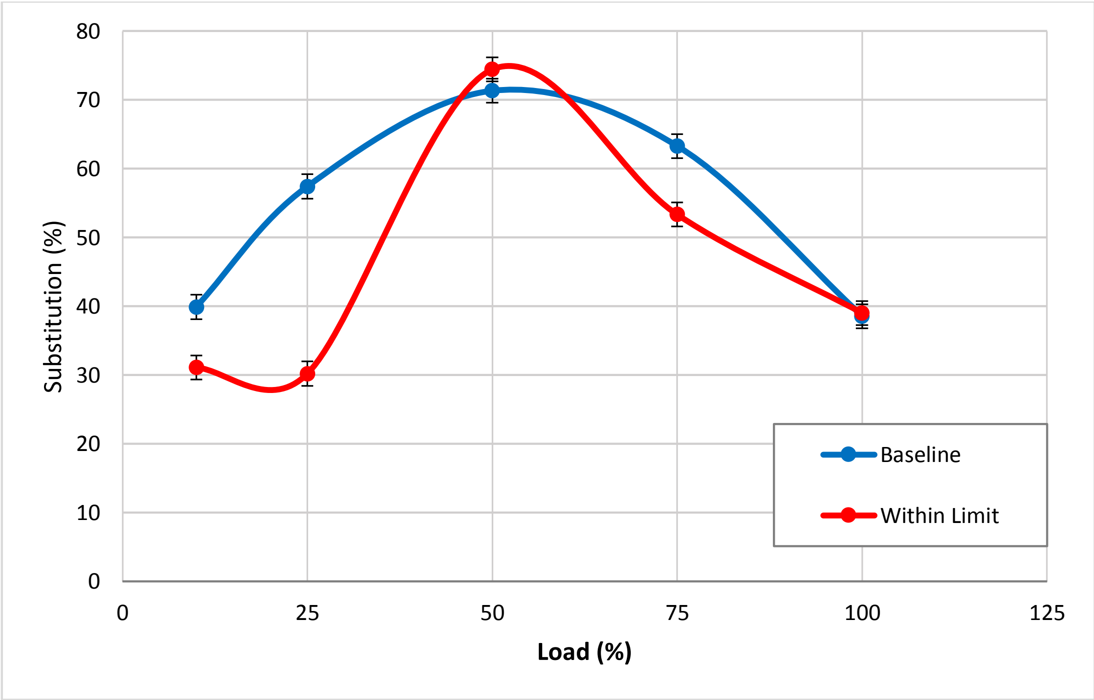

Figure 43: Substitution levels for the baseline commissioning and within limitations without changes to the ECU.

5.1 Knock Limitations

During the baseline commissioning an audible knock was heard while trying to increase substitution at full load. The conditions were replicated after incorporating in-cylinder pressure measurements. A LabVIEW program is used to determine the knock intensity from the in-cylinder pressure sensors [39]. The program uses a band pass filter to isolate the knock frequency from normal combustion events in the knock intensity calculations. A fast Fourier transform (FFT) is performed on the filtered pressure signal for an individual cylinder and individual cycle. The knock intensity is averaged over multiple cycles and is proportional to the amplitude of the knock pressure peak. A theoretical knock frequency is determined based on the trapped mass and mean cylinder temperature by Equations 9-12. The knock frequency equation is

$$f_{knock} = \frac{c}{2b} \tag{9}$$

Where c is the local speed of sound and b is the bore. In Equation (9) the knock frequency f is the frequency of pressure waves sensed by the in-cylinder pressure sensor. The pressure wave created by the knock event is assumed to travel at the speed of sound c at the local temperature. The pressure sensor is assumed to be on one side of the cylinder. The pressure wave travels to the far side of the cylinder and back before it is sensed again. Based on this assumption the distance component of Equation (9) is twice the bore. The speed of sound is calculated from

$$c = \sqrt{\gamma RT} \tag{10}$$

where γ is the ratio of specific heats, R is the gas constant of air, and T is the bulk cylinder temperature at the occurrence of knock. It is determined from the ideal gas equation at the pressure and volume at ignition. The ideal gas relationship is given

$$P_k V_k = mRT_k \tag{11}$$

where P is the pressure, V is the volume and m is the mass. The subscript k denotes the value is taken at the onset of knock. The ideal gas relationship is also solved at the inlet valve closure, denoted by the subscript ivc, as shown by Equation (12).

$$P_{ivc}V_{ivc} = mRT_{ivc} \tag{12}$$

The pressure and volume are known from the pressure sensor, crank angle rotary encoder and geometry of the engine. The temperature at intake valve closure is assumed to be the same as the intake manifold temperature. This allows for the mass of the cylinder to be solved for. Assuming no mass is lost during the compression stroke, the mass used in Equation (11) is the same as the mass solved for at the inlet valve closure. The pressure and volume at ignition are again determined from the in-cylinder pressure and crank angle degree. This allows for an estimated temperature at the onset of the knock event. For the John Deere engine, the knock frequency is approximately 5000Hz at full load. The knock frequency and knock intensity are different for each engine and are used as a comparative tool rather than an absolute indicator of knock. At 100% load the knock intensity was primarily below 100 with some cycles reaching the 200 level for diesel only operation and low substitution levels as shown by Figure 44. Increasing the substitution did not change the knock intensity until a threshold was crossed. Above the threshold the knock intensity for cylinder 1 rose to around 300. At this point it was determined the engine is beginning to knock and will be referred to as incipient knock. The increase in knock intensity is displayed in Figure 45. The substitution for normal knock-free operation is 38% and at incipient knock it is 42%. The substitution was increased further to examine the mechanism of knock. The knock intensity values shown in Figure 46 are more in the 300 to 400 range for the data points collected at the operating condition referred to as moderate knock. The substitution at moderate knock is 46%. The knock intensity values under heavy knock shown in Figure 47 are consistently around 1000 with some cycles even higher. Heavy knock occurred with approximately 51% substitution. The pressure spikes from heavy knock became larger thus resulting in much higher knock intensity values.

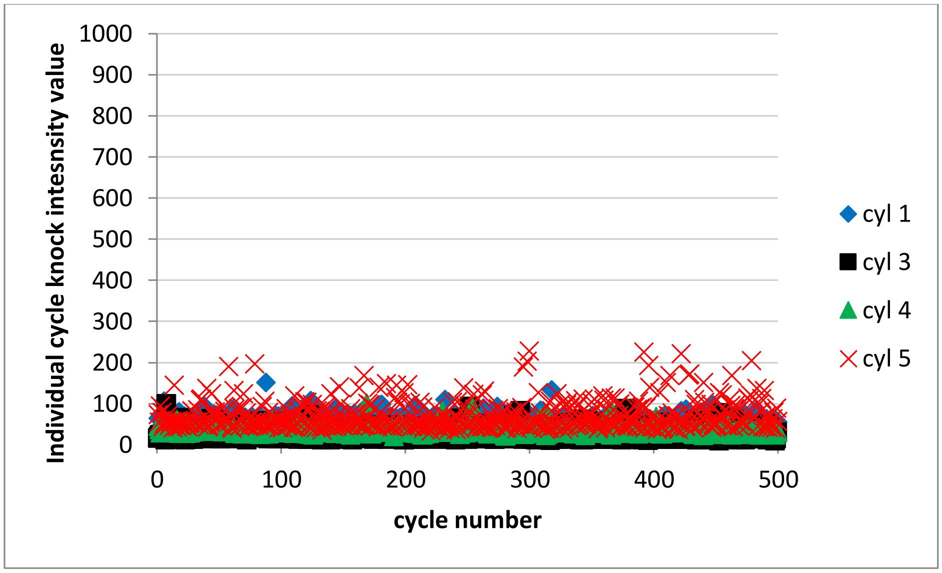

Figure 44: Knock intensity values for single cycles at 100% load in dual fuel operation without knock.

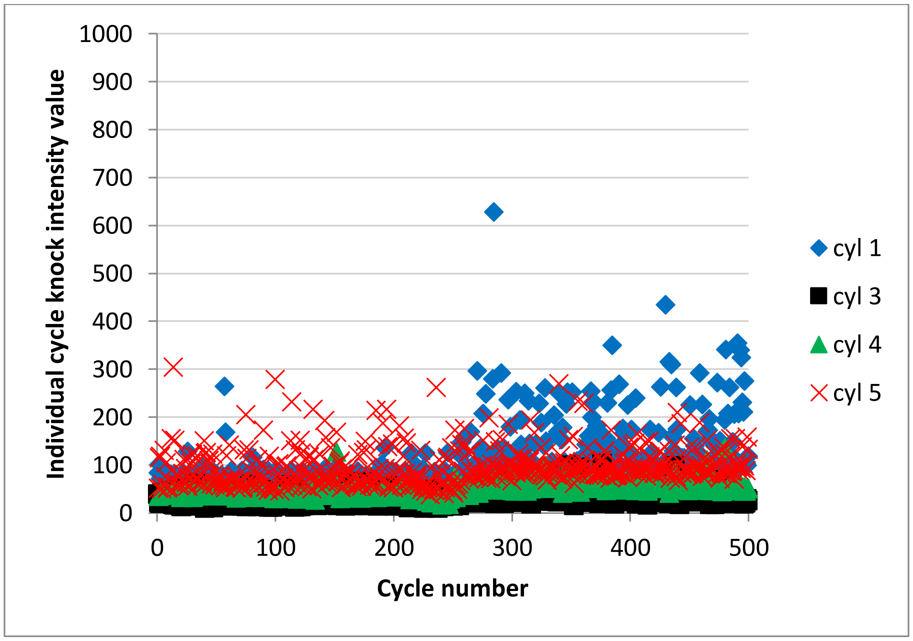

Figure 45: Knock intensity values for incipient knock at 100% load.

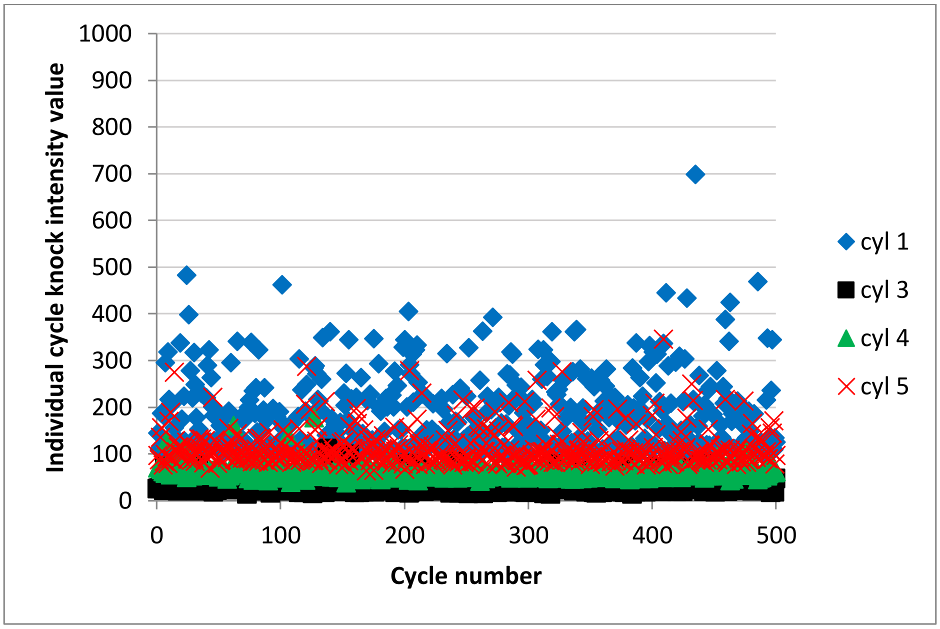

Figure 46: Knock intensity values for moderate knock at 100% load.

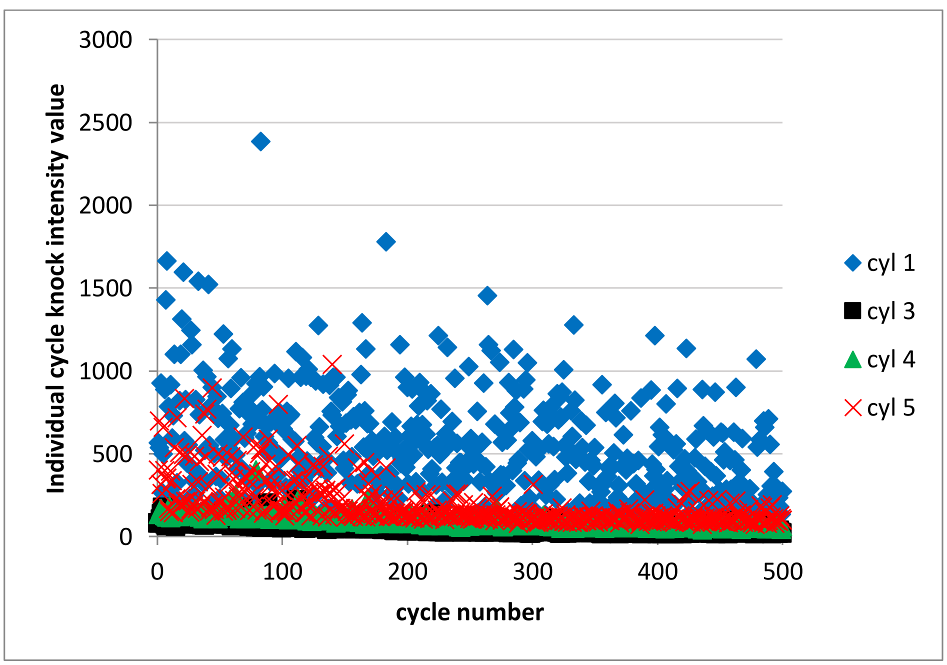

Figure 47: Knock intensity values for heavy knock at 100% load.

The same method for determining knock was used throughout the experiments. For subsequent sections the plots of the knock intensity will be omitted for brevity. The individual cycle pressure traces will be examined providing more insight than the knock intensity values. The knock intensity values show that cylinders one and five are more likely to knock than cylinders three and four. All of the individual cycle traces displayed in the following figures are for cylinder 1.

Figure 48 shows the in-cylinder pressure for the various levels of knock intensity at full load. The heavy knock case shows the fastest pressure rise and earliest peak pressure location, characterized by large pressure oscillations. The moderate and incipient knock cases have similar knock frequencies. The moderate knock peak pressure occurs slightly earlier but the main difference between moderate and incipient knock is the number of occurrences as shown by the knock intensity values versus cycle in Figures 45 and 46. The knock-free case has the lowest pressure rise rate resulting in a later location of peak pressure. The natural gas throttle has 100 discrete positions from closed to wide open. Backing off the natural gas throttle position by just a single discrete point is able to transition from the incipient knock to within knock.

With each level of knock intensity, the pressure reaches a motoring pressure then has a quick pressure rise ensued by the main pressure rise. This consistency of the early stages supports an end gas auto-ignition knock. Pre-ignition and early surface ignition at deposits or hot spots would cause the pressure to increase before reaching the motoring pressure. The initial sharp pressure rise is caused by the ignition of the pre-mixed diesel fuel and some of the natural gas locally around the diesel spray. The main pressure rise is where the differences between the knock events are observed. Heywood describes end gas auto-ignition as a race between the flame front and heat transfer to the unburned mixture [1]. If the end gas receives enough heat transfer the temperature can rise to a level at which auto-ignition can occur. At higher substitutions the premixed equivalence ratios are increased, accelerating end gas reactions. The increased pressure rise rates favor the heat transfer over the flame. The higher equivalence ratio in the end gas lowers the auto-ignition temperature. This causes the auto-ignition to occur earlier and consume more of the fuel before the flame front reaches the region creating larger pressure spikes from the constructive and destructive interference of the two pressure waves traveling in opposite directions. The concave up pressure trace leading to auto-ignition was also observed by Hockett et al [40].

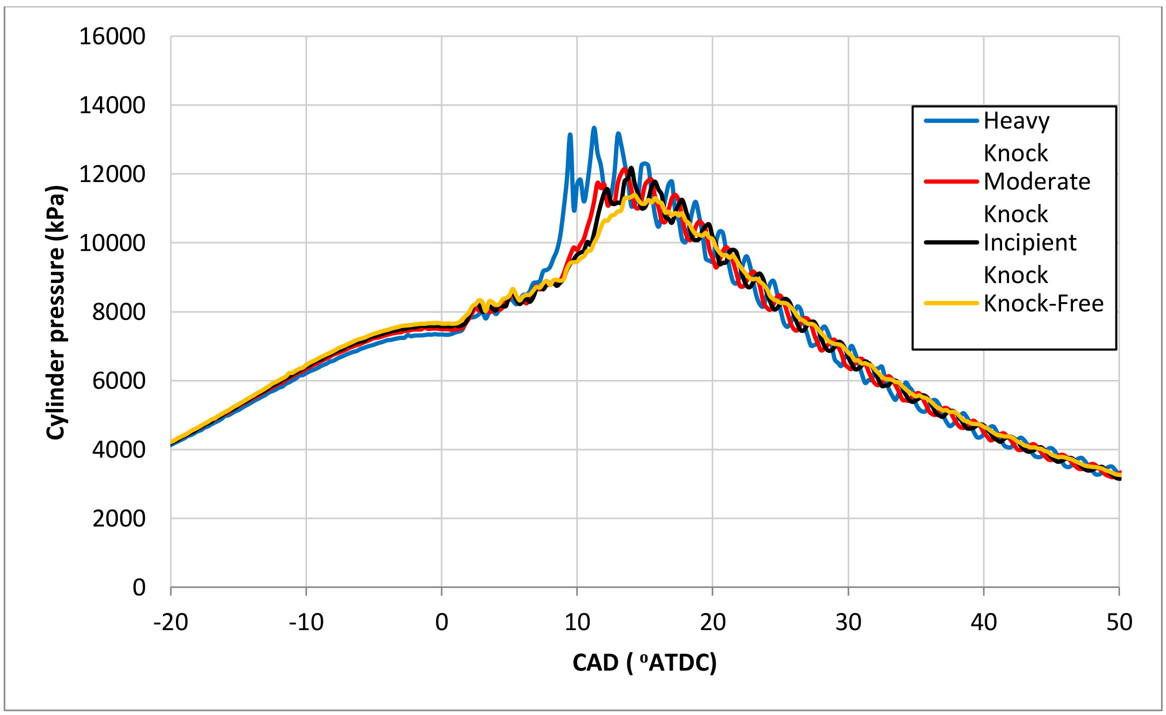

Figure 48: Pressure traces for an individual cycle at varying levels of knock intensity at 100% load.

Another possible mechanism for dual fuel knock is the pre-mixed fuel burning uncontrollably fast immediately following the ignition of the diesel fuel. This would likely be caused by a large diesel jet and high equivalence ratio of premixed fuel. It has been shown that the knocking torque decreases as the diesel fuel rate is increased while maintaining the same natural gas flow rate [24]. Increasing the diesel flow rate can increase the ignition energy, compatible to the spark energy for natural gas in spark ignited engines. Increasing the energy supplied for ignition increases initial reaction rates of the premixed flame. If this rate is high enough, auto-ignition can occur immediately following ignition. The pressure traces shown do not support this theory though. This type of knock event would have pressure oscillations immediately following the sharp pressure rise from diesel ignition. The heavy knock case would be the closest to this type of knock but it still shows a similar pressure rise rate as the no knock case until 5 degrees after top dead center. The incipient and moderate knock cases have a similar pressure rise until almost reaching the peak pressure. This supports the theory of having a flame propagating through the natural gas and some of the fuel auto-igniting before the flame reaches all areas of the cylinder.

The heat release traces presented in Figure 49 further support the theories proposed based on the pressure traces. All of the cases have a similar ignition delay based on degrees after SOI. The ignition delay is dominated by the characteristics of the diesel fuel. This causes the first peak of heat release to be nearly identical for each case. The natural gas surrounding the diesel jet adds to the initial heat release. The peak of the HRR of ignition is the same for each test despite different diesel injection rates and natural gas equivalence ratios. The most noticeable differences are observed in the main energy release. The within knock case displays the lowest heat release rate peak. The lower peak of heat release rate is enough to avoid knock. As the knock events become more severe the peak of the heat release rate occurs earlier. The higher substitution increased the equivalence ratio of the pre-mixed fuel and allowed faster reaction rates. The faster reaction rates cause the pressure to rise quicker and reach the peak pressure closer to top dead center. Reaching the peak pressure earlier results in the peak pressure being higher and increases the combustion temperatures. The equivalence ratio in the end gas is also higher making it easier to auto-ignite. This is likely one of the major factors in the engine knocking as the incipient and within knock cases have similar pressure traces until about 10 degrees after top dead center. The mass fraction burn curves for each knock case are displayed in Figure 50. The moderate and incipient knock cases burn only slightly faster than the within knock case. The increased concentration of natural gas for the heavy knock case burns faster than the other test conditions. All of the dual fuel cases burn significantly faster than diesel only operation.

Both 75% and 100% load share many characteristics and are grouped together as the high loads. They both display characteristics of pre-mixed combustion and flame propagation. This also means they are both knock limited. A major difference between 100% and 75% loads is the injection timing based on the stock ECU settings. At 100% load the diesel flow rate is such that the timing is not advanced relative to diesel only operation. However, at 75% load the diesel flow rates are reduced to rates that trigger advances in the diesel injection timing.

Operating with incipient knock, the SOI is 4.4 degrees before TDC and the corresponding substitution is 53%. When the substitution is increased to 62% the SOI is advanced to 5.9 degrees before TDC. Early injection and high substitution resulted in heavy knock occurred while operating at a substitution of 66%. Initially the incipient knock case was thought to be knock-free. At 100% load reducing the natural gas throttle by a single discrete percent of opening successfully avoided knock. However, at 75% load, the substitution was decreased by around 10% so it was believed this was not knocking despite still showing elevated KI values.

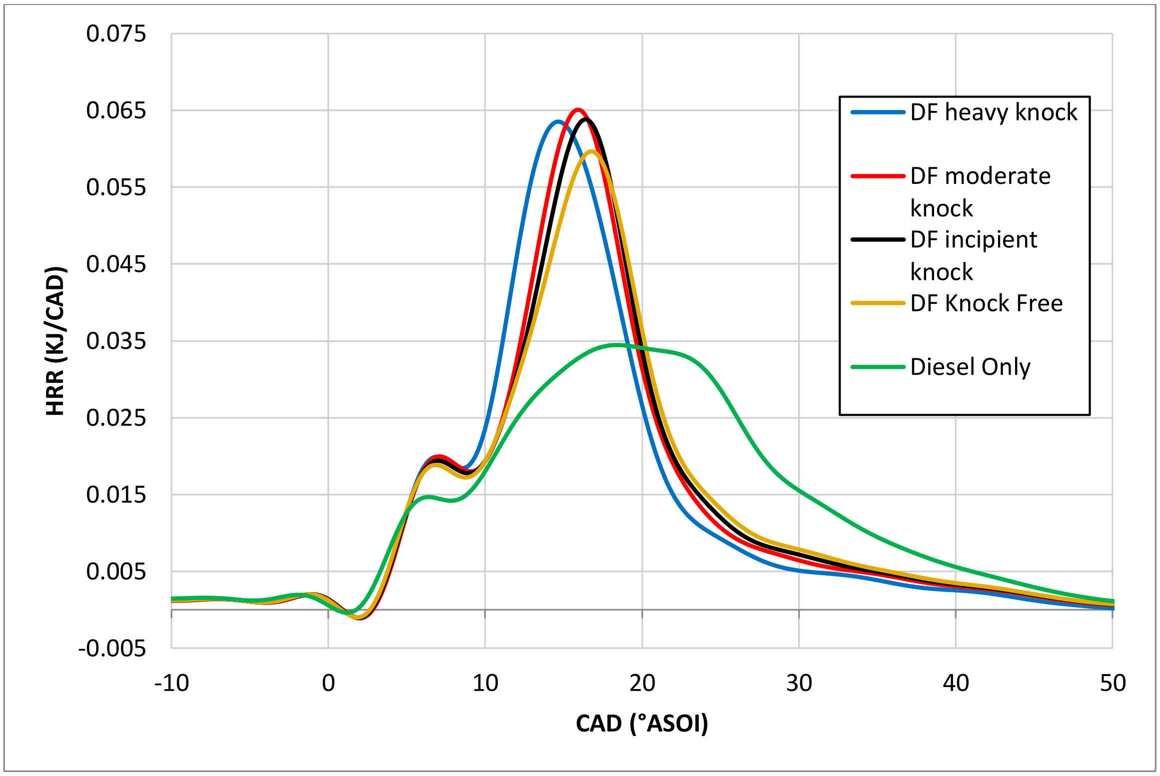

Figure 49: Average heat release rates for 100% load under various knocking conditions.

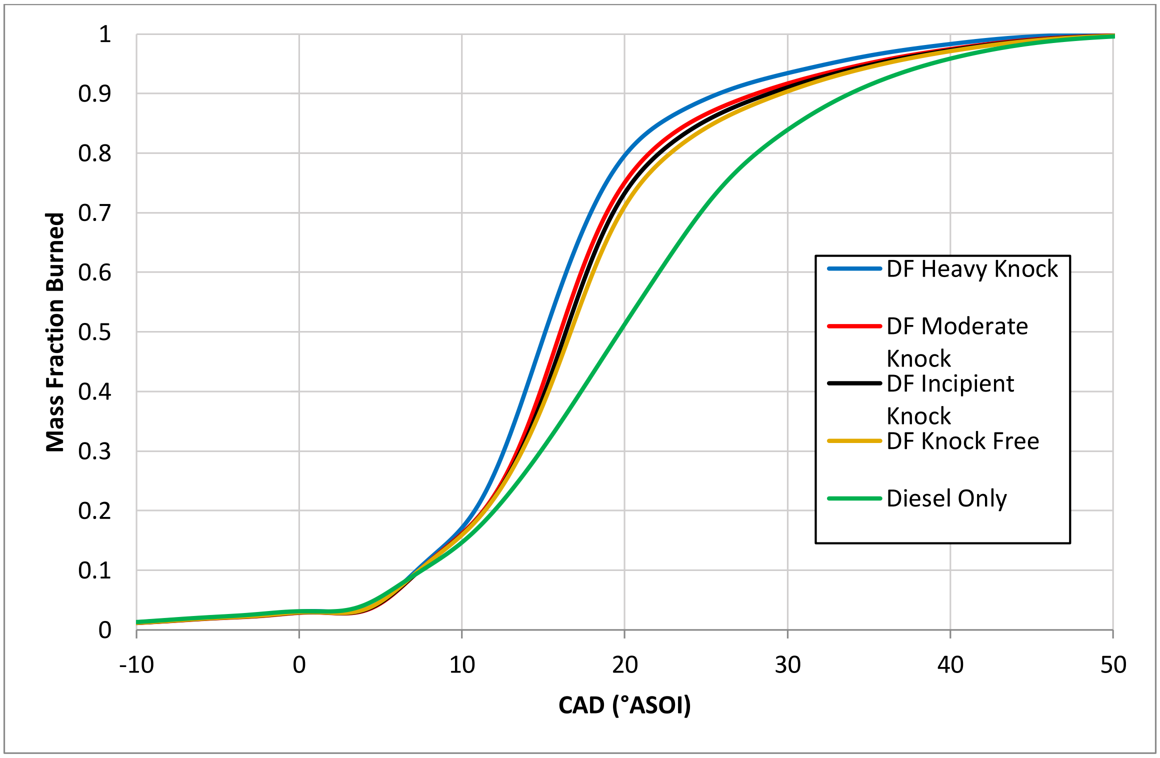

Figure 50: Mass fraction burned curves for various knock intensities at 100% load.

The transition of the KI is not as clearly defined because the geometry of the injector causes multiple premixed flames that can cause a ringing in the pressure trace. After post processing the data it has been determined the engine was in fact knocking even at 53% substitution. The advanced start of injection timing caused a higher likelihood of knocking especially with higher equivalence ratios of the premixed fuel compared to full load. The knock events at 75% load, however, are not audible. The engine noise sounds the same as the diesel combustion noise. This resulted in the substitution being set above the knock threshold during baseline dual fuel commissioning.

Figure 51 shows the knock traces at 75% load. The incipient knock case is the test at 53% substitution originally believed to be knock-free. It clearly shows a high frequency pressure oscillation. Similar to 100% the heavy knock case experiences the peak pressure closest to top dead center. Each pressure trace shows the initial pressure rise from diesel ignition and ensuing flame propagation. After some flame propagation, the pressure spikes because of the ignition of the end gas. There are no signs of pre-ignition even with earlier injection timing. The heat release traces at 75% load vary for each knock condition more than the 100% load tests. The heat release rates for 75% load are presented in Figure 52. The heavy knock case reaches a significantly higher peak than the moderate and incipient knock cases. Although the moderate knock case has a higher heat release peak than the incipient knock case. The ignition delays vary some based on the degrees after SOI because the SOI timing was advanced with increasing substitution levels. Heavy knock and moderate knock occur with a SOI of 5.9 ^{o}btdc. Heavy knock is at 65% substitution while moderate knock is at 62%. Incipient knock occurs with SOI at 4.4 ^{o}btdc and 53% substitution. However, the initial peaks all still resemble a diesel pre-mixed start of ignition augmented by natural gas local to the ignition centers. This is shown by the ignition peak of heat release greater than the diesel only peak.

A common technique to avoiding knock is retarding the ignition timing. The stock ECU responds in an adverse way to avoiding knock by advancing the SOI with increased natural gas usage. The associated heat release rate and mass fraction burned curves are presented in Figures 52 and 53, respectively. The dual fuel mass fraction burn curves speed up significantly after 15 degrees after SOI compared to the diesel only curve. A possible reason for the shorter burn durations compared to diesel only operation is the increased flame surface area of dual fuel combustion.

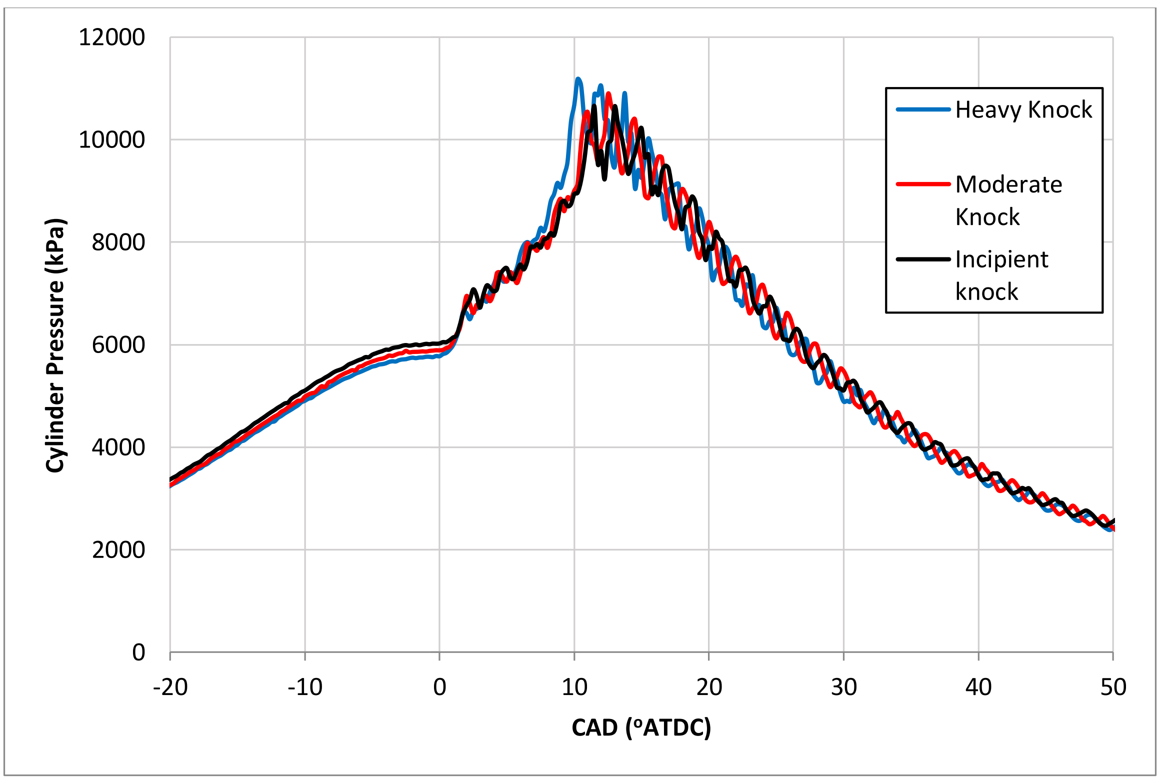

Figure 51: In-cylinder pressure traces for various knocking conditions at 75% load.

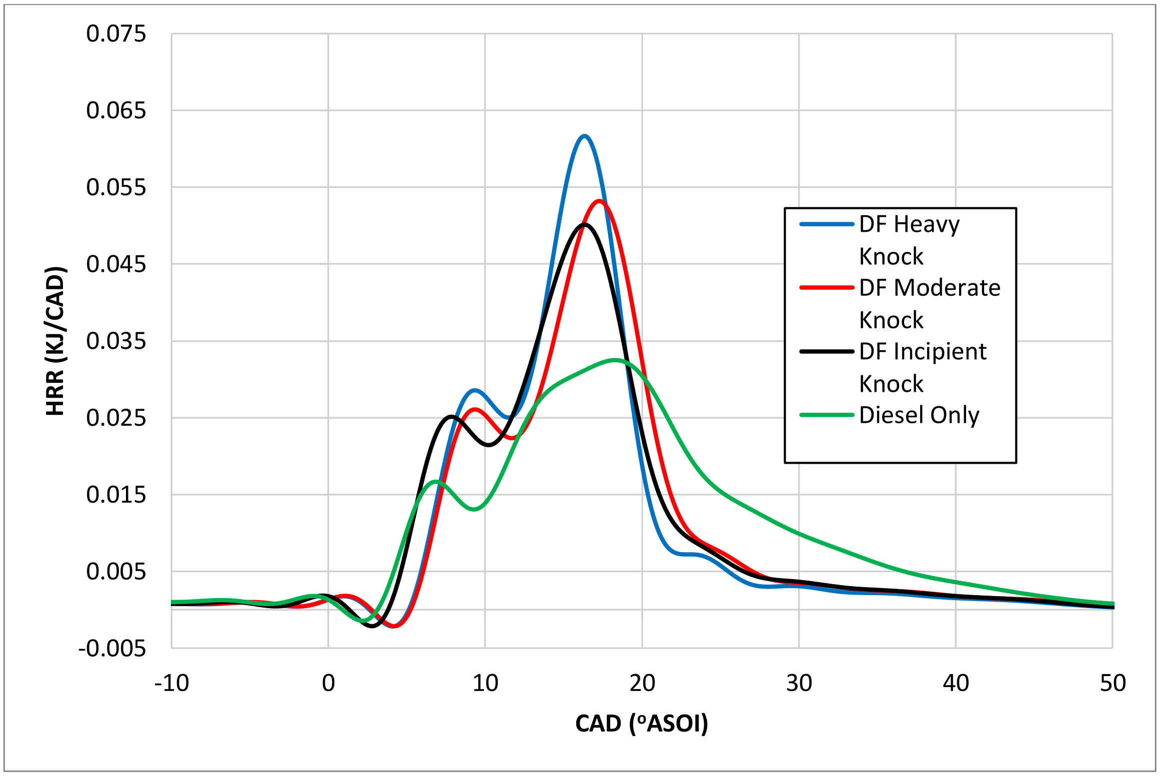

Figure 52: Heat release rates of dual fuel opeartion with varying knock intensities at 75% load.

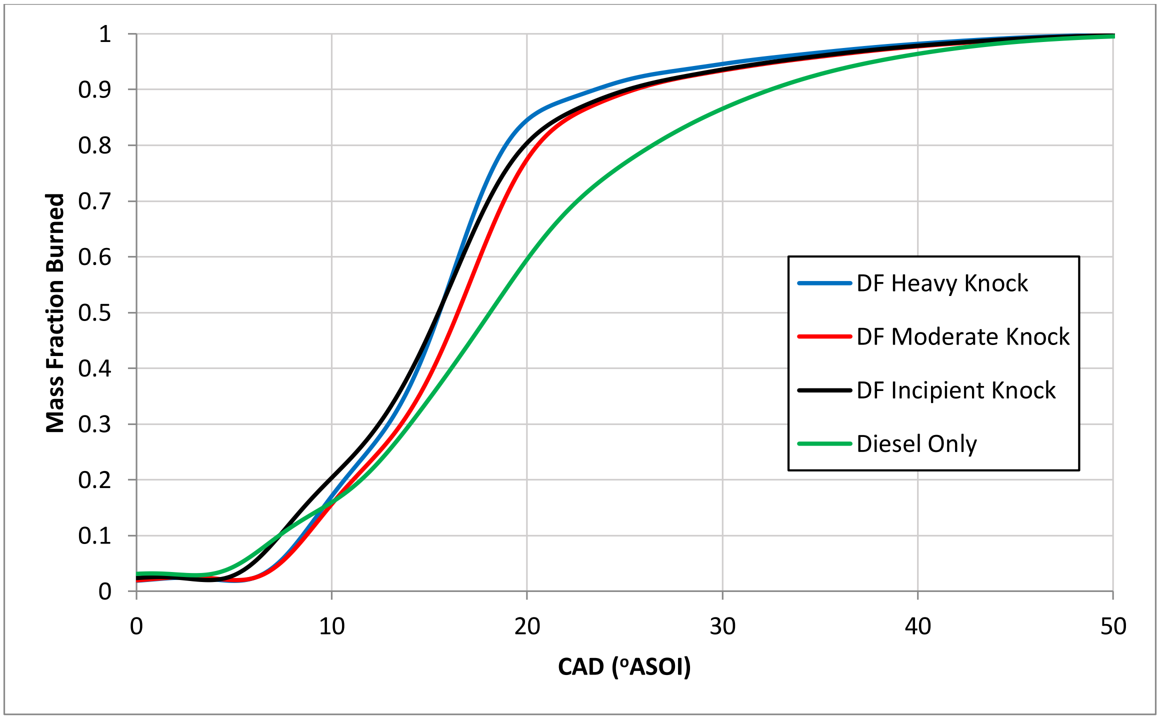

Figure 53: Mass fraction burn curves of dual fuel operation with varying knock intensities at 75% load.

5.2 Limitations at 50% Load

The baseline commissioning chapter describes three different combustion regions within the dual fuel operation: high loads that have flame propagation, a transition region with flame propagation but quenching and low loads without flame propagation. The transition point occurs at 50% load. During the baseline commissioning it is demonstrated that 50% load achieves the highest substitution, yet is still effectively utilizing most of the natural gas in the cylinder. Although there seems to be flame propagation at 50% load, the natural gas throttle is able to be wide open and the engine will not knock. With the maximum flow of natural gas, the combustion was consistent and the engine ran smoothly. The peak pressure shown in Figure 54 is actually lower than the diesel baseline when the natural gas throttle is wide open. Wide open throttle is represented by the TPS (throttle position set point) max series in the figures. The substitution is approximately 90% with the natural gas throttle wide open. The diesel flow is 2.2 L/hr at 90% substitution, which is below the minimum diesel flow considered for the baseline commissioning. The priority is to find the physical limits of the combustion process. Future designs could address the cooling problem with a design change, such as using the excess fuel flow that leaves in the return lines for extra cooling.

The combustion is dependent on the amount of diesel injected as shown by the decrease in knocking torque from increasing the amount of diesel injected while the natural gas flow rate is held constant [24]. The likely cause is the increased diesel fuel provides more energy at ignition accelerating the burning rates of the premixed fuel. This is why some dual fuel experiments show lower peak pressures when operated on dual fuel compared to diesel only operation [28] but most of the test data here displays higher peak pressures from dual fuel operation. The diesel flow for the within emissions case is 5.8 L/hr compared to 2.2 L/hr when the substitution is 90%. The increased diesel flow rate is shown to have a greater pressure rise from the diesel premixed ignition. The within emissions case rises to 5200 kPa from the ignition while the TPS max only reaches 4800 kPa.

The dual fuel pressure traces in Figure 54 show some ringing but it is not the same sharp pressure oscillations as the high load cases. The pressure transducers are mounted in glow plugs, which have a transfer tube. The resonance of the transfer tube can cause ringing in the pressure trace but it is more prominent in the dual fuel tests. A possible explanation for the ringing is the numerous flames found in dual fuel engines. In a typical pre-mixed engine there is a single flame front traveling through the cylinder. In a dual fuel application, there are flames propagating from each diesel jet. This causes numerous flame fronts and pressure waves traversing the cylinder in different directions.

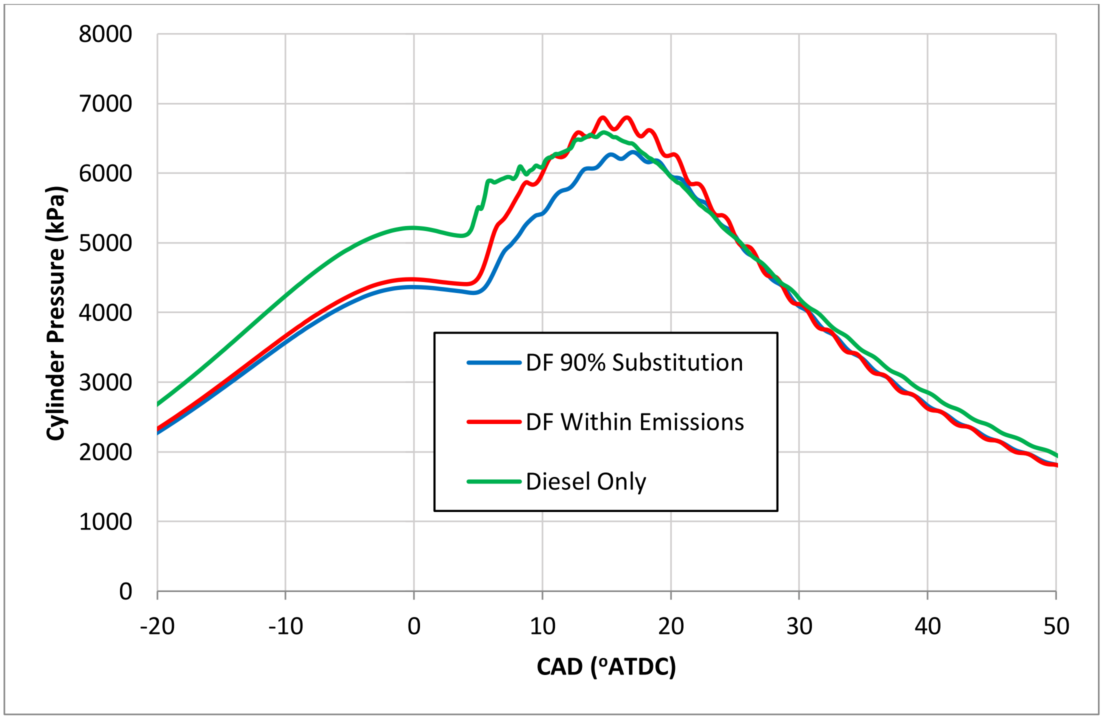

Figure 54: Pressure traces for varying substitution levels at 50% load.

The constructive and destructive interference of the pressure waves causes variations in the pressure sensed in the glow plug adapter. This is the same mechanism causing the pressure oscillations during knock events but can be caused by normal operation of a dual fuel engine based on injector geometries. The injectors used on this engine have six nozzles.

While the engine runs smooth with the throttle wide open, the emissions levels for hydrocarbons and carbon monoxide increase to unsuitable levels. The Tier II engine does not have an after treatment system so the THC emissions are not reduced and the ISO weighted average emissions does not comply with Tier II requirements. An effective 2-way, or oxidation, catalyst might allow substitution levels to approach 90%, but the system was designed to be implemented without adding an after treatment system.

The ISO 8178 weighted average for emissions was discussed in the baseline commissioning chapter. The results show both BSCO and BSNMHC+BSNOx were not able to meet standards using the baseline substitution map. The focus was on reducing BSNMHC+BSNOx to acceptable levels. There are two reasons behind this decision. First, CO and NOx typically have opposing trends to adjusting engine parameters. An attempt to reduce NOx emissions would push CO emissions further out of range. Determining an emissions limit in real time for both exhaust products would be challenging and time consuming. Second, the focus was placed on NMHC+ NOx instead of CO because CO is relatively easy to convert to CO_2 with an oxidation catalyst but an SCR system used for reducing NOx is more complex and more expensive.

The emissions criteria are based on a weighted average of five load break points so a method was developed to determine an emissions limit for a specific load level. The emissions concentrations at 100% and 75% load are close to diesel levels. A reduction in substitution at these loads would have minimal returns in lowering the ISO weighted emission levels. At lower loads, an increase in substitution directly causes an increase in THC emissions and CO. Therefore, low loads were determined to be the set points that could be used to reduce the hydrocarbon emissions. Updated ISO weighted emissions levels were determined using the baseline commissioning levels for the high loads and adjusting values of the light load cases. An iterative method for the emissions levels for 50, 25 and 10% loads was used to compare to the ISO weighted average. The ISO weighted average of the emissions levels from 100 and 75% load from the baseline commissioning and the iterative levels for the low loads were compared against the ISO weighted average Tier II limit. These reduced emissions levels became target levels at an individual load to comply with Tier II requirements. The target NMHC+ NOx emissions levels for 10 and 25% load were 50% of the original commissioning levels. At 50% load the emissions levels were limited to the same value as the original commissioning. The individual load emissions limits are summarized in Table 5.

The 5-Gas emissions analyzer is used for determining the emissions limits in real time. It measures total hydrocarbons in parts per million dry. However, the Tier II emission limit is non-methane hydrocarbons. For a real-time calculation of non-methane hydrocarbons, the fraction of

Table 5: Target Emissions for individual engine loading conditions to meet Tier II regulations.

Engine Loading	NMHC+ NOx target level (g/bkW-hr)
100%	NA
75%	NA
50%	6.80
25%	7.98
10%	25.20

NMHC to THC was assumed to be the same fraction as from the baseline commissioning. This is an estimate but it reduces the test time considerably for exploring different substitution rates. TO compute NMHC values more accurately the FTIR must be used, which has a longer measurement time. For the final value of the emissions NMHC values are computed accurately by combing FTIR and 5-gas analyzer measurements. The FTIR analyzer is capable of measuring parts-per-million of methane wet and the water content of the exhaust products. For the NMHC values, the FTIR measured methane is converted from wet to dry and subtracted from the 5-gas THC. This conversion is given by Equation (13).

$$NMHC[ppmd] = THC[ppmd] - \frac{CH4[ppmw] * 10^6}{10^6 - H2O[ppm]} \quad (13)$$

To meet the emissions target at 50% load the substitution was reduced to 73%. With lower substitution, the heat release is faster than the 90% substitution shown in Figure 55. For other conditions of dual fuel operation, an increase in substitution increases the heat release rates. A possible explanation is the lower substitution cases are releasing energy from both fuels simultaneously. At 90% substitution there is much less energy supplied from the diesel fuel. The energy release is primarily from a pre-mixed propagating flame. The diesel fuel is likely consumed very quickly so the main energy release would be only from propagating flames. A substitution of around 50% has a larger flame surface area if diffusion and pre-mixed flames occur simultaneously allowing faster heat release rates. This benefit is lost when the substitution nears either 0 or 100%. Another possible reason is the smaller diesel jet ignites less of the pre-mixed fuel. This causes the heat release from the pre-mixed flame to lag behind the heat release rate of the lower substitution case. The associated mass fraction burned curve are also presented in Figure 56.

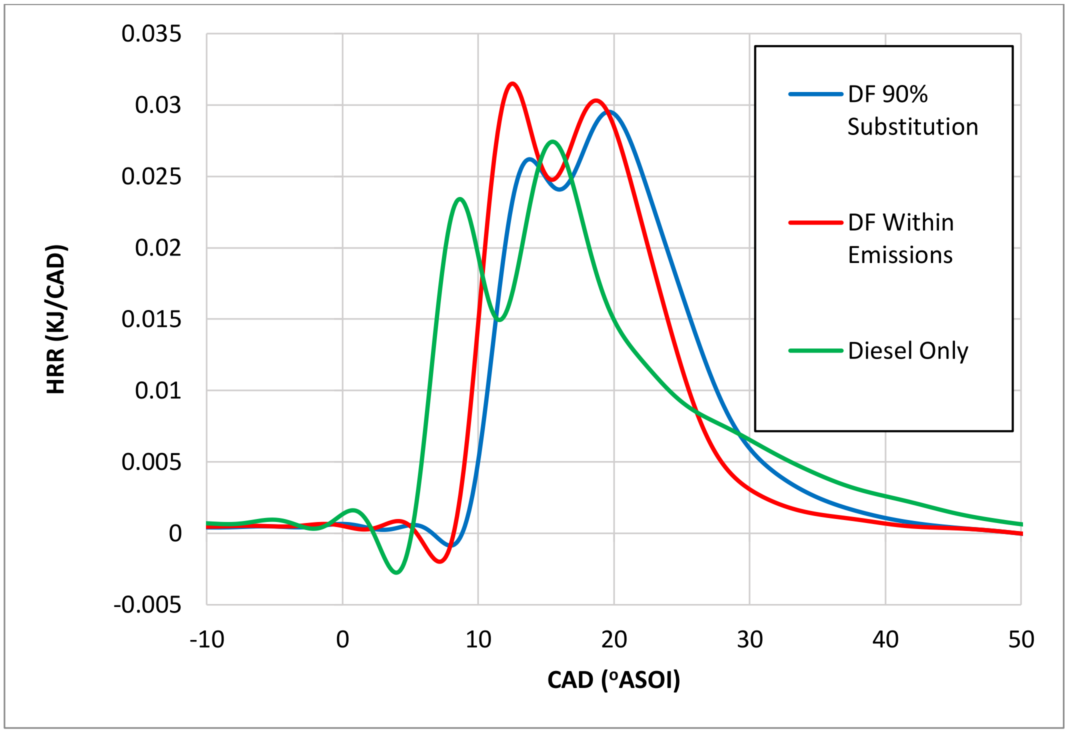

Figure 55: Heat release rates of dual fuel operation compaaring wide open natural gas throttle to conditions able to meet emissions and the diesel baseline at 50% load.

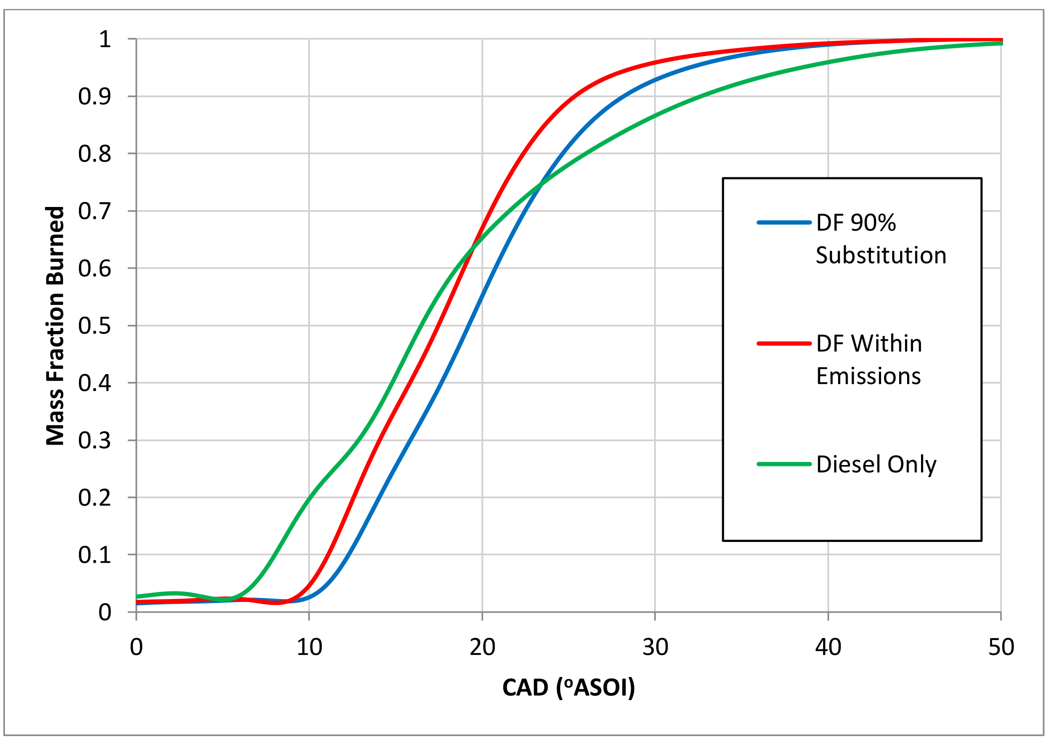

Figure 56: Mass fraction burn curves for 50% load. TPS max is a substitution of 90%. Within emissions is a substitution of 73%.

5.3 Governor Instability

A physical limit to combustion exists at low engine loads. At 50% load enough diesel fuel is being injected to reliably burn even with the natural gas throttle wide open but at light loads governor instability occurs. Figure 57 shows the electrical power exported to the grid by the generator and the engine speed versus time at 25% load. The same behavior is exhibited at 10% load. The power is hunting some with the high substitution level but with a further increase in substitution the speed control becomes erratic because of governor instability.

As a generator set, the engine should always be at 1800 rpm to match the 60Hz electrical frequency on the grid. When the substitution becomes very high the amount of diesel injected each cycle is reduced to a point where combustion becomes inconsistent. Misfiring causes the engine to drop load. Two separate PID's are affected by this. The main controller has a PID for load control and the ECU has a governor PID to maintain speed. When the engine misfires the load control PID demands more diesel fuel to meet the load. The injectors are able to react more quickly than the power from the generator so some of the additional fuel supplied increases the speed. The governor reacts by reducing fuel to keep the speed at a constant rpm. This increase and decrease in fuel demand becomes a cyclical pattern and continues until the substitution is reduced and the diesel consumption rate is high enough to burn consistently.

The results of Badr, Karim and Liu's study were not recreated here [29]. There was no point where the CO and THC emissions decreased with added natural gas addition. The governor would only become a serious limit to substitution if the engine was equipped with a highly effective after treatment system as the emissions are out of range well before governor instability occurs. The substitution levels were significantly decreased compared to the original commissioning conditions to be able to meet Tier II emissions.

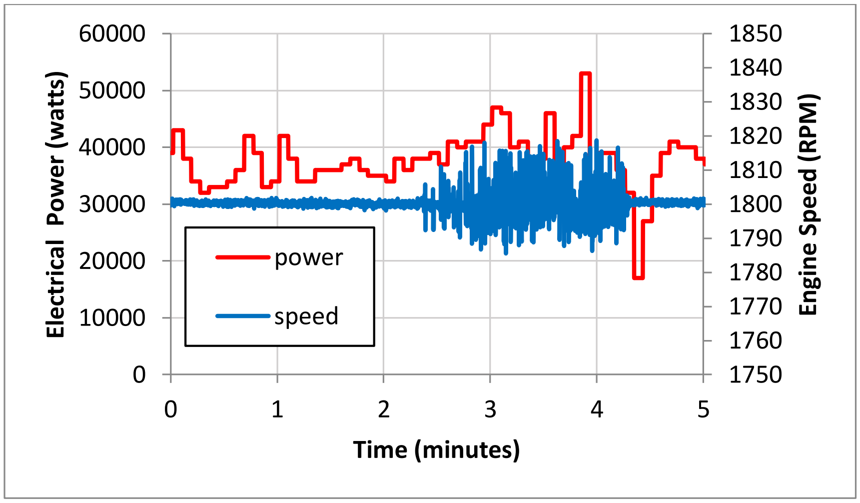

Figure 57: Engine power and speed history as the substitution limit resulting in governor instability at 25% load is reached.

Governor instability occurred at 94% substitution. To be within emissions limits required a substitution of about 30% substitution for both of the light load cases. The natural gas throttle was opened the minimum amount possible, 1% of opening. Some dual fuel applications cut off the natural gas flow at light loads to avoid excessive regulated emissions.

5.4 Emissions Limitations

Figures 58 and 59 show that the pressure and heat release rates with low substitution levels at light loads behave nearly identical to diesel only operation. The diesel jet is reduced during dual fuel operation and the nearby natural gas burns making up the difference in diesel admitted to the cylinder. A substitution of 30% is less of a diesel fuel consumption reduction at light loads than at high loads. Under the baseline commissioning 100% load is the only point tested that diesel provided more energy than natural gas. However, at 10 and 25% loads to meet emissions over 63% of the energy is from diesel fuel. The characteristics of diesel combustion dominate the process when the majority of the energy is provided by diesel. The COV of IMEP was 33% for the governor instability tests at low loads. Figures 58 and 59 are both averaged profiles of the in-cylinder pressure and heat release rate over 500 cycles. They show a slight pressure rise and a slow energy release rate. Over the 500 cycle window there are some cycles that probably experience a complete misfire. In the next few cycles the amount of diesel injected is increased to keep load and speed. The extra energy supplied from diesel is able to ignite causing large variations in the combustion process cycle-to-cycle.

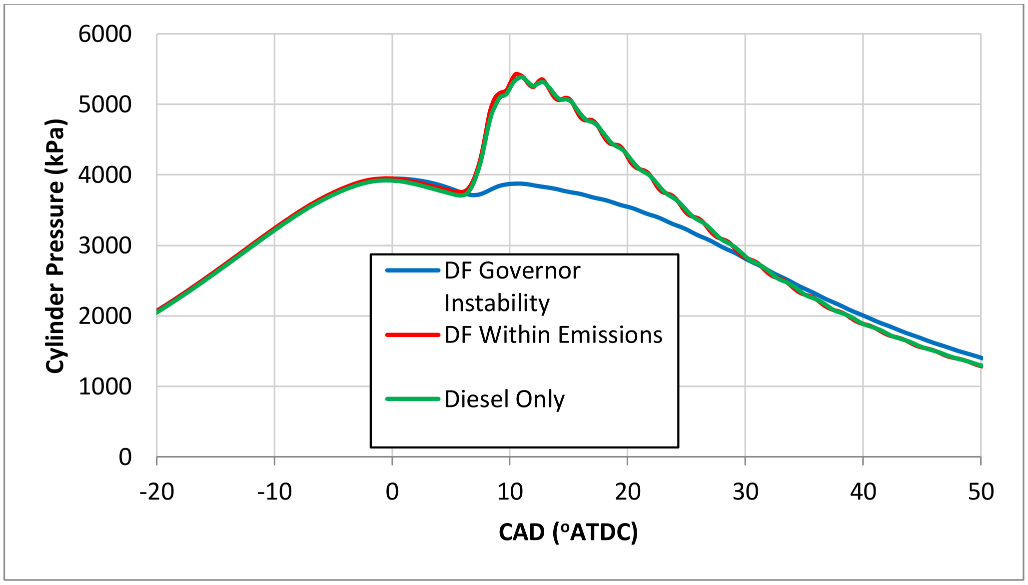

Figure 58: Pressure traces for 25% load with varying levels of substitution. Governor instability is at 94% substitution and within emissions is at 30% substitution.

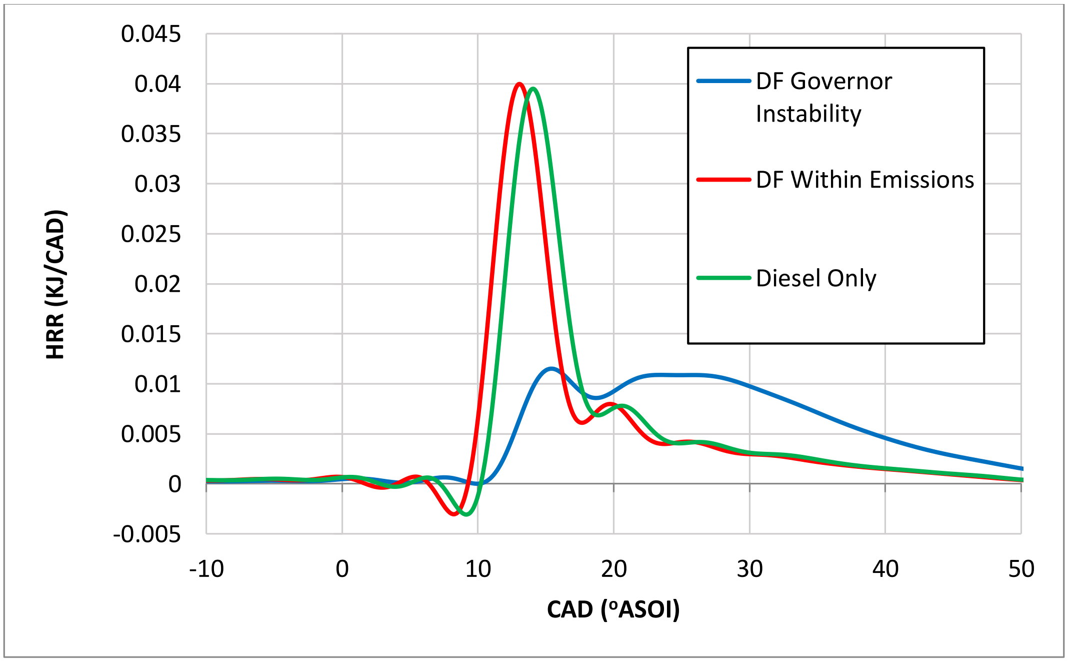

Figure 59: Heat release rate profiles for 25% load at governor instability, within emissions and diesel baseline.

5.5 Summary

Two prominent mechanisms were determined to limit substitution levels. These mechanisms are end gas auto-ignition and flame quenching. Governor instability is another mechanism but is not a major concern because under the engine operating conditions flame quenching caused emissions limitations before the combustion became unstable. To increase the allowable substitution levels in dual fuel engines a diesel injector design with as many injector nozzles as possible is recommended. Heavy engine loading conditions provide conditions suitable for premixed flame propagation. The limits of substitution are set by end gas auto-ignition in the same manor that limits the compression ratio of spark ignition engines.

Engine designs with small bores and fast engine speeds are the most favorable to avoiding knock. Shorter distances for the flame to travel reduces the likelihood the flame compression of the end gas will raise the pressure and temperature to the point of auto-ignition. Higher speeds slow reaction rates because of more expansive cooling. Using an injector with more nozzles will effectively reduce the flame propagation lengths. It is proposed the end gas region where auto-ignition occurs is in the center of the piston bowl between two adjacent jets [34]. This is the last region a propagating flame will reach and is not cooled by the cylinder walls. An injector with 8 nozzle holes reduces the distance between each adjacent diesel jet from which natural gas flames propagate outwards. The overall mass of diesel injected would remain the same but the substitution levels could be extended by delaying the onset of knock.

The conditions at low loads do not allow for premixed flame propagation. At both 10 and 25% load it is proposed that only the natural gas in the immediate vicinity of the diesel jet is being burned. The main problem is getting the natural gas to react. The diesel jets burn consistently and will increase the injection duration to meet load so dual fuel combustion achieves very low COV of IMEP values. However, it is ideal that natural gas provides more of the combustion energy than diesel. At low loads, the equivalence ratios are low so that flame propagation does not occur and little of the natural gas in the cylinder is burned exhausting high levels of hydrocarbons. The substitution rates were increased to try increasing the equivalence ratio and promoting flame propagation but this caused governor instability.

Since flame propagation is not observed, natural gas is diffusing with oxygen to the diesel jet. The way to reduce emissions and increase natural gas substitution thus becomes a way to introduce more natural gas to the diffusion flame. This could be achieved by increasing the swirl levels to increase the transport of the premixed fuel and air or to increase the surface area of the flame. Increasing the surface area of the jets increases the amount of natural that participates in the diffusion flame reactions. Increasing the number of jets increases the surface area to volume ratio so the same mass of fuel injected remains the same while possibly allowing more natural gas to react. The conditions at 50% load allowed the highest substitution levels of any of the loads tested. It seems to have a flame propagating but some flame quenching during the late combustion stages. Reducing the required premixed flame propagating distance could help reduce the THC emissions at 50% load. Finding a single solution for increasing the dual fuel substitution is difficult because opposing characteristics are desired at each end of the load range. The next chapter will examine two methods for extending substitution without major hardware changes.

6. Extending Substitution Limits

With substitution limiting mechanisms clearly identified, the next goal was to evaluate methods for extending substitution limits. Two methods that are examined in this section are start of injecting timing of diesel fuel (SOI) and air manifold temperature. Figure 60 shows substitution levels achieved by adjusting the SOI and air manifold temperatures compared to the baseline and within limits substitution maps. The within limit and baseline commissioning match substitution levels at 100% load. It was found that the engine was actually knocking at 75% load under baseline commissioning so the knock-free substitution is lower for the within limits map. The substitution at 50% load is the highest without any adjustments to the operating conditions. The baseline commissioning substitution map is higher than any of the other tests were able to achieve at low 25 and 10% loads.

End gas auto-ignition limits substitution at 100 and 75% load. Retarding the injection timing is successful in extending the substitution at high loads by delaying the onset of knock. Lowering air manifold temperatures is even more effective than adjusting the SOI, increasing the allowable substitution by 10 percentage points or more over the baseline commissioning. With lower air manifold temperatures, the substitution at 75% load is able to reach the highest of any operating condition at nearly 80%. The engine runs well with high levels of substitution at 50% load without knocking or governor instability. However, the emissions become a limitation because of excessive THC and CO emissions. Adjustments to the SOI and air manifold temperatures are not able to increase the substitution levels at 50% load. Advancing the SOI at 25 and 10% loads is not able to increase the substitution. Preheating the intake charge is an effective way to convert more of the premixed fuel and reduce THC emissions allowing higher substitutions. However, this was not able to be achieved at 10% load. The subsequent sections examine the effects of adjusting the SOI timing and air manifold temperatures on the engine performance.

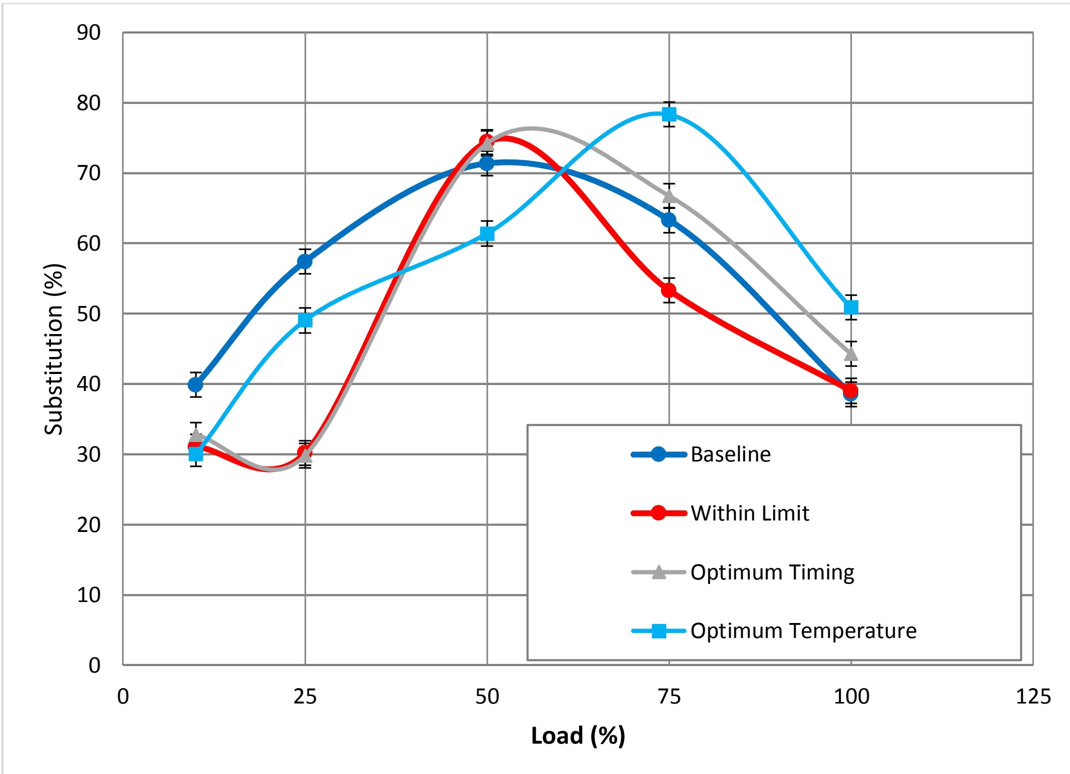

Figure 60: Comparison of substitution levels at baseline commissioning, within limit and at the optimum SOI timing.

6.1 Start of Injection Timing

In the last chapter, results showed stock ECU settings can make timing changes that hinder natural gas utilization. This was observed primarily at 75% load where the injection timing was advanced with increasing substitution leading to knock. Changes to the ECU are not typically made when retrofit kits are installed. It is desired to know how much the substitution can be increased if access is available to adjust some ECU parameters. It is also of note that the John Deere engine tested uses multiple injections only during the warm up period. Once the engine is warm, a single injection of diesel is used. A pilot injection would be equivalent to advancing the SOI. Most new diesel engines sold use a multi-injection scheme as a solution to comply with Tier IV emissions requirements. Installing a dual fuel retrofit kit on such an engine could benefit from access to the ECU to enable a single injection scheme.

6.1.1 100% Load

The normal SOI timing for dual fuel operation at 100% load is 3^{o}btdc. The timing was retarded to 2 ^{o}btdc. The later ignition adjusts the combustion phasing later after tdc. Exhaust gas temperatures limit how late the injection timing can be set. Late injection timings result in burning later into the exhaust stroke and elevate the exhaust gas temperature. The engine manufacturer recommends keeping the exhaust temperatures below 700^{o}C and performing a shutdown at 750^{o}C. Each cylinder had an exhaust port thermocouple to monitor these temperatures. With a SOI of 2 the exhaust gas temperatures measured in the exhaust runners to the manifold were around 700^{o}C for each cylinder with the highest reaching 715^{o}C.

Figure 61 shows retarding the injection timing shifts the combustion process later after top dead center. Other than being shifted, the SOI 2 and no knock cases are very similar. A timing change of 1^{o} is a small change so there are minimal effects to the profile of the pressure trace. The main difference between the two curves is the standard timing curves leads the retarded SOI timing curve by 1^{o} throughout the power stroke. The pressure rise rates are similar as well as the peak pressures sensed. The substitution is four percentage points higher with the retarded timing. The expansive cooling lowers the end gas temperature enough to avoid auto-ignition before the flame reaches the final regions of the combustion chamber. A knock case with the retarded timing is also plotted for comparison. A higher substitution increases the equivalence ratio of the end gas and results in the classic knock pressure trace.

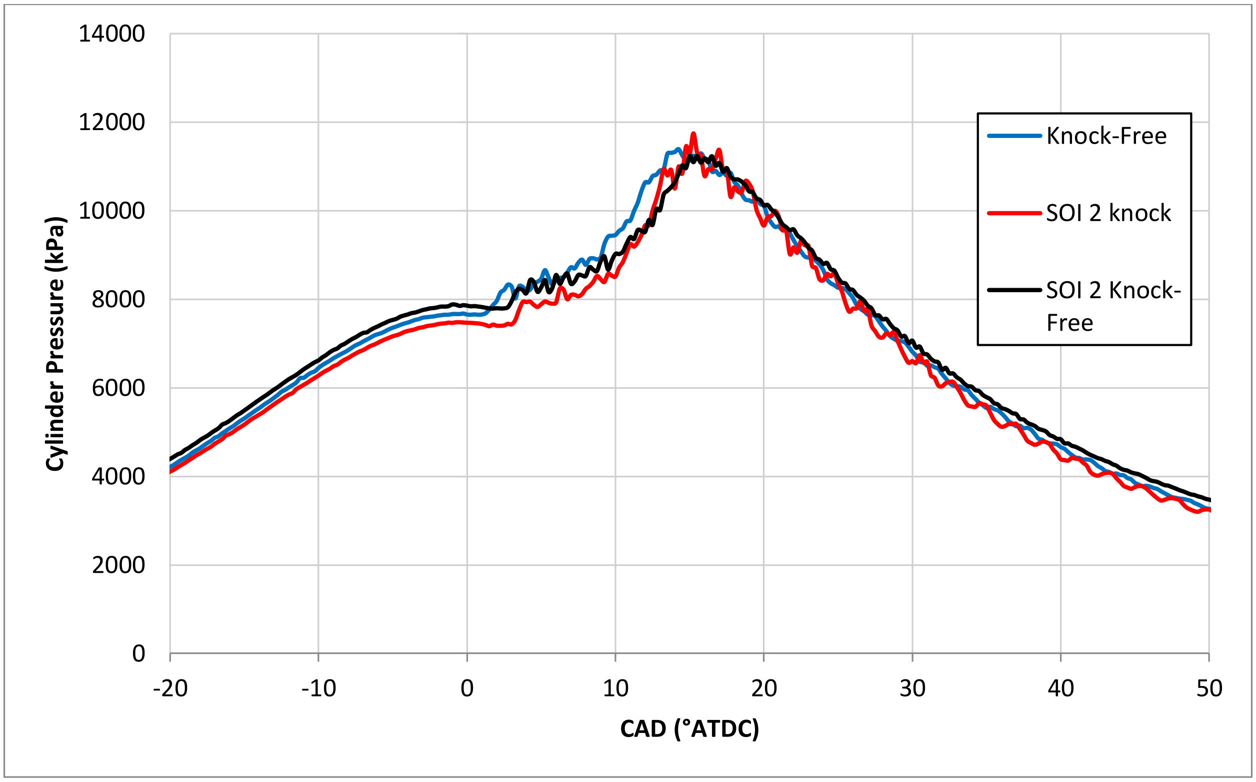

Figure 61: In-cylinder pressure traces for 100% load at the substitution limit under normal ECU settings and retarded SOI. The air manifold temperature is 60C just as during the baseline commissioning.

6.1.2 75% Load

The largest benefit from adjusting SOI timing is observed at 75% load. The amount of diesel fuel injected during dual fuel operation is typical of operation at a lower load if running on only diesel. At lower loads it is typical to advance the timing but this action is problematic for increasing substitution. Operating with a substitution of 66% and normal ECU settings, the SOI is advanced to 5.9 ^{o}btdc and resulted in heavy knock. Retarding the timing to 3 ^{o} BTDC enabled the engine to operate safely at 66% substitution. This is a greater increase in the substitution than achieved from retarding the timing at 100% load. Figure 62 shows individual cycle pressure traces for knock under normal timing and the retarded timing cases. With a SOI of 5.9^{o}, the pressure rise due to combustion starts at tdc. Retarding the timing pushes the initial pressure rise later after tdc. With combustion occurring later in the expansion stroke the peak pressure is reduced by around 2000 kPa. There are some pressure oscillations in the SOI 3^o case but not the large pressure oscillations at the peak pressure. At the 50% load case, the pressure ringing can be caused by normal dual fuel operation because of the geometry and multiple flames propagating from around each diesel jet.

The heat release rates are presented in Figure 63. Reaction rates are controlled by temperature, pressure and concentrations of fuel and radicals. The heavy knock case and the SOI 3^o case have similar substitution levels so concentrations of fuels are similar. After the ignition delay, the pressures are similar for the knock and SOI 3^o case also. The difference between the two cases is the phasing of the SOI 3^o case is later in the expansion stroke so the cylinder volume is larger. The greater cylinder volumes during the combustion process results in lower temperatures and thus lower heat release rates. The heat release rates are still much higher than diesel only operation at 75% load. The exhaust gas temperatures in the ports were between 650-670^{o}C so retarding the timing did not pose a risk of damaging the exhaust valves or turbocharger components.

The heat release traces presented in Figure 63 show dual fuel operation has a longer ignition delay compared to diesel only operation. The diesel is injected in a jet spray that is locally rich and above the rich flammability limit. Some of the diesel must mix with air to a state that is within the flammability limit. In diesel only operation, the mixing is only between diesel and air. During dual fuel operation, some of the air in the cylinder is displaced with natural gas. This makes the mixing duration longer to achieve a fuel to air ratio below the upper limit of ignition. The ignition delay is independent of injection timing. The initial stage of heat release for the heavy knock cases, which diesel is injected at 5.9 ^{o}btdc, and the retarded timing case line up well when plotted versus crank angle degrees after SOI. The first peak of the heat release is

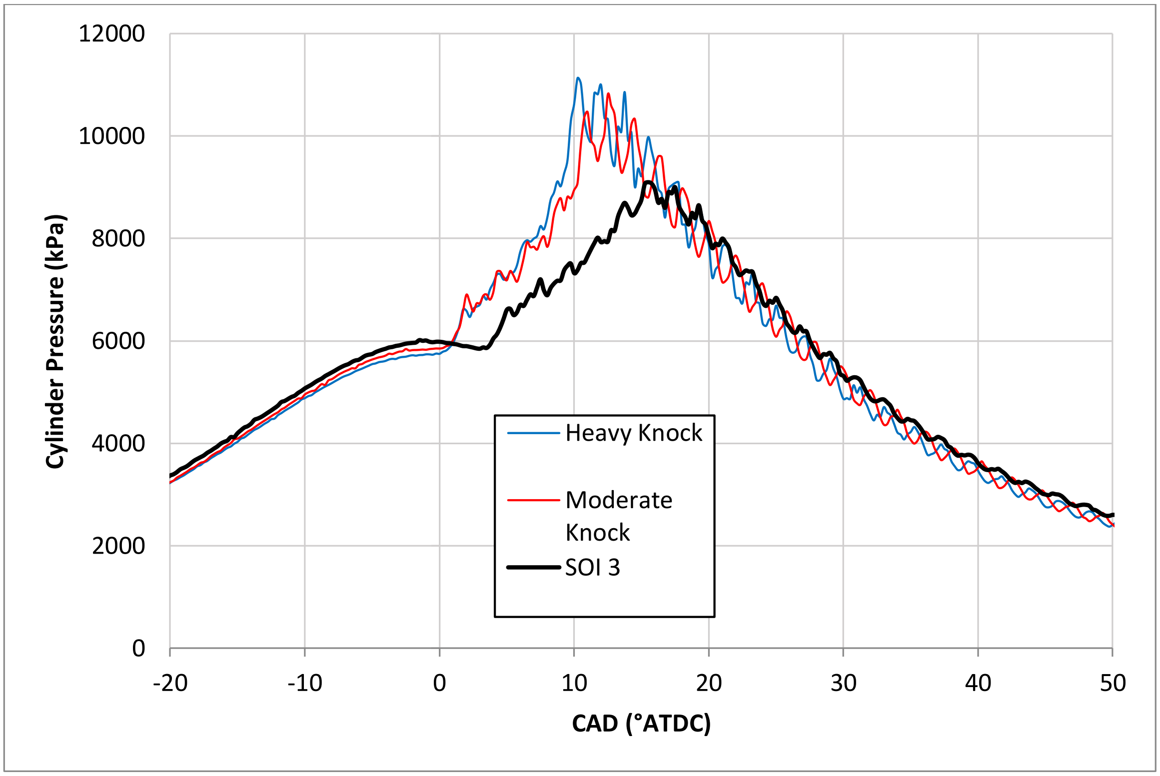

Figure 62: Individual cycle pressure traces at 75% load with normal and retarded SOI timing.

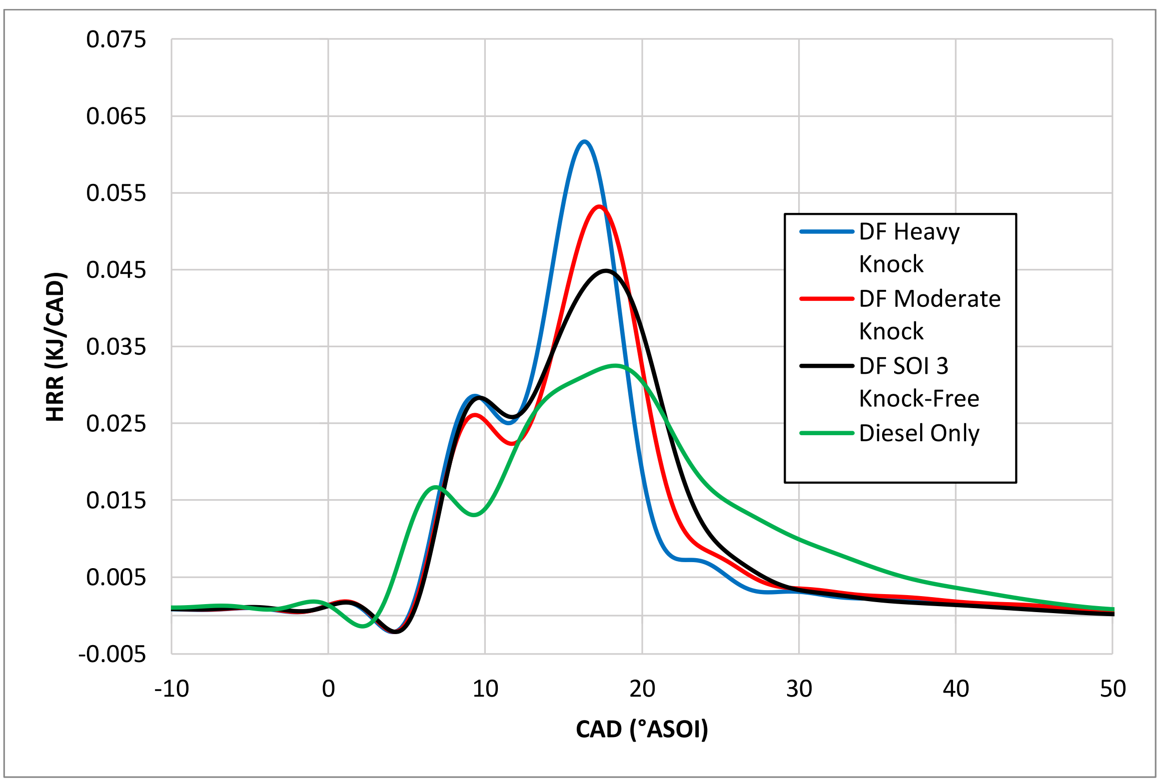

Figure 63: Heat release rate profiles for 75% load with various SOI timing.

larger for dual fuel operation than diesel only operation. The natural gas is adding to some of this early heat release. Also, since the mixing takes longer to reach flammable conditions, the transport of the jet entrains a larger area of the combustion chamber. The heat release rate of the heavy knock and no knock case with a SOI of 3 start to deviate after 12 crank angle degrees. The knock case reaches a peak heat release rate of .015 KJ/CAD greater than the SOI 3 case. The substitution levels are the same the slower heat release rates are due to expansive cooling slowing the chemical reaction rates.

6.1.3 50% Load

During the substitution limit testing, it was found that 50% load is able to operate with the natural gas throttle wide open. This much natural gas flow led to an emissions limited condition for the engine rather than an abnormal combustion event. The Tier II limit combines oxides of nitrogen and non-methane hydrocarbons into one category. From the baseline commissioning results it was determined the NOx levels were nearly the same as diesel only operation but the hydrocarbon levels were elevated. This makes reducing THC emissions the focus of adjusting the SOI timing. The initial prediction was advancing the timing will reduce the hydrocarbon levels. Advancing the injection timing moves the peak pressure location closer to top dead center and typically raises the peak pressure. A higher peak pressure earlier in the expansion stroke is accompanied by elevated cylinder temperatures for longer durations that should promote burning of the pre-mixed natural gas. While this result was observed, the effect was marginal. In fact, the increase in NOx from advancing the timing was more than the reduction in hydrocarbon emissions. A test was performed at retarded timings to test the opposite of the initial hypothesis. Retarding the timing reduced NOx with a slight increase in hydrocarbons. This result was unexpected because the best of the conditions tested is a start of injection timing of 2 ^{o}btdc. The brake specific emissions at 50% load are presented in Figure 64.

The BSTHC and BSCO emissions are much higher for dual fuel operation compared to diesel only operation. Adjusting the SOI is able to reduce the NOx levels below the diesel only operation levels though. The BSNMHC and NOx levels are brought closest to diesel only values by retarding the SOI. The within emissions limit and SOI 2 case were operating at a substitution of 74%. For comparison, advancing the SOI to 8 ^{o}btdc and operating at a substitution of 74%, the emissions were 3.22, 6.44, 7.62 and 8.98 g/bkW-hr for BSTHC, BSCO, BSNOx and BSNMHC+ BSNOx, respectively. The increase in NOx is the largest change in emissions levels from advancing the timing.

Observing the 5-Gas exhaust concentrations live during the testing revealed a few consistent trends. First, the concentration of hydrocarbons in the exhaust stream is controlled primarily by the substitution level while having little response to SOI timing changes. At the

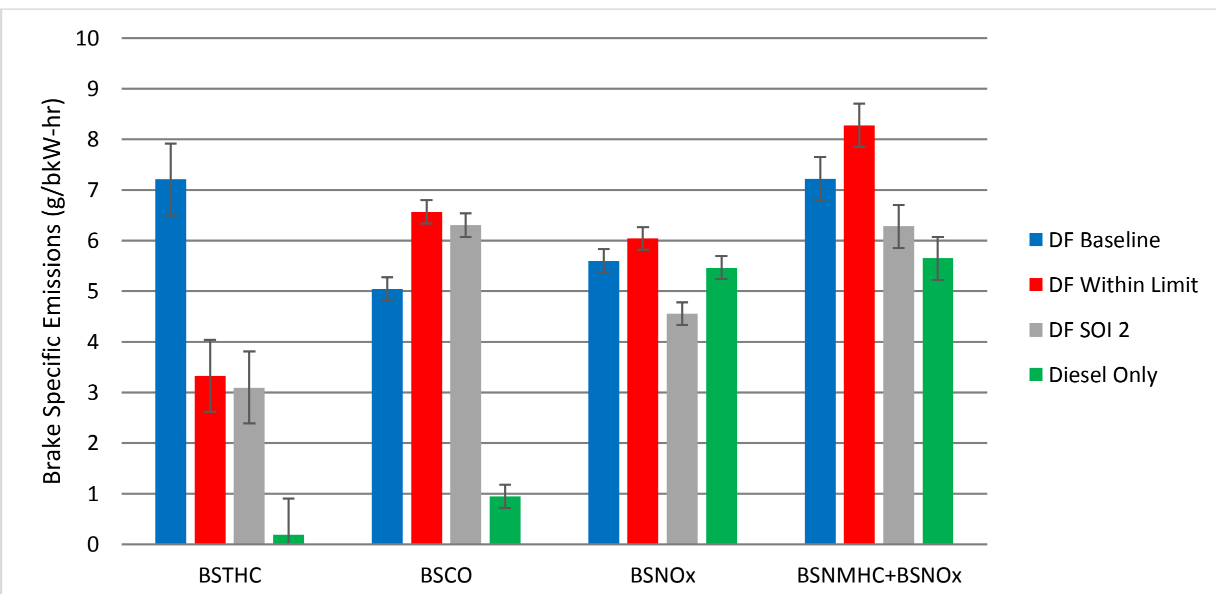

Figure 64: Brake specific emissions levels for 50% load at various SOI timing and substitution levels.

same substitution, the THC concentration in the exhaust was 3.22, 3.32 and 3.09 with injection timings of 8, 5.8 and 2 ^{o}btdc, respectively. Second, concentrations of NOx are controlled primarily by SOI timing and are largely unaffected by substitution levels.

The in-cylinder pressure traces and heat releases rates are presented in Figures 65 and 66. The highest level of substitution corresponds to the most natural gas fumigated into the intake and reduces the ratio of specific heats more than any other test condition. The difference in motoring pressure is just less than 1000 kPa compared to diesel only operation where only air is compressed. With dual fuel operation and standard ECU settings for SOI, the initial pressure rise occurs at nearly the same time as diesel only operation. When the SOI is retarded, the cylinder pressure drops by approximately 500 kPa from peak motoring pressure before combustion starts to increase the pressure at 10^{o}. The peak pressure of dual fuel operation with stock timing is greater than diesel only operation but the peak pressure of dual fuel operation with retarded timing is less than that of diesel by over 1000 kPa.

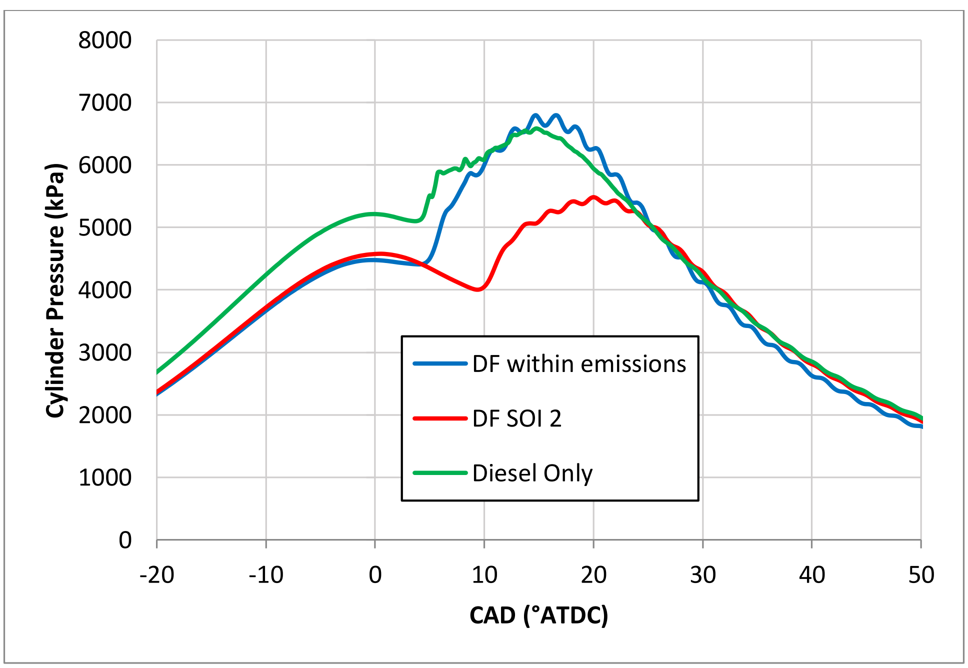

Figure 65: Average in-cylinder pressure traces at 50% load with various DF and diesel only conditions.

The heat release rates for dual fuel with stock timing and a SOI of 2° have similar profiles when normalized to degrees after SOI as shown by Figure 66. Again, the ignition delay is longer because of a longer mixing process to achieve flammable conditions. A major difference between dual fuel and diesel only operation at 50% load is the first peak of heat release compared to the second peak. For dual fuel operation, the first peak of heat release rate is greater than the second peak of heat release. For diesel only operation, the second peak of the heat release rate is greater than the first peak. At high loads, the first peak in the heat release rate of dual fuel combustion is greater than diesel only operation because some natural gas is being entrained in the premixed diffusion combustion of diesel. At 50% load, this has a large enough effect that the peak is higher than the primary heat release rate. Typically, only low loads so this characteristic where the first peak is the primary energy release. Although the heat release rates are similar for both dual fuel cases, the pressure rise rate is slower in Figure 65 for the later injection timing. This is also shown by the second peak of the heat release lagging farther behind the standard injection timing case than at the first peak.

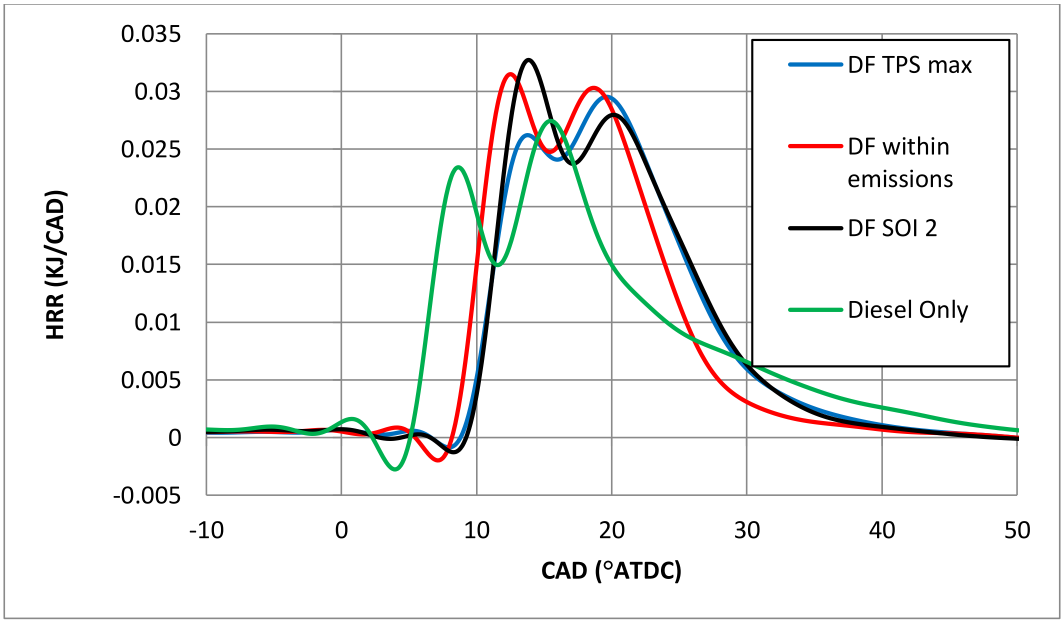

Figure 66: Average heat release rates at 50% load with various DF and diesel only conditions.

6.1.4 25 and 10% Load

Quarter load shows a similar trend to half load based on SOI's timing effects on emissions. The substitution controls THC emissions while timing primarily controls NOx emissions. However, at a lower load the combustion pressures and temperatures are lower so the effect of timing on NOx emissions is diminished. The emissions at 25% load are presented in Figure 67. The optimum timing was 2 °btdc at 50% load but it is earlier at 25% load. Advancing the timing has a more substantial effect on THC compared to NOx at light load so the optimum timing is at 5 °btdc, which was the same timing as under normal ECU configurations. For this reason, the emissions for the within limit case and SOI 5 tests are nearly the same.

The substitution is reduced to around 30% to be able to meet emissions. With this little energy coming from natural gas, the THC emissions are greatly reduced. Carbon monoxide emissions also are reduced from less natural gas usage. The NOx formation is greater than the baseline commissioning that operated at higher substitution levels.

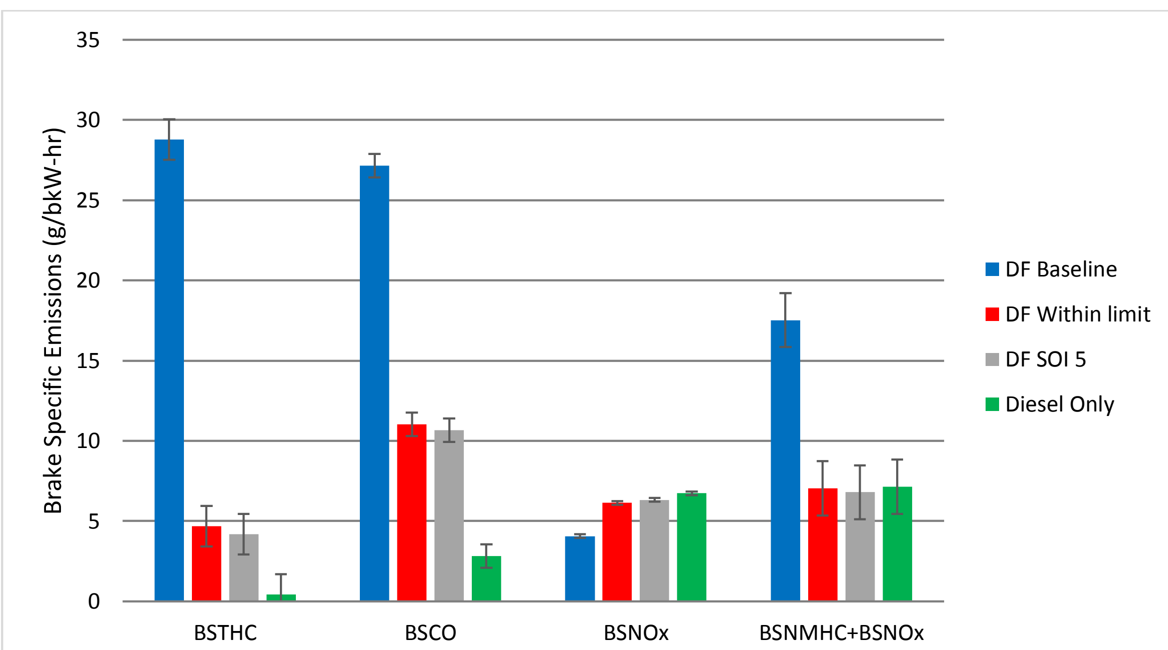

Figure 67: Brake specific emissions at 25% load for various conditions.

The combustion is primarily from diesel combustion and at stock injection timings so the NOx emissions are similar to diesel only operation. The BSNMHC+ NOx emissions are only slightly higher than the BSNOx levels. This shows most of the THC's exhausted from the engine are methane molecules. Without flame propagation at low loads, most of the THC emissions are due to fuel in regions away from the diesel jets that are not introduced to the diesel combustion by methods of air turbulence such as swirl. The energy release is primarily from the diesel fuel so the pressure trace is nearly identical to a diesel only pressure trace. The pressure traces and heat release rates are shown in Figures 68 and 69. The dual fuel tests do not show an increase in the height of the peak HRR as shown at higher loads. The heat release is slightly faster though for dual fuel cases than the diesel baseline initially and then slightly slower towards the end of combustion.

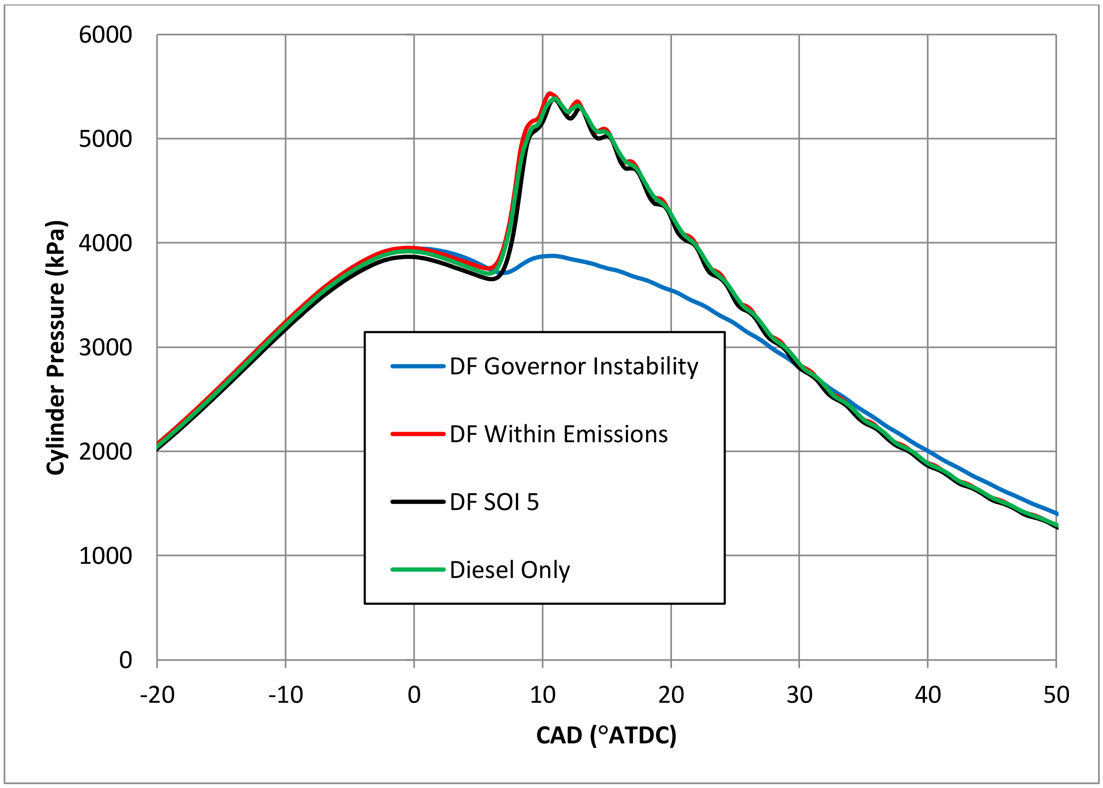

Figure 68: Average in-cylinder pressure traces at 25% load

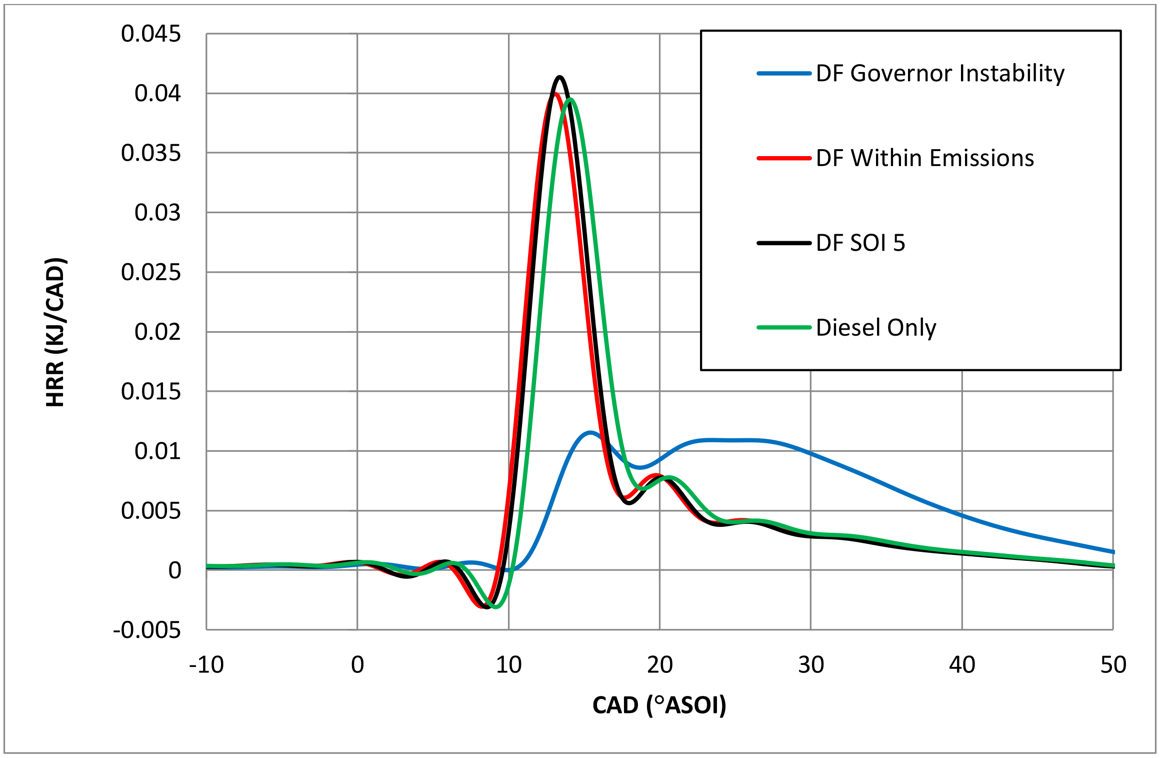

Figure 69: Average heat release rate at 25% load

The lowest pressures and temperatures of any of the engine operating conditions are at 10% load. Because NOx is exponentially dependent on temperature, the SOI timing changes had the least effect on NOx. It is determined that a start of injection timing of 9 ^{o}btdc was able to reduce the combined NMHC + NOx emissions the most. However, even with an advanced injection timing any increase in substitution would increase the hydrocarbon emissions above the defined emissions limit at 10% load of 25.2 g/bkW-hr for NMHC+ NOx. A timing change was not able to increase substitution levels at either 25 or 10% load because of this. The brake specific emissions at 10% load are shown in Figure 70. The diesel flow rate is marginally reduced when running on dual fuel at 10% load. The substitution rate is around 30% so the energy release is primarily from diesel fuel. The pressure trace again is very similar to the diesel baseline trace. The in-cylinder pressure and HRR are presented in Figures 71 and 72.

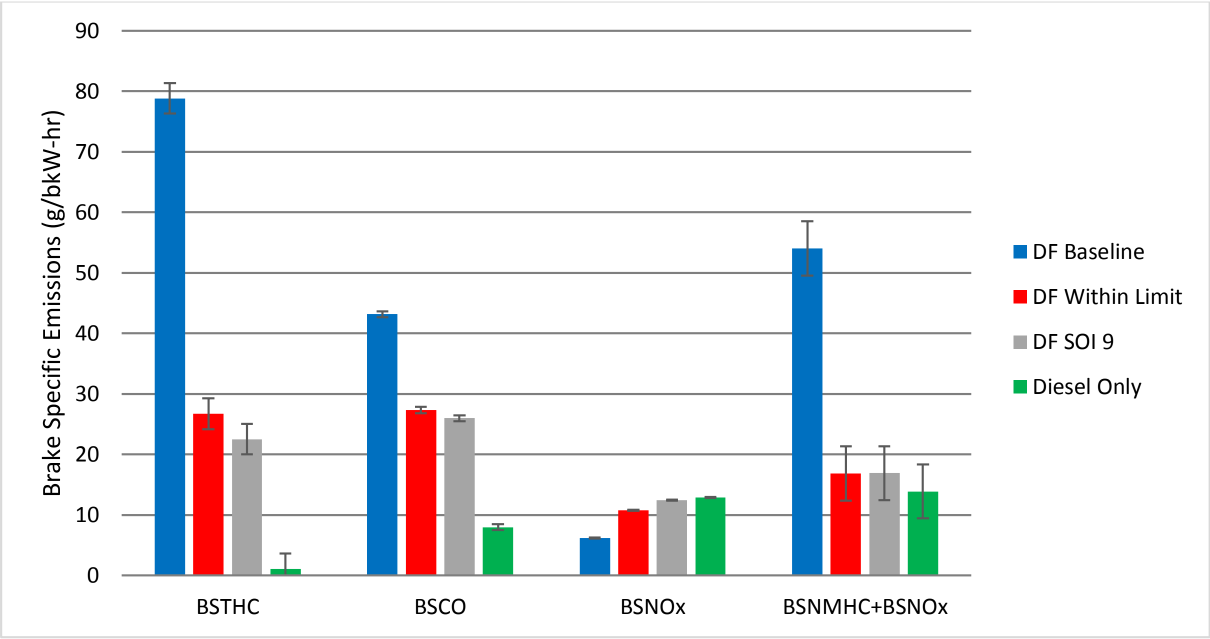

Figure 70: Brake specific emissions at 10% load with various DF and diesel only conditions

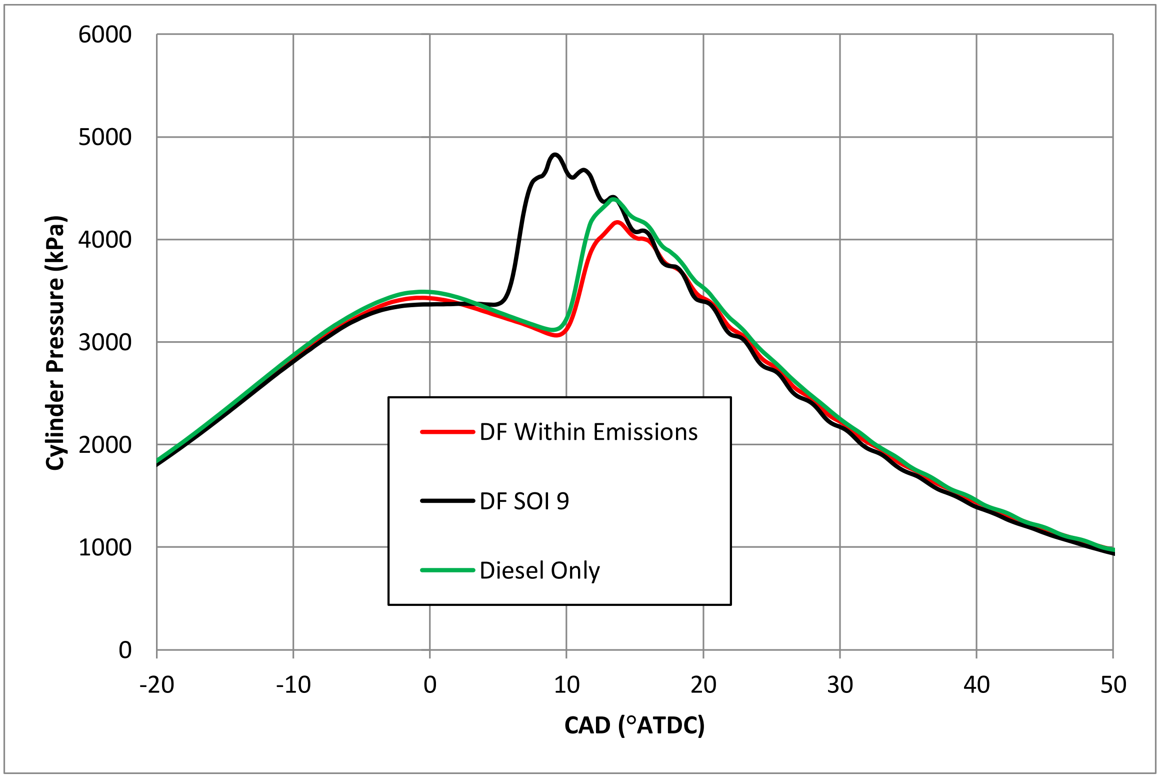

Figure 71: Average In-cylinder pressure at 10% load with advanced injection timing.

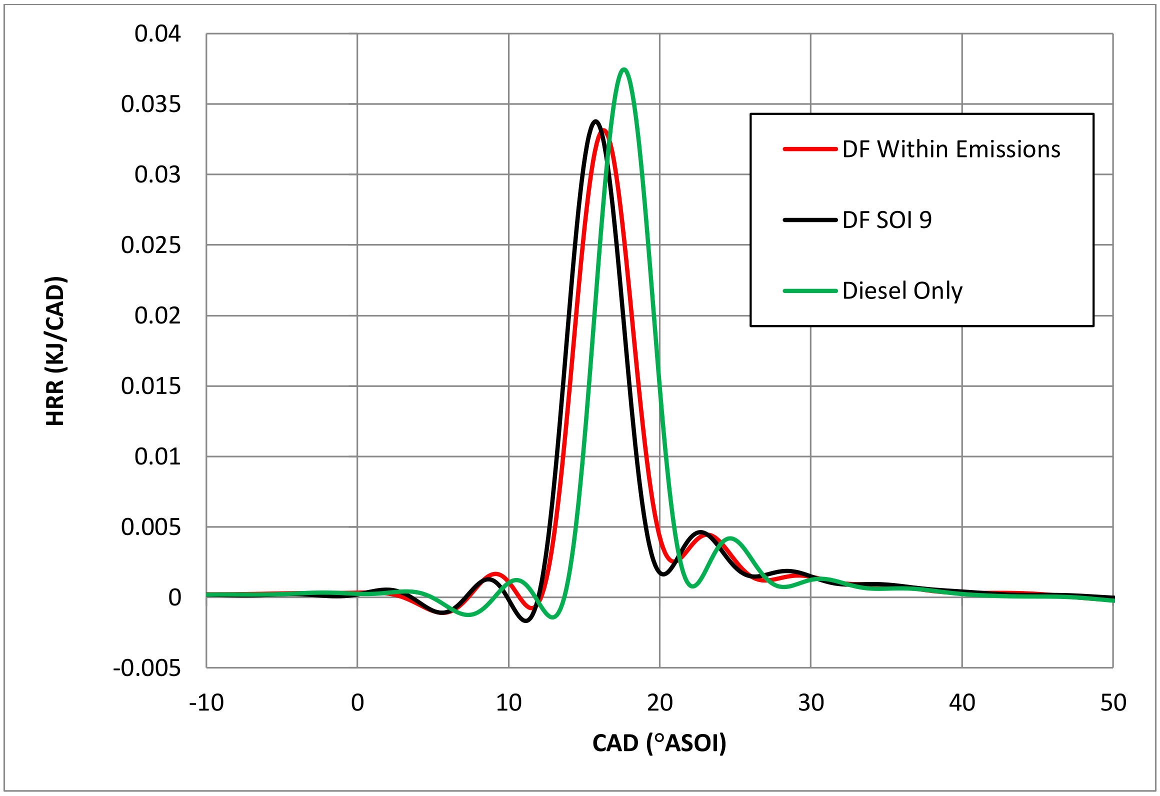

Figure 72: Average heat release rates at 10% load with advanced SOI timing.

Advancing the SOI timing from 6.5 to 9 ^{o}btdc shifts the peak pressure closer to top dead center. It is possible advancing the SOI timing even earlier would be able to decrease THC emissions. However, it is likely that the equivalence ratio of the premixed fuel would have to be much higher. The equivalence ratio for both dual fuel tests presented in Figure 70 are around 0.12. The heat release rate for each dual fuel case is similar despite phasing. The earlier injection results in a peak pressure that is approximately 500 kPa higher than at stock timing. This is about the same amount the pressure decreases from the peak motoring pressure to the start of the pressure rise from combustion around 10 ^{o}atdc.

6.1.5 Summary

Adjusting the start of injection timing of an engine affects the combustion in two ways. First, the volume at which combustion is occuring is changed. Second, the duration of combustion can be increased or decreased depending on the direction of the timing change. The first mechanism changes the pressure and temperature by the ideal gas law. An increase in volume decreases the pressure, disregarding any reactions. A decrease in pressure will also decrease the temperature of the gas. The lower temperature will result in slower chemical reaction rates. This is a desired effect at high loads where addition of natural gas increases reaction rates. The fast reaction rates cause very fast pressure rises and increases in temperature of the unburned fuel air mixture. This can lead to end gas auto-ignition that damages primary engine compinents. Slowing the reaction rates allows more natural gas to be added while avoiding abnormal combustion events. An increase in chemical reaction rates is desired at light loads to promote burning of all of the premixed fuel. Advancing the SOI timing should increase the pressure and temperatures of combustion. The combustion duration also becomes an important factor for low loads, especially for premixed flames. If the duration is too long, the volume of the combustion chamber can increase too much allowing flame quenching to occur and exhausts high levels of pollutant emissions. Advancing the SOI can increase the combustion duration before conditions favor quenching and thus utilizing more of the fuel. This mechanism does not affect diesel combustion though because of the burning characteristics of diesel. It also does not affect dual fuel combustion when the equivalence ratio is too low for flame propagation. This is case of dual fuel operation at light loads and why advancing the SOI at light loads is not able to provide increases in substitution levels. The primary effect of changing the injection timing for dual fuel operation is making small adjustments to the combustion temperatures.

6.2 Air Manifold Temperature

During the baseline commissioning the dependence of substitution on air manifold became evident while a thermostat setting was being selected for use throughout the rest of the testing. The setting was desired to be a temperature that is reasonable for field operation. This was originally set to 65°C but then lowered to 60°C. Even this small change allowed a substantial increase in natural gas utilization. This is the intake manifold temperature used throughout baseline testing and the variation of SOI timing testing. Testing to examine the magnitude of this effect at different loads was performed. The hypothesis is lowering the air manifold temperature slows the reaction rates delaying knock. At low loads, heating up the incoming charge adds energy to the fuel air mixture reducing the amount of energy needed from combustion of diesel to burn the surrounding natural gas.

6.2.1 100% Load

The results shown in Figure 73 confirm the prediction that reducing the air manifold temperature allows an increase in natural gas usage. The accompanying overall equivalence ratio and pre-mixed equivalence ratio are shown in Figure 74. The overall equivalence ratio generally decreases as the substitution increases because air is displaced by natural gas. The natural gas equivalence ratio increases with the substitution level as the AMT is lowered. The testing to extend substitution limits by retarding the SOI timing at nominal 60° C are also plotted. After determining the increase in substitution based on only changing the air manifold temperature, the SOI timing was also changed as an optimum case. An air manifold temperature of 32°C and SOI timing of 2 °btdc enabled the engine to safely operate with close to 55% substitution. At the same air manifold temperature changes to the SOI timing increase the substitution by approximately four percentage points. Decreasing the temperature by close to 30°C extends the substitution limit by ten percentage points.

The thermostat controlling air manifold temperature is able to control from 47 to 92°C so most of the cases the thermostat is at a controlled condition. The tests with temperatures below

47°C, the thermostat was adjusted to be wide open the entire time so the intercooler was operating at maximum cooling capacity. The coolant temperature slowly increases from this and is not able to hold the very low temperatures indefinitely. The temperature rise was slow enough to take short test points to demonstrate the trend.

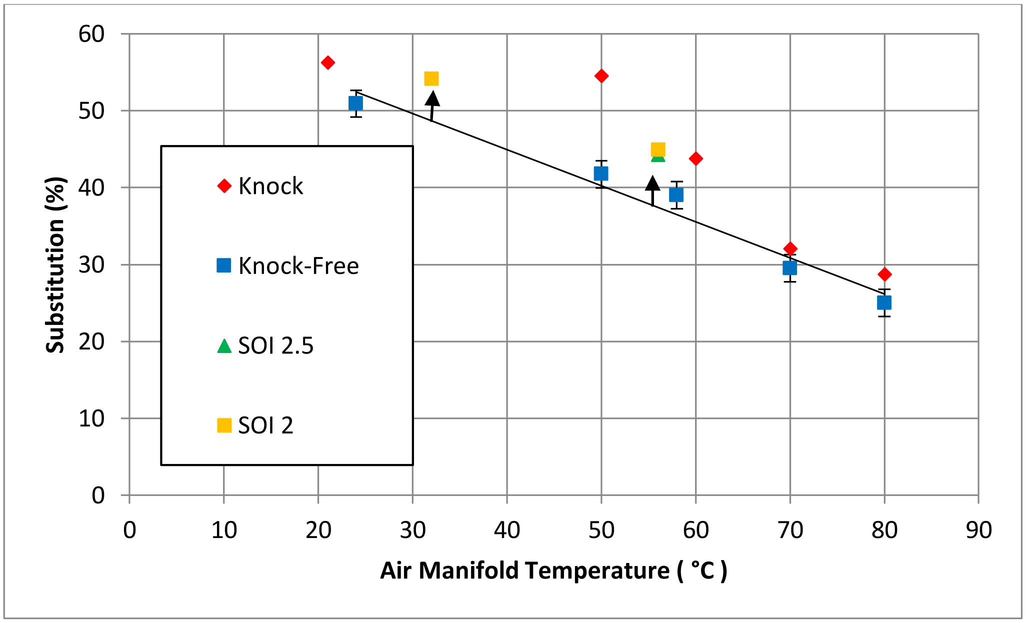

Figure 73: Substitution levels versus air manifold temperature at 100% load. The arrows represent the increase in substitution from retarding injection timing.

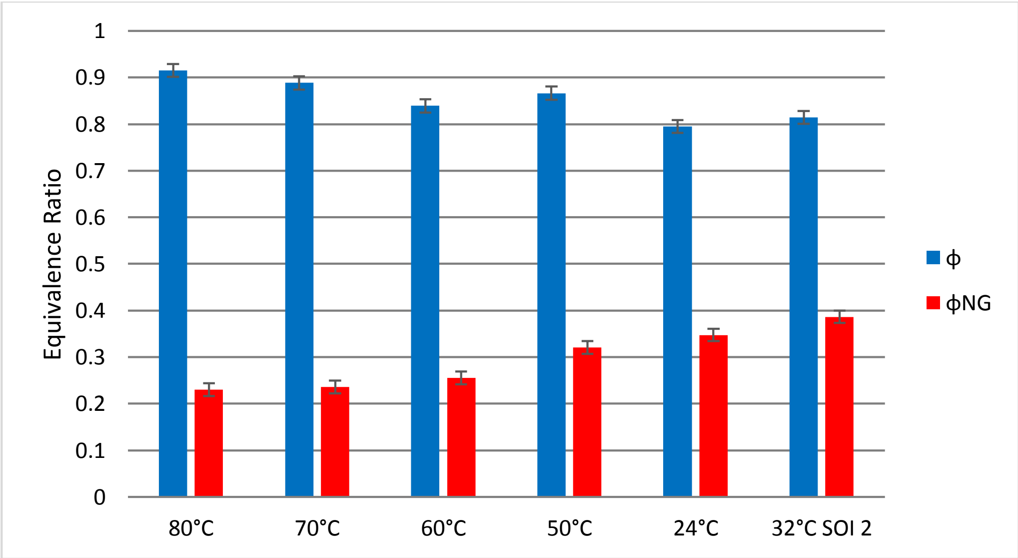

Figure 74: Overall equivalence ratio and pre-mixed natural gas equivalence ratio at 100% load with varying intake air manifold temperatures. The substitution and ΦNG increase with decreasing air manifold temperature. for the data presented in Figure 7.

The pressure traces for the varying temperature cases are presented in Figure 75. The 80ºC test has the lowest substitution level but the highest peak pressure. The peak pressure decreases with air manifold temperature and occurs later after top dead center. The test at 24ºC has a significant change in the pressure trace compared to most of the high load dual fuel tests. The pressure resembles the diesel baseline more than the typical dual fuel pressure traces. The peak pressure is significantly lower than the other dual fuel cases although still higher than the diesel baseline. The pre-mixed equivalence ratio is the highest for the test at cold air manifold temperatures and lowest at high temperatures. This shows the reaction rates have a stronger dependence on temperature than the concentration levels.

The motoring pressure is lower for the dual fuel cases compared to the diesel only case because of natural gas lowering the ratio of specific heats. The difference in peak motoring pressure is about 500 kPa. The ignition delay is the longest for the 24ºC test. It also has the

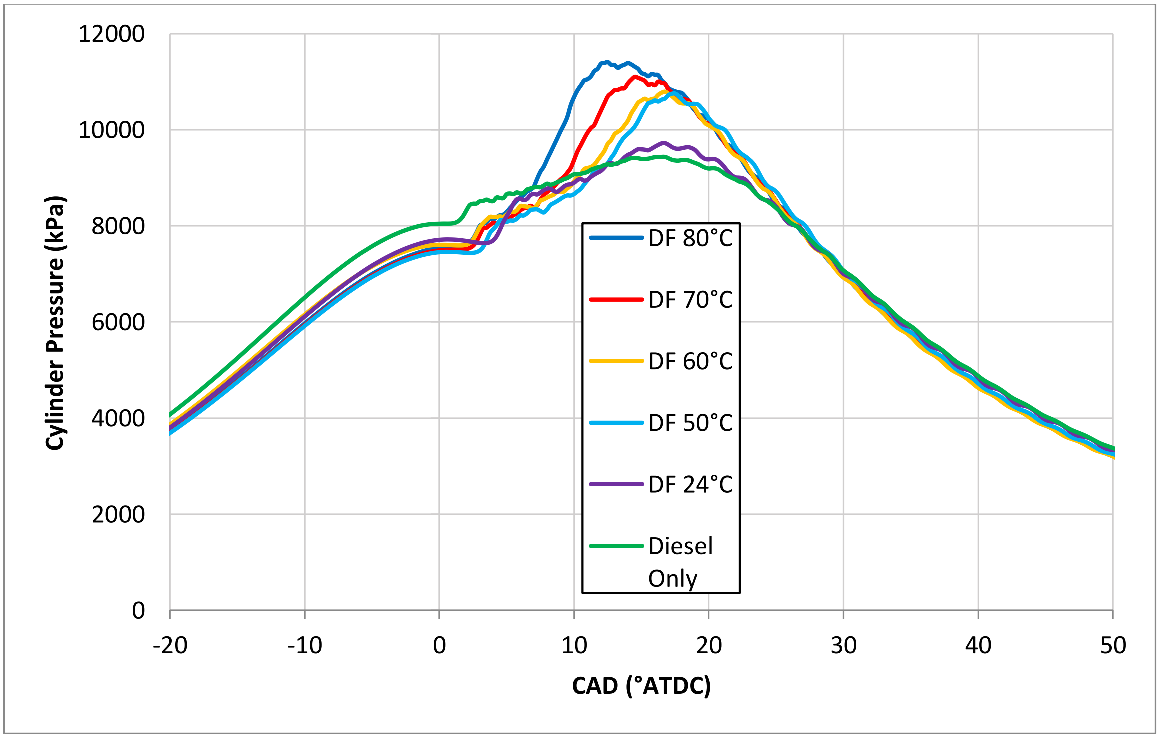

Figure 75: In-cylinder pressure traces for 100% load with varying air manifold temperatures.

largest initial pressure rise. The location of peak pressure moves later after tdc when the charge temperatures are lowered. The location of peak pressure does not move farther back than about 17 ^{o}atdc though. The difference in peak location is small between the 60 and 50^{o}C tests and the peak location is the about the same between the 50 and 24^{o}C. As the crank moves further from top dead center the change in volume with every CAD increases. Around 17 ^{o}atdc the volume expansion is likely causing more expansion cooling that the heat release rates start decreasing if not already slowing due to reduced radical concentrations.

The heat release traces for each test with varying charge temperatures are presented in Figure 76. The initial heat release traces are similar for 80, 70 and 60^{o}C. At 80^{o}C, the combustion quickly transitions from the initial energy release to the main phase of combustion. The test at 70^{o}C has a longer delay before the primary energy release. The test at 60^{o}C has an even longer delay with a slight reduction in heat release rate before ramping back up. This delay period is the main differentiation point between the higher temperature cases. In dual fuel literature the dip in the heat release curve is termed the primary fuel ignition delay [22]. However, the peak of the initial heat release for each dual fuel test is greater than diesel only initial peak. Therefore, some of the natural gas is being entrained in the diffusion mixing of the diesel with air and releasing more energy in the initiation phase of combustion. A possible cause of the delay in the heat release is the diesel fuel transition from premixed to diffusion burning. A limited amount of fuel is readily available to be burned in the premixed initiation of combustion. When this fuel is depleted, there is a delay for more fuel to be mixed at a proper air to fuel ratio. When the intake charge temperatures are low, the natural gas burning during this transition period does not fully make up the difference for diesel not burning. As the charge temperature increases, more combustion of natural gas will take place limiting the delay time. The profile of

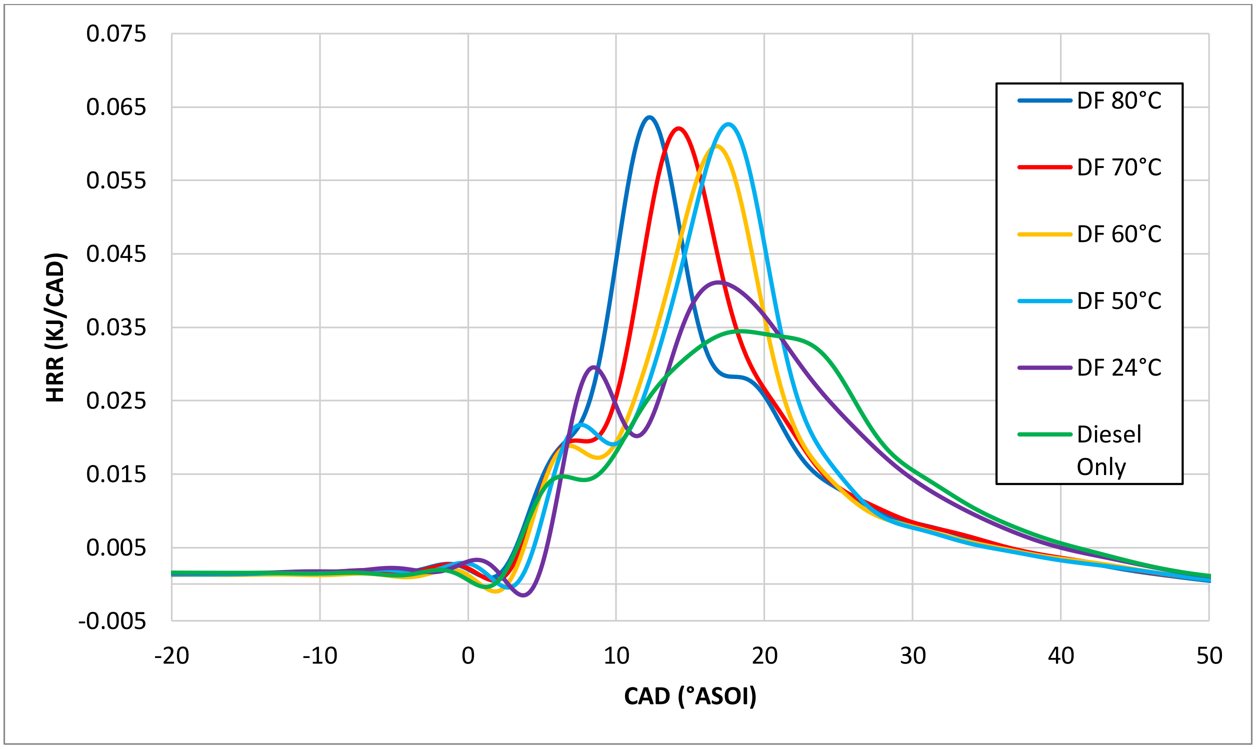

Figure 76: Heat release rates for 100% load at varying air manifold temperatures.

the remainder of the heat release profiles are similar but shifted based on the delay around 8^{o}atdc, except for the 24^{o}C profile.

A possible explanation for the is based on the fact that natural gas is pre-mixed with air before the turbocharger so the elevated temperatures from compression are also heating the natural gas. The intercooler is adjusted to only bring the temperatures down a certain amount. At elevated temperatures the natural gas undergoes more thermal decomposition reactions before the start of combustion creating a larger radical pool. The larger radical pool reduces the delay from the ignition of the diesel fuel to burning of the majority of the natural gas. A theoretical temperature at top dead center can be computed using Equation 14 assuming isentropic compression.

$$T_2 = T_1 r^{\gamma-1} \quad (14)$$

T_2 is the temperature at tdc. T_1 is assumed to be the air manifold temperature controlled by the thermostat. This neglects heat transfer to the charge in the block and cylinder. The compression ratio is given by r, which is 17 for this engine. The ratio of specific heats, γ, is assumed to be 1.35. For an air manifold temperature of 80°C, T_2 is 215°C using these assumptions. At 60°C intake, T_2 is 161°C. When the air manifold temperature is 24°C, T_2 is only 65°C, almost the same as air manifold temperatures. At this low of a temperature it is unlikely much thermal decomposition is creating a radical pool.

At 50°C the ignition delay starts to lengthen. The longer the ignition delay results in a higher first peak in the heat release rate. The longer ignition delay allows more time for transport of the diesel spray and more natural gas entrainment resulting in more fuel readily available to ignite at the time of ignition. The delay to the primary heat release is similar to at 60°C but occurs later due to the longer ignition delay. The most interesting observation here is the heat release rate of the 24°C test. The lower pre-mixed temperature causes an even longer ignition delay. The peak of the heat release rate is much lower than the higher temperature dual fuel tests. The higher temperature tests rise to the peak of heat release and then fall just as quickly as they rose but the test at 24°C shows more of a sustained burn. The accompanying mass fraction burned curves are presented in Figure 77 for the varying air manifold temperature tests.

After achieving high substitution levels by cooling the intake air manifold temperature, the SOI was also retarded to try for a best operating condition point. The in-cylinder pressure traces are shown for this test in Figure 78. The lower temperatures result in combustion similar to the diesel only test. Typical dual fuel combustion at high loads shows larger increases in peak pressures. Operating with cold air manifold temperatures and retarding the SOI to 2 °btdc results

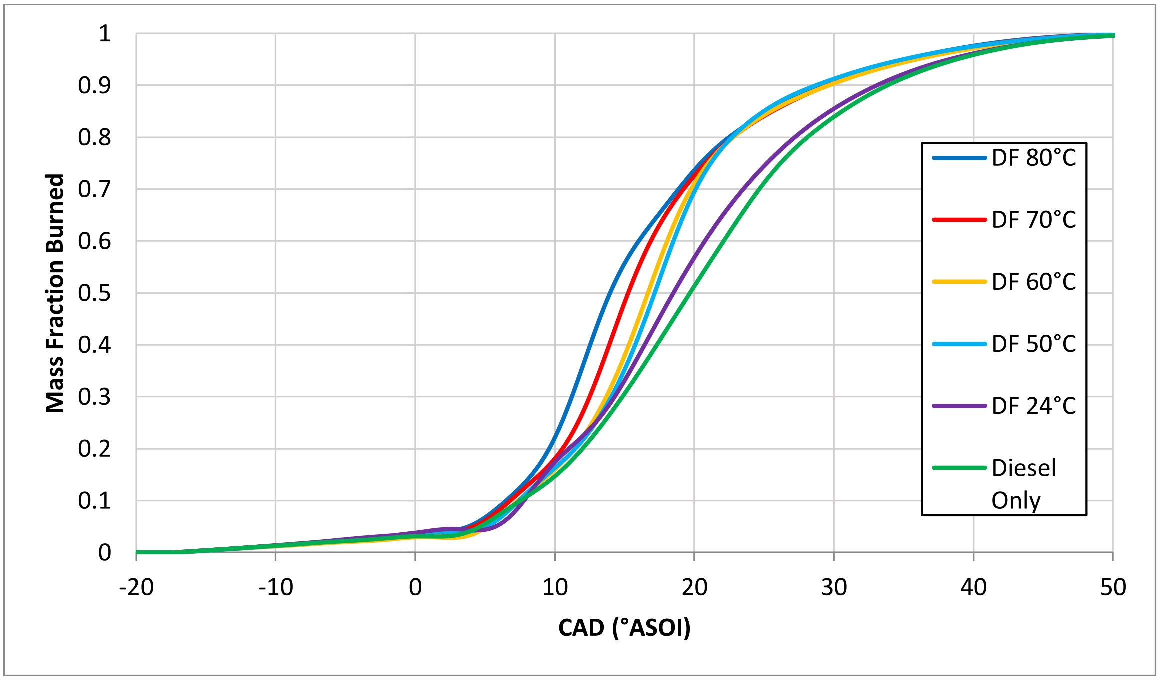

Figure 77: Mass fraction burn curves at 100% load at varying air manifold temperatures.

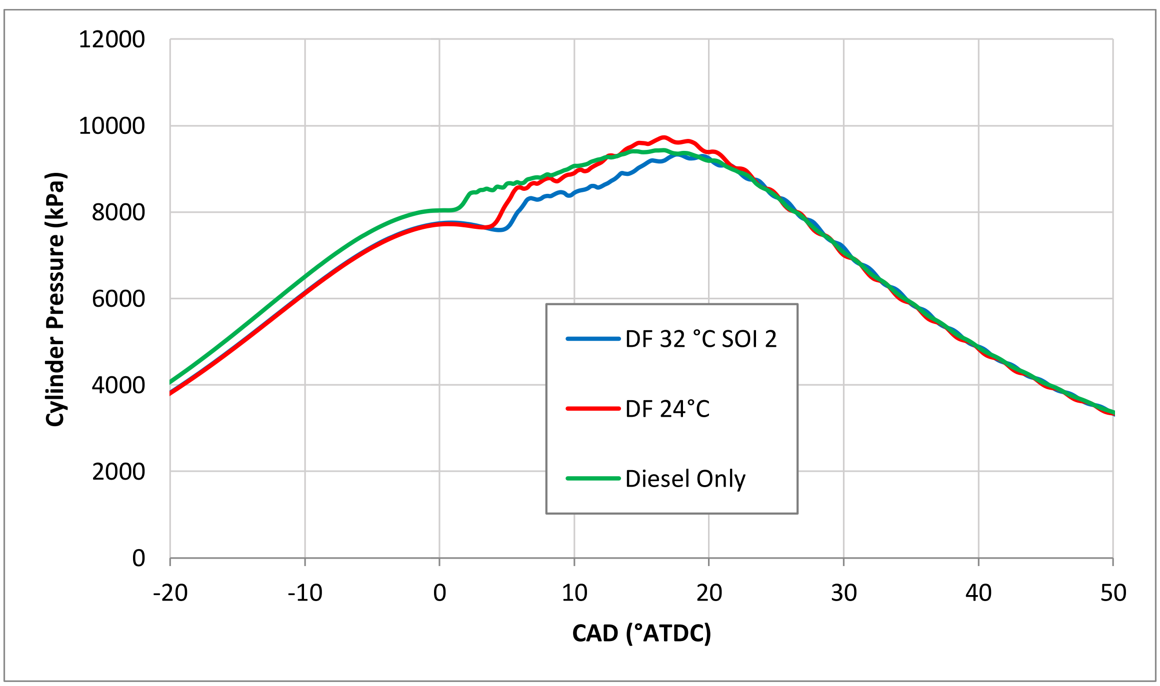

Figure 78: In-cylinder pressure traces at 100% load with the air manifold temperatures as cooled to the maximum capacity of the intercooler and adjusting SOI.

in a dual fuel peak pressure below the diesel only peak pressure. The heat release rate curves are more rounded.

The emissions levels were also recorded during the air manifold temperature testing. The THC emissions decreased with temperature, shown in Figure 79. The actual causality is believed to be an increase of THC with an increase of substitution. This was the trend observed during the substitution limit testing previously. During the injection timing adjustment experiment the THC levels were minimally changed based on injection timing but largely changed with substitution. Likewise, the NOx levels were unchanged from increases in substitution. The results for this test show a decrease of NOx with decreases of air manifold temperature. The combustion pressures and temperatures increase with the air manifold temperature, which favors NOx formation. Typically, changes that increase NOx decrease CO emissions. However, the CO emissions increase as the air manifold temperature increases. This is likely because the additional gaseous fuel that is burned is disassociating but CO oxidation is quenched. CO oxidation is one of the slowest combustion reactions because there is only one possible reaction to form CO_2. The final stages of combustion when CO_2 is formed occur late in the expansion stroke so the reaction rates are slow even though the overall combustion is lean and there is excess oxygen available for the reaction. The test at 60^{o}C was performed on a different day than the rest of the plotted results. Each day had different natural gas constituents and associated methane numbers causing a shift in the trend relative to the other tests. The natural gas compositions are presented in Table 6.

The tests at 60^{o}C in Figure 79 was performed on the substitution limit testing day. Table 6 shows the natural gas composition varied between the substitution limit testing and air manifold temperature testing. The air manifold temperature testing had higher concentrations of ethane, propane and butane. The substitution limit testing day has the highest concentration of any of the test days of Nitrogen. Simulations of dual fuel combustion have only included the Zeldovich NOx mechanism [34] [35], which does not take into account Nitrogen present in either fuel source. Figure 79 shows a significant increase in NOx compared to the trendline from the other data points at various air manifold temperatures. This signifies NOx formation from Nitrogen in the fuel is a significant contributing factor.

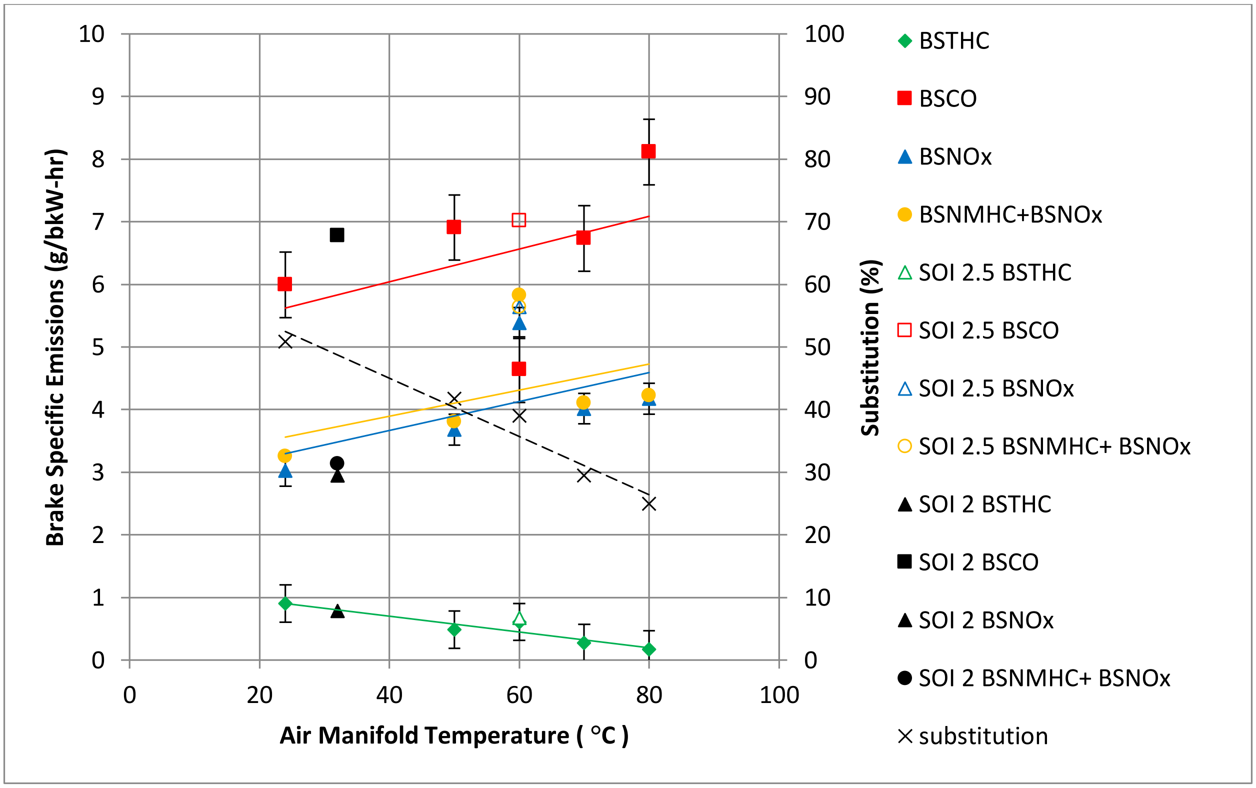

Figure 79: Variations of brake specific emissions levels at 100% load with different air manifold temperatures.

Table 6: Natural Gas Composition During Different Tests

	Baseline Commissioning	Substitution Limit Testing	Air Manifold Temperature Testing
	Mole fraction	Mole Fraction	Mole Fraction
CH_4	.862	.899	.861
C_2H_6	.114	.076	.093
C_3H_8	.006	.005	.020
C_4H_{10}	.001	.0003	.003
N_2	.005	.011	.009
CO_2	.011	.007	.014
Methane Number	76.6	81	75

6.2.2 75% Load

At 75% load the substitution level (Figure 80) increases with a decrease in air manifold temperature just as at full load. The trend appears to be linear. Figure 81 displays average cylinder pressure traces at 75% load. At 100% load the peak pressure decreases with temperature, which is also seen for the high temperature tests in Figure 81. However, at 75% load the tests at the lowest possible air manifold temperatures (17°C) reach the highest peak pressures. Also, at the lowest possible temperature the combustion becomes emissions limited. The natural gas throttle position is able to be opened to wide open without knocking but the THC emissions rose significantly at the lower temperatures and higher substitution rates. The SOI timing was advanced to bring the hydrocarbons down. This brought the THC levels down slightly but increased the NOx levels more.

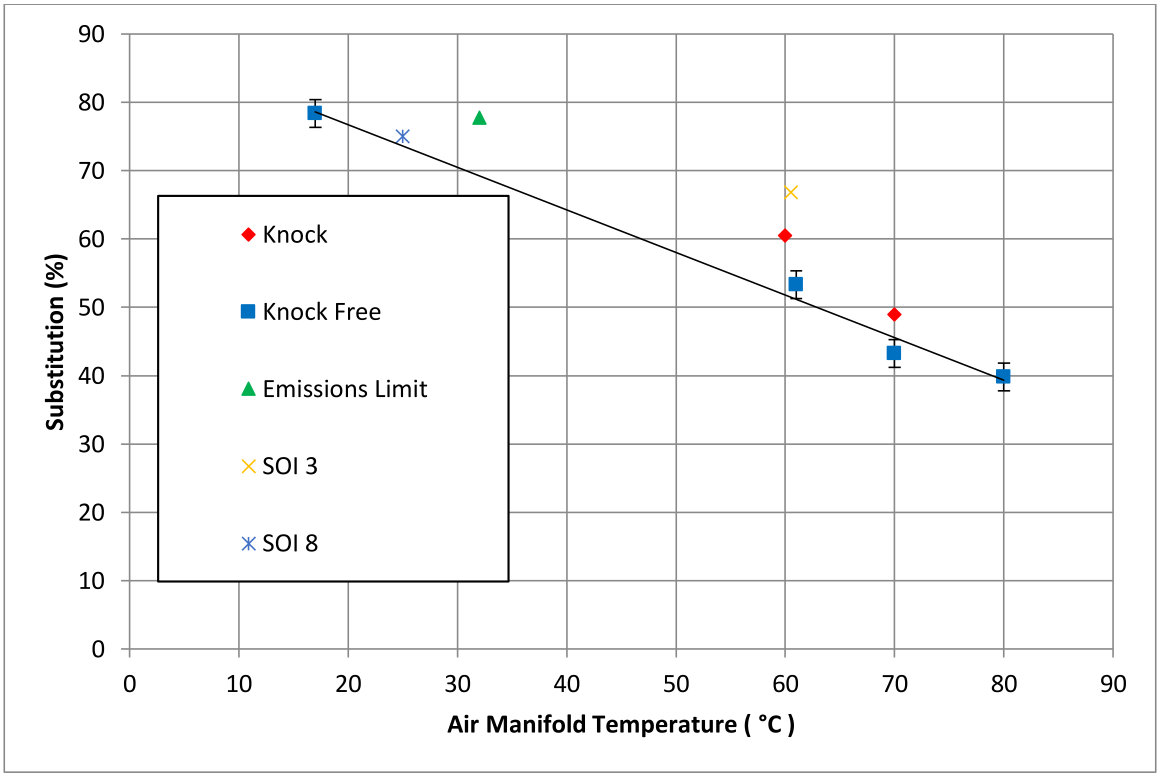

Figure 80: Dependence of substitution levels on air manifold temperature

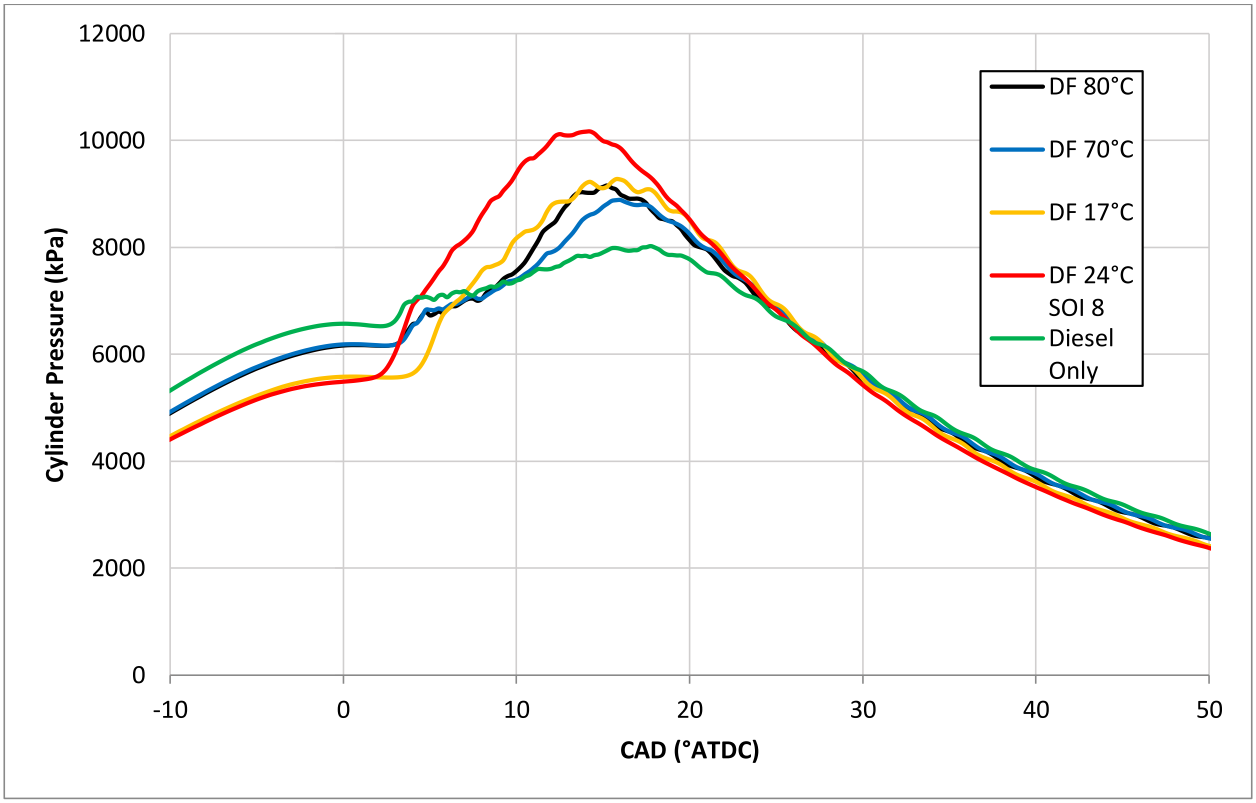

Figure 81: Average in-cylinder pressure traces for 75% load with varying air manifold temperatures. Start of injection for 80C is 3.16, 70C is 4.1, as cold as possible is 6.5and for the diesel baseline is 2.9 degrees before TDC.

The heat release rates plotted in Figure 82 shows a drastic change in dual fuel energy release. At 80 and 70°C the heat release displays the two peak dual fuel characteristics that have been shown in previously at high loads. The cold manifold temperatures cause a long ignition delay. At lower temperatures the first peak of the heat release rate is about twice as large as the 80 and 70°C cases. There is still a second peak in the heat release rate that is slightly higher than the initial peak. Examining the mass fraction burned curves in Figure 83 shows the cold cases are the last to reach 10% of the mass burned yet the first to reach 90% of the fuel burned.

It has been discussed previously that a possible reason for the faster heat release rates for dual fuel operation are due to increases flame areas. It has also been discussed that dual fuel operation has longer ignition delays because of longer times for diesel premixing. When the engine operates at 75% load and 17°C intake manifold temperatures, the substitution level is

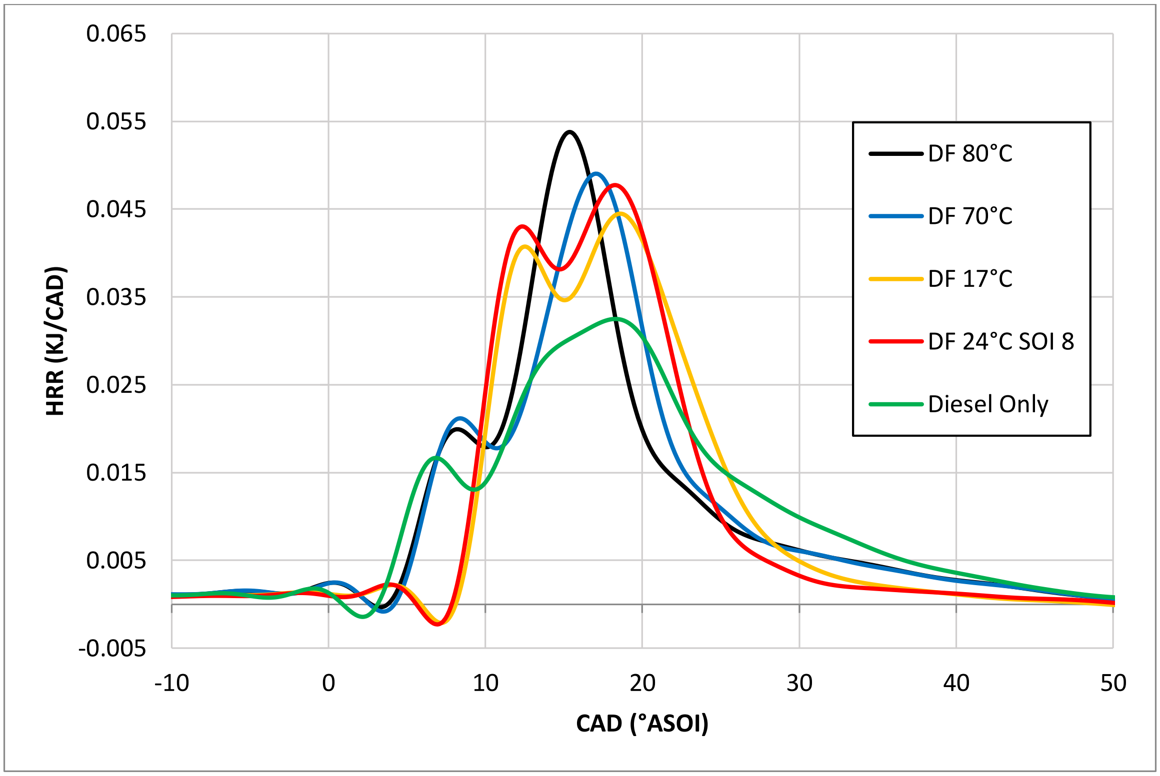

Figure 82: Heat release rates at 75% load with varying air manifold temperatures

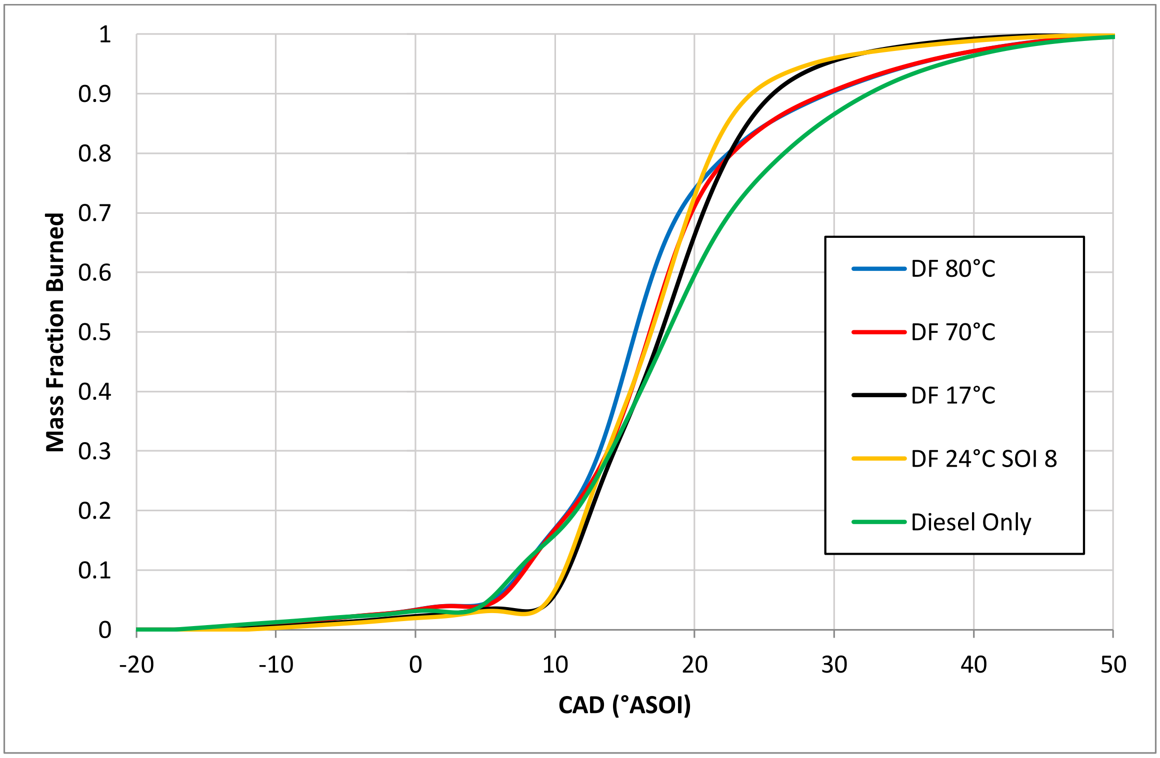

Figure 83: Mass fraction burned curves at 75% load with varying air manifold temperatures.

increased to approximately 75% and the natural gas equivalence ratio is 0.58. At this substitution gaseous fuel displaces much of the air that would normally be in the combustion chamber. As a result, the diffusion time for the diesel jet to be below the rich flammability limit increases by close to 5°. The extra time allows for the spray to reach a larger area of the combustion chamber because of the transport properties of the diesel jet. The reaction rates may be slower because of the lower temperatures but it seems the larger entrainment area increases the energy release during the early stages of combustion compared to the higher temperature cases. The earlier energy release results in the higher peak pressures.

The emissions levels associated with these dual fuel tests are shown in Figure 84. Based on time limitations emissions were not sampled at every temperature set point. The tests conditions of the most interest were the tests at low air manifold temperatures where the substitution was the highest.

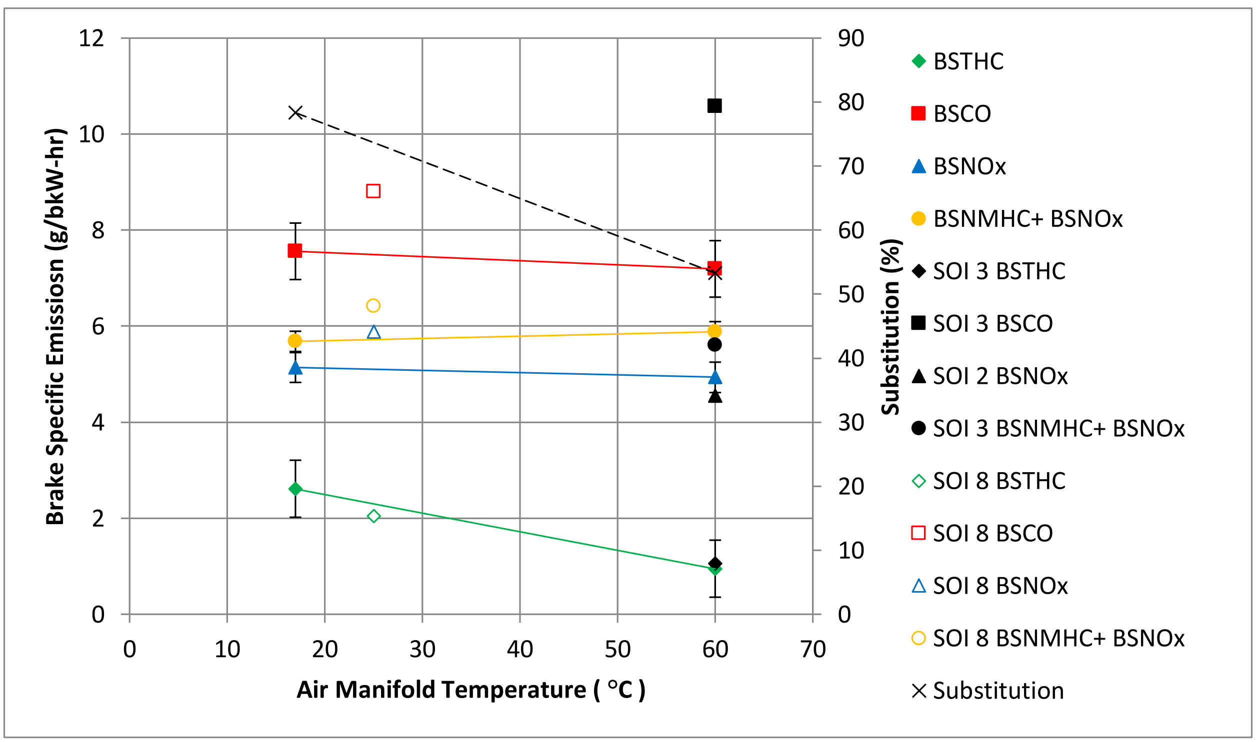

Figure 84: Brake specific emissions at 75% load with varying air manifold temperatures.

The lower temperatures allowed for operation with a wide open natural gas throttle but THC emissions levels became unacceptable. The substitution level was adjusted so the NMHC+ NOx was around the same levels as the baseline commissioning. The trends here are similar to what have been shown throughout the dual fuel results. An increase in substitution increases the THC emissions but not NOx. Advancing the SOI timing in an attempt to reduce THC emissions resulted in a larger increase of NOx emissions. Advancing the SOI also increases the CO levels.

6.2.3 50% Load

Moderate loads are able to operate with the highest substitution levels. Under baseline commissioning testing the substitution was set to 70%. Adjusting the SOI was not able to significantly increase the substitution. Trying both preheating and cooling of the intake air was tested at 50% load with little success. Figure 85 shows there is no clear trend of the allowable substitution versus air manifold temperature. Around 75-80% may be an upper substitution limit for dual fuel operation where the natural gas fuel is fumigated into the intake air stream. The SOI timing is close to tdc so most of the combustion occurs during the expansion stroke. Moderate loads start to experience an increase in THC emissions because of late flame quenching. The later the burning continues into the expansion stroke the higher chance of quenching due to expansive cooling. Injecting the diesel fuel earlier should result in the fuel burning earlier and avoiding late quenching. This was tested by advancing the SOI but was shown to have adverse effects on NOx emissions. The second method tested is promoting better flame propagation by increasing the intake temperature. Chemical reaction rates are exponentially dependent on temperature so increasing the temperature should increase reaction rates. This was also found to be unsuccessful because of the adverse effects on NOx emissions. Figure 86 shows there is an even trade-off between THC and NOx emissions. NOx formation is also strongly dependent on temperature so the reduction in THC was offset by NOx.

Various SOI timings were evaluated at each air manifold temperature to try to find an optimum operating condition. Figure 87 shows emissions levels at various SOI timings while at a constant manifold temperature and substitution. Retarding the timing did not change the hydrocarbon emissions but significantly reduced the NOx emissions. This demonstrates that THC are unaffected by the injection timing at these conditions. However, THC levels are directly proportional to substitution though. Substitution has little effect on NOx formation. Substitution and SOI seem to each independently control THC and NOx, respectively. Dual fuel operation at 50% load had the highest substitution and highest equivalence ratio of natural gas ranging from 0.37 -0.5. This would seem to enable flame propagation throughout all of the premixed fuel.

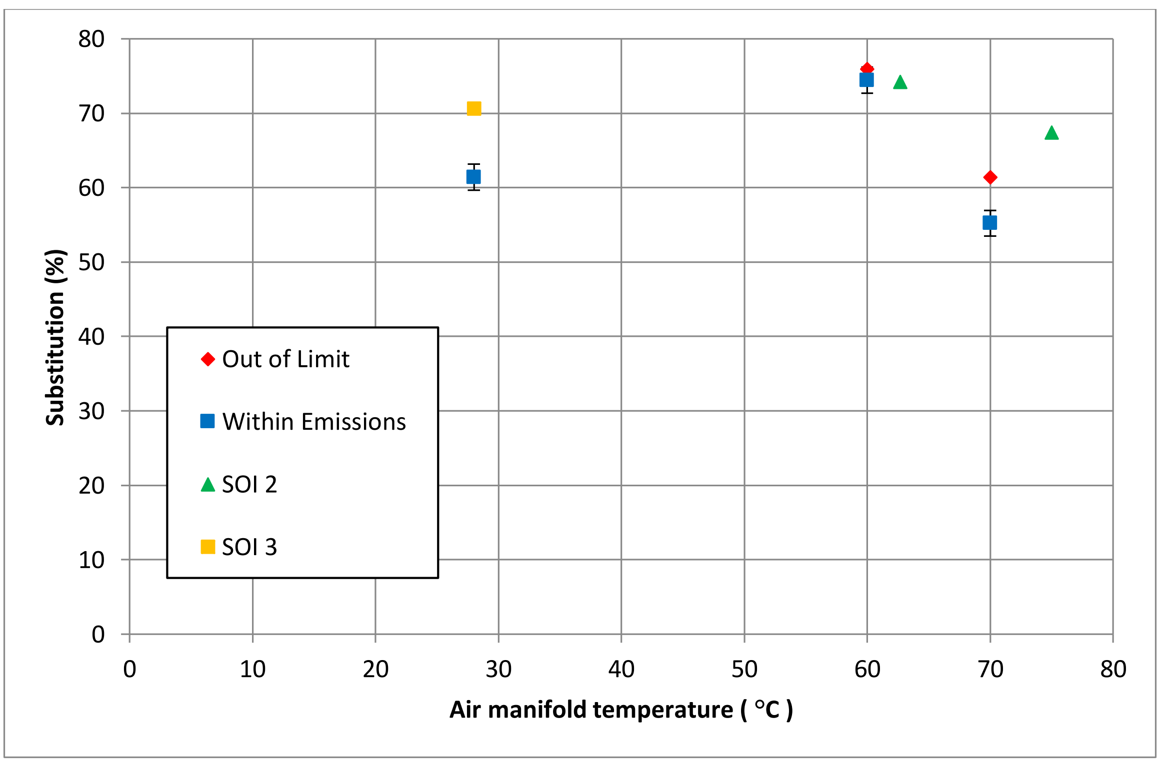

Figure 85: Substitution versus air manifold temperature at 50% load.

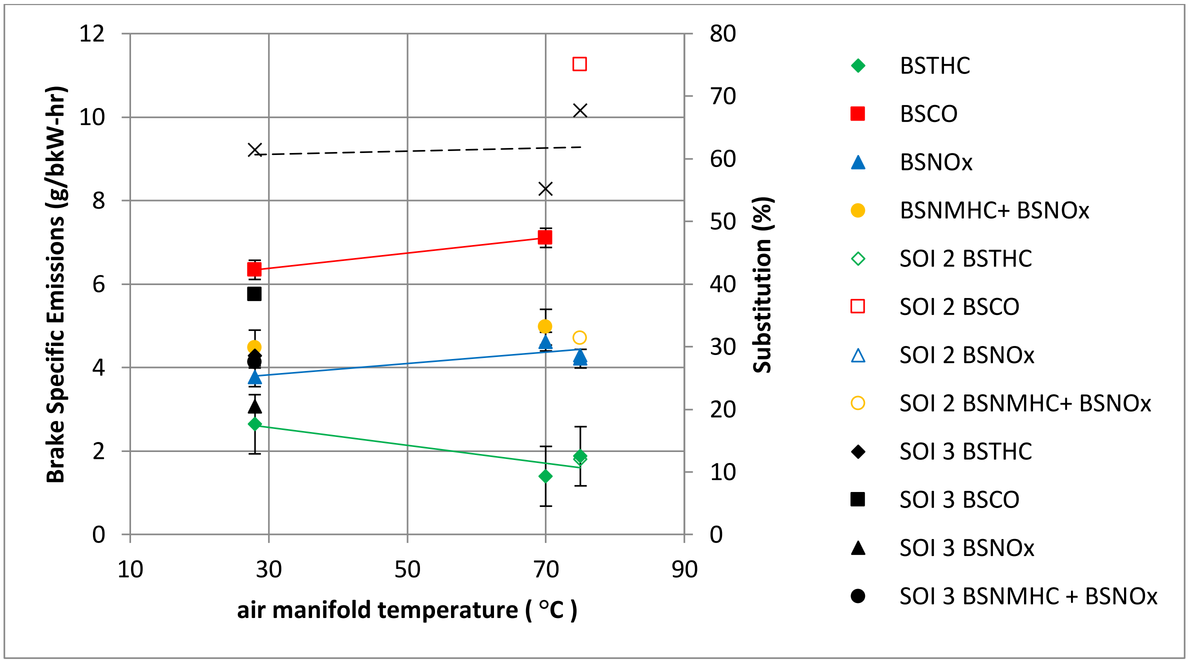

Figure 86: Brake specific emissions versus air manifold temperature at 50% load.

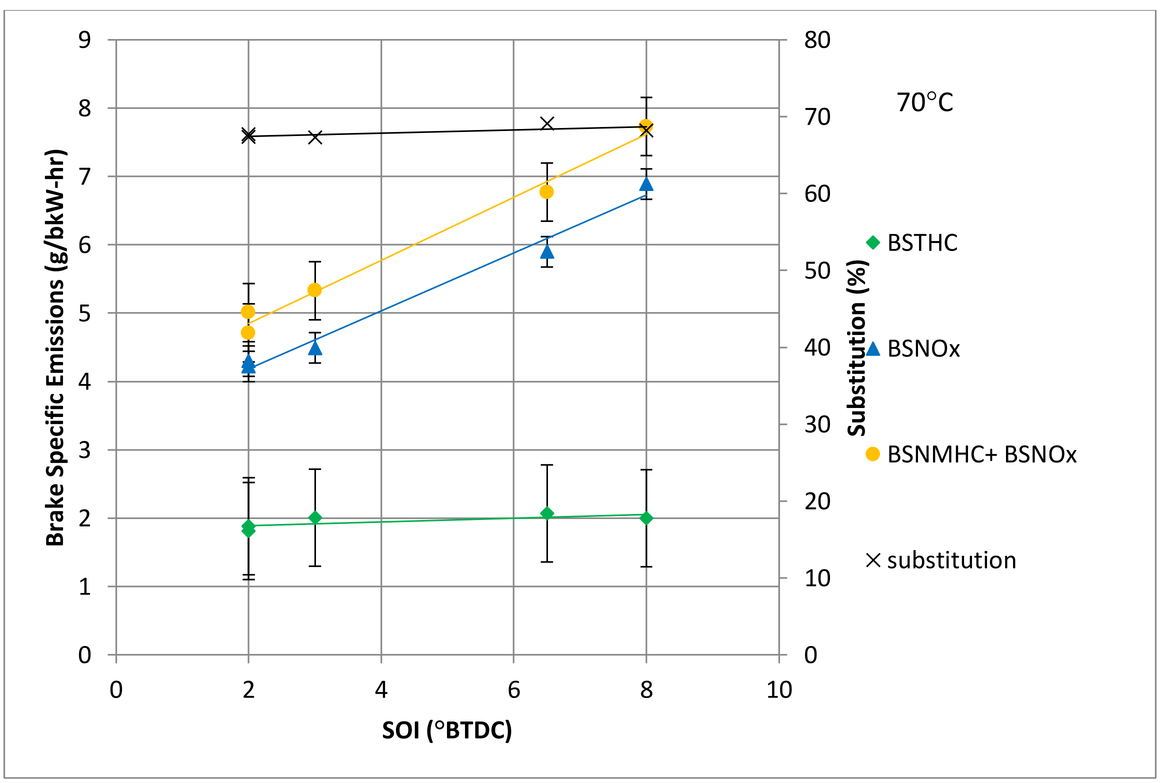

Figure 87: Brake specific emissions at 50% load, 70C, and 68% substitution with different SOI timings.

The equivalence ratio of natural gas is similar for 50% load to 75% load at the baseline conditions, which exhausts low levels of THCs. The combustion pressures are higher at 75% load though resulting in higher bulk temperatures. The 50% load conditions produce lower pressures and bulk temperatures so quenching due to expansive cooling occurs, which was not observed at 75% load. Advancing the SOI raises the peak pressure and increases local flame temperatures promoting NOx formation but the bulk gas temperatures are not elevated enough to completely burn the gaseous fuel.

6.2.4 25% Load

Generally, at low loads conversion of fuel to exhaust products is incomplete. The hypothesis was that elevating the air manifold temperature increases the reaction rates and will promote burning of more of the premixed fuel. Under normal operating conditions the thermostat does not control the air manifold temperature at 25 and 10% loads. There is not enough of a temperature rise due to compression by the turbocharger to increase the temperature above the thermostat set point. The uncontrolled air manifold temperature is around 53^{o}C at 25% load. To test the hypothesis, the chilling loop through the intercooler was adapted to accept hot jacket water from the engine. Pipe tee’s and valves were placed in the loop to stop the flow of the coolant in the chilling loop and allow flow from the jacket water. The chilling loop is typically between 20- 30^{o}C depending on ambient conditions while the jacket water is controlled to 80^{o}C. In-engine thermostats open at 80^{o}C for warming up the coolant at low loads and chilling it once the engine is warm. If the jacket water temperature is below 80^{o}C, the thermostats are closed returning coolant back through the engine, bypassing the chilling loop heat exchanger. The engine was operated at higher loads to ensure the jacket water temperatures were above 80^{o}C and then reduced to the desired set point.

Figure 88 shows the substitution level is increased by approximately 20 percentage points by heating the intake air from 55 to 70°C. The substitution levels are limited by NMHC+ NOx emissions. Figure 89 shows the NMHC+ NOx levels remaining constant but the substitution being increased with temperature. The SOI was also varied while preheating the incoming charge to determine an optimum condition allowing the most natural gas flow. Figure 90 shows the emissions levels while varying the SOI timing. The advanced injection timings are able to reduce THC but at the expense of NOx. Also note the test at 6.5 °btdc, the higher substitution creates a disproportionate change in THC and NOx. This is more evidence supporting THC emissions are directly related to substitution and NOx is primarily controlled by SOI timing.

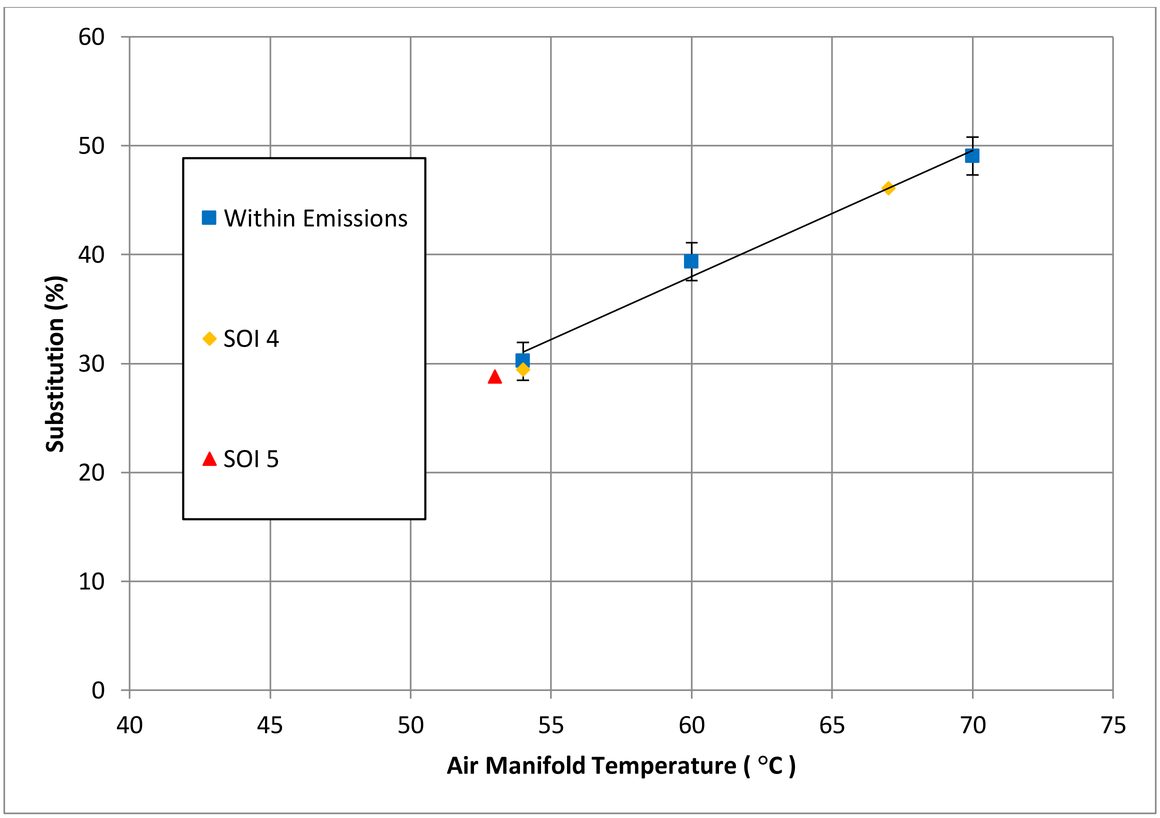

Figure 88: Substitution levels versus air manifold temperature at 25% load.

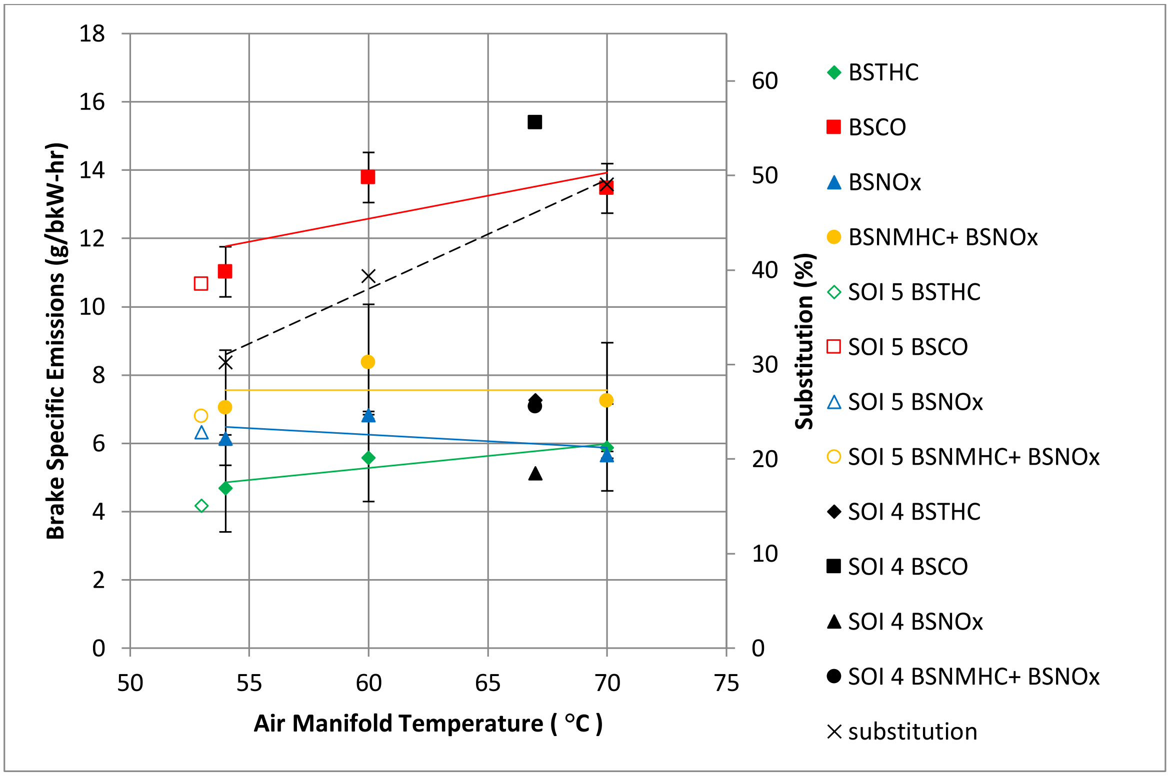

Figure 89: Brake specific emissions levels for dual fuel operation at 25% load with various air manifold temperatures.

This shows the stock injection timing of 5 ^{o}btdc is actually the optimal injection timing for dual fuel combustion at 25% load. Since the THC were unaffected by SOI, at elevated temperatures the SOI was retarded to reduce NOx emissions as much as possible. The late injection timings do not significantly affect NOx or THC emissions. The in-cylinder pressure traces at 25% load with intake preheating are shown in Figure 91. The pressure traces shown for dual fuel operation and diesel only operation are similar. The extra energy in the charge is not enough to provide conditions for flame propagation. The natural gas is likely only burning in the surrounding areas of the diesel jet. The equivalence ratio is around 0.27 at a substitution of 50% when the charge air is at 70^{o}C. This seems to be below the lean propagation limit of natural gas at the cylinder conditions.

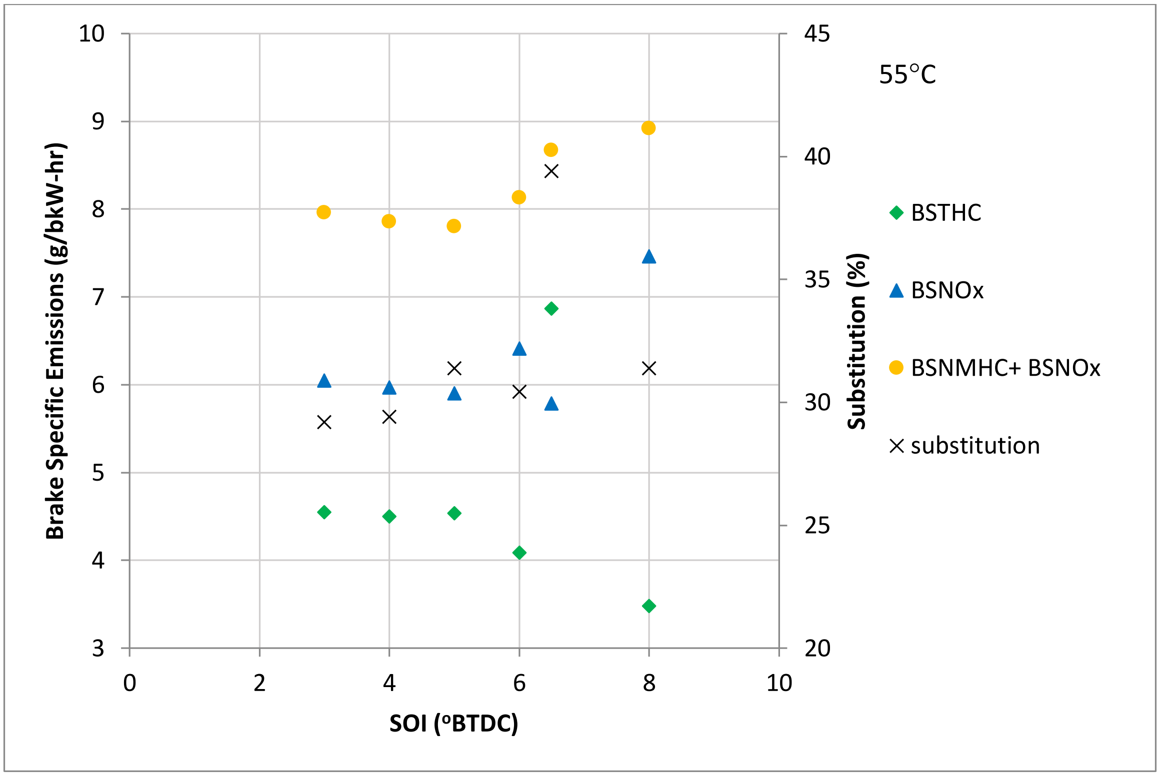

Figure 90: Brake specific emissions at 25% load at 55C and various SOI timings.

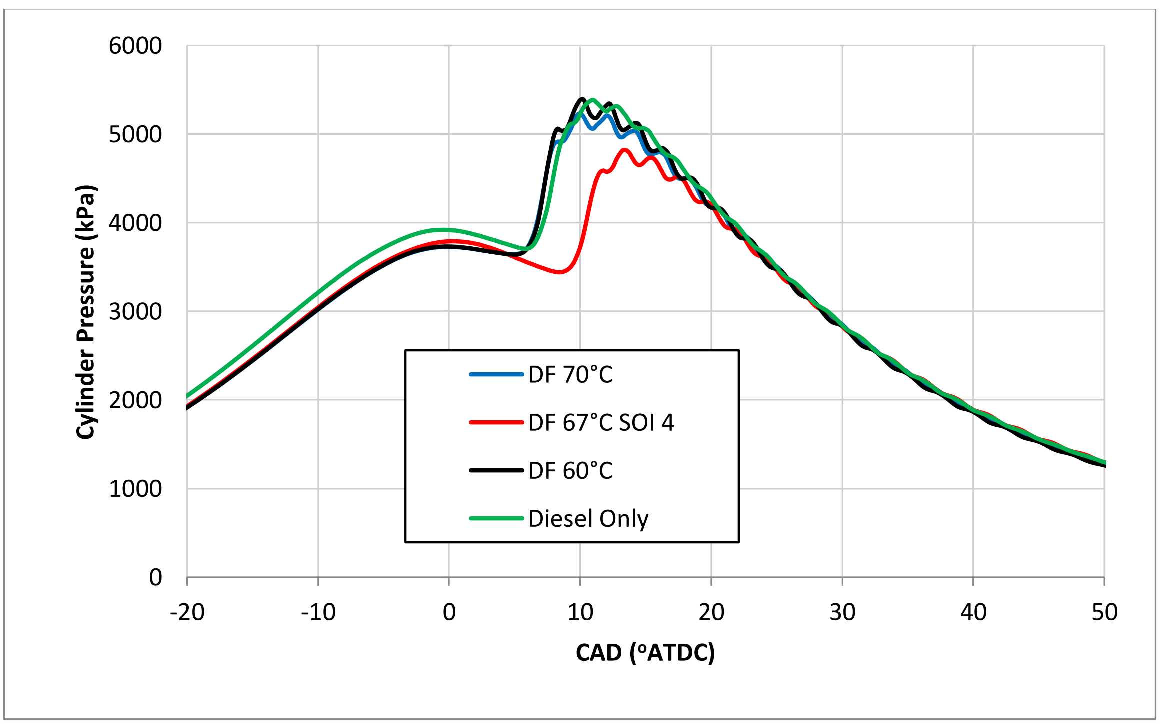

Figure 91: In cylinder pressure traces at 25% load with various air manifold temperatures.

Figure 92 shows the heat release rates at 25% load while utilizing preheating to increase substitution. The natural gas equivalence ratio is low so that the ignition delay is not increased for dual fuel operation as it is at higher loads. The heat release profiles are nearly the same for dual fuel and diesel only. The primary fuel consumption is from diesel fuel at low loads. With higher temperatures increasing the reaction rates of natural gas, the surrounding gaseous fuel region that burns is expanded. This supplies the same amount of energy as is normally released when only diesel is burned. There are two different ways to examine this. The first is there is no flame propagation so only the natural gas surrounding the diesel fuel increases the flame thickness. Increasing the premixed temperature allows for greater flame thicknesses so the natural gas is substituted for diesel. The second way to examine this is by comparing the combustion to the quenching distance. This assumes a flame propagates through the natural gas but quenches at a certain point due to the mixture becoming leaner with distance from the diesel jet.

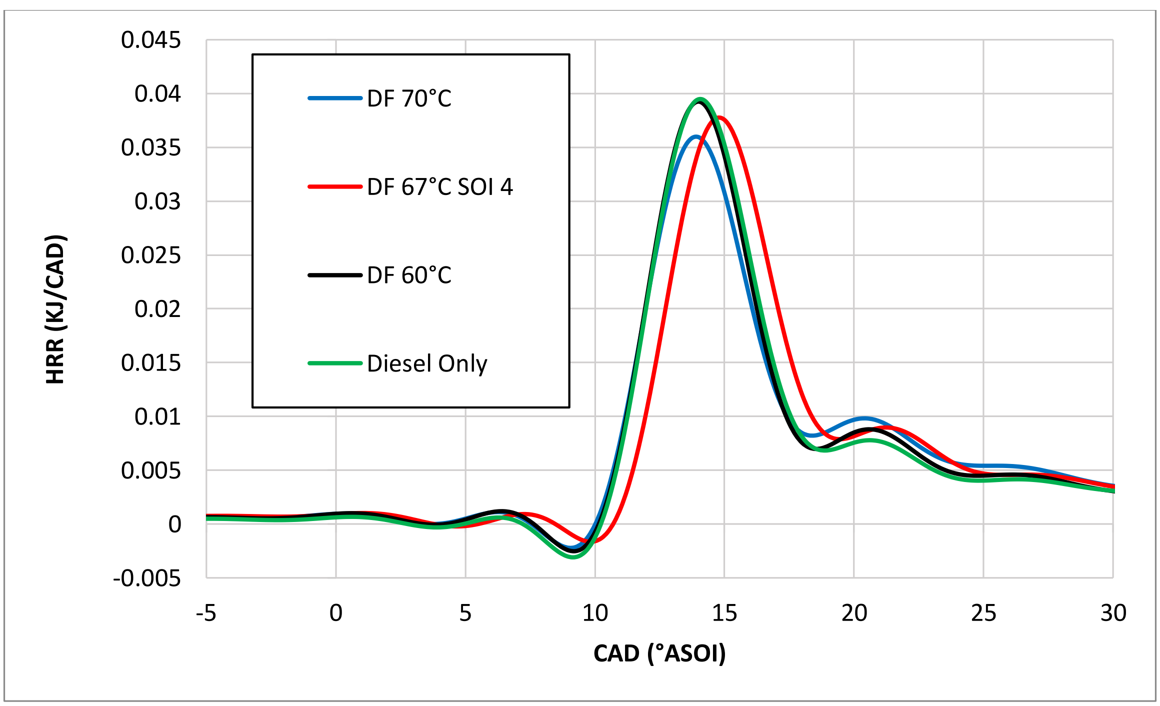

Figure 92: Heat release rates at 25% load with various levels of preheating the intake air.

Both methods of understanding low load dual fuel combustion are similar in respect to the importance of the distance the flame can reach from the diesel jet. Either this is a true diffusion flame and natural gas is being burned as it diffuses to the flame or it is starting to propagate but quenching soon after it starts. Further increases in substitution decrease the diesel fuel injected. The area of the jet decreases so less natural gas is entrained and slow the chemical reaction rates leaving large quantities of natural gas unreacted. Benefits of pre-heating the air intake at 10% load were not able to be realized. The equivalence ratios are too low at 10% load for a relatively small temperature increase to increase the flame thickness or decrease the quenching distance.

6.3 Evaluation of Methods for Extending Substitution Limits

The two main limiting mechanisms to substitution are end gas auto-ignition and NMHC+NOx emissions. To avoid end gas auto-ignition it is desired to slow the combustion process. To reduce hydrocarbons at low loads, however, it is desired to speed up the reaction rates. Without introducing additives to the fuel to adjust the burning characteristics, adjusting the temperature is the primary way for changing the rate of combustion in dual fuel combustion. This was achieved by two methods in this experiment, by adjusting the combustion phasing relative to tdc and directly adjusting the temperatures of the mixture entering the cylinder. Adjusting the combustion phasing by advancing or retarding the SOI showed limited success, primarily at high loads. Figure 93 shows the optimum timing was not able to increase substitution levels at or below 50% load. A small increase in substitution is achieved at full load by delaying knock via retarded injection timing. Retarding the injection timing more is limited by elevated exhaust gas temperatures that may cause damage to exhaust valves or the turbocharger. The greatest increase of substitution due to retarding the SOI is realized at 75%

load. This is primarily a result of the stock ECU adjusting the combustion phasing earlier as substitution is increased.

Adjusting the intake air-fuel mixture proved to have a greater impact on extending the substitution limit than changing the SOI timing. At high loads, slowing the chemical reaction rates by cooling the intake increases the substitution by 10 percentage points over adjusting the SOI only. Significant increases were not achieved by either method at 50% load. A physical limit to substitution may exist around 75% substitution for dual fuel applications that fumigate the gaseous fuel into the air stream. The conditions at moderate loads are near the lean flammability limit of natural gas and attempts to use all of the natural gas negatively affect NOx emissions before benefitting late combustion reactions of hydrocarbons. Preheating the intake air is able to achieve an increase in substitution at light loads where SOI was not able to extend the

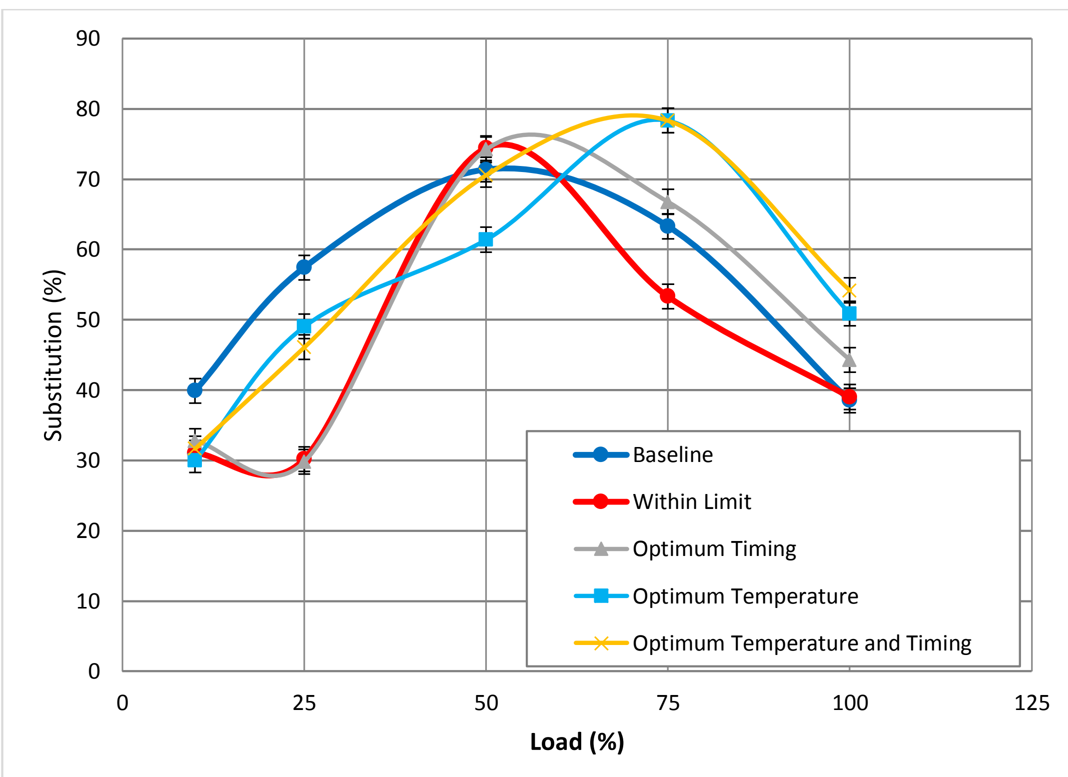

Figure 93: Substitution levels across the load range for various methods of extending substitution limits.

substitution limit. An increase of 15% of substitution was achieved at 25% load preheating the natural gas air mixture. At 25% load, preheating increases the bulk gas temperatures and benefits the natural gas utilization but the local flame temperatures are not affected as much so the NOx formation does not increase as much as at 50% load. Neither of the two methods were able to extend the substitution limit at 10% load. Some dual fuel applications shut off the natural gas flow at low loads because of the poor use of natural gas.

Adjusting the temperature and timing successfully brings the NMHC+ NOx emissions within Tier II limits while extending the substitution. Figure 94 shows the ISO weighted emissions levels for the Tier II regulated emissions for the various testing conditions. The real time emissions calculation for a substitution limit is approximate for NMHC + NOx emissions. This is why the within limit case is actually slightly above the limit. The NMHC + NOx emissions are reduced to within the limit for each method used for extending the substitution and less than the diesel only operation. Adjusting the SOI mainly benefits the high load conditions, which are fairly low THC emissions on a g/bkW-hr basis.

Adjusting the temperature is also able to benefit some of the low loads cases. The low loads are the primary problem for excessive NMHC+ NOx levels so the benefits at 25% load make an appreciable difference in the ISO weighted average emissions level. Particulate matter was not sampled at every test condition because of time constraints so an ISO weighted emissions value cannot be calculated. However, based on the low weighted average of the baseline dual fuel case, all dual fuel cases are expected to be well below the Tier II PM limit.

For each dual fuel test the CO emissions exceed the Tier II limit. Conversion of CO to CO_2 is one of the slowest reactions in the combustion process. The conversion is primarily time dependent. However, even with promoting more burning of natural gas, the hydrocarbons may be oxidized to CO but stop reacting before being converted to CO_2. This is caused by quenched reactions occurring late in the expansion stroke. Most premixed combustion engines have early spark timing so the fuel begins to burn in a shrinking volume. For the dual fuel application much of the burning occurs in the expansion stroke and causes incomplete combustion due to expansive cooling. An oxidation catalyst would be required for the engine to comply with pollutant emissions levels. Previous research has shown high oxidation efficiencies of CO when the temperature is over 350°C [41]. Exhaust temperatures from dual fuel operation are typically higher than diesel only operation so it is expected the oxidation efficiencies for use with the dual fuel engine would be high. A possible field implementation for dual fuel operation could adjust the timing and temperatures to reduce NOx emissions levels as much as possible.

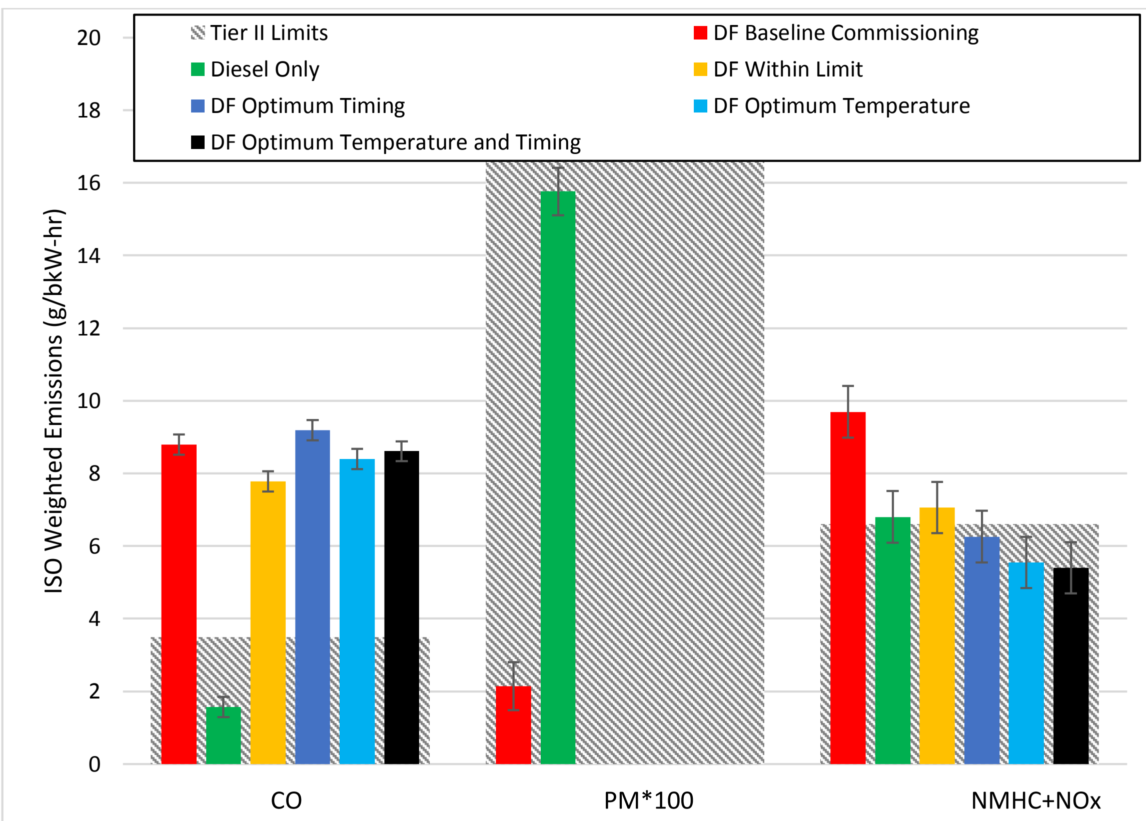

Figure 94: ISO weighted emissions for each method of extending substitution limits.

For this engine, substitution might be able to be pushed further and accept higher THC emissions that are reduced to acceptable levels with a two-way oxidation catalyst since a catalyst is already required to reduce CO. On higher Tier engines, it is possible the NOx can be reduced to acceptable levels without installing an SCR system. At the very least, adjusting the temperature and timing can reduce the diesel exhaust fluid consumed for reduction reactions with NOx.

Both of the methods evaluated for extending substitution limits affected the combustion by changing the temperatures in the cylinder. Combustion duration did not seem to have an effect on whether or not complete combustion was achieved. The SOI was advanced at low loads to try increasing the combustion duration and oxidize more of the THC and CO. This was shown to slightly decrease THCs exhausted but sometimes increase the CO. This showed the more important factor in dual fuel combustion is the temperatures of the reactants. There is a distinct difference in how this is achieved though by the two methods evaluated in this study. Adjusting the SOI affects local flame temperatures more than it affects temperatures of the bulk premixed air fuel mixture. Adjusting the gas mixture temperature entering the intake manifold greatly changes the bulk gas temperature as shown by the discussion of the temperature at the peak motoring pressure with varying intake temperatures. In dual fuel engines, controlling the reaction rates of the premixed gas becomes the most important factor on natural gas utilization. Cooling the intake fuel air mixture is able to slow chemical reactions more than the effects of expansive cooling from retarding injection timing. This allows for greater increases in substitution at high loads by avoiding knock. Then, at low loads, increased chemical reaction rates are needed to be able to extend substitution limits that are caused by emissions levels. Advancing the SOI at 25 and 50% load increases local temperatures from higher peak pressures so NOx formation is increased. The bulk gas temperature is not elevated much from this so the THC emissions are not reduced. Preheating the incoming fuel air mixture does not increase the combustion pressures but increases the temperatures of premixed natural gas and therefore more fuel is burned. This is able to increase the substitution at 25% load where adjusting the SOI was not able to increase substitution. Over the entire load range, adjusting the air manifold temperature was shown to be more effective at extending dual fuel substitution limits.

The diesel flow rate and substitution levels for the tests can be combined to one weighted average by using the same weighting scale used for the weighted emissions based on ISO 8178. This summarizes the effectiveness of operating in dual fuel mode to reduce diesel flow. These results are displayed in Table 7. For this comparison, two categories are considered. First, maximizing the substitution and the engine operates safely. Second, maximizing the substitution while still meeting Tier II emissions limits. For this category, it is assumed the PM emissions under dual fuel operation are within the limits and CO emissions would be reduced by an oxidation catalyst. Thus, the primary criteria for meeting Tier II limits is NOx+ NMHC emissions. Table 7 shows the baseline commissioning achieved a weighted substitution of 60% and reduced the weighted diesel flow to 100 g/bkW-hr. Adjusting the air manifold temperature and SOI timing is able to increase substitution levels above the baseline levels at 100 and 75% load. Using the substitution set points and the baseline commissioning substitution levels for 10 and 25% load, the weighted average substitution is increased to 66%. An increase in substitution at high loads is a larger decrease in diesel fuel consumption compared to low load cases so the average weighted diesel flow is significantly reduced to 79 g/bkW-hr. The second category focuses on reducing emissions to acceptable levels while utilizing the most natural gas as possible.

Table 7: Summary of Reduced Diesel Flow and Substitution to maximize substitution and to maximize substitution while meeting emissions.

	Test Case	ISO weighted Diesel Flow (g/bkW-hr)	ISO weighted Substitution (%)
	Diesel Only	274	0
Maximizing Substitution	DF Baseline Commissioning	100	60
	Maximum	79	66
Within Tier II Limit	DF Reduced Substitution from Baseline	107	53
	DF Optimum Temperature and Timing	89	60

Without making any changes to the operating parameters except the substitution level, the substitution levels have to be reduced to 53% to comply with Tier II limits. This was achieved primarily by reducing the substitution at 10 and 25% load as the gaseous fuel is not effectively used under the cylinder conditions. Adjusting the timing and air manifold temperature is able to increase the average substitution level to the same average substitution level as the baseline commissioning but also meet Tier II emissions. Significant substitution gains are achieved at high loads, which provide a greater reduction in diesel fuel flow. This is why the Optimum temperature and timing case has a lower ISO weighted diesel flow but the same ISO weighted substitution as the baseline commissioning. In either category, the average substitution can be increased by 6-7% by adjusting the air manifold temperature and SOI timing appropriately.

7. Conclusion

In this work John Deere 6068 Tier II diesel engine was commissioned as a generator set to study mechanisms limiting diesel substitution for natural gas-diesel dual fuel operation. The diesel engine was converted to a dual fuel engine by installing an aftermarket dual fuel retrofit kit from Eden Innovations. The dual fuel kit was originally commissioned Eden based on field experience. After programming the Eden PLC for dual fuel control for baseline commissioning the combustion characteristics were examined with in-cylinder pressure sensors and exhaust gas pollutant emission measurements. Then testing was performed to clearly identify the mechanisms prohibiting further natural gas utilization. With the mechanisms identified, two methods were examined for extending the substitution limits. The two methods examined were adjusting SOI timing and air manifold temperature. Emissions produced during dual fuel operation with baseline commissioning exceeded Tier II emissions limits. Substitution levels were reduced compared to the baseline to be able to comply with Tier II limits. Adjusting the air manifold temperature and SOI enabled increased substitution levels relative to baseline commissioning while meeting Tier II emissions limits. Based on the test data the following conclusions are made:

1. Operating as a dual fuel engine significantly increases THC and CO emissions compared to diesel only operation. The baseline commissioning increased the THC emissions by 16 to 80 times that of diesel operation. CO emissions are increased by 2-5 times that of diesel.

2. Particulate matter emissions are reduced from dual fuel operation compared to diesel only operation. PM emissions from diesel are 3-8 times greater than from dual fuel operation.

3. Dual fuel operation results in higher peak pressures at each load point. Adjustments to the SOI and temperature can reduce the peak pressure to be lower than diesel under some conditions.

4. Substitution at high loads are limited by knock. At 75% load the knock is not audible but observed on the pressure traces.

5. The engine is able to operate at moderate loads with high levels of substitution without knocking or showing signs of combustion instability. The substitution limit is an emissions limit.

6. At light loads the substitution levels can be increased to where combustion becomes inconsistent and governor instability occurs. However, this limit is well beyond the emissions limit.

7. Retarding the injection timing delays the onset of knock at high engine loads. This was observed to be the most prominent at 75% load.

8. Adjusting the injection timing affects NOx emissions more THC emissions; however, varying substitution impacts THC emissions more than NOx emissions.

9. Lowering the air manifold temperature allows greater substitution levels before the onset of engine knock at high loads; air manifold adjustment is more effective than retarding the injection timing. At 100% load, the rate of substitution increase is 0.44%/^{o}C decrease and 4%/^{o}SOI retard. The air manifold temperature has a larger adjustment range allowing it to be more effective.

10. Preheating the intake fuel air mixture is able to increase the substitution level at 25% load without NMHC+NOx emissions becoming excessive. The substitution increases by about 1.25%/^{o}C increase. Adjusting SOI timing is not able to increase substitution levels while maintaining suitable emissions levels.

The combustion reaction rates of dual fuel combustion are important for avoiding knock at high loads and complete combustion at low loads. The effect of swirl levels on dual fuel combustion has yet to be shown and can significantly change reaction rates. Increasing the fuel air mixture temperature is an effective technique to increase substitution at low loads. More work is needed to develop an intake air manifold temperature control scheme suitable for field use. Engine loads can be changed quickly in field operation and a fast response temperature control system is needed. Electric heaters could be implemented to improve control. It is recommended that different preheating techniques be evaluated. Additionally, higher temperatures should be tested to explore the upper limit of allowable preheating before pre-ignition or end gas auto-ignition of the gaseous fuel occurs.

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9. Appendix A: Relevant Figures

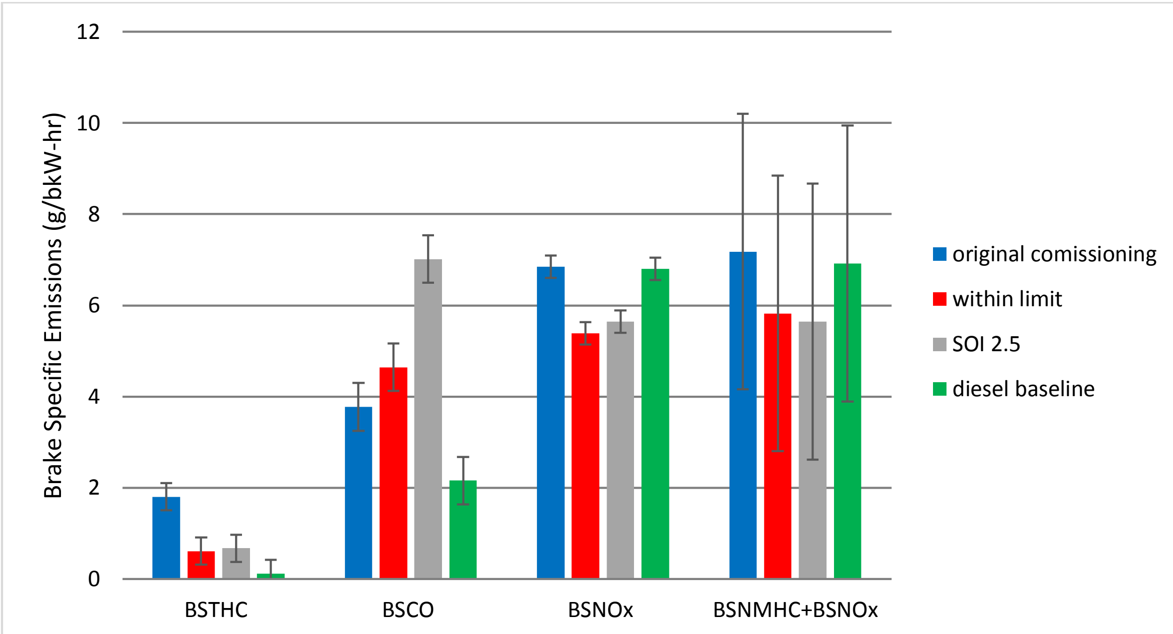

Figure 95: 100% load emissions with various conditions

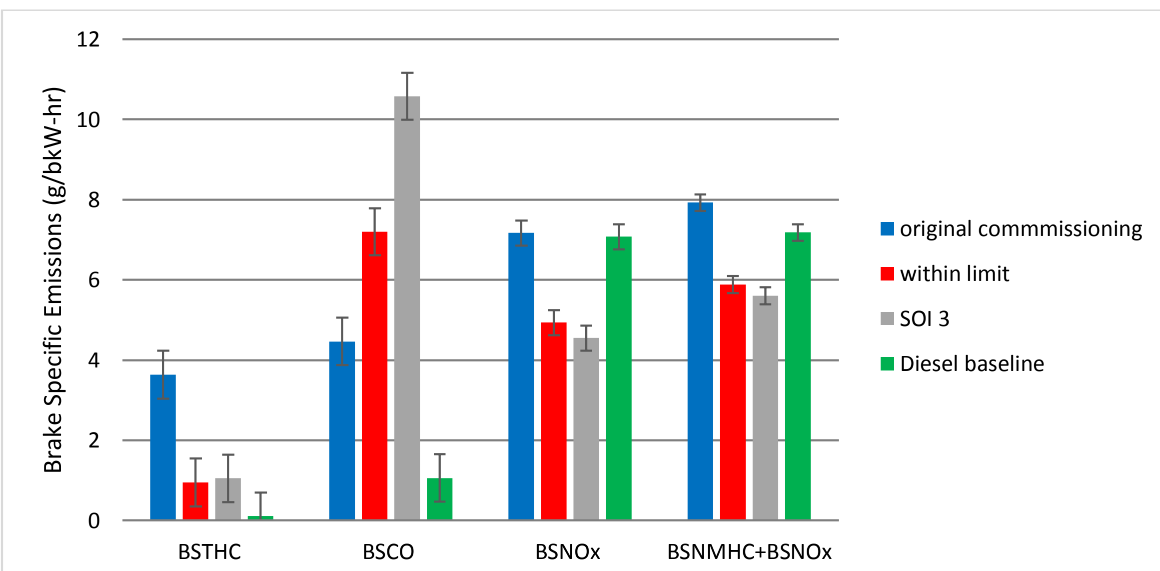

Figure 96: brake specific emissions at 75% load with various conditions

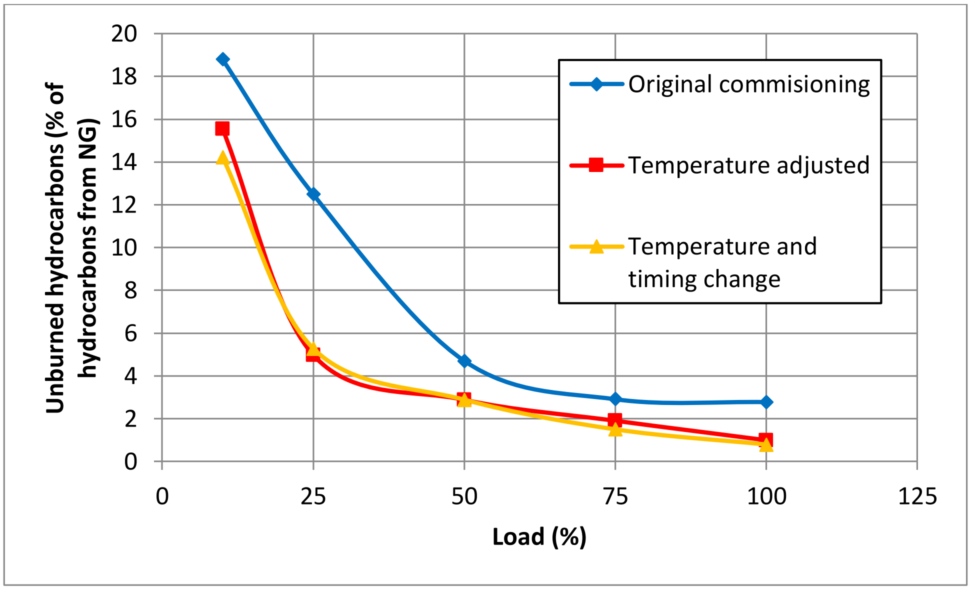

Figure 97: Percent of gaseous fuel unconverted for dual fuel testing at various conditions

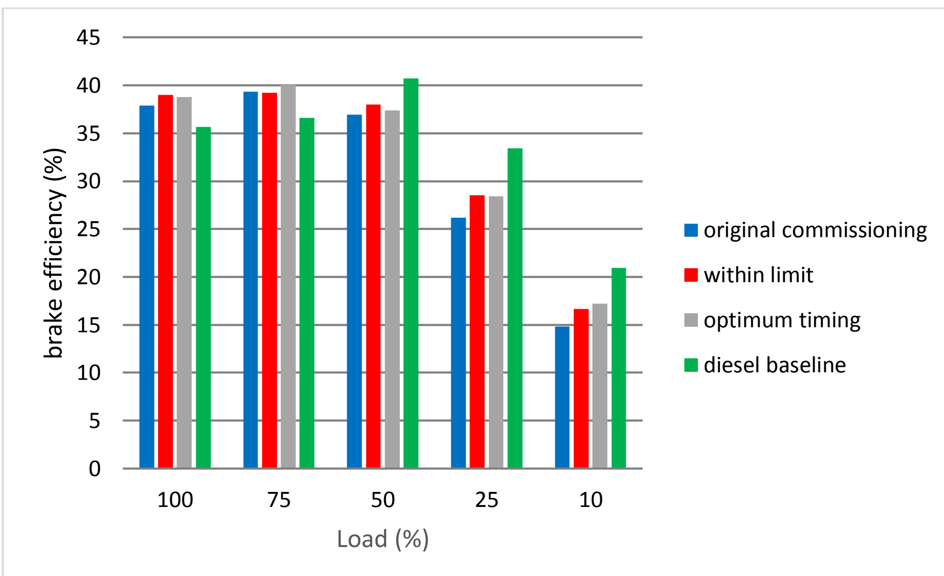

Figure 98: Brake Efficiency across the load range for various testing conditions

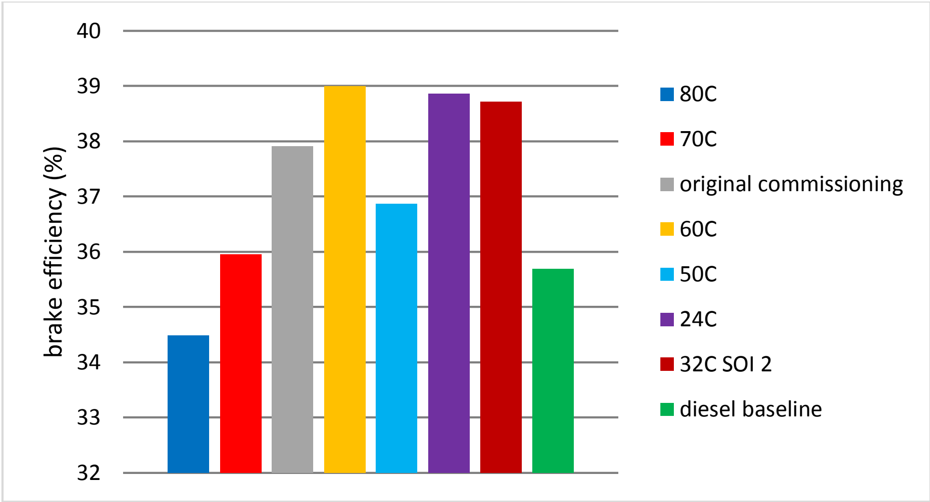

Figure 99: Brake Efficiency at 100% load at various air manifold temperatures

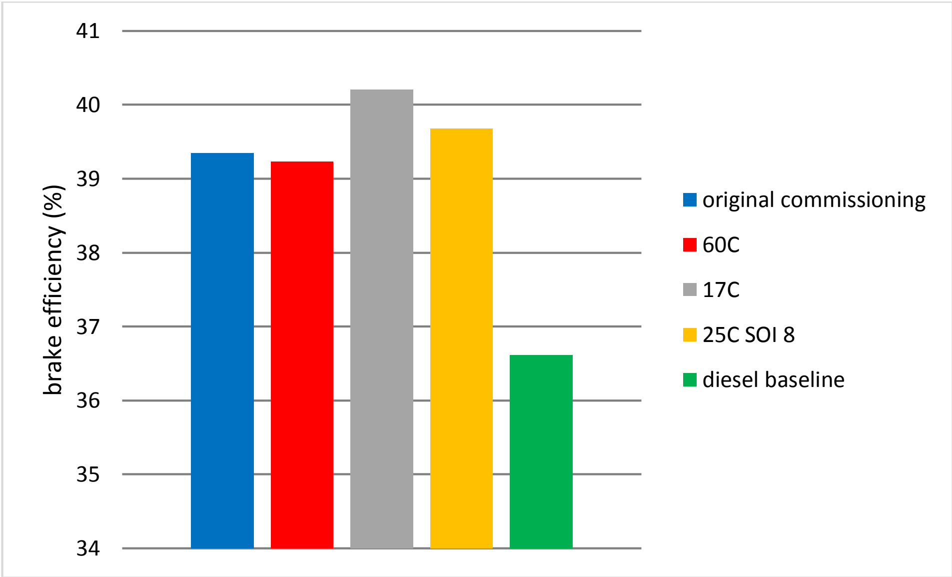

Figure 100: Brake Efficiency at 75% load with varying air manifold temperatures

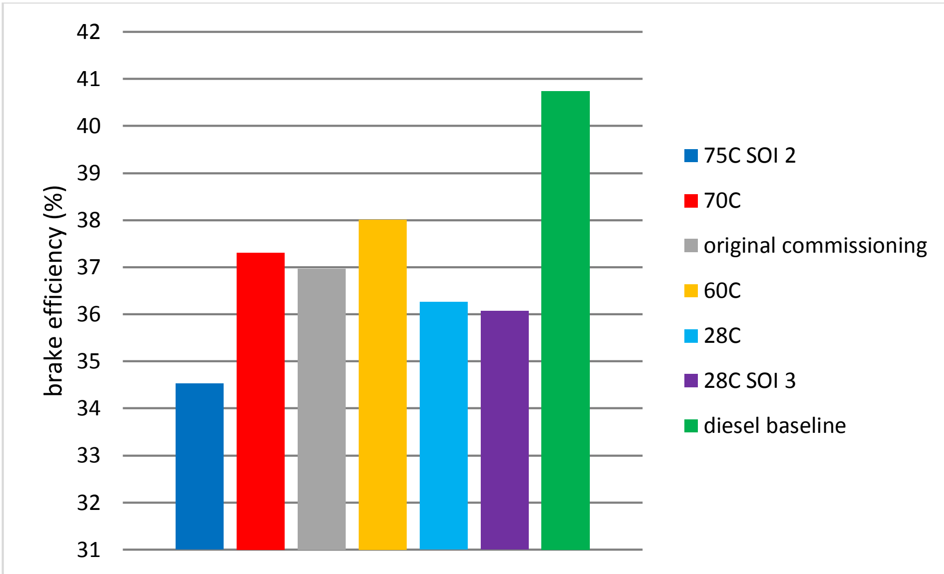

Figure 101: Brake Efficiency at 50% load with various air manifold temperatures

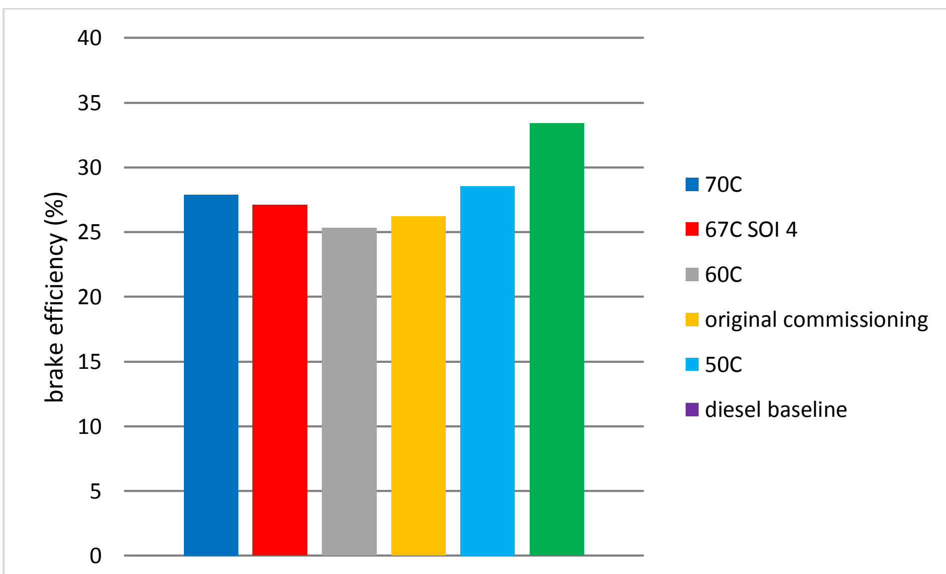

Figure 102: Brake Efficiency at 25% load with various air manifold temperatures

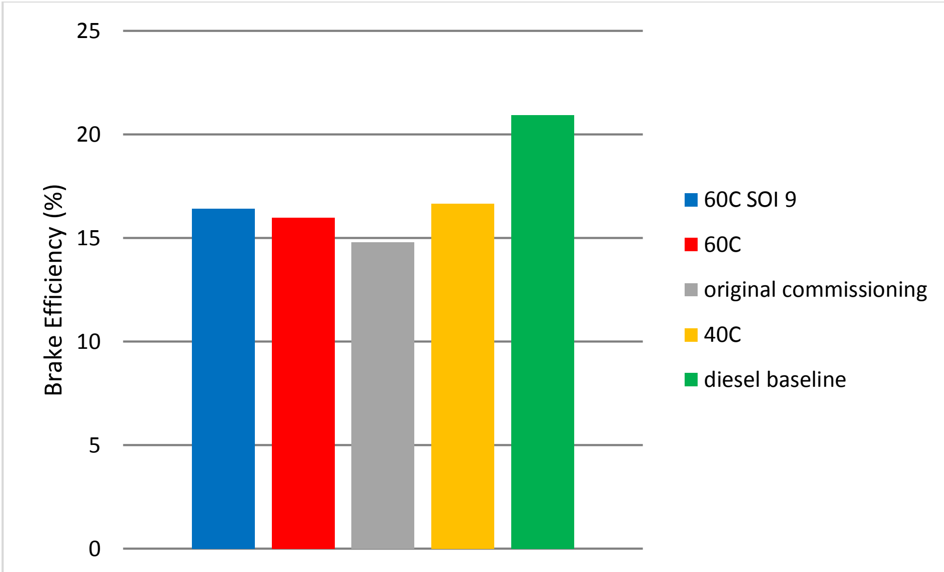

Figure 103: Brake Efficiency at 10% load with various air manifold temperatures

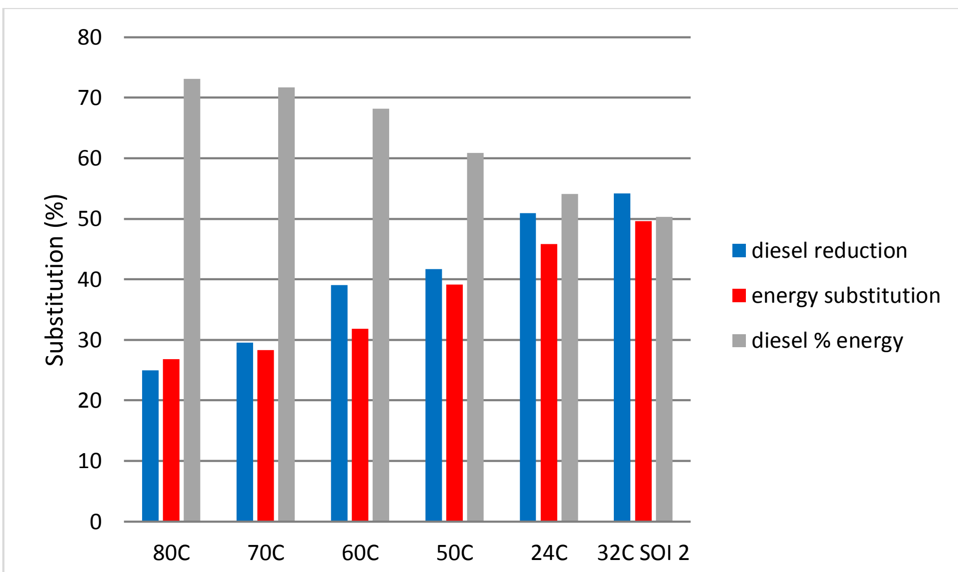

Figure 104: Substitution levels at 100% load based on diesel reduction and energy supplied

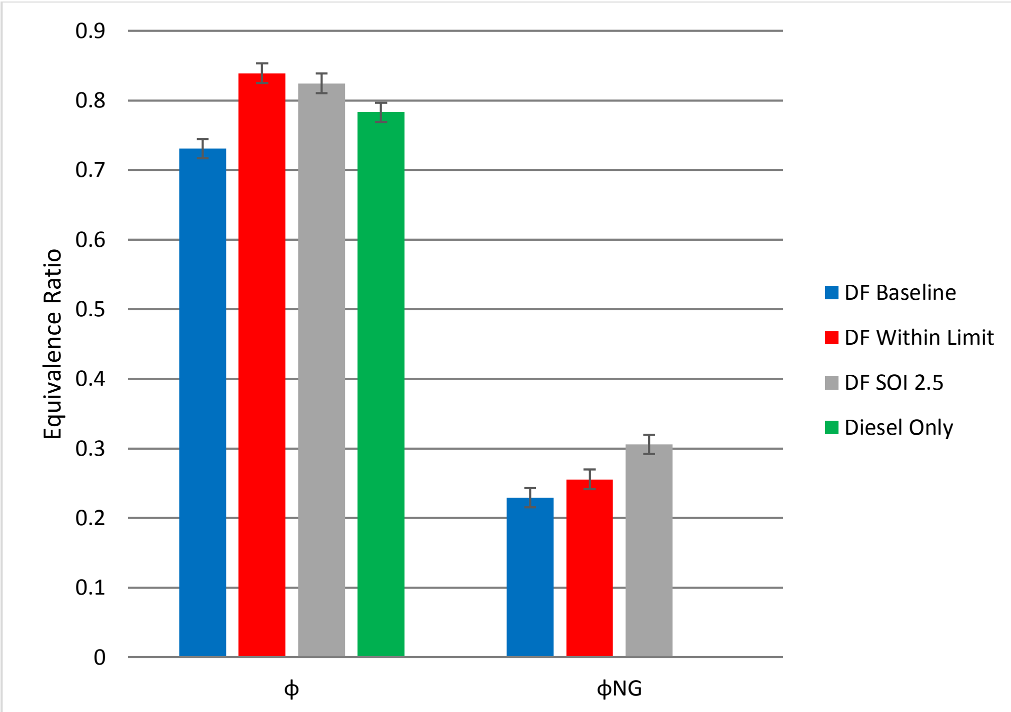

Figure 105: Equivalence ratios associated with various tests at 100% load

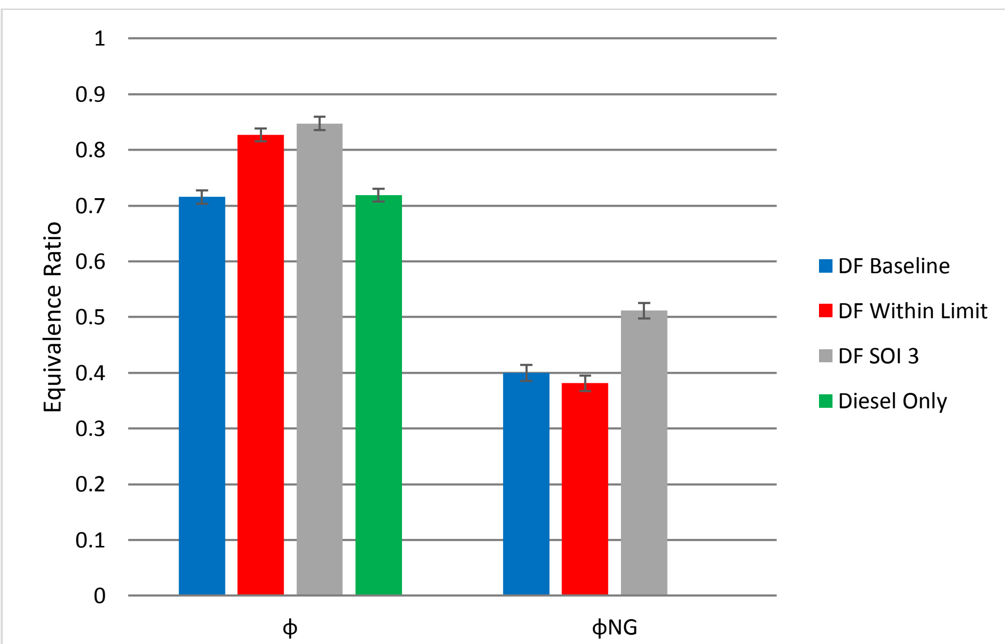

Figure 106: Equivalence ratio associated with various tests at 75% load

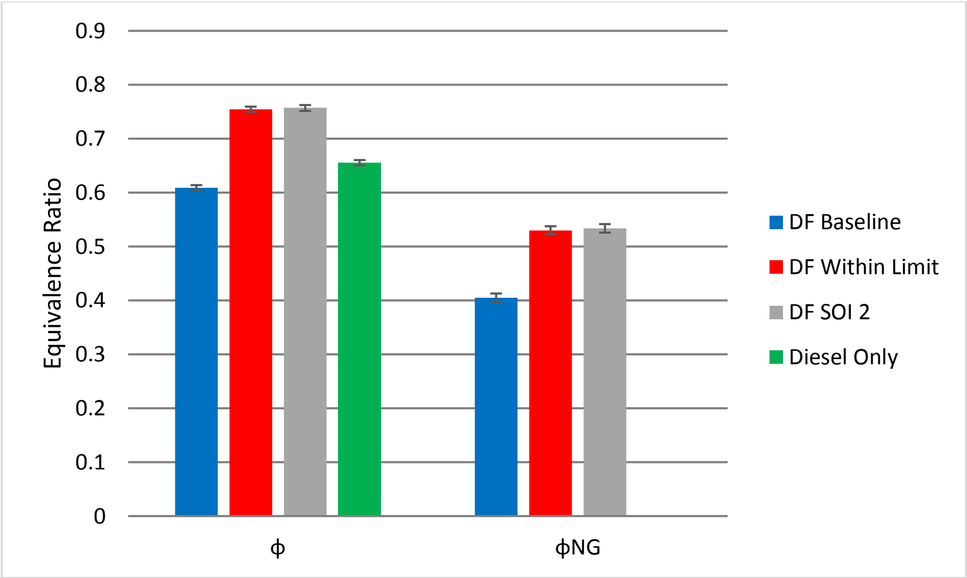

Figure 107: Equivalence ratios associated with various tests at 50% load

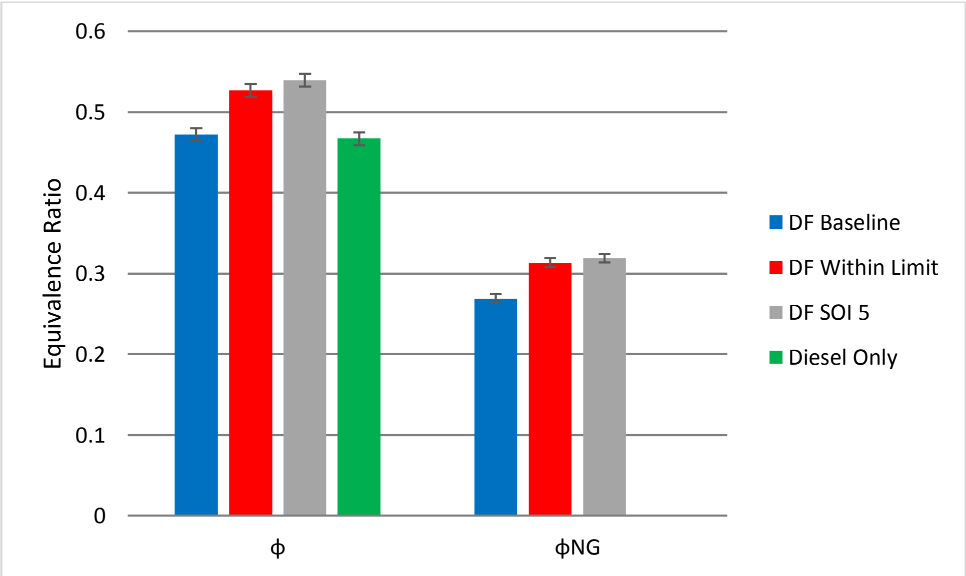

Figure 108: Equivalence ratios associated with various tests at 25% load

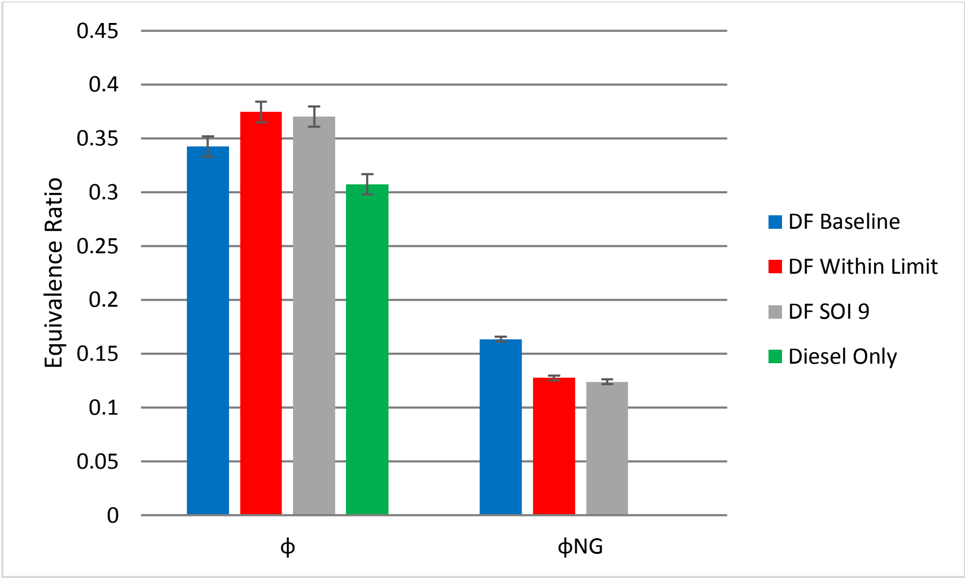

Figure 109: Equivalence ratios associated with various tests at 10% load

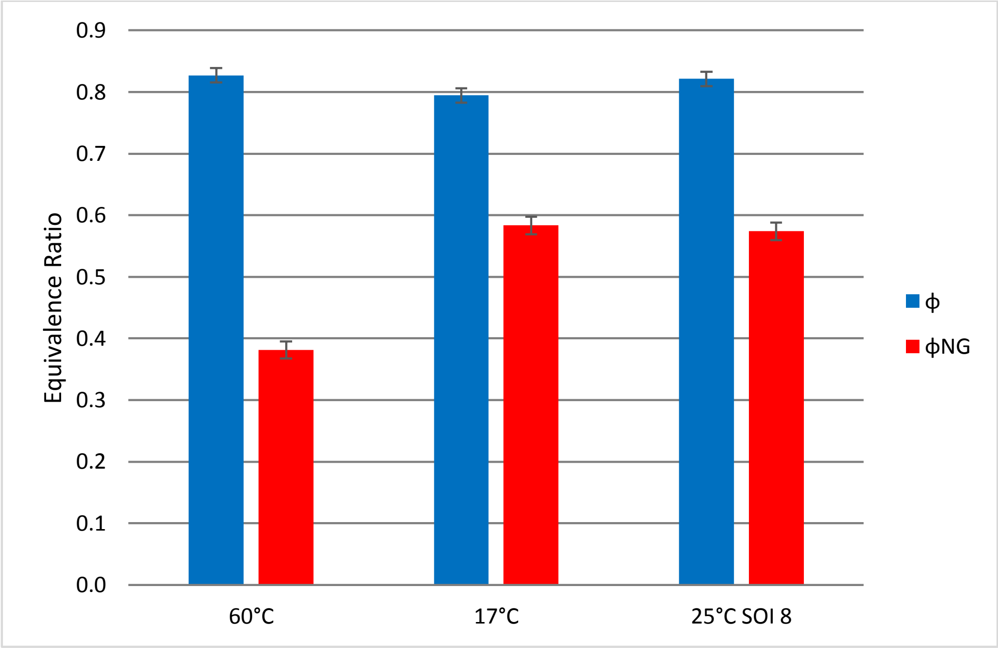

Figure 110: Equivalence ratios at 75% load while adjusting air manifold temperature and SOI timing

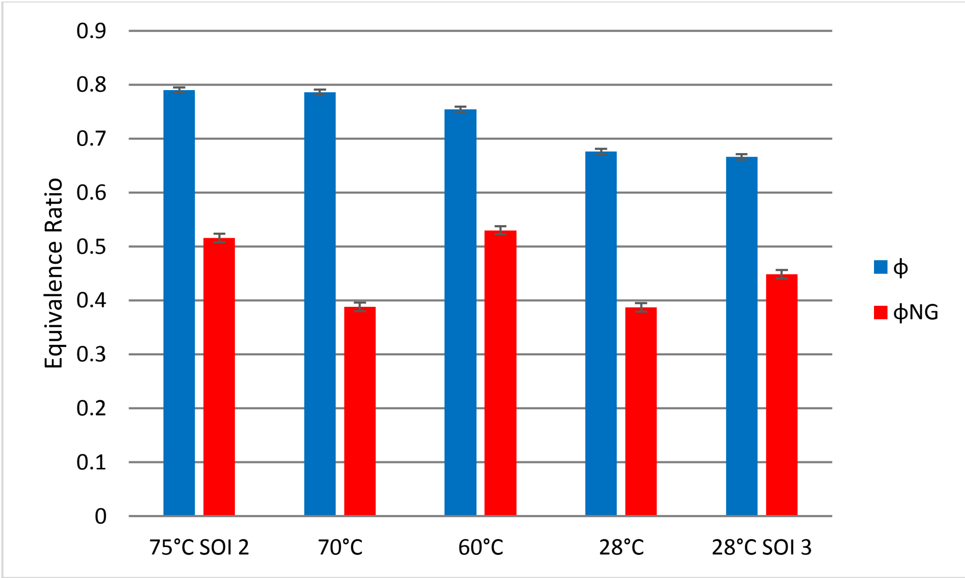

Figure 111: Equivalence ratios at 50% load while adjusting air manifold temperature and SOI timing

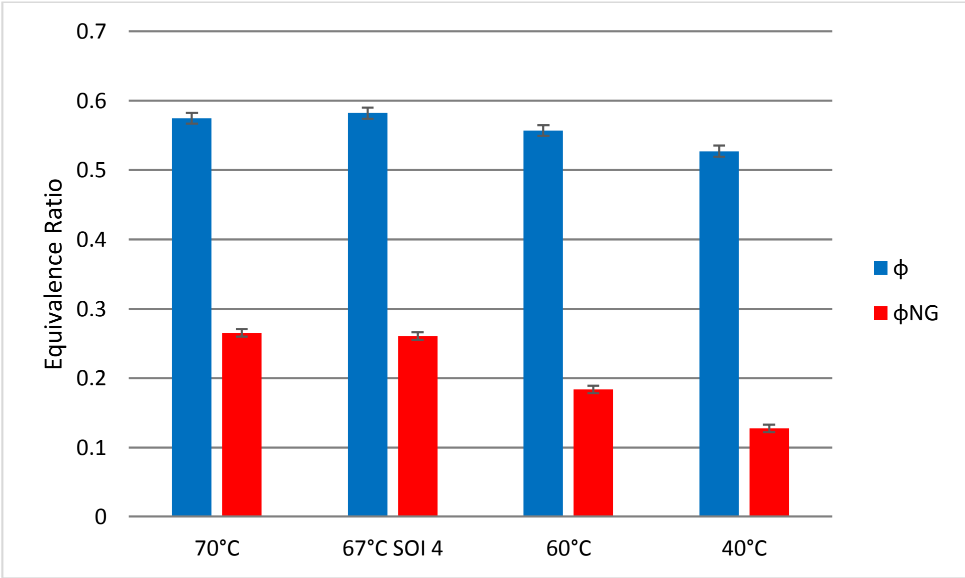

Figure 112: Equivalence ratios at 25% load while adjusting air manifold temperature and SOI timing

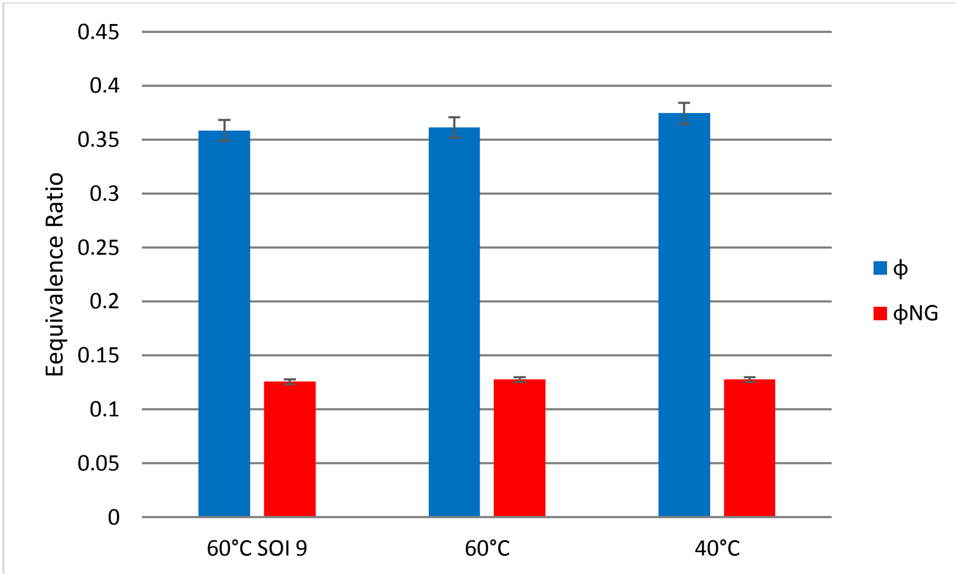

Figure 113: Equivalence ratios at 10% load while adjusting air manifold temperature and SOI timing

10. List of Abbreviations

AMT- air manifold temperature

BMEP- brake mean effective pressure

BS- brake specific

BSFC- brake specific fuel consumption

CAD- crank angle degrees

CO_2 – carbon dioxide

CO-carbon monoxide

COV- coefficient of variation

DF- dual fuel

HRR- heat release rate

IMEP- indicated mean effective pressure

MFB-mass fraction burned

NG- natural gas

NMHC- Non-methane hydrocarbons

NOx- oxides of nitrogen

^{o} atdc- degrees after top dead center

^{o} btdc- degrees before top dead center

^{o}ASOI- degrees after start of injection

OEM- original equipment manufacturer

PM- particulate matter

SOI- Start of injection

THC- total hydrocarbons