

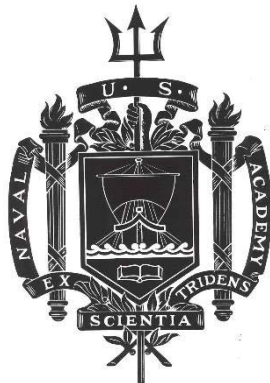
A TRIDENT SCHOLAR PROJECT REPORT

NO. 474

**A Partially Premixed Combustion Application for
Power Improvement in Military Diesel Engines**

by

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UNITED STATES NAVAL ACADEMY
ANNAPOLIS, MARYLAND

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14. ABSTRACT Due to increasing weight in military platforms, engine power needs to be increased in order to maintain performance. Diesel engine power is limited by soot formation, which is an indicator of incomplete fuel combustion due to lack of oxygen and poor mixing of the fuel and air. Once the soot limit is reached in a conventional diesel engine, further fuel increases will not result in more engine power since both the time for combustion (i.e. engine RPM) and oxygen are limited. An alternative approach is needed to both deliver and convert fuel energy in a diesel engine's combustion chamber. Partially Premixed Combustion (PPC) allows for better mixing of the air and fuel in the combustion chamber, leading to lower combustion temperatures and higher flame speed (shorter burn duration) as compared to conventional diesel combustion. PPC delivers additional fuel to the combustion chamber in internal combustion engines through the air intake system in addition to the in-cylinder (i.e. combustion chamber) injection event, allowing for increased power opportunities. This project will improve the specific power gains in three distinct engines by retrofitting each with port injection to achieve PPC. This project fundamentally characterizes achievable power gains in a flexible Waukesha Diesel CFR research engine that allows for the manipulation of combustion phasing-timing, compression ratio (CR), and maximum baseline load. The conditions to achieve optimal combustion phasing will be determined. Fuels evaluated include conventional Navy JP-5 and less reactive, non-JP-5 fuels via port injection (potentially leading to increased pre-mixing with further power gains). In essence, this study sought to explore whether or not a two-fuel PPC approach might be worth the additional fuel complexity when compared to a conventional diesel approach or a single-fuel PPC approach, based on power improvements from the high load extension of the exhaust sooting limit. Based on these results, PPC was then applied to a small Navy diesel generator, and Marine Corps special operations Humvee engine in order to quantify actual practical power gains, using both single and dual fuel approaches. In the Waukesha CFR engine, it was seen that power levels were then able to increase from -2 to 27% (at CR 21.5) over conventional diesel combustion without a soot opacity penalty. In the Yanmar L100V6 engine-generator, power levels of 9.3 kW to 11.3 kW were achieved compared to 8.5 kW at conventional operation without a soot penalty. In the Humvee engine, power improvements of 7% and 8% were shown with JP-5 and iso-octane. Early heat release behavior was seen with both JP-5 and iso-octane, leading to longer burn durations and less soot-reduction benefit than expected.					
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**A PARTIALLY PREMIXED COMBUSTION APPLICATION FOR
POWER IMPROVEMENT IN MILITARY DIESEL ENGINES**

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USNA-1531-2

Abstract

Due to increasing weight in military platforms, engine power needs to be increased in order to maintain performance. Diesel engine power is limited by soot formation, which is an indicator of incomplete fuel combustion due to lack of oxygen and poor mixing of the fuel and air. Once the soot limit is reached in a conventional diesel engine, further fuel increases will not result in more engine power since both the time for combustion (i.e. engine RPM) and oxygen are limited. An alternative approach is needed to both deliver and convert fuel energy in a diesel engine's combustion chamber.

Partially Premixed Combustion (PPC) allows for better mixing of the air and fuel in the combustion chamber, leading to lower combustion temperatures and higher flame speed (shorter burn duration) as compared to conventional diesel combustion. PPC delivers additional fuel to the combustion chamber in internal combustion engines through the air intake system in addition to the in-cylinder (i.e. combustion chamber) injection event, allowing for increased power opportunities. Similar to the four-stroke Diesel cycle (conventional compression ignition, CI), PPC port injects fuel during the intake stroke *and* at the top of the compression stroke. With a PPC fueling approach, by the end of the intake stroke, some fuel (a minority fraction up to one-half of the total injection amount) and air have been mixed in the engine's cylinder. Then a fuel injection event at the top of the compression stroke auto ignites, starting the overall combustion event, which then forces the piston down for the power stroke. The premixing improves the combustion stoichiometry, but this small amount of fuel is not enough to cause early ignition, so that the fuel ignites during the main injection event. Thus, the main in-cylinder injection event controls combustion timing. The lower temperature and improved flame region (air-to-fuel ratios) lead to lower soot and NO_x (oxides of nitrogen) in the exhaust in comparison to standard diesel combustion.

This project will improve the specific power gains in three distinct engines by retrofitting each with port injection to achieve PPC. This project fundamentally characterizes achievable power gains in a flexible Waukesha Diesel CFR research engine that allows for the manipulation of combustion phasing-timing, compression ratio (CR), and maximum baseline load. The conditions to achieve optimal combustion phasing will be determined. Fuels evaluated include conventional Navy JP-5 and less reactive, non-JP-5 fuels via port injection (potentially leading to increased pre-mixing with further power gains). In essence, this study sought to explore whether or not a two-fuel PPC approach might be worth the additional fuel complexity when compared to a conventional diesel approach or a single-fuel PPC approach, based on power improvements from the high load extension of the exhaust sooting limit. Based on these results, PPC was then applied to a small Navy diesel generator, and Marine Corps special operations Humvee engine in order to quantify actual practical power gains, using both single and dual fuel approaches.

In the Waukesha CFR engine, it was seen that power levels were then able to increase from -2 to 27% (at CR 21.5) over conventional diesel combustion without a soot opacity penalty. In the Yanmar L100V6 engine-generator, power levels of 9.3 kW to 11.3 kW were achieved compared to 8.5 kW at conventional operation without a soot penalty. In the Humvee engine, power improvements of 7% and 8% were shown with JP-5 and iso-octane. Early heat release behavior was seen with both JP-5 and iso-octane, leading to longer burn durations and less soot-reduction benefit than expected.

Keywords

“Partially Premixed Combustion”

“port fuel injection”

“Navy JP-5”

“Waukesha CFR”

“Yanmar L100”

“AM General HMMWV Engine”

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Introduction

Military platforms continue to increase in weight due to technological and personnel safety requirements. For example, over the last decade, the High Mobility Multipurpose Wheeled Vehicle (HMMWV, “Humvee”) has increased in weight by over 700 lbs. [1]. The same is true in the commercial sector, with most model-year vehicles gaining approximately 500 lbs. per decade [1]. In order to maintain mobility and performance, engine specific power must be increased. However, diesel engine power is limited by soot formation due to poor mixing of the air and fuel at high engine power. More fuel cannot be delivered and mixed in the combustion chamber since the time of the piston stroke is very small at high RPM and oxygen is limited at high engine loads. Power could be added by installing a larger, more powerful engine, but this is expensive and would only continue to add weight. A better solution, the solution pursued in this study, is to retrofit existing engines for increased power.

An advanced combustion mode called Partially Premixed Combustion (PPC) delivers a portion of the fuel to the cylinder via the engine air intake system, leading to the opportunity for increased specific power. This can be accomplished with a relatively straightforward approach with the application of port fuel injection. This project characterizes power gains in a research engine due to the additional delivery of fuel to the combustion chamber through a Partially Premixed Combustion approach. Based on these initial analysis and results, PPC was then

applied to representative diesel engines in order to quantify actual attainable specific power gains.

There are many terms and abbreviations in this report unique to the field of engine combustion. These terms are defined within the report itself, but also in a glossary at the end of the report.

Background

Both diesel and gasoline reciprocating internal combustion engines convert fuel into energy through a series of combustion events. The major difference between diesel and gasoline is the way these combustion events develop. In a gasoline engine, fuel is premixed with air, compressed by a piston, and ignited by a high voltage electrical arc from the spark plug. In a conventional diesel engine, only air is compressed first, and then fuel is injected directly into the combustion chamber (direct injection) [1]. It is the heat of the compressed air that ‘auto ignites’ the fuel in a diesel engine.

Most Navy and commercial diesel engines use a four-stroke combustion cycle. The four strokes are:

1. **Intake stroke:** The intake valve opens up, drawing in air as the piston moves down.
2. **Compression stroke:** The piston moves back up and compresses the air.
3. **Combustion stroke:** As the piston reaches the top of its stroke, fuel is injected and ignites, forcing the piston back down.
4. **Exhaust stroke:** The piston moves back to the top, pushing the exhaust created from combustion out the exhaust valve [1].

When the piston is at its uppermost point, it is said to be at Top Dead Center (TDC), when it is at its lowermost point, it is at Bottom Dead Center (BDC). For reciprocating engines, instead of expressing the position of the piston with respect to time, it is expressed in terms of Crank Angle Degrees (CAD) of the engine’s crankshaft, which connects to the piston via the connecting rod and converts the liner motion of the piston to rotational motion. The four-stroke cycle is broken into two 360-degree full rotations with each of the four strokes taking one-half of an engine crankshaft revolution. The position of the piston is then referenced from its position away from TDC of the compression stroke. Thus, a piston’s position is said to be a certain number of CAD before top dead center (deg. BTC) or after top dead center (deg. ATC).

Conventional Diesel Cycle

Current Compression Ignition (CI) engines, seen in Figure 1, use diesel (or jet) fuel which auto ignites very soon after it is injected at the top of the compression stroke. At this early point in the combustion event, very little fuel has had a chance to mix with oxygen in the cylinder. This lack of mixing leads to large fuel droplets and highly fuel rich regions in the combustion chamber, leading to high combustion temperatures with correspondingly slow flame speed (long burn duration as compared to premixed gasoline combustion). The high temperature and stoichiometric imbalance (fuel rich) leads to high soot and NO_x (oxides of nitrogen) in the exhaust.

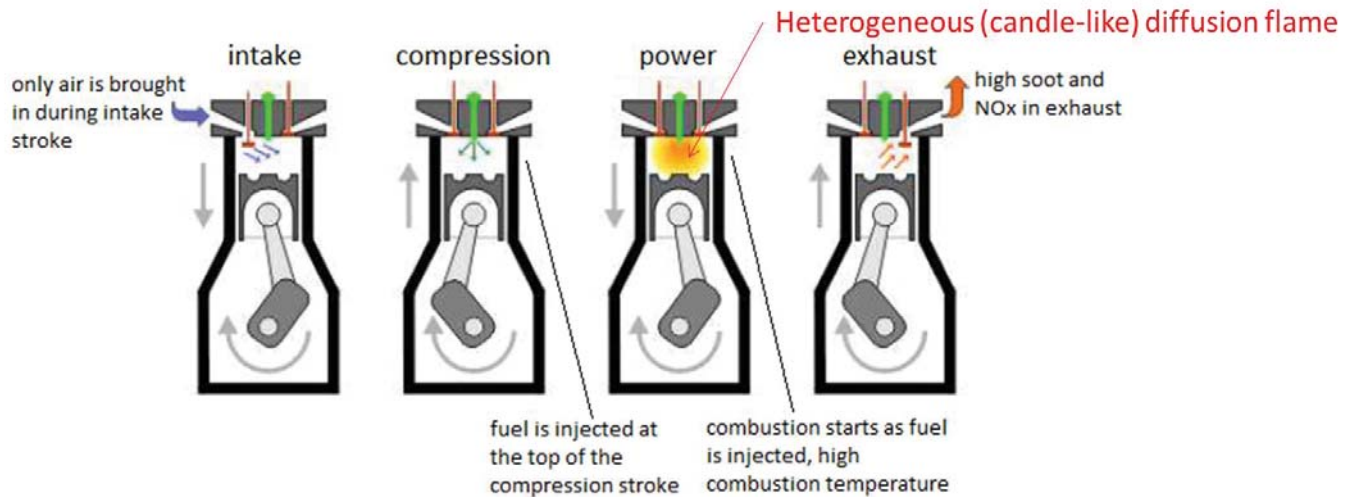


Figure 1. Conventional Compression Ignition (CI) cycle [2]

Homogeneous Charge Compression Ignition

Homogeneous Charge Compression Ignition (HCCI) is an alternative engine combustion concept that has received much research attention over the past decade. It is currently only in the research phase due to challenges discussed below. As seen in Figure 2, HCCI functions similarly to the four-stroke Diesel cycle. However, instead of injecting fuel at the top of the compression stroke, fuel is injected during the air intake stroke. A port injection system or carburetor system injects fuel just prior to or during the intake stroke (outside the cylinder). All of the fuel to be burned enters the cylinder during the intake stroke. By the end of the intake stroke, the fuel and air have been fully introduced and well mixed in the cylinder's combustion chamber. The piston begins to move back up during the compression stroke, and once sufficient heat (e.g. thermal energy) and pressure has accumulated, the fuel and air mixture spontaneously combusts (no spark necessary), forcing the piston down for the power stroke.

The HCCI approach allows for much better mixing of the air and fuel in the combustion chamber (as compared to conventional diesel combustion), seen in Figure 3, leading to low combustion temperatures [3]. Additionally, because of this improved stoichiometric balance in mixing, the combustion flame is able to propagate across the combustion chamber more quickly, thus it has higher flame speed (short burn duration) than conventional diesel combustion. The lower temperature and improved stoichiometric balance leads to lower soot and NO_x in the exhaust [3]. HCCI engines can operate at high compression ratios, thus potentially operating at higher efficiency than conventional diesel engines [3].

Because there is no injection event, the combustion timing of HCCI is very difficult to control. Often the fuel will begin to burn too early during the compression stroke, leading to poor efficiency due to increased compression work. If the compression of the fuel-air charge is too much, the fuel-air mixture spontaneously auto ignites before TDC, leading to a strong 'knocking' sound due to poorly phased combustion. The compression of the fuel-air mixture, in conjunction with the specific fuel used, limits the compression ratio of the engine. Because it causes excessive energy release (very fast overall rate), knocking can damage the engine if it is intense or sustained [1]. Early ignition timing relative to TDC is the fundamental challenge with HCCI, as there is a need to control combustion phasing (i.e. when combustion occurs). Thus, HCCI has

a limited operational range with the onset of auto ignition, and is very difficult to operate at higher engine loads/powers. Furthermore, increases in fuel consumption, unburned hydrocarbons (UHCs) and carbon monoxide (CO) have often accompanied this approach when HCCI operates with non-ideal combustion timing-phasing [4].

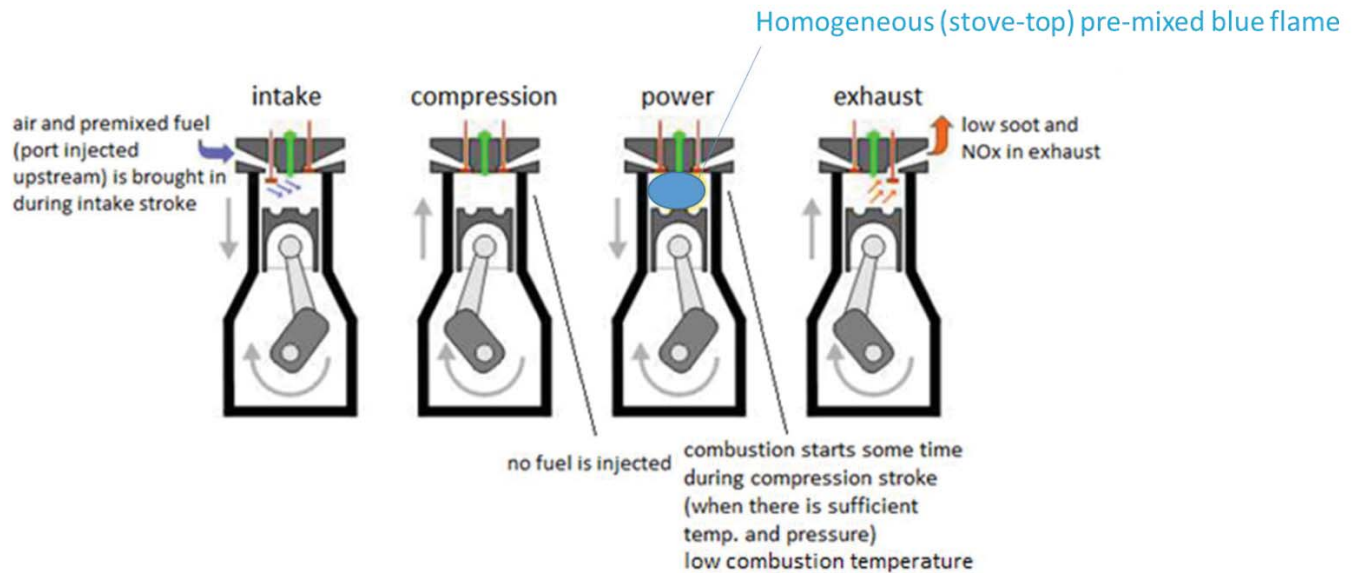


Figure 2. Homogeneous Charge Compression Ignition (HCCI) cycle [2]

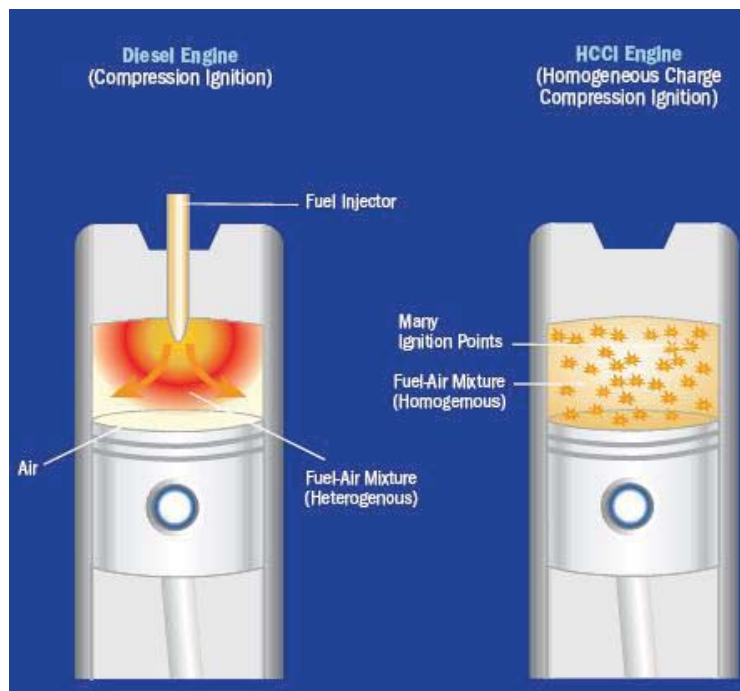


Figure 3. Conventional CI and HCCI combustion comparison [1]

Partially Premixed Combustion

Partially Premixed Combustion (PPC), also known as Partially Premixed Compression Ignition (PPCI), seen in Figure 6, has been proposed and investigated as a solution to the problem of controlling combustion phasing-timing in HCCI engines [2]. During PPC, a portion (partial

fueling) of the operational fuel is port injected during the air intake stroke *and* then the remainder of the fuel is directly injected at the top of the compression stroke. By the end of the intake stroke, the partial fuel added to the inlet air stream along with the inlet air have been well mixed in the engine cylinder. Then the injection at the top of the compression stroke (remainder of fuel to be burned) auto ignites, controlling the combustion phasing.

PPC is a practical way of achieving low NO_x and low soot in CI engines. Similar to HCCI, NO_x is reduced due to a low combustion temperature and soot is reduced due to the leaner air-fuel mixture in the combustion zone [3]. The equivalence ratio (ϕ) is the ratio of the actual fuel-air ratio to the stoichiometric (all oxygen consumed) fuel-air ratio, seen in Equation (1). In essence, an equivalence ratio of one denotes a stoichiometric mixture, less than one denotes a fuel lean mixture, while greater than one denotes a fuel rich mixture. Typically soot is reduced when the local equivalence ratio is less than 2 in the region of the cylinder where combustion is occurring [2]. The PPC flame is very complex, but generally retains the high flame speed (short burn duration) seen in HCCI, allowing for the potential to still run at higher efficiency.

$$\phi = \frac{F/A_{actual}}{F/A_{stoic}} \quad (1)$$

Because diesel engine power is limited by soot formation, further fuel addition will not result in more fuel conversion and greater engine power. This is because as engine speed (RPM) increases, turbulence in the combustion chamber increases and there is less time during the intake stroke for oxygen to enter the cylinder. If increased fueling were pursued once the combustion chamber starts to soot, only further soot and increased carbon monoxide (CO) and unburned fuels (UHCs) would result, with no power increase.

Figures 4 and 5 show well established characterizations of PPC based on previous work. Figure 4 shows the ability of PPC (denoted at PCCI, Premixed Charge Compression Ignition) to reduce both soot and NO_x. With increase premixing, the local equivalence ratios are decreased, as are local combustion temperatures, thereby moving combustion outside of the soot and NO_x formation regions. Figure 5 compares soot formation in diesel flames, showing a substantial decrease in soot concentration in PPC flames.

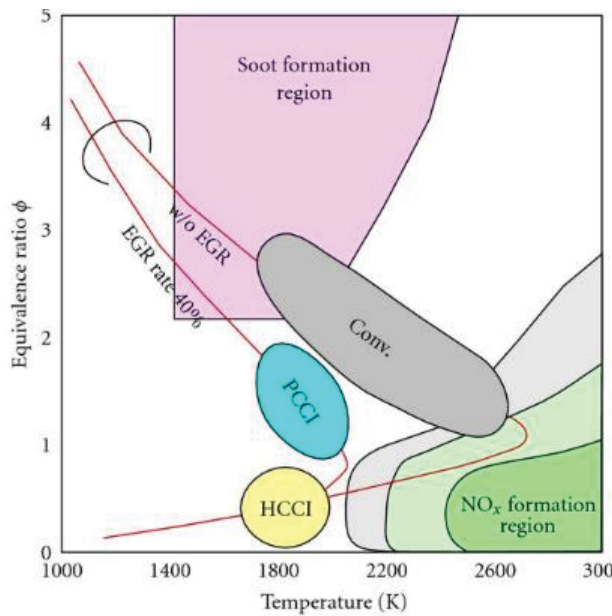


Figure 4. Emissions Map for HCCI, PPC, and Conventional Compression Ignition [3]

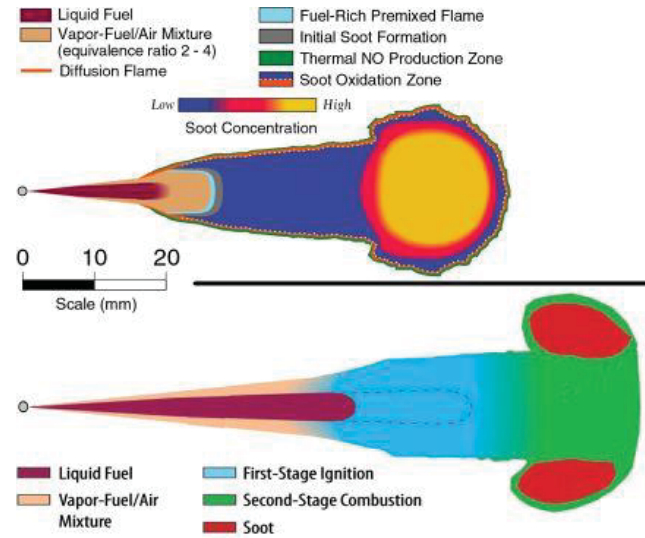


Figure 5. Flame characterization for Conventional Compression Ignition (top) and PPC (bottom) as fuel is injected [3]

Partially Premixed Combustion (PPC), with additional fuel delivered to the incoming air stream, allows for more fuel in the combustion chamber, and may allow increased specific power opportunities due to improved fuel-air interaction during the combustion event. Modest increases in UHCs and CO have accompanied this approach because the cylinder walls are cold relative to the rest of the chamber during combustion [4]. Thus, some premixed fuel at the periphery of the combustion chamber does not burn. Additionally, moderate increases in noise, vibration, and harshness (NVH) have accompanied the PPC approach because the combustion event is more violent and peak in-cylinder pressures are slightly elevated.

In summary, PPC is a hybrid approach combining conventional CI and HCCI to get the benefits of HCCI combustion (low NO_x, low soot, and high efficiency) while retaining control over combustion phasing. The inhomogeneity in the cylinder during the injection also allows the engine to be run at load conditions where HCCI combustion would not be possible. As described next, PPC has been studied principally in the context of commercial diesel engines in which regulatory pressures have pushed for significantly reduced emitted soot levels.

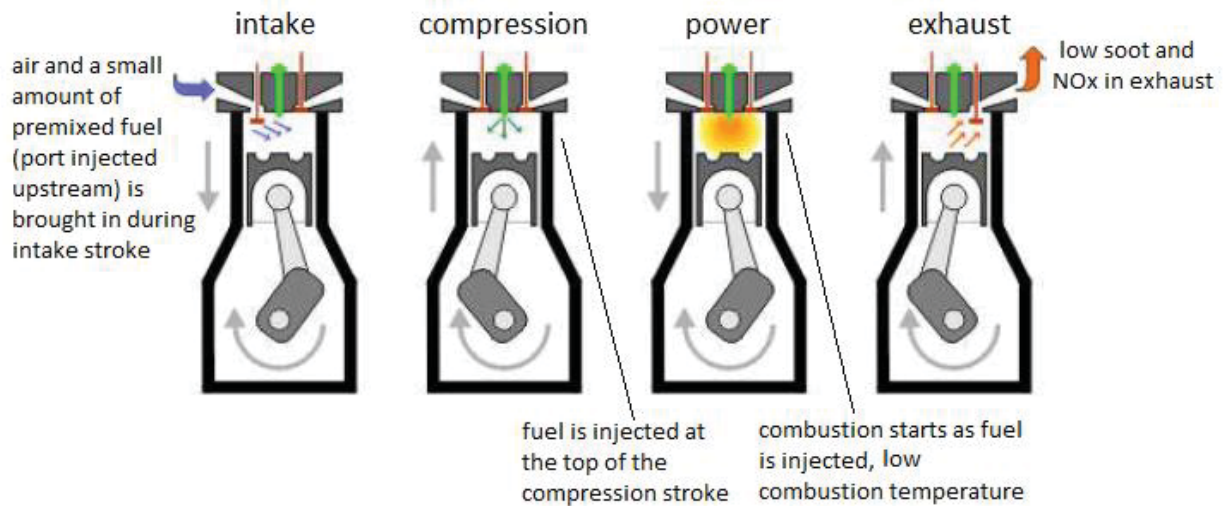


Figure 6. Partially Premixed Combustion cycle [2]

Part 1: Power Improvement with JP-5 in the Waukesha CFR

Summary

A partially premixed combustion (PPC) approach was applied in a single cylinder diesel research engine in order to characterize engine power improvements. PPC is an alternative advanced combustion approach that generally results in lower engine-out soot and NO_x emission, with a moderate penalty in engine-out unburned hydrocarbon (UHC) and carbon monoxide (CO) emissions. In this study, PPC is accomplished with a minority fraction of jet fuel injected into the intake manifold, while the majority fraction of jet fuel is delivered directly to the combustion chamber near the start of combustion (SOC). Four compression ratios (CR) were studied. Exhaust emissions plus exhaust opacity and particulate measurements were performed during the experiments in addition to fast in-cylinder combustion metrics. It was seen that as CR increased the soot threshold equivalence ratio decreased for conventional diesel combustion, however this afforded an increased opportunity for higher levels of port-injected fuel leading to power increases from 5 to 23% as CR increased from 14 to 21.5. PPC allowed for these power increases (defined by a threshold opacity level of 3%) due to smaller particles (and lower overall number of particles) in the exhaust that influence measured opacity less significantly than larger and more numerous conventional diesel combustion exhaust particulates. Carbon monoxide levels at the higher PPC driven power levels were only modestly higher, although NO_x was generally lower due to the overall enriched operation.

Literature Review

The objective of Part 1 of this study was to apply a PPC approach and the resulting reduction in engine-out particulate matter (soot) to extend the high load limit of a single cylinder diesel research engine based on exhaust opacity. Lim et al. [3], in a principally modeling based study, optimized a dual-fuel, dual-injection reactivity controlled compression ignition (RCCI) strategy of iso-octane then normal heptane to achieve 21 bar indicated mean effective pressure (IMEP) operation with low engine-out NO_x, soot, and carbon monoxide emissions. Using the same fuels in a similar RCCI model, Klos and Kokjohn [4] found that injection timing had significant tradeoffs between engine efficiency, emissions, and combustion instability. A near top dead center (TDC) injection timing significantly reduced combustion instability, while the emissions

and efficiency dropped to near conventional direct injection levels. More advanced phasing was even more stable, but produced high peak pressure rise rates, more NO_x and lower efficiency due to increased heat transfer losses.

Manente et al. [5] used a single-fuel, PPC approach at 11.5 bar gIMEP and found that high octane number fuels, like ethanol and gasoline, were able to achieve a reduction of soot and NO_x with a lower amount of exhaust gas recirculation (EGR) as compared to diesel, with NO_x and UHCs generally below Euro 5 and 6 emissions standards. Splitter [6], using a dual-fuel RCCI approach with gasoline and ethanol, found that UHC emissions were a function of piston crevice and squish geometry, and CO emissions were sensitive to local equivalence ratio. He predicted that by maintaining lean conditions with sufficient fully premixed fuel, simultaneous reductions in heat transfer and incomplete combustion could be achieved.

Butanol has been studied as a promising biofuel to use for PPC due to prolonged ignition delay, low cetane number, and high volatility fuel characteristics. Cheng et al. [7] achieved PPC with butanol-diesel blends, showing soot emissions could be reduced by 70%, while NO_x increased at low loads. Leermakers et al. [8] showed extremely high soot reduction with moderate butanol-diesel blends compared to diesel only operation using a PPC approach. They showed butanol-diesel blends to be a viable fuel, with stable operation at 50-70% butanol by volume, and 50% average gross indicated efficiency over the whole load range.

Noehre et al. [9] mapped out the operating range for PPC using standard diesel fuel under a range of engine operating conditions including CR, EGR, speed, and boost at loads up to 15 bar gross indicated mean effective pressure (gIMEP). The combination of low compression ratio, high EGR, and engine operation close to stoichiometric conditions resulted in premixed and diffusion combustion that enabled simultaneous NO_x and soot reduction, though a decrease in efficiency and a rise in UHC and CO emissions was noted. They concluded that soot emission reduction was a result of good premixing, long ignition delay, and low local combustion temperature due to high EGR. Additionally, Dempsey et al. [10] found, for a PPC gasoline approach, that when the premixed fuel percentage was low, the NO_x emissions began to increase due to the existence of high equivalence ratio regions in the combustion chamber at the start of combustion. Kalghatgi et al. [11] demonstrated the ability to use the soot reduction of PPC for high load extension. Using PPC with gasoline, they reached mean gIMEP of 15.95 bar and smoke opacity of 0.33%, while to attain the same level of smoke with diesel fuel, gIMEP had to be below 6.5 bar.

Zha et al. [12] compared a high cetane number (CN) jet propulsion '8' (JP-8) fuel with that of a low CN Sasol JP-8 under PPC mode. They found that low CN Sasol JP-8 exhibits longer ignition delay, lower premixed peak heat release and later combustion phasing compared to JP-8 combustion. Low CN Sasol JP-8 was shown to be a satisfactory under light-load condition, but with higher UHC emissions and lower fuel efficiency compared to high cetane JP-8 fuel. This investigation seeks to expand on the work of Zha et al. in using jet fuel for a PPC approach.

From a military standpoint, due to the significant logistical burden of a dual-fuel RCCI approach, this study focuses on a single-fuel PPC approach. This is consistent with Department of Defense (DoD) Directive 4140.43, the 'One Fuel Forward' initiative, meant to reduce the logistics burden associated with bringing multiple fuels to the battlefield. The fuel used in this study was Navy jet propulsion '5' (JP-5), a fuel commonly used in many Navy and Marine Corps platforms due

to its low volatility for safety purposes. Thus, the objective of this work is to work is to principally focus on approaches to quantify the PPC based power improvements based on the high load extension of the exhaust sooting limit in a compression ignition diesel engine as compared to conventional diesel operation.

Experimental Details

The engine used in this study was the Waukesha Cooperative Fuels Research (CFR) diesel engine, shown in Figure 7. This engine has an indirect injection (IDI) combustion chamber, shown in Figure 8, and has mechanically variable compression ratio (CR), injection amount, and injection timing. For this PPC alternative combustion mode study, the IDI pre-chamber fuel injection system was supplemented with a Cole-Parmer peristaltic pump delivering jet fuel directly to the intake manifold. In-cylinder pressure was measured with a Kistler 6125C piezoelectric sensor. Engine position was measured using a BEI dual channel shaft encoder coupled directly to the crankshaft. The CFR cetane rating engine operates at a fixed 900 rpm. Experimental details can be found in other references [13, 14]. An ETAS-Bosch wide-band oxygen sensor and lambda meter provided overall air-fuel ratio information. Exhaust emissions (via a heated Unique Heated Products filter) were measured with an Infrared Industries FGA4000S analyzer. Opacity was measured with a Wager 6500 opacity meter. During some of the experiments, exhaust particulate measurements were acquired with a TSI 3090 EEPS unit via a Matter Engineering diluter. A schematic of instrumentation is shown in Figure 9.

The general categories of components that are present in JP-5 were determined using gas chromatograph/mass spectrometry (GC/MS) analysis. An Agilent 5975 inert mass selective detector was used in conjunction with an Agilent 6890N Gas Chromatograph System under the same conditions utilized in previous studies [15]. The GC was equipped with a Zebron phase ZB-5ms® column (30 m, 0.25 mm, 0.25 μ m, 5% diphenyl-arylene/95% dimethyl polysiloxane) and operated at a helium flowrate of 1.5 mL/min. The compounds were separated using a temperature ramping program that started at 60.0 °C and climbed at a rate of 10 °C/min to 250 °C. An electron impact ionization (EI) method was used for the mass spectrometer with an m/z scan from 30 to 600. To prevent the detector from being saturated, all fuel samples were diluted in n-hexane (1/100 dilution), and the mass spectral analysis of the fuel sample was delayed until the solvent peak had passed. The fragmentation patterns of peaks found at several retention times were compared to those found in the NIST database to determine potential compounds present.

The speed of sound of the fuels were measured using an Anton Paar DSA 5000 Density and Sound Analyzer. Certified reference standards were used to test the accuracy of these instruments, which were recalibrated if they failed testing. The bulk modulus at 20°C was calculated by multiplying the density by the speed of sound squared. The other properties were determined using ASTM techniques specified in the military specifications.



Figure 7. Waukesha Diesel CFR Engine

Table 1. Waukesha DCFR Engine parameters

Engine type	Single cylinder, Indirect-injection, in-line cam-driven pump
Bore × Stroke	82.6 mm × 114 mm
Displacement	610 cc (0.610 L)
Speed	900 RPM fixed
Coolant	Water reflux, 100°C
Intake Air	Lab ambient, 20°C

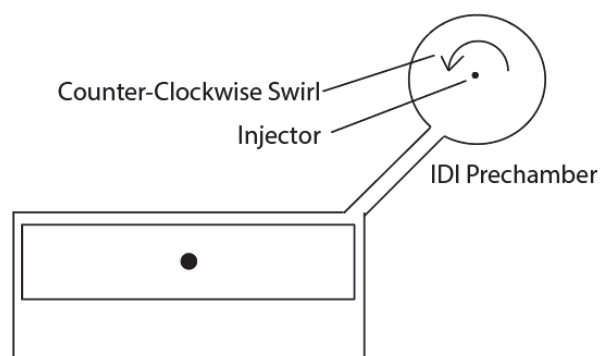


Figure 8. Indirect Injection Combustion Chamber geometry

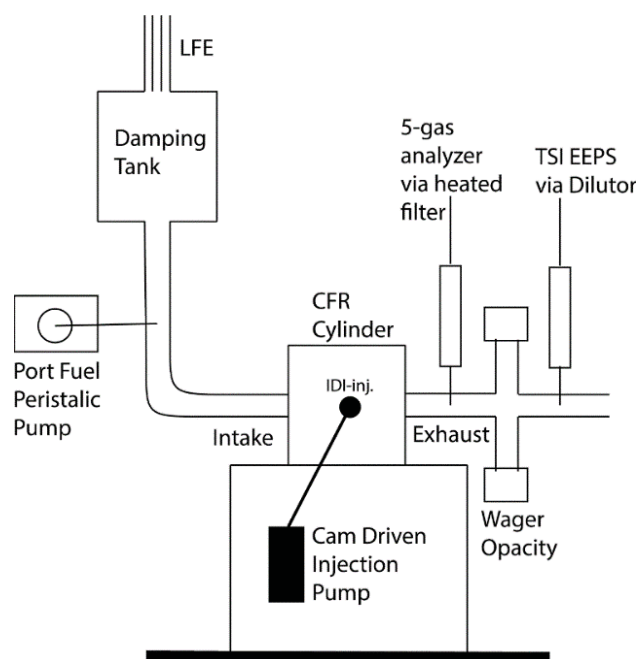


Figure 9. Experimental Setup

Fuel Properties

Navy jet propulsion ‘5’ (JP-5) fuel was used in this investigation. This jet fuel has a moderately higher flash point (less volatile) as compared to conventional commercial jet fuels. A summary of key properties is shown in Table 2.

The chromatogram of the fuel is shown in Figure 10. Based on a NIST database search and comparing to original standards, the major peaks at 5.5, 7.1, 8.7, 10.1, 11.5, and 12.7 minutes are decane, undecane, dodecane, tridecane, tetradecane, and pentadecane, respectively. Other components identified by the NIST database include branched alkanes, aromatic compounds, alkylcyclohexanes, and indenenes. The specific compounds were not confirmed by original standards.

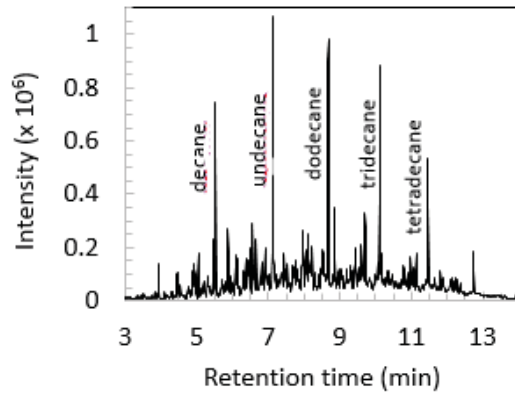


Figure 10. Chromatogram of JP-5

Table 2. Navy jet JP-5 fuel properties

Test	Units	Value
Cetane Index		45.3
DCN		46.5
Density 15°C	kg/L	0.8068
Distill. 10%	°C	191
Distill. 50%	°C	208.6
Distill. 90%	°C	239.4
Aromatics	vol%	18.1
Olefins	vol%	1.1
Saturates	vol%	80.8
Flash Point	°C	66
Heating Value	MJ/kg	43.16
Hydrogen%	mass %	13.74
Sulfur XRF	mass %	0.0953
Viscosity 40°C	cSt	1.389
Surface Tension	dyne/cm	28.7
Speed of sound 20°C	m/s	1319
Bulk Modulus 20°C	MPa	1397

Experimental Approach

After the CFR engine was warmed up using conventional diesel combustion with Navy jet fuel, the desired compression ratio (CR) was adjusted followed by baseline conventional IDI data collection. Engine load was set starting in the mid-load range of the diesel CFR (~5 to 6 bar gIMEP), then incrementally increased until relatively high levels of CO were observed. At each operating point a real time fast heat release analysis was performed in order to maintain the crank angle location for 50% heat release (CAD50) nominally around 10°ATC with slight start of injection (SOI) adjustments.

After the conventional IDI fuel-load sweep was performed, the SOI and fuel injection amount (e.g. load) were reset to the condition where opacity was nominally 3%. At this point two different PPC port jet fuel injection levels were introduced. Data were collected at each operating point. Since JP-5 is very reactive, only a minority of fraction of this jet fuel could be applied to the intake port due to premature combustion during the compression stroke. The maximum port fuel injection level was empirically determined by the loss of SOI control. Another paper by the authors outlines this auto-ignition nature of JP-5 at various compression ratios [16]. It should be noted that this approach does not seek to optimize the PPC combustion by also adjusting SOI, but rather to see the power increase effects with PPC added to a conventional diesel engine without adjustment (e.g. retro-fit).

At each stabilized operating point, twenty five seconds of in-cylinder pressure data were collected at a 50 kHz sampling frequency using MATLAB data acquisition software and National Instruments hardware. A conventional single zone heat release analysis approach was then applied to the data as outlined in references [17, 18].

Results and Analysis

A peristaltic pump was used to deliver port fueling. The pump was calibrated to determine fuel volume as a function of time at various pump settings. The fuel was continuously delivered to the intake pipe runner approximately 25 cm upstream of the cylinder head. In applying the PPC approach, the amount of port fuel injection was steadily increased until the start of combustion (SOC) was unable to be controlled by the in-cylinder injection event at SOI due to pre-ignition of the port injected charge, where ignition delay (IGD) was slightly negative (in terms of degrees ATC). Figure 11 shows the two PPC operating fueling amounts used in this study as a function of CR. The upper line is the maximum port fuel fraction added at the threshold of port-injected fuel pre-ignition. The lower line is an operating condition slightly less of maximum where combustion phasing is still controllable with SOI adjustments.

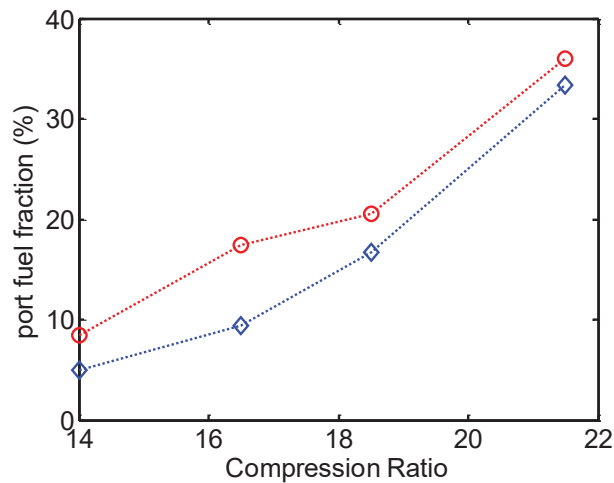


Figure 11. Percentage of total fuel port injected for CRs

The amount of port fuel injection to the intake manifold was determined from the peristaltic pump calibration. This value was also similar to the air-fuel equivalence ratio (λ) measured from the ETAS-Bosch oxygen sensor at the maximum Conventional IDI (Conv. IDI) operating point (3% opacity) of the same CR to determine the port fuel fraction (port fuel on v. off). As seen in Figure 11, more fuel was able to be port injected at higher compression ratios since the overall high load Conv. IDI limit became leaner as CR increased (results to be discussed shortly).

Next, Figure 12 and Figure 13 focus just on the specific CR 16.5 operating point; operation through the range of compression ratios will be discussed later.

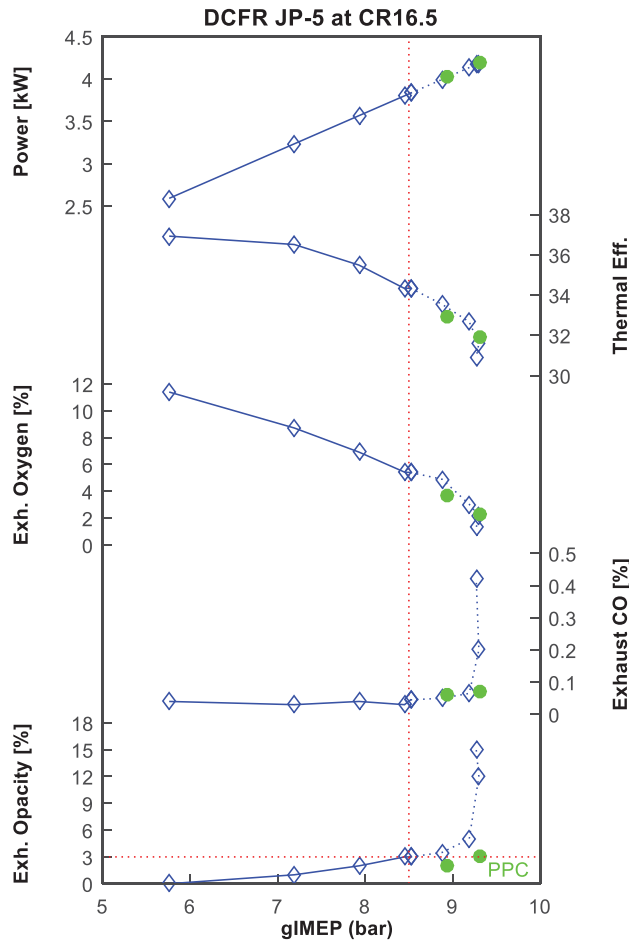


Figure 12. PPC and Conv. IDI operation comparison of power, efficiency, and exhaust characteristics at CR 16.5

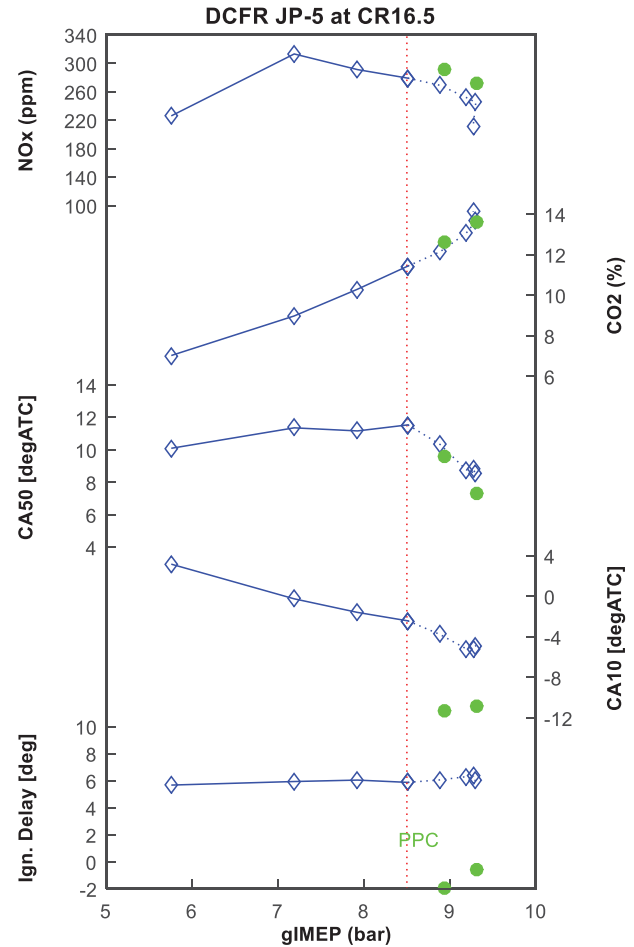


Figure 13. PPC and Conv. IDI operation comparison of combustion phasing and exhaust characteristics at CR 16.5

Figure 12 compares PPC to Conventional IDI operation at CR16.5. The blue diamond symbols solid and dashed lines show Conv. IDI operating mode. The solid line connects the data below the opacity threshold, the dashed line data shows conventional engine operation in the high load excessively sooty regime. The green solid ‘dots’ shows PPC operation. On the x-axis, gIMEP increases directly with the amount of fuel injected for fixed engine speed. A dashed horizontal red line marks the 3% exhaust opacity threshold defined to be the point of maximum engine load, such that engine soot is too great for further enriched operation. A vertical dashed red line marks the engine load (gIMEP) where conventional operation reaches this 3% threshold, approximately 8.5 bar. As seen in Figure 12 (bottom graph), PPC operation does not reach this 3% threshold until a higher engine load, 9.3 bar, because more fuel was able to be added via port injection without a significant increase in exhaust soot.

The top of Figure 12 shows that the power for Conv. IDI operation at the opacity threshold (red line) was 3.8 kW, while the maximum PPC point achieved 4.25 kW, an 11% improvement. In Figure 12 (second from bottom) shows low levels of exhaust CO between the Conv. IDI and PPC operating modes at the 3% opacity threshold. Such low levels of CO indicate nearly complete combustion (as CO levels are just beginning a dramatic rise with Conv. IDI).

Exhaust oxygen levels (Figure 12 middle) were lower in the PPC case compared to the Conv. IDI case at the opacity limit indicating that more of the available oxygen was being consumed in partially premixed combustion due to the increased fuel addition and port injection air-fuel mixing. Thermal efficiency (Figure 12 second from top) was moderately lower in the PPC case compared to the Conv. IDI case at the opacity limit due to the increase in specific fuel consumption. However, as seen in the Conv. IDI operating points above the opacity threshold, there was a similar decrease in thermal efficiency. Thus, this loss in efficiency is not a problem endemic only to PPC operation.

Figure 13 continues the comparison of PPC to Conventional IDI operation at CR16.5. Ignition delay (Figure 13 bottom) was insensitive for Conv. IDI operation across the range of engine loads. As discussed in Figure 11, PPC was set such that IGD was near zero or negative in degrees ATC, limited by pre-ignition of the fuel-primed mixture from port injection.

The 10% heat release location (CAD10) shown second from the bottom in Figure 13 was defined as the SOC in this study. For Conv. IDI operation, CAD10 steadily advanced as gIMEP increased. This is because SOI was advanced to inject more fuel to achieve higher gIMEP keeping CAD50 approximately consistent. For PPC, SOI was set to -10 degrees ATC (essentially stock Conv. IDI setting – not optimized for PPC), and CAD10 (and thus SOC) was reached almost immediately after injection due to pre-mixing and heat release of the port injected fuel. The 50% heat release location (CAD50, Figure 13 middle) was kept near 8-10 degrees ATC for both operating modes across the range of engine loads to retain ideal combustion phasing for best torque. As gIMEP increased, especially with PPC, CAD50 advanced due to the increased fuel injection.

Exhaust carbon dioxide (CO_2 , Figure 13 second from top) increased with gIMEP because as more fuel was added to achieve higher engine loads, more CO_2 was formed in combustion. NO_x , shown at the top of Figure 13, followed the characteristic curve for Conv. IDI operation. NO_x peaked at mid-load due to the steady increase in combustion temperature and steady decrease in available oxygen as gIMEP increases. NO_x for the PPC case was slightly elevated due to a modest increase in combustion temperature, shown later from slightly greater peak combustion pressures.

A characteristic in-cylinder pressure and cumulative heat release curve for both Conv. IDI (9.2 bar) and PPC (9.3 bar) are shown next in Figure 14. The PPC data (green) clearly shows an early energy release period (~330 deg.) followed by the main fuel injection energy release period taking off at ~345 deg. Due to the minority fuel-energy fraction of the PPC port fueling, only a modest low level of early PPC energy release is observed. The Conv. IDI heat release shows some slight evaporative cooling (~350 deg.) followed by the fuel injection driven energy release period. Figure 15 through Figure 27 compare PPC and Conventional IDI engine operation over four compression ratios. As above, solid large dots indicate PPC operation, open diamonds indicate Conventional IDI operation. The start of the dashed line indicates the point at which Conv. IDI operation reaches the 3% exhaust opacity threshold similar to the above composite figures at CR 16.5.

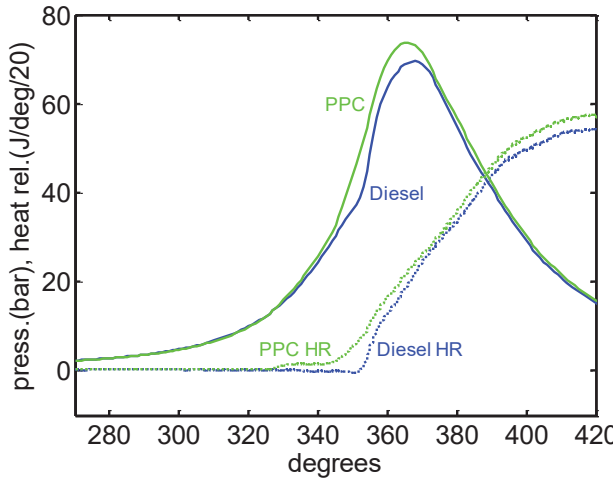


Figure 14. In-cylinder pressure trace and cumulative heat release for 9.2-9.3 bar cases

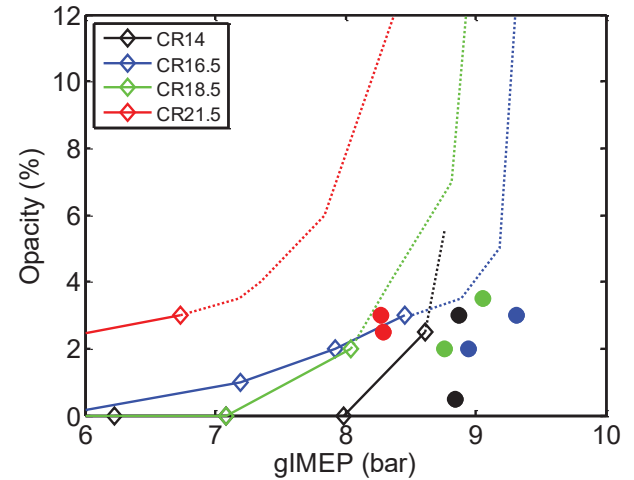


Figure 15. Exhaust opacity as a function of engine load

The engine soot limit was set at 3% exhaust opacity. Any further increase in engine load, shown by the dashed lines in Figure 15, was considered outside of the normal Conv. IDI operating range of the engine. Data were collected at these operating points for the sake of comparison to high load PPC operation. Figure 15 shows that significant sooting was delayed until much higher loads using PPC. PPC allowed for high load gIMEP extension past the maximum point of Conv. IDI engine operation. Exhaust opacity for the Conventional IDI operating points at high gIMEP were significantly greater than the PPC operating points at the same load, across the range of compression ratios.

Figure 16 shows an advance in SOI for all cases with increasing gIMEP such that CAD50 was kept nominally constant ATC for optimal torque-combustion phasing. SOI was advanced for higher CRs, because at higher CRs, IGD was shorter due to the elevated in-cylinder pressures. Shorter IGD led to less time for premixing and thus longer 10-90% burn durations, shown in Figure 15. While PPC operating points maintained relatively similar SOI to Conv. IDI operation, a -10° ATC SOI is likely the earliest possible timing due to the onset of pre-ignition.

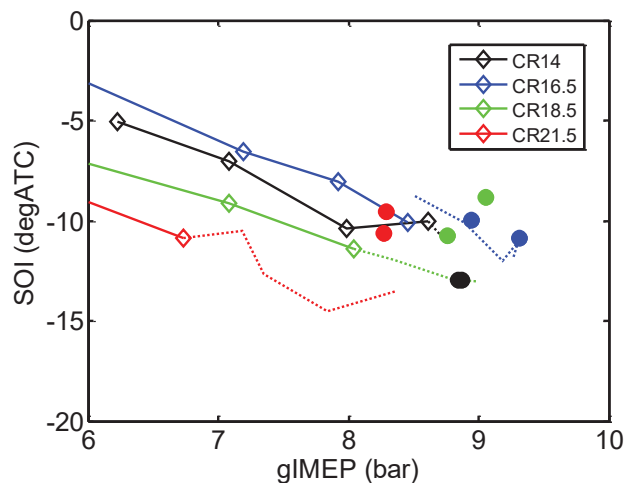


Figure 16. Start of Injection as a function of engine load

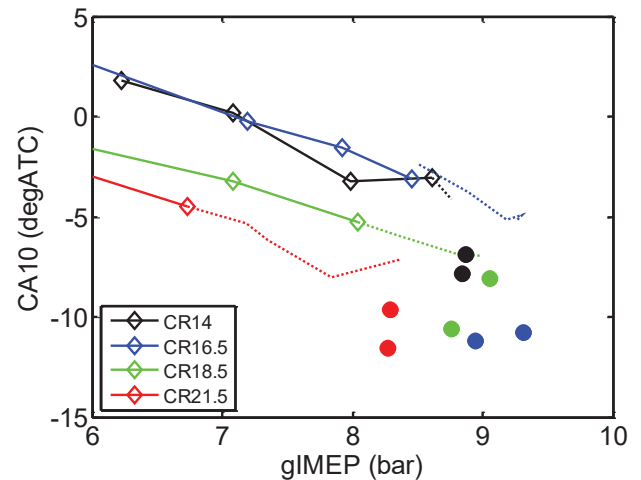


Figure 17. 10% heat release location as a function of engine load

Shown in Figure 17, CAD10 is roughly 6 degrees earlier for PPC. Figure 17 shows CAD10 following the same trends as SOI. For higher gIMEP, fuel injection amount was increased, thus SOI must advance leading to earlier CAD10 and SOC. At higher CRs, CAD10 advanced due to increased pressures at SOI leading to earlier SOC.

Ignition delay, in Figure 18, was defined as the difference between SOI and CAD10 (SOC). For Conv. IDI operation, the CR14 case was expected to have the coolest in-cylinder temperatures and lowest pressures during compression and thus the longest IGD. The PPC operating points at CR16.5, 18.5, 21.5 had IGDs of slightly negative or near zero, thus the pre-mixed fuel began to burn before or during the direct injection event. PPC at CR14 had IGDs close to those of Conv. IDI operation.

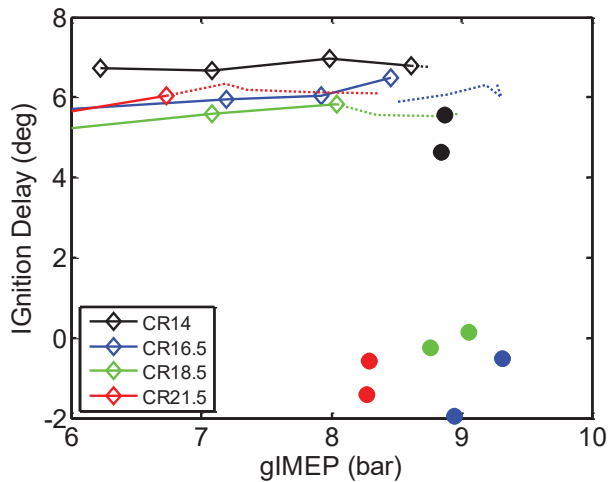


Figure 18. Ignition delay as a function of engine load

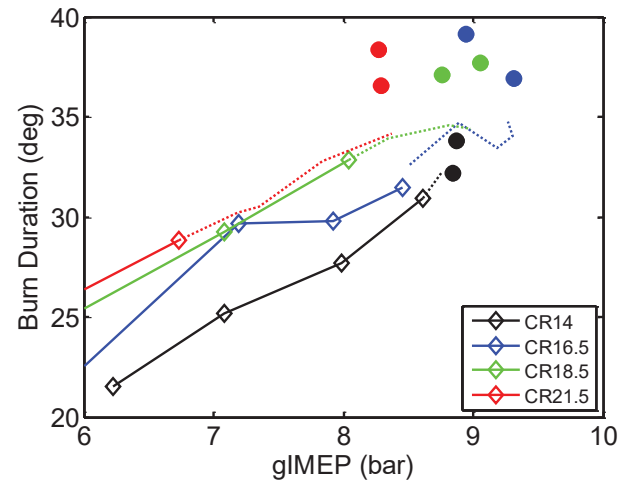


Figure 19. 10-90% heat release duration as a function of engine load

The burn duration, shown in Figure 19, increased with increasing CR and increasing gIMEP. To achieve higher engine loads, more fuel was injected leading to a longer combustion event. At higher CRs, IGD was shorter, and these shorter IGDs led to less time for premixing and thus longer burn durations. Despite the port injection, PPC operating points had even longer burn durations. The authors hypothesize that the port injected charge was burning early (indicated by the negative IGD), thereby consuming oxygen in the direct injection flame region. This small oxygen deficiency led to a slower, diffusion-dominated flame.

Figure 20 shows increasing peak in-cylinder pressures as CR and load (fueling) were increased, in accordance with that expected by the ideal gas law. Peak combustion temperatures normally tend to increase with peak combustion pressures for a given CR. PPC peak pressures were close to those of Conv. IDI operation. Thus, it would be expected that NO_x increases with PPC due to the higher peak pressures. However, as seen in Figure 21, this was not the case.

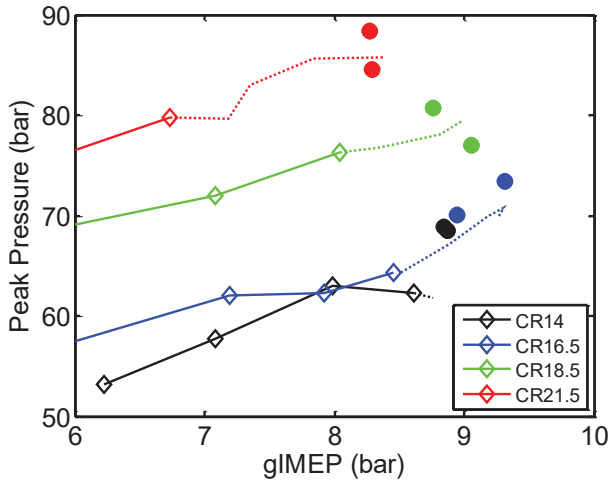


Figure 20. Peak in-cylinder pressure as a function of engine load

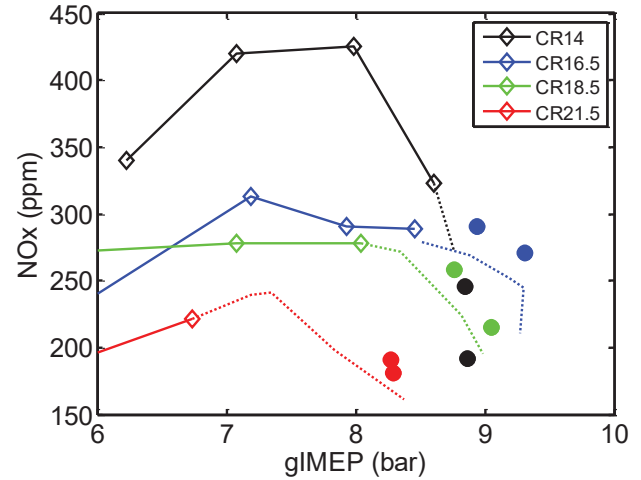


Figure 21. Exhaust NOx concentration as a function of engine load

Figure 21 shows that exhaust NOx concentration followed the characteristic curve for Conv. IDI operation as a function of load. NOx peaked at mid-load due to the steady increase in combustion temperature and steady decrease in available oxygen as gIMEP increased. PPC, operating at high loads, followed this same NOx trend as Conv. IDI operating above the 3% opacity threshold (dashed lines). For PPC, the authors believe the premixed charge consumes oxygen in combustion outside of the flame zone, in the cylinder periphery, indicated by the short IGDs, leading to the low levels of NOx. Overall, as CR increases, exhaust NOx concentrations decrease, despite the greater peak pressures at high CR. The authors hypothesize that (to be shown in Figure 23) at higher CRs, in-cylinder temperatures do not increase, but rather moderately decrease, with the increase in CR and peak pressures. This is because, for this fixed displacement engine, the higher compression ratios lead to smaller combustion chamber volumes which tend to lower combustion temperatures (to be further discussed shortly). Thus, the moderately lower temperature combustion event at increased CR reduced NOx levels.

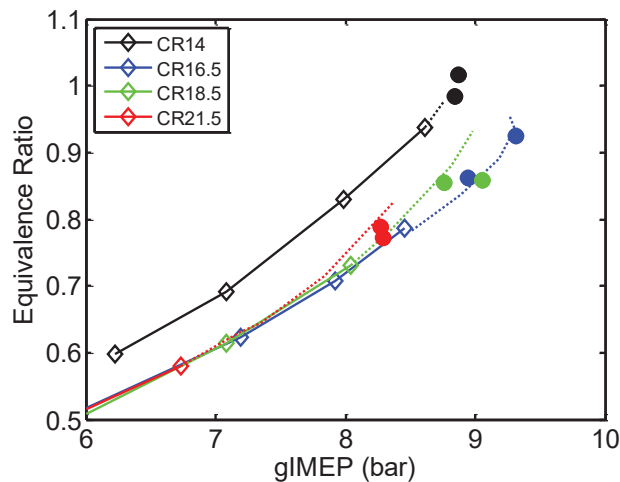


Figure 22. Overall fuel-air equivalence ratio as a function of engine load

The overall equivalence ratio increased at higher gIMEP due to the greater fuel addition. Because PPC allowed for the addition of more fuel during port injection, the overall equivalence

ratio was even higher for the PPC cases, shown in Figure 22. At CR14, PPC allowed operation near stoichiometric conditions. However, shown in Figure 25, at CR14, thermal efficiency was significantly lower due to the decrease in expansion work during the combustion stroke.

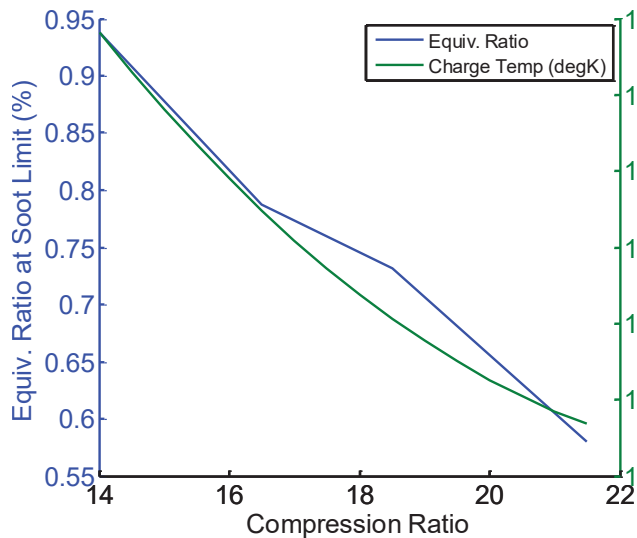


Figure 23. Equivalence ratio at the 3% soot limit and peak average predicted combustion temperature as f(CR)

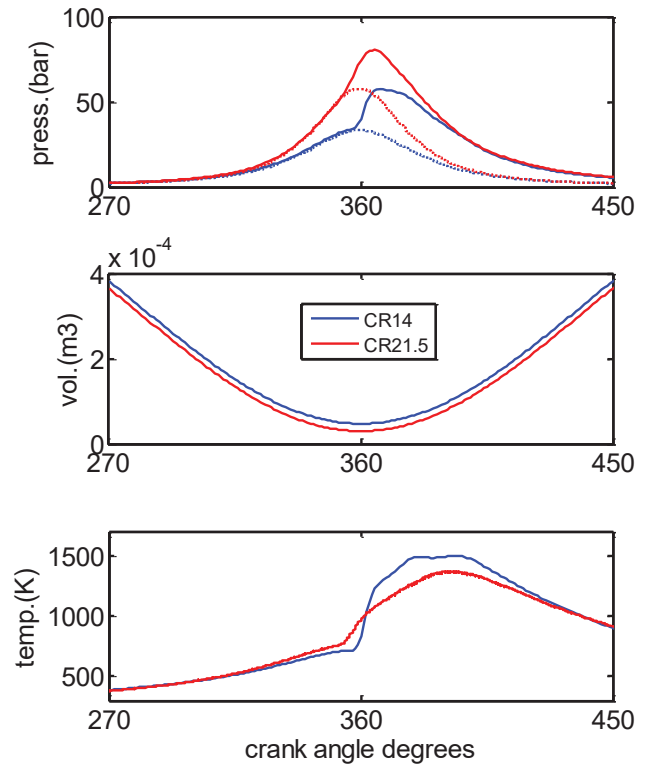


Figure 24. CR21.5 (red) and CR14 (blue) in-cylinder pressure, volume and average predicted temperature

The equivalence ratio at the soot-limit (nominally 3%) is shown in Figure 23 as a function of CR. It is seen that as CR increases the corresponding maximum fueling (e.g. equivalence ratio) that can be fueled, must decrease significantly. When the peak combustion pressure is converted to an average charge temperature using the ideal gas law and cylinder volume at the time of combustion, the peak charge temperature is predicted (air only gas constant), shown in Figure 23. Because the combustion chamber volume decreases with an increasing CR for a fixed displaced volume, the peak temperature also decreases as the airflow rate was constant with increasing CR. The increase in combustion pressure with increasing CR does not offset the reduced combustion chamber volume, thus average combustion temperatures moderately fall with increasing CR. This helps explain why the NO_x levels decreased with increasing CR.

As this result is somewhat counterintuitive, more detail is shown in Figure 24. A representative in-cylinder pressure trace is shown for from the 7.15 bar gIMEP cases in the top segment. For reference a dotted ideal polytropic ($n = 1.33$) reference line is shown for both compression ratios. As expected, the CR21.5 case has much higher compression and combustion pressure as compared to the CR14 case. The middle segment of this figure shows the total cylinder volume as a function of engine position. The CR21.5 case has a smaller volume throughout the cycle due to the smaller clearance volume at this high CR. The lower segment shows a ‘single zone’ temperature estimate using the ideal gas law. It is seen that during compression both cases have

similar predicted charge temperatures. Near the later phase of compression, the CR21.5 case is hotter than the CR14 case, with combustion starting slightly earlier as described above. Then, quite interestingly, the CR14 single zone average charge temperature during combustion exceeds that of the CR21.5 case. This result is due to the larger charge volume (nearly 1.6 times larger than the high CR case) that exists at CR14 despite its lower overall pressure.

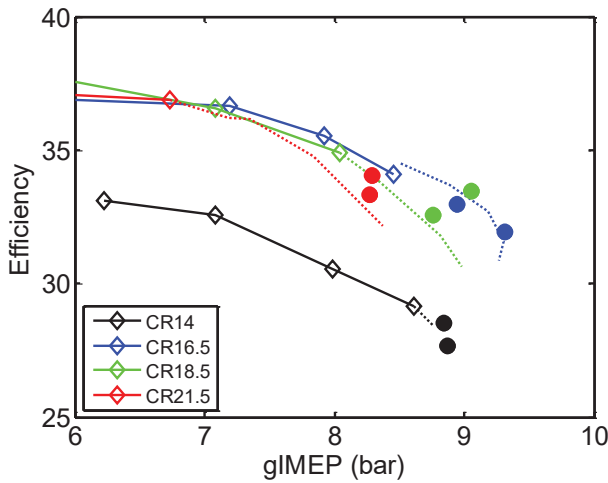


Figure 25. Thermal efficiency as a function of engine load

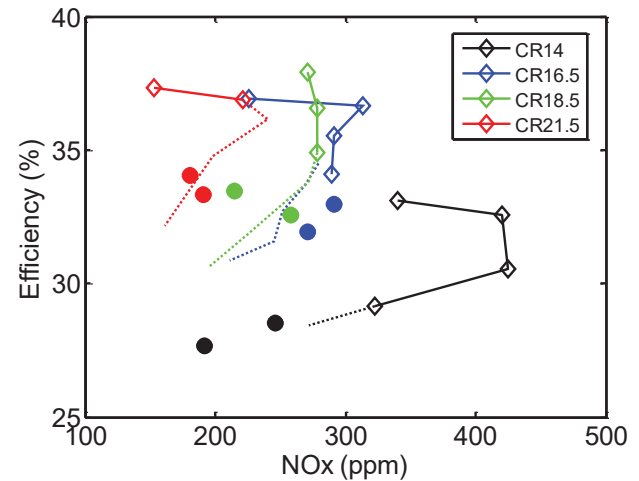


Figure 26. Indicated Efficiency v. NOx

Next, in looking at engine indicated efficiency, the PPC cases resulted in a moderate decrease in thermal efficiency at high loads compared to the Conv. IDI high load limit, shown in Figure 25. This is because at higher loads there were increased heat losses as more fuel was added and there was a lower, less advantageous ratio of specific heats at these PPC richer conditions. However, as seen in the Conventional IDI operating points above the opacity threshold, there was a similar decrease in thermal efficiency. Thus, this loss in efficiency is not a problem endemic only to PPC operation. It is interesting to note that the higher compression ratios have very similar thermal efficiencies. While higher compression ratio allow for increased expansion work, it also results in higher compression/combustion temperatures, which increase heat losses. As CR increases, these effects offset. The effects of engine efficiency with engine-out NOx are shown in Figure 26. In general, NOx peaks near the highest measured efficiency (for a given CR). This corresponds to a mid-high load point. NOx and efficiency then fall with increased enrichment and with PPC. The increased enrichment reduces available oxygen necessary for NOx formation.

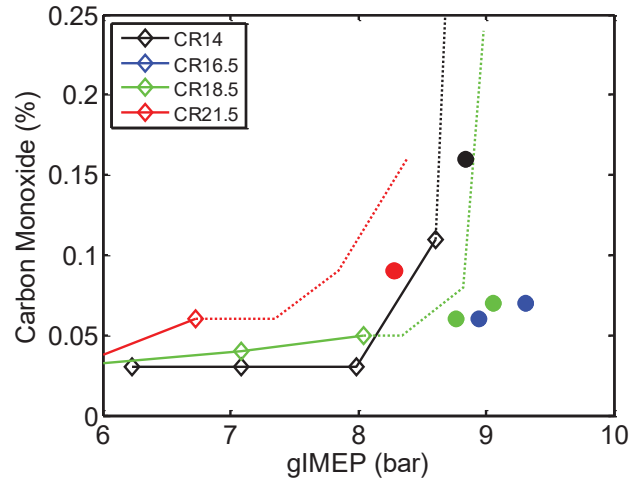


Figure 27. Exhaust CO concentration as a function of engine load

Exhaust carbon monoxide concentration in Figure 27 follows similar trends as exhaust opacity in Figure 15. The port injection event of PPC allowed for greater air-fuel mixing prior to SOC, leading to more complete combustion and a reduction of CO at high loads. Time for mixing during Conventional IDI was limited due to engine speed. Port injection allowed for mixing in the intake manifold and during the intake and compression strokes, leading to a more partially ‘homogeneous’ pre-mixed charge prior to the main injection event.

Figure 28 shows the exhaust soot-particulate concentration (numbers of particles per volume) as a function of particle size. Due to instrument saturation limits, these data were collected at high load just near or below the minimum detection limit of the opacity meter. The integrated results for each curve are converted to a total mass concentration assuming spherical carbon particles. Two cases are run for both Conv. IDI as well as PPC at a CR14, with equivalence ratios of 0.85 and 0.89. Only the 0.89 case with Conv. IDI having just exceeded the minimum opacity threshold ($\sim 1\%$). The two PPC cases were expected to be well below the minimum opacity detection level.

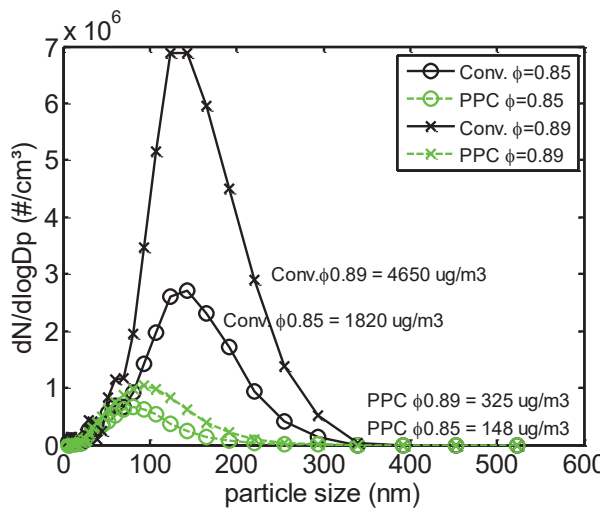


Figure 28. Exhaust soot-particulate concentration as a function of particle size

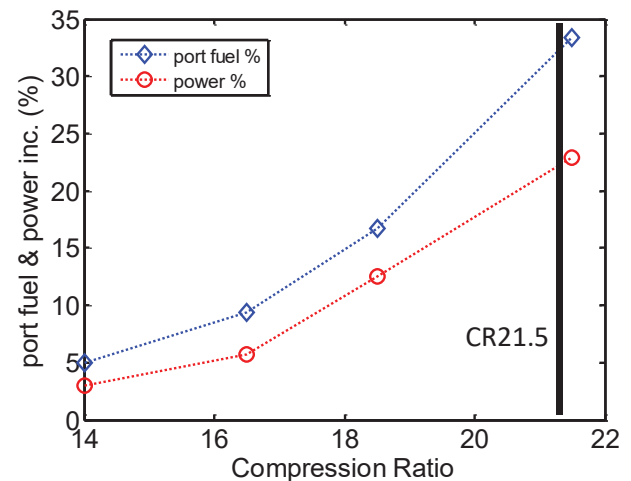


Figure 29. Maximum PPC port fuel injection fraction and PPC specific power increase over Conv. IDI operation

Figure 29 illustrates the key finding of this study. More fuel was able to be added using PPC at higher compression ratios because the 3% exhaust opacity soot limit was delayed until higher engine loads compared to Conventional IDI operation. At fixed engine speed, this torque improvement translated into a moderate power improvement over Conventional IDI diffusion operation. Power increased from 3% to 23% at the highest CR. The solid black vertical line on Figure 29 marks a compression ratio of 21.5. This is the compression ratio for both the Yanmar gen-set and the Humvee engine. Thus it is expected to see power increases of approximately 20% in those studies.

Conclusions

This study applied a PPC approach with a minority fraction port fuel injection in order to extend the high load limit based on exhaust soot opacity. Jet fuel was used as the same fuel for both port and in-cylinder injection. It was seen that the Conventional IDI soot limit occurred at leaner overall maximum fuel-air equivalence ratios as CR was increased. This afforded a greater opportunity to increase port injected fuel levels with increased CR. It was thus seen that power levels were then able to increase from 5 to 23% (at CR 21.5) over conventional diesel combustion by applying port injected companion fuel (PPC) without a soot opacity penalty.

Part 2: Power Improvement Comparison with non-JP-5 fuels in the Waukesha CFR

Summary

Partially Premixed Combustion (PPC) approaches were applied in a single cylinder diesel research engine in order to characterize engine power improvements. PPC (dual fuel) and PPC (single fuel) are alternative advanced combustion approaches that generally result in lower engine-out soot and NO_x emissions, with a moderate penalty in engine-out unburned hydrocarbon (UHC) and carbon monoxide (CO) emissions. In essence, this study sought explore whether or not a two-fuel PPC approach might be worth the additional fuel complexity when compared to a conventional diesel approach or a single-fuel PPC approach, based on power improvements from the high load extension of the exhaust sooting limit. In this study, PPC was accomplished with a minority fraction of fuel (isobutanol, iso-octane and jet JP-5) injected into the intake manifold, while the majority fraction of jet JP-5 fuel was delivered directly to the combustion chamber near the start of combustion (SOC). Four compression ratios (CR) were studied. Exhaust emissions plus exhaust opacity and particulate measurements were performed during the experiments in addition to fast in-cylinder combustion metrics. It was seen that as CR increased, the soot threshold equivalence ratio decreased for conventional diesel combustion; however, this afforded an opportunity for increased levels of port fuel leading to a range of power increases from -2 to 27% as CR increased from 14 to 21.5. PPC allowed for these power increases (defined by a threshold opacity level of 3%) due to smaller particles (and lower overall number of particles) in the exhaust that influenced measured opacity less significantly than larger and more numerous conventional diesel combustion exhaust particulates. Carbon monoxide levels at the higher driven power levels were only modestly higher for the PPC combustion modes, although NO_x was generally lower due to the overall enriched operation.

Literature Review

Due to technological and personnel safety requirements, Military vehicles and other platforms have significantly increased in weight over the past decades. In order to maintain mobility and

performance, engine specific power must be increased. However, diesel engine power is limited by soot formation, an indicator of locally high fuel-air equivalence ratio, due to poor mixing of the air and fuel at high engine loads. More fuel cannot be delivered and reacted in the combustion chamber since the allotted diffusion time is very small at high RPM and oxygen is limited at high engine loads. Alternative advanced combustion modes called partially premixed combustion (PPC, also known as Partially Premixed Compression Ignition, PPCI, and Premixed Charge Compression Ignition, PCCI) and reactivity controlled compression ignition (RCCI) deliver a portion of the fuel to the cylinder via the engine air intake system, which generally leads to lower engine-out particulate. This approach has the potential to lead to increased specific power at the same soot-opacity level.

In the case of RCCI, the process involves introduction of a low reactivity fuel (various fuels) into the cylinder during intake in order to create a well-mixed charge of low reactivity fuel and air. The normal reactivity fuel is injected directly into the combustion chamber before ignition of the premixed fuel occurs.

In relations to Part 2 of this study, normal butanol and isobutanol have been studied as promising biofuels to use for advanced combustion modes due to prolonged ignition delay, low cetane number, and high volatility fuel characteristics. Cheng et al. [19] achieved PPC with butanol-diesel blends, showing soot emissions could be reduced by 70%, while nitrogen oxide concentration (NO_x) increased at low loads. Leermakers et al. [20] showed extremely high soot reduction with moderate butanol-diesel blends compared to diesel only operation using a PPC approach. They showed butanol-diesel blends to be a viable fuel, with stable operation at 50-70% butanol by volume, and 50% average gross indicated efficiency over the whole load range.

DeVescovo et al. [21], using an RCCI approach with a net load of 6 bar indicated mean effective pressure (IMEP), found that due to isobutanol's high octane number (low cetane number, CN), a significantly larger quantity of direct-injected fuel was required to match combustion phasing, resulting in higher NO_x emissions and decreased gross thermal efficiency. Wang et al. [22] modeled RCCI combustion using isobutanol combined with a cetane improver over wide operating conditions. Due to the low reactivity of isobutanol, a relatively high amount of cetane improver was needed to enhance the reactivity of the mixture. Zheng et al. [23] studied n-butanol, isobutanol, and octanol (2-ethylhexanol) under conventional diesel operation. They found that due to their oxygen content, these fuels produced much lower emissions of soot and carbon monoxide than were achieved with diesel fuel alone, with a modest increase in NO_x emissions and similar hydrocarbon emissions. Possibly because of its branched molecular structure, the isobutanol blends yielded slightly higher soot emissions than the n-butanol blends.

Lim et al. [24], in a principally modeling based study, optimized a dual-fuel, dual-injection RCCI strategy of iso-octane (intake port) then normal heptane (direct in-cylinder) to achieve 21 bar gIMEP operation with low engine-out NO_x, soot, and carbon monoxide emissions. Using the same fuels in a similar RCCI model, Klos and Kokjohn [25] found that injection timing had significant tradeoffs between engine efficiency, emissions, and combustion instability. A near top dead center (TDC) injection timing significantly reduced combustion instability, while the emissions and efficiency dropped to near conventional direct injection levels. More advanced phasing was even more stable, but produced high peak pressure rise rates, more NO_x, and lower

efficiency due to increased heat transfer losses. Splitter [26], using a dual-fuel RCCI approach with gasoline and ethanol, found that unburned hydrocarbon (UHC) emissions were a function of piston crevice and squish geometry, and carbon monoxide (CO) emissions were sensitive to local equivalence ratio. They predicted that by maintaining lean conditions with sufficient premixed fuel, simultaneous reductions in heat transfer and incomplete combustion could be achieved.

While RCCI has been studied quite extensively, there is a need for more research especially at high load conditions. Due to the heavy-duty nature of most military engines, it is of particular interest to research the opportunities that RCCI type combustion may increase an engine's maximum load and increase power without a soot penalty. One objective of this study was to apply a RCCI type approach (although in a simpler mechanical military context engine) to extend the high load limit of a diesel research engine based on exhaust opacity. It should be noted that commonly RCCI is performed with dual-fuel and high levels of EGR with a flexible fuel injection system. In this study, the engine is naturally aspirated and mechanical injection is used.

In a previous study by the authors [27] a single-fuel PPC approach was explored with a minority fraction of jet fuel injected into the intake manifold (retro-fit type application), while the majority fraction of jet fuel is delivered directly to the combustion chamber near the start of combustion (SOC). It was seen that as CR increased the soot threshold equivalence ratio decreased for conventional diesel combustion, however this afforded an increased opportunity for higher levels of port-injected fuel leading to power increases from 5 to 23% as CR increased from 14 to 21.5. PPC allowed for these power increases (defined by a threshold opacity level of 3%) due to smaller particles (and lower overall number of particles) in the exhaust that influence measured opacity less significantly than larger and more numerous conventional diesel combustion exhaust particulates. Carbon monoxide levels at the higher PPC driven power levels were only modestly higher, although NO_x was generally lower due to the overall enriched operation. In this study, engine control parameters were left at the base operational state. Manente et al. [28] used a single-fuel, PPC approach at 11.5 bar gIMEP and found that high octane number fuels, like ethanol and gasoline, were able to achieve a reduction of soot and NO_x with a lower amount of exhaust gas recirculation (EGR) as compared to diesel, with NO_x and UHCs generally below Euro 5 and 6 emissions standards.

Noehre et al. [29] mapped out the operating range for PPC using standard diesel fuel under a range of engine operating conditions including CR, EGR, speed, and boost at loads up to 15 bar gross indicated mean effective pressure (gIMEP). The combination of low compression ratio, high EGR, and engine operation close to stoichiometric conditions resulted in premixed and diffusion combustion that enabled simultaneous NO_x and soot reduction, though a decrease in efficiency and a rise in UHC and CO emissions were noted. They concluded that soot emission reduction was a result of good premixing, long ignition delay, and low local combustion temperature due to high EGR. Additionally, Dempsey et al. [30] found, for a PPC gasoline approach, that when the premixed fuel percentage was low, the NO_x emissions began to increase due to the existence of high equivalence ratio regions in the combustion chamber at the start of combustion.

Kalghatgi et al. [31] demonstrated the ability to use the soot reduction of PPC for high load extension. Using PPC with gasoline, they reached mean gIMEP of 15.95 bar and smoke opacity of 0.33%, while to attain the same level of smoke with diesel fuel, gIMEP had to be below 6.5 bar.

Zha et al. [32] compared a high cetane number (CN) jet propulsion '8' (JP-8) fuel with that of a low CN Sasol JP-8 under PPC mode. They found that low CN Sasol JP-8 exhibits longer ignition delay, lower premixed peak heat release and later combustion phasing compared to JP-8 combustion. Low CN Sasol JP-8 was shown to be a satisfactory under light-load condition, but with higher UHC emissions and lower fuel efficiency compared to high cetane JP-8 fuel. This investigation also seeks to expand on the work of Zha et al. in using jet fuel for a PPC approach.

Motivation

This study explores both a single and dual-fuel PPC approach in order to modify the nature of diesel engine combustion via some port delivered fuel. The fuel chosen for direct injection in this study was Navy jet propulsion '5' (JP-5), a fuel commonly used in many Navy and Marine Corps platforms due to its low volatility for safety purposes. From a military standpoint, it is an attractive choice to reduce the logistical burden of bringing multiple fuels to the battlefield (Department of Defense (DoD) Instruction 4140.25 [33]), via minimizing the type of fuel required. However, in this study, isobutanol and iso-octane (2,2,4-Trimethylpentane) along with JP-5 for reference, were studied in order to characterize the power gains that might be possible with a PPC approach where higher levels of port fuel are possible, especially with the low reactivity fuels. These fuels were chosen due to their high volatilities and low reactivities so as to increase in-cylinder fuel-air equivalence ratios without a loss of control due to autoignition before the direct injection event. An oxygenated fuel like isobutanol was chosen in the hope of reducing soot by adding oxygen to the cylinder periphery. Additionally, the high latent heat of evaporation of isobutanol could further help delay autoignition due to evaporative cooling effects.

Thus, the overall objective of this work is to focus on approaches to quantify PPC based (single and dual fuel) power improvements based on the high load extension of the exhaust sooting limit in a compression ignition engine as compared to conventional operation. In essence, this study seeks to initially explore whether or not a two-fuel PPC approach might be worth the additional fuel complexity when compared to a conventional diesel approach or a single-fuel PPC approach, based on power improvements from the high load extension of the exhaust sooting limit. Diesel engine exhaust soot visibility is tactically important in the military context.

Experimental Details

The engine used in this study was the Waukesha Cooperative Fuels Research (CFR) diesel engine (bore = 82.6 mm and stroke = 114.3 mm). This indirect injection (IDI) engine has mechanically variable compression ratio (CR from 8:1 to 36:1), injection amount, and injection timing. This engine does not have EGR. Reference [34] provides details and combustion chamber cross-sectional information on this diesel CFR engine. The single pintle injector has an orifice of nominally 1 mm with an injector opening pressure of 100 bar.

For this alternative combustion mode study, the IDI pre-chamber fuel injection system was supplemented with a Cole-Parmer peristaltic pump delivering fuel directly to the intake air pipe approximately 25 cm upstream of the cylinder head via a #13 (0.8 mm ID) flexible tubing. The tubing was inserted directly into the air stream – no attempt was made to create a spray. This port delivered fuel was delivered continuously during engine operation. The pump was calibrated to determine fuel volume as a function of time at various pump settings.

In-cylinder pressure was measured with a Kistler 6125C piezoelectric sensor (located in the pre-chamber of this IDI engine). Engine position was measured using a BEI dual channel shaft encoder coupled directly to the crankshaft. This CFR cetane rating engine operates at a fixed 900 rpm. An overall schematic can be seen in Figure 30, and further experimental details can be found in other references by the authors [35, 36]. A Meriam LFE and air-damping tank provided engine airflow measurement. An ETAS wide-band lambda meter provided overall air-fuel ratio information. Exhaust emissions (via a heated Unique Heated Products filter) were measured with an Infrared Industries FGA4000S analyzer. Opacity was measured with a Wager 6500 opacity meter. During some of the experiments, exhaust particulate measurements were acquired with a TSI 3090 EEPS unit via a Matter Engineering diluter. The engine was operated under stable laboratory conditions with ambient pressure intake and exhaust, and air temperature of nominally 20 °C.

Table 3. Waukesha DCFR Engine parameters

Engine type	Single cylinder, Indirect-injection, in-line cam-driven pump
Bore × Stroke	82.6 mm × 114 mm
Displacement	610 cc (0.610 L)
Speed	900 RPM fixed
Coolant	Water reflux, 100°C
Intake Air	Lab ambient, 20°C

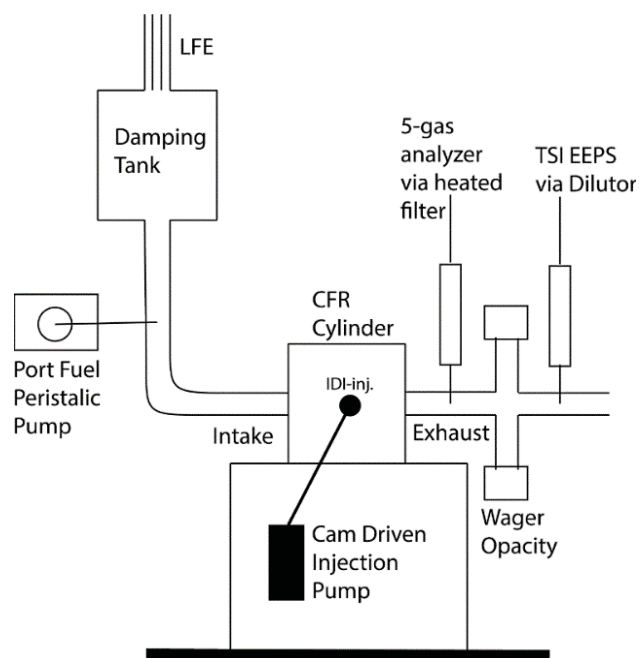


Figure 30. Experimental Setup

The general categories of components that are present in JP-5 were determined using gas chromatograph/mass spectrometry (GC/MS) analysis. An Agilent 5975 inert mass selective detector was used in conjunction with an Agilent 6890N Gas Chromatograph System under the same conditions utilized in previous studies [37]. The GC was equipped with a Zebron phase ZB-5ms® column (30 m, 0.25 mm, 0.25µm, 5% diphenyl-arylene/95% dimethyl polysiloxane) and operated at a helium flowrate of 1.5 mL/min. The compounds were separated using a temperature ramping program that started at 60.0 °C and climbed at a rate of 10 °C/min to 250 °C. An electron impact ionization (EI) method was used for the mass spectrometer with an m/z

scan from 30 to 600. To prevent the detector from being saturated, all fuel samples were diluted in n-hexane (1/100 dilution), and the mass spectral analysis of the fuel sample was delayed until the solvent peak had passed. The fragmentation patterns of peaks found at several retention times were compared to those found in the NIST database to determine potential compounds present.

For all of the fuels tested in this study, a range of physical properties was evaluated. The speed of sound and density of the fuels were measured using an Anton Paar DSA 5000 Density and Sound Analyzer. Certified reference standards were used to test the accuracy of these instruments, which were recalibrated if they failed testing. The bulk modulus at 20°C was calculated by multiplying the density by the speed of sound squared. The surface tension was measured using an axisymmetric drop shape analyzer (Kruss DS100), and distillation curve information was measured using a Grabner MINIDIS ADXpert automatic distillation apparatus that measures the distillation information per specifications of ASTM D7344-14 with a correlation to ASTM D86 when using small volumes. The other properties were determined using ASTM techniques specified in the military specifications.

Fuel Properties

Navy jet propulsion ‘5’ (JP-5) fuel, isobutanol, and iso-octane (2,2,4-Trimethylpentane) were used in this investigation. This JP-5 fuel has a moderately higher flash point (less volatile) as compared to conventional commercial jet fuels (aromatic content is 17%). Isobutanol and iso-octane were chosen due to their high volatilities and low reactivities so as to increase in-cylinder fuel-air equivalence ratios without a loss of control due to autoignition before the main cylinder injection event. An oxygenated fuel like isobutanol was chosen in the hope of reducing soot by adding oxygen to the cylinder periphery. Additionally, the high latent heat of evaporation of isobutanol could further help delay autoignition due to evaporative cooling effects. A summary of key properties is shown in Table 2. The reported cetane numbers from the literature were measured using ASTM D6890, and from the distillation curve, data are given T-10, T-50, and T-90.

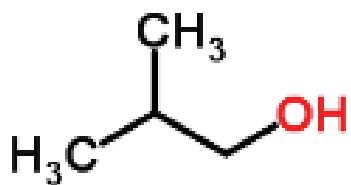


Figure 31. Structure of Isobutanol (isoB)

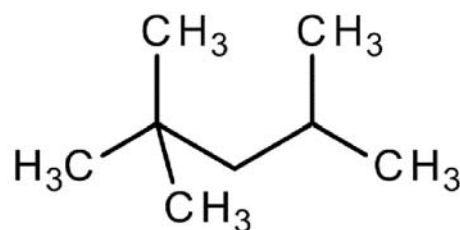


Figure 32. Structure of Iso-octane (iC8)

The chromatogram of JP-5 is shown in Figure 10. The analytical results are based on a NIST database search and comparing to original standards, the major peaks at 5.5, 7.1, 8.7, 10.1, 11.5, and 12.7 minutes are decane, undecane, dodecane, tridecane, tetradecane, and pentadecane, respectively. Other components identified by the NIST database include branched alkanes, aromatic compounds, alkylcyclohexanes, and indenes. The specific compounds were not confirmed by original standards.

Experimental Approach and Analysis

After the CFR engine was warmed up using conventional diesel combustion with Navy jet fuel JP-5, the desired compression ratio (CR) was adjusted followed by baseline conventional IDI data collection. Engine load was set starting in the mid-load range of the diesel CFR (~5 to 6 bar gross IMEP), then the fuel IDI amount (e.g. load) was incrementally increased up to the condition where an opacity of nominally 3% and then 6% was observed. At each operating point a real time fast heat release analysis was performed in order to maintain the crank angle location for 50% heat release (CAD50) nominally around 8°ATC with slight start of injection (SOI) adjustments (with a few exceptions noted below). This CAD50 point was chosen as it provides best torque timing in this engine at higher engine loads.

After the conventional IDI fuel-load sweep was performed, this process was repeated for PPC operation with isobutanol at two port injection levels and iso-octane at a single injection level. Next, this sweep was repeated for PPC operation with a single JP-5 port fuel injection level. Again, engine load was set starting in the mid-load range of the diesel CFR (~5 to 6 bar gross IMEP), then the fuel IDI amount (e.g. load) was incrementally increased up to the condition where an opacity of nominally 3% and then 6% was observed. Data were collected at each operating point. In a previous study by the authors [38], the maximum port fuel injection levels were empirically determined by the loss of SOI control, and then in this study the intake port fueling levels were reduced so that SOI directly affected combustion phasing. Another paper by the authors outlines this auto-ignition nature of JP-5 at various compression ratios [38]. At the higher compression ratios, CAD50 control was often lost before reaching 3% and 6% opacity, thus only the lowest isobutanol injection level is shown in the results at the higher CRs. Table 4 shows the nominal operational mass fuel fractions (port and IDI in-cylinder) used in this study. In general, the overall operating equivalence ratio was measured. Then in conjunction with the LFE measurement, total engine fuel flow was determined. The calibrated peristaltic pump provided direct intake port fuel flow rates. Thus, the in-cylinder fuel injection quantity was analytically determined.

At each stabilized operating point, twenty five seconds of in-cylinder pressure data were collected at a 50 kHz sampling frequency using MATLAB data acquisition software and National Instruments hardware. A conventional single zone heat release analysis approach was then applied to the data as outlined in references [39, 40]. As the combustion pressure was measured in the engine's pre-chamber, the indicated metrics may be modestly different as that experienced on the piston surface.

Table 4. Experimental Matrix Overview: nominal fuel fractions for the different operating conditions.

	Intake Port Fuel	IDI Pre-Chamber Fuel
Part 2a	isobutanol (30%)	JP-5 (70%)
	isobutanol (50%)	JP-5 (50%)
Part 2b	JP-5 (15%)	JP-5 (85%)
	iso-octane (30%)	JP-5 (70%)

Results and Analysis

Part 2a: Isobutanol PPC with Conv. JP-5 IDI

Figure 33 compares the exhaust opacity of two isobutanol PPC fueling levels to conventional ‘base’ IDI operation at CR14, 16.5, 18.5, and 21.5. Two isobutanol port injection levels, denoted ‘isoB 30%’ and ‘isoB 50%’, correspond to mass flow rates of 88 mg/s and 150 mg/s, respectively. Baseline operation was achieved with only JP-5 Conv. IDI. The lower isobutanol level was chosen to be similar to iso-octane PPC described in Part 2b. A higher level (without bulk compression autoignition) was also chosen in order to evaluate the effects with 50% fuel addition from isobutanol to the intake manifold. At CR 21.5, the higher PPC fueling level could not be achieved without the loss of control over CAD50 by changing the SOI of the IDI event or without the onset of early autoignition with respect to combustion phasing (limited by pre-ignition of the fuel-primed mixture from port injection during the compression stroke). In essence, CAD50 could not be kept nominally around 8°ATC with SOI adjustments, thus these cases are not shown on their respective graphs. On the x-axis, gIMEP increases directly with the amount of fuel injected for fixed engine speed. A 3% exhaust opacity threshold was defined as the point of maximum engine load, such that engine soot is too great for further enriched operation. At CR14, conventional ‘base’ operation reaches this 3% opacity threshold at approximately 8 bar gIMEP. Both cases of PPC operation do not reach this 3% threshold until a higher engine load, 8.6 bar, because more fuel was able to be added via intake port injection without a significant increase in exhaust soot. Thus, in this sense PPC can be seen to extend the high load limit of compression ignition (CI) engine operation. At CR16.5, this extension is less well defined for the ‘isoB 50%’ port fueling case, but shows even greater improvement with the ‘isoB 30%’ case. While Conv. IDI reaches this threshold near 8.3 bar gIMEP, the ‘isoB 30%’ case reaches 3% opacity at 8.9 bar. For CR18.5 and 21.5, high load extension is not apparent in the PPC cases, and perhaps even causes the opposite effect, reaching 3% opacity at the same or lower loads than Conv. IDI.

As seen in Figure 33, the ‘isoB 30%’ isobutanol PPC fueling case shows the same or better improvement in high load extension as the ‘isoB 50%’ PPC case. When also taking into account the greater fuel consumption, the ‘isoB 30%’ case is thus considered the best PPC case in this isobutanol investigation. It is believed that the higher PPC case is causing fuel over-enrichment in the cylinder periphery, creating fuel rich regions that are commonly associated with soot, and thereby failing to reduce exhaust opacity capable of extending the engine high load limit.

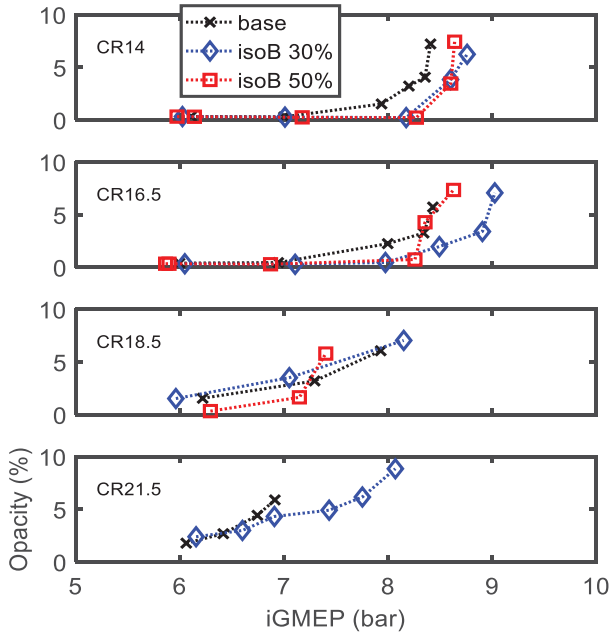


Figure 33. Exhaust opacity as a function of engine load at four compression ratios (CR) and two isobutanol port injection levels (with JP-5 Conv. IDI) in comparison to baseline JP-5 Conv. IDI.

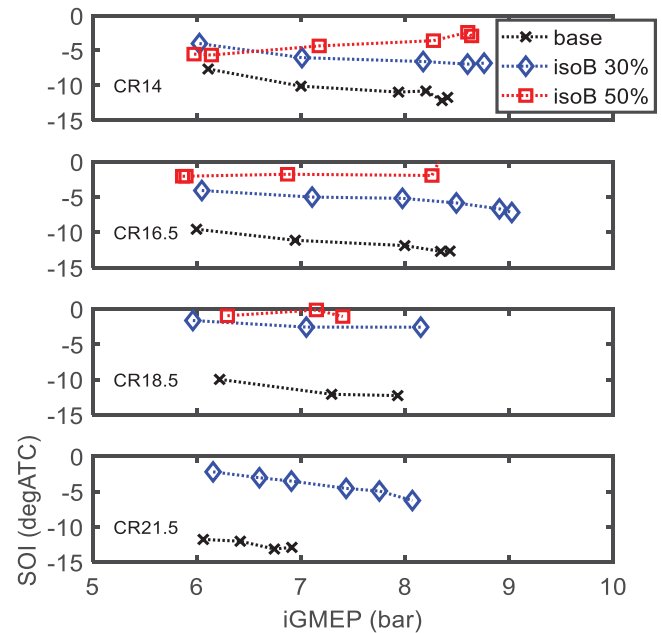


Figure 34. Start of Injection (SOI) as a function of engine load at four compression ratios (CR) with the two isobutanol PPC port injection levels in comparison to baseline JP-5 Conv. IDI.

Figure 34 shows a gradual advance (more negative) in start of ignition (SOI) for all cases with increasing gIMEP such that CAD50 was kept nominally constant for optimal torque-combustion phasing. For Conv. IDI with JP-5, the ‘base’ case, SOI was advanced as CR increased because Ignition Delay (IGD) shortens with increased compression (shown in next figure), leading to less fuel-air premixing and longer overall burn durations as will be seen later. SOI advanced for increasing CRs due to the elevated in-cylinder pressures causing an advance in the start of combustion (SOC). The ‘isoB 30% and 50%’ PPC isobutanol cases show more delayed-retarded SOI as compared to the ‘base’ operation due to intake port fuel-air premixing.

Ignition Delay (IGD) in Figure 35 was defined as the difference between SOI and CAD10 (SOC). In general, IGD shortens with increasing CR as expected. For Conv. IDI operation, IGD was initially slightly faster (smaller crank angle degrees) at the lowest CR14 as compared to the PPC cases. This effect is likely due to the importance of evaporative cooling with the isobutanol PPC cases leading to charge cooling and longer IGDs. At CR16.5, both the base conventional IDI IGDs and isobutanol PPC cases have similar IGDs across the load range. At the two highest CRs, the isobutanol PPC cases have shorter IGDs. As will be seen, at these higher CRs, the port-injected isobutanol is starting to exhibit two-stage ignition behavior at higher CRs due to the

greater combustion chamber charge density. Shorter IGD, in general, led to less time for premixing and thus longer 10-90% burn durations as will be shown later.

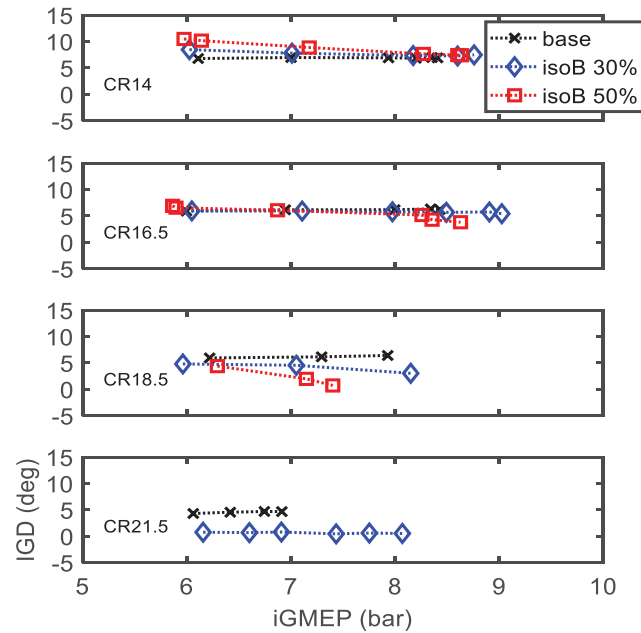


Figure 35. Ignition Delay (IGD) as a function of engine load at four compression ratios (CR) with PPC isobutanol port injection levels in comparison to baseline JP-5 Conv. IDI.

Figure 36, Figure 37 and Figure 38 show in-cylinder pressure and instantaneous heat release rate at CR14, 16.5, and 18.5 respectively for the different isobutanol port fuel injection levels as well as the ‘base’ Conv. IDI case. All of these operating points are from engine operation with nominally 3% exhaust opacity. SOI is shown on the x-axis as an open symbol. As more isobutanol is added to the intake port airflow, longer IGDs result, and a stronger maximum rate of heat release is observed. It is believed that the addition of increased intake port fuel displaced air in the cylinder periphery, leading to a different flame behavior as compared to the base. Furthermore, increasing intake port fuel leads to increasing overall charge evaporative cooling, further delaying ignition. The overall fueling amounts in Figure 37 correspond to equivalence ratios (ϕ) of 0.86 (base), 0.91 (isoB 30%), 0.85 (isoB 50%). The overall fueling amounts in Figure 38 correspond to a ϕ of 0.75 (base), 0.72 (isoB 30%), and 0.68 (isoB 50%).

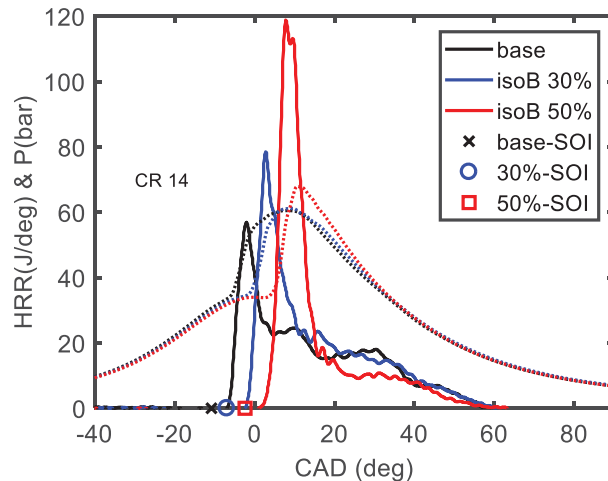


Figure 36. Heat Release Rate and in-cylinder Pressure as a function of piston position (CAD) at a CR of 14 with two isobutanol port injection levels (with JP-5 Conv. IDI) in comparison to baseline JP-5 Conv. IDI. TDC is shown at 0 CAD.

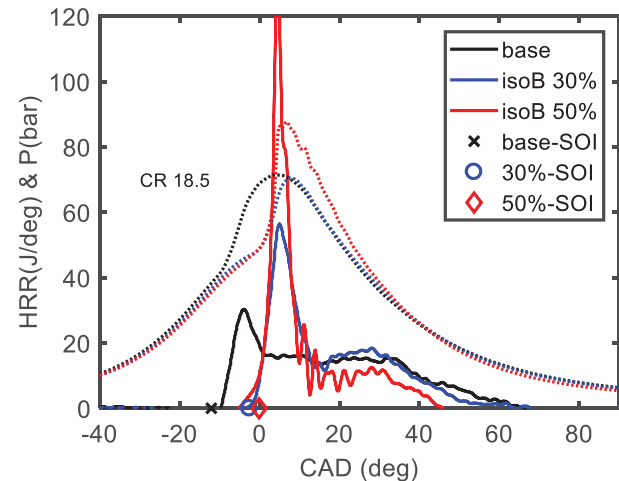


Figure 37. Heat Release Rate and in-cylinder Pressure as a function of piston position (CAD) at a CR of 18.5 with two isobutanol port injection levels (with JP-5 Conv. IDI) in comparison to baseline JP-5 Conv. IDI. TDC is shown at 0 CAD.

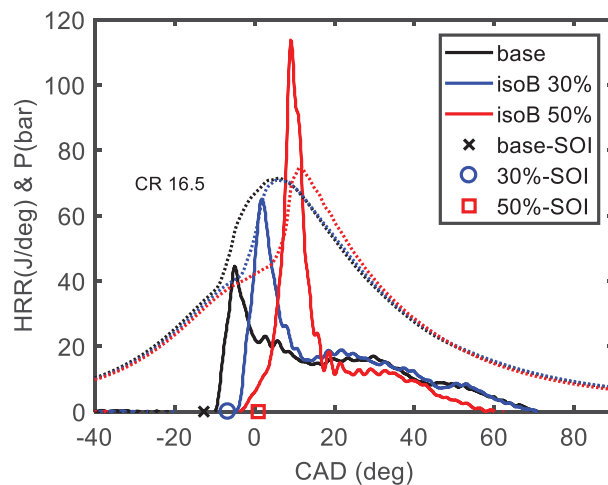


Figure 38. Heat Release Rate and in-cylinder Pressure as a function of piston position (CAD) at a CR of 16.5 with two isobutanol port injection levels (with JP-5 Conv. IDI) in comparison to baseline JP-5 Conv. IDI. TDC is shown at 0 CAD.

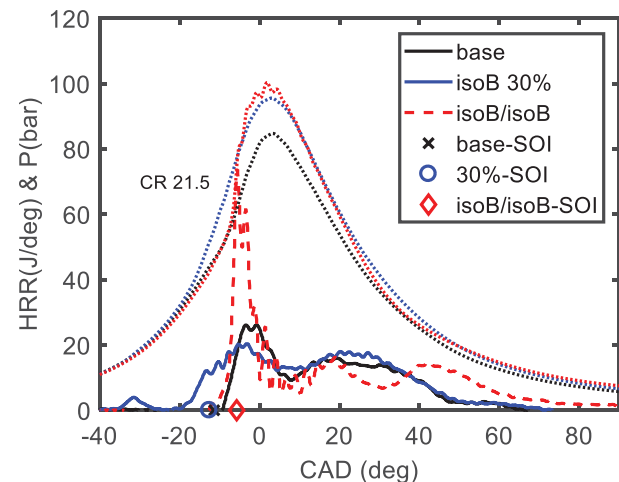


Figure 39. Heat Release Rate and in-cylinder Pressure as a function of piston position (CAD) at a CR of 21.5 with the 'isoB 30%' isobutanol port injection level in both operating modes in comparison to baseline JP-5 Conv. IDI. TDC is shown at 0 CAD.

Figure 39 shows in-cylinder pressure and heat release rate at CR21.5 for the baseline Conv. IDI case (black data) with JP-5 fuel, also a 'isoB 30%' isobutanol PPC fueling case (with JP-5 IDI: blue data), and additionally a PPC isobutanol application for comparison (isobutanol port and isobutanol IDI). As seen above, increasing intake port fuel addition leads to increasing evaporative cooling, further delaying ignition. The baseline case yielded a 6.5 bar gIMEP (ϕ of 0.61), while the isobutanol PPC case was 7.8 bar gIMEP (ϕ of 0.77), and the isobutanol/isobutanol PPC case was 8.3 bar gIMEP (ϕ of 1.06). Thus it is notable that by only fueling with isobutanol (both intake port and directly into the combustion chamber),

stoichiometric and fuel rich conditions were achieved with no soot-opacity penalty relative to the other combustion modes. Additionally, Figure 39 shows that the ‘isoB 30%’ isobutanol PPC case exhibited early low temperature heat release prior to TDC in the premixed charge at this highest CR case tested.

Figure 40 shows the burn duration for all the isobutanol PPC port fueling levels across the CR range tested. In general, burn duration lengthens modestly with increasing levels of fueling, except at the highest CR21.5. As CR increased, IGD shortened which led to less time for premixing and thus longer burn durations for both combustion modes. Despite the port injection, PPC operating points displayed a wide range of burn durations. The authors hypothesize that some of the port injected charge was burning early (indicated by the negative IGD at the highest CR), thereby consuming oxygen in the direct injection flame region. This small oxygen deficiency led to a slower, diffusion-dominated flame for lower CRs and an almost instantaneous, HCCI-like combustion event for higher CRs.

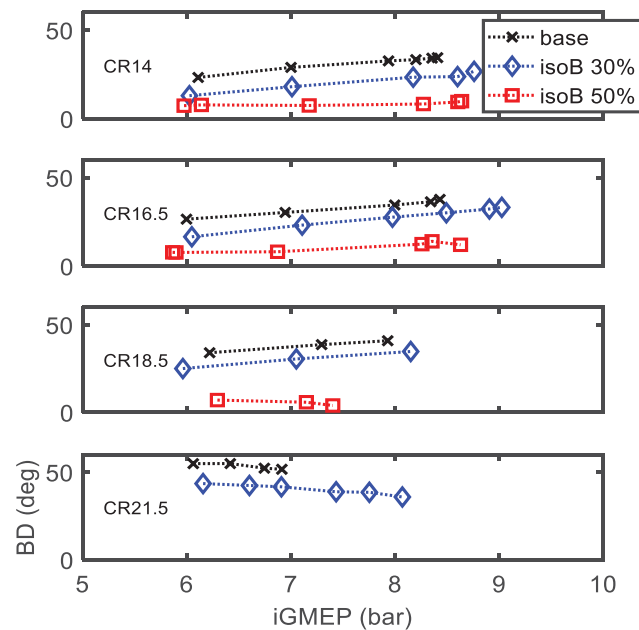


Figure 40. Burn Duration (BD) as a function of engine load at four compression ratios (CR) and three isobutanol port injection levels (with JP-5 Conv. IDI) in comparison to baseline JP-5 Conv. IDI.

Oxides of Nitrogen (NO_x), in Figure 41, show that exhaust NO_x concentration followed the characteristic curve for Conv. IDI operation as a function of load. NO_x peaked at mid-load due to the steady increase in combustion temperature and steady decrease in available oxygen as iGMEP increased. PPC isobutanol does not follow this trend, but rather increases with increasing load at high CR and falls off with increasing load at low CR. In general, PPC NO_x levels are moderately lower as compared to the base cases due to the increased overall fuel-air charge premixing.

Overall, as CR increases, exhaust NO_x concentrations decrease, despite the greater peak pressures at higher CRs. This effect has also been observed in other work by the authors using this engine [38]. The authors believe that at higher CRs in this CFR engine, the combustion ‘average’ in-cylinder equivalence ratio is overall richer at higher CRs. This is due to the shorter

IGDs at higher CRs, which lead to longer burn durations as seen in previous figures. It is believed that more diffusion control burning is leading to lower NO_x formation at higher CRs.

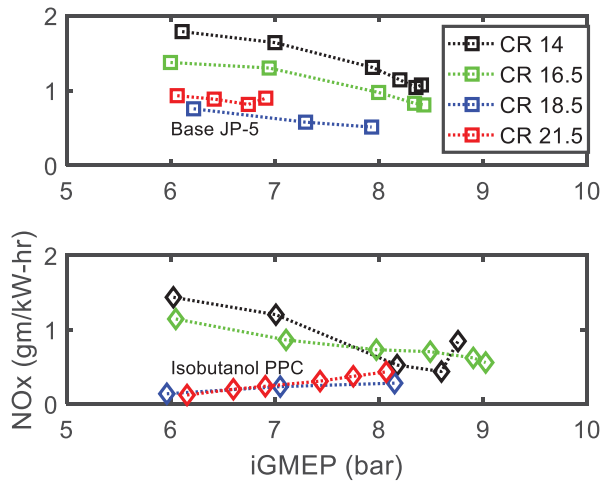


Figure 41. Oxides of Nitrogen (NO_x) as a function of engine load at four compression ratios (CR) with the 'isoB 30%' isobutanol PPC port injection level in comparison to baseline JP-5 Conv. IDI.

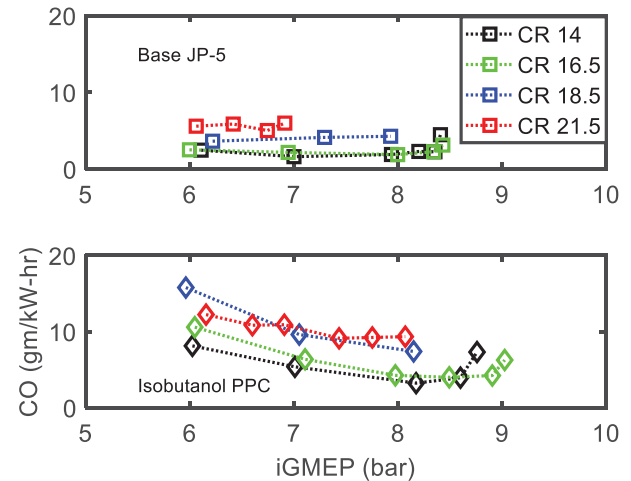


Figure 42. Carbon Monoxide (CO) as a function of engine load at four compression ratios (CR) with the 'isoB 30%' isobutanol PPC port injection level in comparison to baseline JP-5 Conv. IDI.

Exhaust carbon monoxide (CO) concentration in Figure 42 follows similar trends as exhaust opacity earlier. The port injection event of isobutanol PPC allowed for greater air-fuel mixing prior to SOC at CR 14 and 16.5, leading to more complete combustion and a reduction of CO at high loads. Time for mixing during Conventional IDI was limited due to engine speed. Port injection PPC allowed for mixing in the intake manifold and during the intake and compression strokes, leading to a more partially 'homogeneous' pre-mixed charge prior to the main injection event. CO was higher with CR 18.5 and 21.5, but these two operating points were shown to exhibit low NO_x in Figure 41.

Part 2b: Iso-octane PPC, jet JP-5 PPC, and Conv. base jet JP-5 IDI

In Part 2b of this study, port injected iso-octane PPC and port-injected JP-5 PPC (with the main in-cylinder IDI always using JP-5 fuel) are compared to baseline conventional IDI operation. Figures 43 through 54 show engine operation with iso-octane PPC at a single port fuel injection level, nominally 30% port fuel (by mass) and JP-5 PPC operation with a single JP-5 port fuel injection level (15% by mass). The single iso-octane port injection rate was 81 mg/s. The JP-5 port injection level rate was 41 mg/s.

Figure 43 compares these three fueling-combustion methods with respect to exhaust opacity. On the x-axis, gIMEP increases directly with the amount of fuel delivered for fixed engine speed. A 3% exhaust opacity threshold was defined as the point of maximum engine load, such that engine soot is too great for further enriched operation (although one more experimental operating point at nominally 6% exhaust opacity was performed). At CR14, all the combustion modes reach this threshold at approximately the same load, with iso-octane PPC showing modestly higher engine loads. However, at higher CRs, it can be seen that the JP-5 PPC and to even a greater extent the iC8-PPC cases extend the load limit before reaching the 3% threshold. This is because more fuel

was able to be added via port injection without a significant increase in exhaust soot. Thus, in this sense iC8 PPC can be seen to significantly extend the high load limit of compression ignition (CI) engine operation.

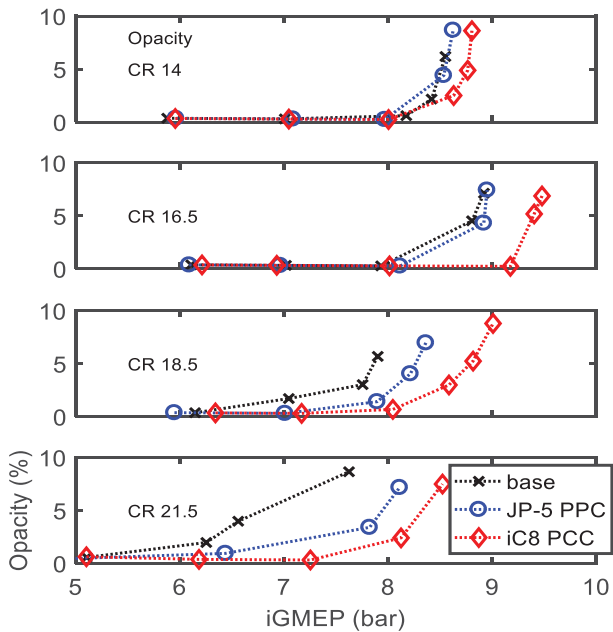


Figure 43. Exhaust opacity as a function of engine load at four compression ratios (CR) with JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI.

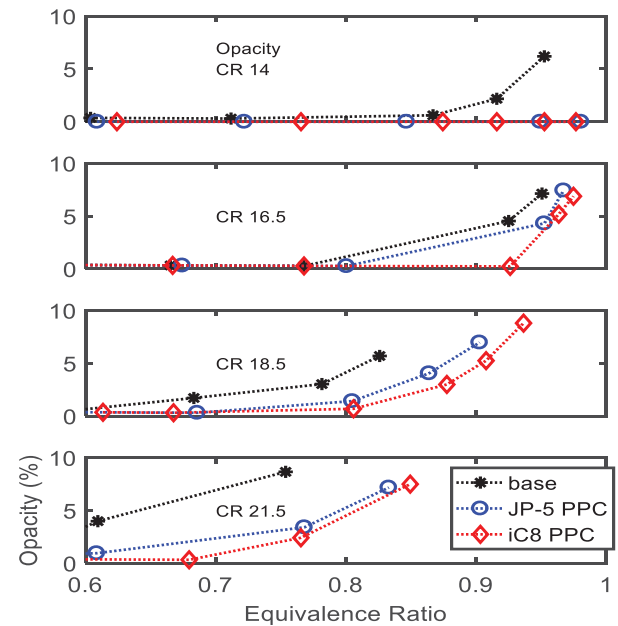


Figure 44. Opacity as a function of Fuel-air equivalence ratio (ϕ , phi) at four compression ratios (CR) with JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI.

Next, Figure 44 shows the operating overall equivalence ratio (ϕ) with the resulting exhaust opacity for the three combustion approaches. It can be seen that the baseline opacity consistently starts to rise at lower-leaner equivalence ratios as compared to JP-5 PPC and iso-octane PPC (iC8 PPC). This result is expected due to the partial fuel pre-mixing that these alternative combustion modes provide, leading to a leaner overall combustion event. The JP-5 PPC and iC8 PPC behaviors are similar; however, the iso-octane results show moderately lower opacity at a given overall equivalence ratio.

Figure 45, Figure 46, Figure 47, and Figure 48 show in-cylinder pressure and heat release rate at CR14, 16.5, 18.5, and 21.5 respectively for the two PPC and Conv. IDI cases. All of these cases are nominally at the 3% opacity operating condition. For all CRs, the JP-5 PPC exhibits negative temperature coefficient (NTC) behavior with early heat release followed by an endothermic period before the main phase heat release. This early heat release before SOI, distorts burn durations in Figure 41, making it look longer than expected. JP-5 has a relatively high cetane number (CN), but coupled with the low temperature heat release, the main injection event occurs in a warmer combustion chamber, leading to shorter IGD and less time for premixing of the main in-cylinder injection event, thus the main phase heat release peak (the premixed heat release phase) is lower than the base Conv. IDI. Because less premixing occurs coupled with lower initial heat release, JP-5 PPC shows primarily a diffusion controlled burn with more of the heat release near and after TDC (the diffusion heat release phase). However, as CR decreases, the in-

cylinder temperatures decrease and this peak heat release increases in magnitude for all combustion modes (increased pre-mixed combustion phase). Iso-octane, due to its low cetane number, allows for more time for mixing prior to SOC, thus for all CRs there is a large initial phase heat release near TDC and less diffusion heat release after TDC during the expansion stroke.

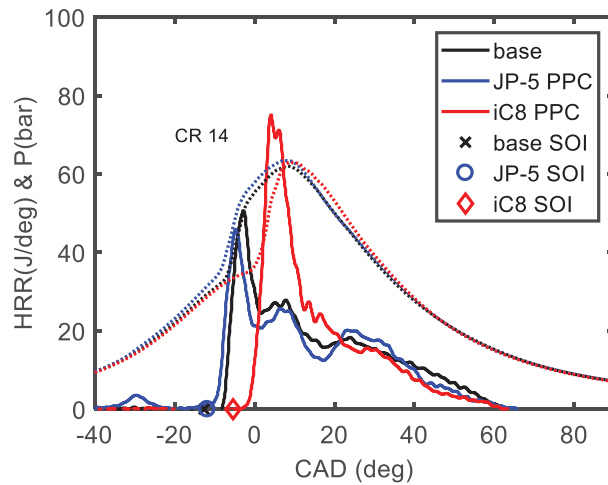


Figure 45. Heat Release Rate and in-cylinder Pressure as a function of piston position (CAD) at a CR of 14 with JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI. TDC is shown at 0 CAD.

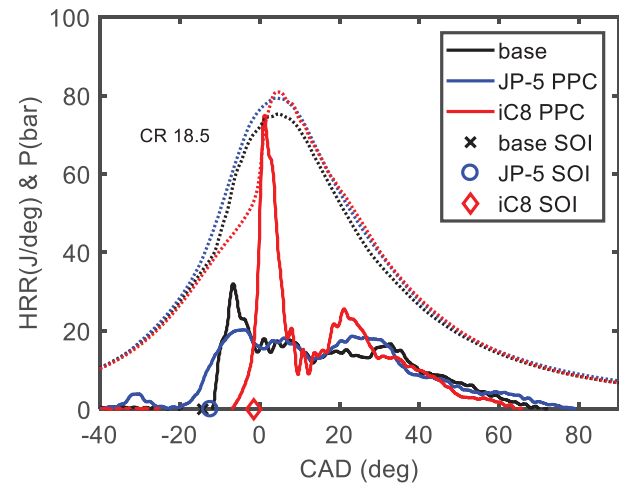


Figure 46. Heat Release Rate and in-cylinder Pressure as a function of piston position (CAD) at a CR of 18.5 with JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI. TDC is shown at 0 CAD.

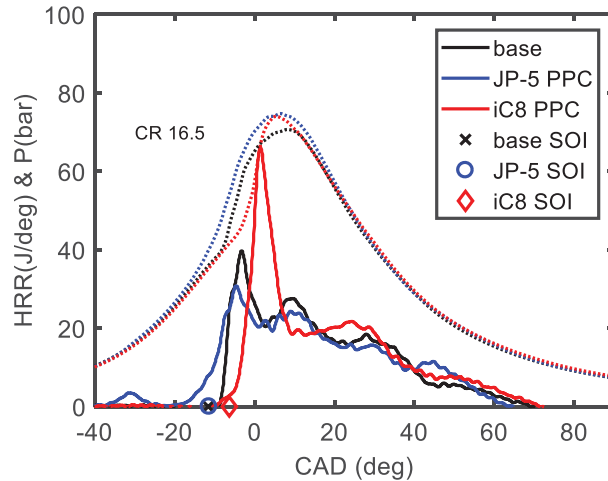


Figure 47. Heat Release Rate and in-cylinder Pressure as a function of piston position (CAD) at a CR of 16.5 with JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI. TDC is shown at 0 CAD.

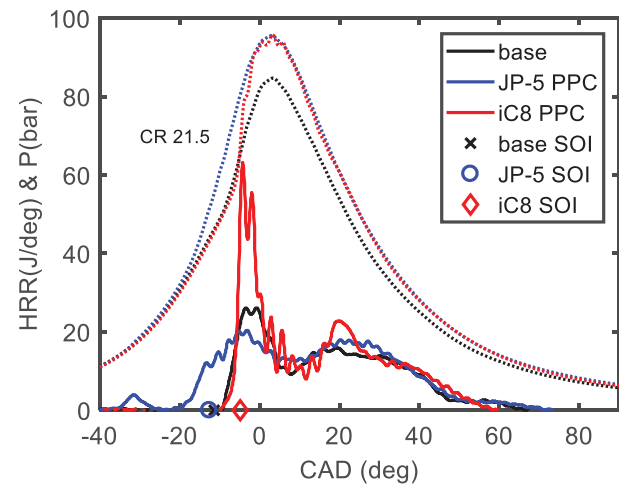


Figure 48. Heat Release Rate and in-cylinder Pressure as a function of piston position (CAD) at a CR of 21.5 with JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI. TDC is shown at 0 CAD.

Figure 49 shows a gradual advance in start of ignition (SOI) for all cases with increasing gIMEP such that CAD₅₀ was kept nominally constant 8°ATC for optimal torque-combustion phasing. For Conv. IDI with JP-5 'base' case, SOI was advanced slightly as CR increased because IGD shortens with increased compression (shown in next figure), leading to less fuel-air premixing

and longer overall burn durations as will be seen later. SOI advanced for increasing CRs due to the elevated in-cylinder pressures causing an advance in the start of combustion (SOC). The PPC iso-octane case shows more delayed-retarded SOI as compared to the ‘base’ operation due to intake port fuel-air premixing, while the jet PPC case falls in between the other combustion modes due to the premixing of the higher cetane number port injected fuel.

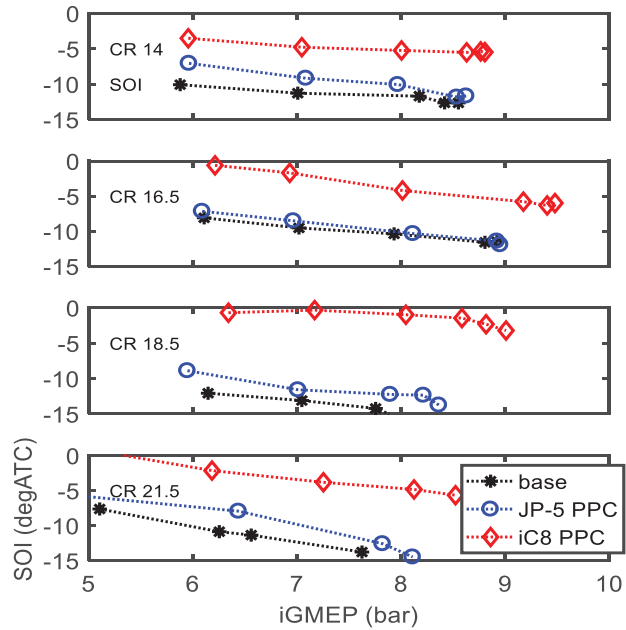


Figure 49. Start of Injection (SOI) as a function of engine load at four compression ratios (CR) with JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI.

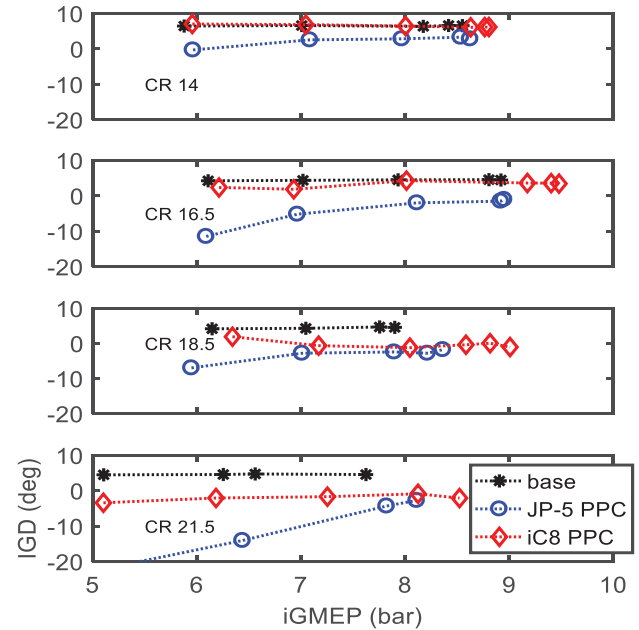


Figure 50. Ignition Delay (IGD) as a function of engine load at four compression ratios (CR) with JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI.

Ignition Delay (IGD) in Figure 50 was defined as the difference between SOI and CAD10 (SOC). In general, IGD shortens with increasing CR as expected, but this is especially true for the JP-5 PPC case. This is because as shown in Figures 35 through 38, the early heat release drastically advances SOC, particularly at CR 21.5. Thus in the cases shown with negative IGD, SOC occurs before SOI. At CR14 and 16.5, both the base Conv. IDI IGDs and iso-octane PPC cases have similar IGDs across the load range. At CR18.5 and 21.5 iso-octane PPC has slightly shorter and negative IGD. This can be attributed to air-fuel premixing after port injection. Thus, unlike for isobutanol PPC, iso-octane PPC and JP-5 PPC show very little evaporative cooling effects to lengthen IGD. Shorter IGD, in general, led to less time for premixing and thus longer 10-90% burn durations as will be shown next.

Figure 51 shows the burn duration for all three combustion modes across the CR range tested on a single graphic pane so that the relative differences across the combustion modes are clearer. The CRs are denoted as CR14 dotted line, CR16.5 dash-dot line, CR18.5 dashed line, and CR21.5 solid line. As CR increased, IGD shortened which led to less time for premixing and thus longer burn durations for both base and PPC combustion modes. Again the longer JP-5 PPC burn duration compared to the other modes can be attributed to the early heat release, which ‘triggered’ the very early SOC (including some ‘negative’ IGDs), thus leading to longer overall burn durations. The authors hypothesize that some of the port-injected charge was burning early,

thereby consuming oxygen in the direct injection flame region. This small oxygen deficiency led to a slower, diffusion-dominated flame for lower CRs and an almost instantaneous, HCCI-like combustion event for higher CRs. Despite the port injection, PPC operating points displayed burn durations moderately less than base conventional IDI. The increased premixing without an early heat release led to a faster combustion event without preignition, as shown by IGD near zero. Overall burn duration increased with increasing gIMEP. This can be attributed to the fact that there is greater fuel mass in the cylinder, and thus it takes longer to burn.

Exhaust carbon monoxide (CO) emissions results shown in Figure 52 are slightly elevated for the higher load PPC modes of combustion. At lighter loads, as characteristic of more homogeneous compression ignition combustion, the CO levels increase relative to conventional diesel combustion. This is due to some port-injected fuel at the periphery of the combustion chamber that is only partially burned during the combustion event. CO follows similar trends as exhaust opacity in Figure 53. CO changes very little with increasing CR, and reaches a minimum approaching max-load. This is because, as shown in Figure 54, near overall stoichiometric conditions are reached at the highest loads achieved in this study where the cylinder becomes starved of oxygen and incomplete combustion occurs.

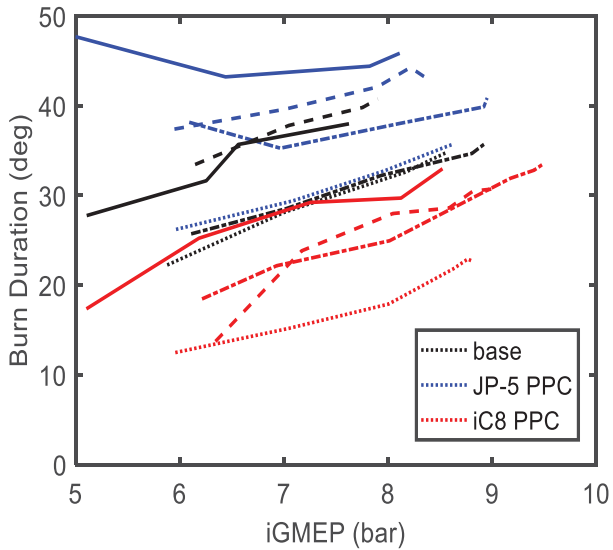


Figure 51. Burn Duration (BD) as a function of engine load at four compression ratios (CR) with JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI. The CRs are denoted as CR14 dotted line, CR16.5 dash-dot line, CR18.5 dashed line, and CR21.5 solid line.

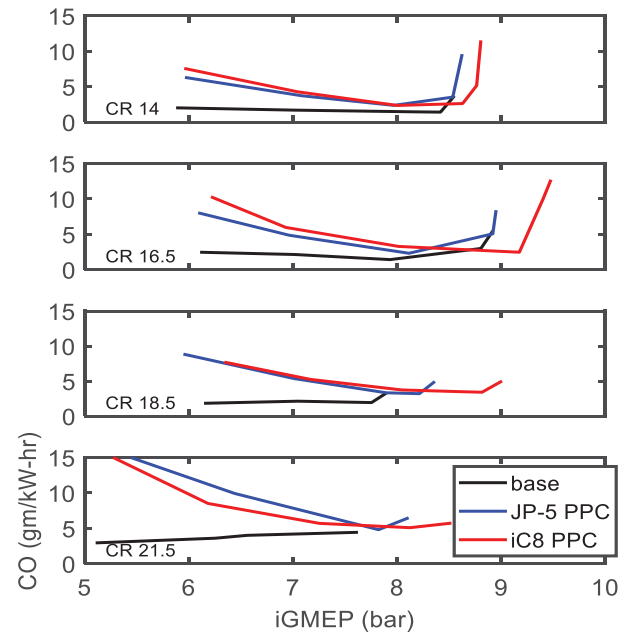


Figure 52. Carbon Monoxide (CO) as a function of engine load at four compression ratios (CR) with JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI.

Oxides of Nitrogen (NO_x), in Figure 53, show that all three combustion modes have similar NO_x levels across CRs and load range. NO_x stayed relatively constant with increasing load and decreased in magnitude with increasing CR. Again, decreasing NO_x with increasing CR is counterintuitive, but has been observed in other work by the authors using this engine [38]. Overall, as CR increases, exhaust NO_x concentrations decrease. As described above, the authors believe that decreasing IGD and associated longer burn durations lead to this behavior.

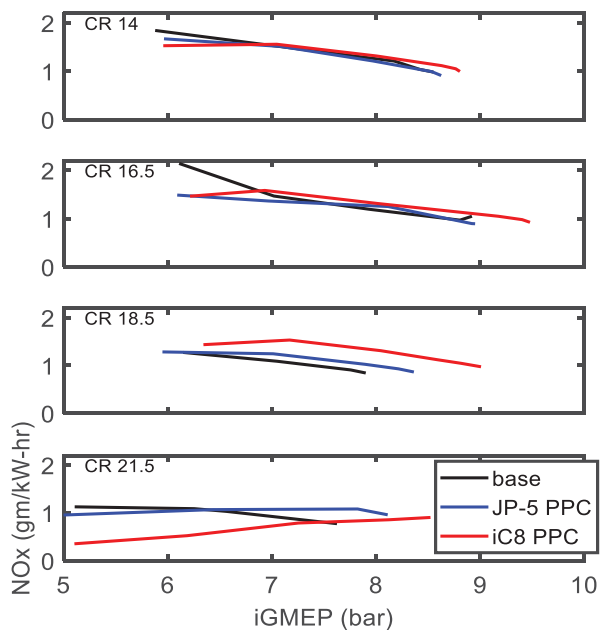


Figure 53. Oxides of Nitrogen (NO_x) as a function of engine load at four compression ratios (CR) with JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI.

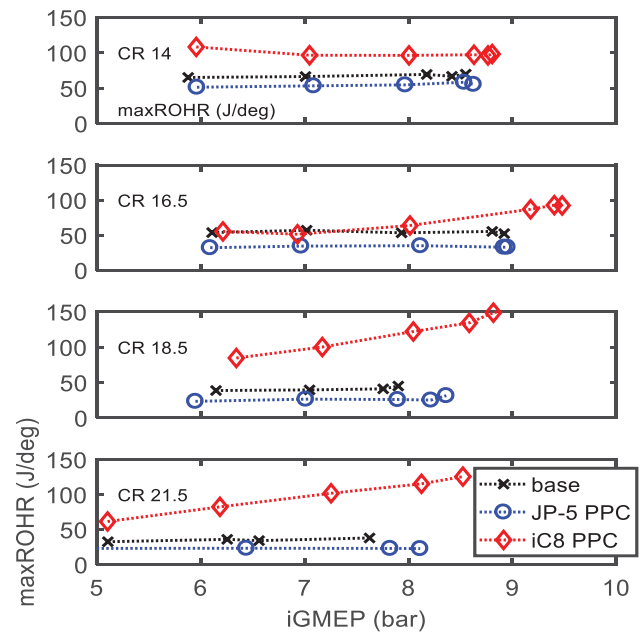


Figure 54. Maximum Rate of Heat Release (maxROHR) as a function of engine load at four compression ratios (CR) with JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI.

Figure 54 shows the maximum rate of heat release (maxROHR) for the three combustion modes across the load and CR ranges tested. For all CRs, the JP-5 PPC exhibits a negative temperature coefficient, then slightly endothermic heat release before the main phase heat release.

Thus, the heat release of JP-5 PPC is more spread out (e.g. longer overall burn durations) than the other modes. Because less premixing occurs and there is less initial heat release, it shows the characteristics of a diffusion controlled burn with more of the heat release near and after TDC (the diffusion heat release phase). However, as CR decreases, the in-cylinder pressures decrease and this peak-max heat release metric increases in magnitude for all combustion modes. Iso-octane, due to its low cetane number, allows for more time for mixing prior to SOC, thus for all CRs there is a large initial phase heat release near TDC and less diffusion heat release after TDC during the expansion stroke. Thus, the peak ROHR for iso-octane PPC is greatest due to the highest level of premixing with this combustion mode, and the magnitude of this increases with increasing CR due to the reduced combustion chamber volume. Because CAD50 was fixed, maxROHR for Conv. IDI stays relatively constant as CR increases and across the load range due to longer burn durations at these higher Conv. IDI CRs.

Figure 55 shows the exhaust soot-particulate concentration (numbers of particles per volume) as a function of particle size. Due to instrument saturation limits, these data were collected at high load just near or below the minimum detection limit of the opacity meter. The integrated results for each curve are converted to a total mass concentration assuming spherical carbon particles. Two cases are run for both Conv. IDI as well as JP-5 PPC at a CR14, with equivalence ratios of 0.85 and 0.89. The 0.89 case with Conv. IDI just exceeded the minimum opacity threshold

(~1%). The two JP-5 PPC operating points were expected to be well below the minimum opacity detection level. It is apparent that PPC reduces the number of soot particles and particle size compared to the Conv. IDI case for constant fuel-air equivalence ratio.

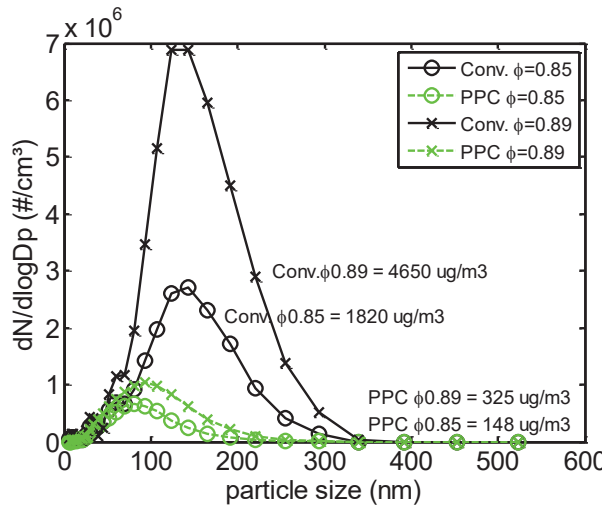


Figure 55. Exhaust particulate-soot concentration as a function of particle size.

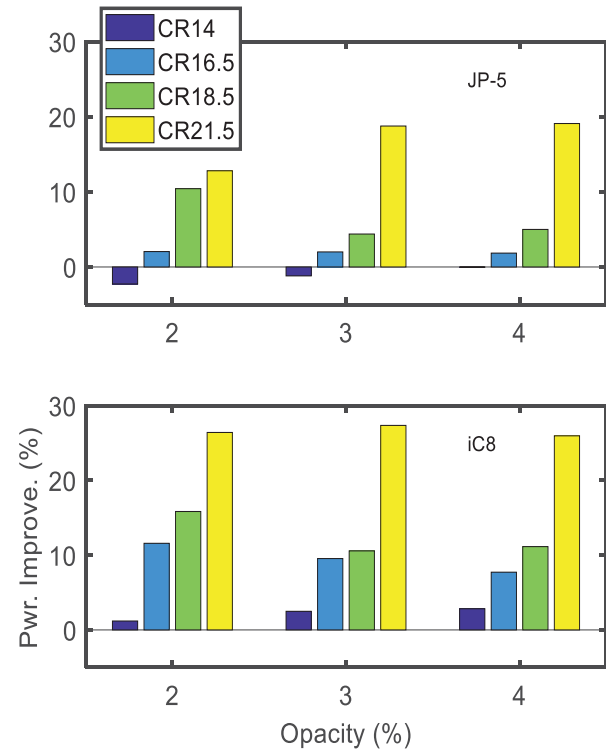


Figure 56. Power improvement (%) of JP-5 PPC and iso-octane PPC operating modes in comparison to baseline JP-5 Conv. IDI at four compression ratios (CR).

Finally, Figure 56 illustrates the key finding of this study, relative torque/power improvements with JP-5 PPC and iC8 PPC. The CRs are denoted as CR14 dark blue, CR16.5 light blue, CR18.5 green, and CR21.5 yellow. More fuel was able to be added using JP-5 PPC leading to increased torque levels without increased soot (upper figure). The lower figure shows even greater improvements with iC8 PPC, especially at the higher compression ratios because the 3% exhaust opacity soot limit was delayed until higher engine loads compared to base conventional IDI operation. At fixed engine speed, this torque improvement translated into a moderate power improvement over base conventional IDI operation. Power increased from -2% to 27% at the highest CR. Figure 56 shows that overall, iso-octane performed better in increasing power than JP-5, primarily due to the early heat release of JP-5.

Conclusions

This study applied a PPC advanced combustion approach using a minority fraction of port fuel injection in order to extend the diesel engine high load limit based on exhaust soot opacity. Isobutanol, iso-octane, and jet JP-5 fuel were used for port injection, while jet JP-5 fuel was the only fuel used for in-cylinder injection. It was seen that the conventional IDI soot limit occurred at leaner overall maximum fuel-air equivalence ratios as CR was increased. This afforded a greater opportunity to increase port injected fuel levels with increased CR. Thus, power levels

were then able to increase from -2 to 27% (at CR 21.5) over conventional diesel combustion by applying port injected companion fuel without a soot opacity penalty. Iso-octane provided the best improvements. PPC allowed for these power increases (defined by a threshold opacity level of 3%) due to smaller particles (and lower overall number of particles) in the exhaust that influence measured opacity less significantly than larger and more numerous conventional diesel combustion exhaust particulates. Carbon monoxide levels at the higher driven power levels were only modestly higher for the advanced combustion modes, although NO_x was generally lower due to the overall enriched operation.

Part 3: Power Improvement with a Yanmar Engine-Generator

Summary

An advanced combustion mode, Partially Premixed Combustion (PPC), approach was applied in a Yanmar L100V6-L single cylinder diesel engine gen-set (electrical generator set with engine) under high operating loads in an effort to increase maximum power. Direct in-cylinder fueling was conventional Navy Jet Fuel (JP-5). Port injected fuel consisted also of JP-5, however, additional low reactivity fuels were also explored, neat isooctane (iC8) and iso-butanol. PPC (both single and dual fuel) modes are alternative advanced combustion approaches that generally result in lower engine-out soot. In this study, these modes were accomplished with port fuel injection through the intake manifold while the majority fraction of JP-5 fuel was delivered directly to the combustion chamber near the start of combustion (SOC). Fast in-cylinder pressure based combustion metrics were acquired along with exhaust soot opacity.

Results show that engine Indicated Gross Mean Effective Pressure (iGMEP) from 7.5 to 8.5 bar could be attained with PPC combustion as compared to only 7 bar with conventional operation. This led to power levels of 9.3 kW to 11.3 kW compared to 8.5 kW at conventional operation (gen-set rated for 6.7 kW) without a soot penalty. In general, JP-5/JP-5 (in-cylinder/port fuel) PPC led to the lowest power increase due to early combustion initiation. JP-5/iC8 provided for the highest power improvement due to the low reactivity port injected iC8 fuel, which resulted in later start and bulk combustion as compared to conventional operation and the other PPC modes. Iso-butanol PPC provided moderate power improvements.

Literature Review

Portable and independent electric power generation is a core need of the military, particularly when operating in remote and isolated environments. Due to the power needs of modern technology and weapons systems, the demands of electric power continue to grow in these operational environments. A rapid and cost effective solution to meeting this need may be found in applying a retrofit approach to existing diesel generators in order to increase engine specific power. However, diesel engine power is limited by soot formation, an indicator of locally high fuel-air equivalence ratio, due to incomplete mixing of the air and fuel at high engine loads. More fuel cannot be delivered and reacted in the combustion chamber since the allotted diffusion time is very small at high engine RPM, and oxygen is limited at high engine loads. An alternative advanced combustion mode called partially premixed combustion (PPC, also known as Partially Premixed Compression Ignition, PPCI, and Premixed Charge Compression Ignition, PCCI) delivers a portion of the fuel to the cylinder via the engine air intake system, which

generally leads to lower engine-out particulate. This approach has the potential to lead to increased specific power at the same soot-opacity level.

As it relates to Part 3 of this study and engine-generator power, normal butanol and isobutanol have been studied as promising biofuels to use for advanced combustion modes due to prolonged ignition delay, low cetane number, and high volatility fuel characteristics. Cheng et al. [41] achieved PPC with butanol-diesel blends, showing soot emissions could be reduced by 70%, while nitrogen oxide concentration (NO_x) increased at low loads. Leermakers et al. [42] showed extremely high soot reduction with moderate butanol-diesel blends compared to diesel only operation using a PPC approach. Zheng et al. [43] studied n-butanol, isobutanol, and octanol (2-ethylhexanol) under conventional diesel operation. They found that due to their oxygen content, these fuels produced much lower emissions of soot and carbon monoxide than were achieved with diesel fuel alone.

In a previous study by the authors [44] a single-fuel PPC approach was explored with a minority fraction of jet fuel injected into the intake manifold (retro-fit type application), while the majority fraction of jet fuel is delivered directly to the combustion chamber near the start of combustion (SOC). This afforded an opportunity for high levels of port-injected fuel leading to power increases from 5 to 23% as CR increased from 14 to 21.5. PPC allowed for these power increases due to smaller particles (and lower overall number of particles) in the exhaust that influence measured opacity less significantly than larger and more numerous conventional diesel exhaust particulates.

Manente et al. [45] used a single-fuel, PPC approach at 11.5 bar gIMEP and found that high octane number fuels, like ethanol and gasoline, were able to achieve a reduction of soot and NO_x with a lower amount of exhaust gas recirculation (EGR) as compared to diesel. Noehre et al. [46] mapped out the operating range for PPC using standard diesel fuel under a range of engine operating conditions including CR, EGR, speed, and boost at loads up to 15 bar gross indicated mean effective pressure (gIMEP). They concluded that soot emission reduction was a result of good premixing, long ignition delay, and low local combustion temperature due to high EGR. Kalghatgi et al. [47] demonstrated the ability to use the soot reduction of PPC for high load extension. Using PPC with gasoline, they reached mean gIMEP of 15.95 bar and smoke opacity of 0.33%, while attaining the same level of smoke with diesel fuel, gIMEP had to be below 6.5 bar.

While PPC has been studied quite extensively, there is a need for more research especially at high load conditions. This study explores both a single and dual-fuel PPC approach in order to modify the nature of diesel engine combustion via some port delivered fuel. The fuel chosen for direct injection in this study was Navy jet propulsion '5' (JP-5), a fuel commonly used in many Navy and Marine Corps platforms due to its low volatility for safety purposes. From a military standpoint, it is an attractive choice to reduce the significant logistical burden of bringing multiple fuels to the battlefield (Department of Defense (DoD) Directive 4140.43, the 'One Fuel Forward' initiative [48]). However, in this study, isobutanol and iso-octane along with JP-5 for reference, were studied in order to characterize the power gains that might be possible with PPC port injection. In essence, this study seeks to explore whether or not a port injection PPC approach is a viable option for increasing diesel generator power with various fuel combinations.

Experimental Setup: Engine Details and Fuels

The engine used in this study was a Yanmar L100V6-L four-stroke single-cylinder diesel engine rated at 6.7 kW at 3600 RPM with specifications listed in Table 5. This engine is naturally aspirated and utilizes a Bosch Unit Pump System with a nominal mechanical injector opening pressure of 200 bar and nominally fixed start-of-injection (SOI) timing at 14 crank angle degrees (CAD) before top dead center (BTC). To attain the PPC alternative combustion mode, port injection was accomplished with a Cole-Parmer peristaltic pump continuously delivering fuel directly to the intake manifold immediately upstream of the cylinder head. The engine was coupled to a Central Maine Diesel S16W-150 generator controlled by an Inter-Lok control unit using constant speed control. The engine was outfitted with a fast Kistler in-cylinder (6125C) piezoelectric pressure sensor as well as a high-pressure injection line sensor (4067C2000) located near the inlet to the fuel injector. Engine position was measured using a BEI dual channel shaft encoder coupled directly to the crankshaft. An ETAS wide-band lambda meter provided overall air-fuel ratio information. Exhaust opacity was measured with a Wager 6500 opacity meter.

Navy Jet Propulsion ‘5’ (JP-5) fuel, isobutanol, and iso-octane (2,2,4-Trimethylpentane) were used in this investigation. For all of the fuels tested in this study, a range of physical properties were evaluated in previous work [44]. The general categories of components that are present in JP-5 were determined using gas chromatograph/mass spectrometry (GC/MS) analysis in these same studies. This JP-5 fuel has a moderately higher flash point (less volatile) as compared to conventional commercial jet fuels (aromatic content is 17%). The analytical results of JP-5 chromatography correspond principally to the normal alkanes: decane, undecane, dodecane, tridecane, tetradecane, and pentadecane. Isobutanol and iso-octane were chosen due to their high volatilities and low reactivities so as to increase in-cylinder fuel-air equivalence ratios without a loss of control due to autoignition before the main cylinder injection event. An oxygenated fuel, isobutanol was chosen in the hope of reducing soot by adding oxygen to the cylinder periphery. Additionally, the high latent heat of evaporation of isobutanol could further help delay autoignition due to evaporative cooling effects. A summary of key fuel properties is shown in Table 7.



Figure 57: Yanmar L100V6

Table 5. Engine specifications and details

Engine type	Yanmar L100V6-L
Bore × Stroke	86 mm × 75 mm
Displacement	435 cc (0.435 L)
Compression ratio	20.6
Combustion chamber	Re-entrant bowl, single cylinder
Net power rating	6.7 kW @ 3600 RPM

Table 6. Experimental Matrix

Port Fuel	DI Fuel	Power loads (kW)
(0%)	JP-5 (100%)	4, 5.5
JP-5 (22%)	JP-5 (78%)	5.5
iC8 (22% & 44%)	JP-5 (78% & 56%)	4, 5.5, 7
isoB (20% & 25%)	JP-5 (80% and 75%)	4, 5.5, 7

Table 7. Fuel Properties

Test	Units	JP-5 [44]	iC8 [13,14,15]	isoB [14]
Density 20°C	kg/L	0.80	0.68	0.80
Flash Point	°C	66	-4	30
Viscosity 40°C	cSt	1.389	0.724	3.71
Surface Tension	dyne/cm	28.7	18.7	22.4
Speed of sound 20°C	m/s	1319	1103.2	1205.1
Bulk Modulus 20°C	MPa	1397	842	1165
Cetane number	DCN	46	17-19	8.5
T-10	°C	190	99	110
T-50	°C	208	99	110
T-90	°C	240	99	110

Experimental Matrix

The Yanmar engine was warmed up using conventional diesel combustion with Navy jet fuel JP-5. Engine load was set at 4 kW, then increased to 5.5 kW and 7 kW. This was accomplished by adding these electric loads to the generator with standard electrical plug and socket. First, a Conv. DI maximum-load operating point was determined so that engine speed was stable at rated RPM. Next, PPC operation was performed with JP-5 at a single port injection level, iso-octane (iC8) at two port injection levels, and isobutanol (isoB) at three injection levels. Nominal fuel fractions and loads for the different operating conditions are shown in Table 6 (previous work by the authors showed these intake port fueling levels to function well in a Waukesha diesel CFR engine without significant compression stroke fuel energy release [44]). For the baseline DI and JP-5 PPC (JP-5 DI and JP-5 port) engine speed could be seen to diminish before the higher load cases, thus these load cases are not shown in Figures 58 - 65. DI JP-5 SOI was not adjusted in this study (stock configuration only) to maintain constant CAD50 despite the port injection event. At each stabilized operating point, twenty five seconds of pressure data were collected at a 50 kHz sampling frequency using MATLAB data acquisition software and National Instruments hardware. A single zone heat release analysis was then applied as outlined in references [17, 18].

Results and Discussion

In-cylinder instantaneous heat release rates (HRR) for each of the four combustion modes at the mid-load (5.5 kW) case are shown in Figure 58. Stronger maximum rate of heat release is observed with iC8 PPC and isoB PPC compared to Conv. DI operation. It is believed that the

addition of increased intake port fuel displaced air in the cylinder periphery, leading to a different flame behavior as compared to the base. The JP-5 PPC exhibits negative temperature coefficient (NTC) behavior with early heat release followed by an endothermic period before the main phase heat release. JP-5 has a relatively high cetane number (CN), but coupled with the low temperature heat release, the main injection event occurs in a warmer combustion chamber, leading to shorter IGD, shown in Figure 60, and less time for premixing of the main in-cylinder injection event, thus the main phase heat release peak (the premixed heat release phase) is lower than the Conv. DI. Because less premixing occurs coupled with lower initial heat release, JP-5 PPC shows primarily a diffusion-controlled burn with more of the heat release after TDC. IsoB and iC8 PPC, due to their lower cetane numbers, allow for more time for mixing prior to SOC, thus there is a large initial phase heat release near TDC and less diffusion heat release. Figure 59 shows the 50% heat release location (CAD50) as a function of engine load. Load is expressed as gross indicated mean effective pressure (gIMEP), which is the integrated work over the closed engine cycle period divided by the displaced volume of the cylinder. For all cases, CAD50 occurs within an acceptable window of 11-14 degrees after top dead center (ATC) for ideal combustion phasing. Notably CAD50 for the PPC cases is similar to that of Conv. DI, thus the comparison of power improvement is valid since combustion phasing is similar. Peak in-cylinder pressure is not shown but remained in an acceptable range of 65 to 70 bar across the range of power loads and combustion modes.

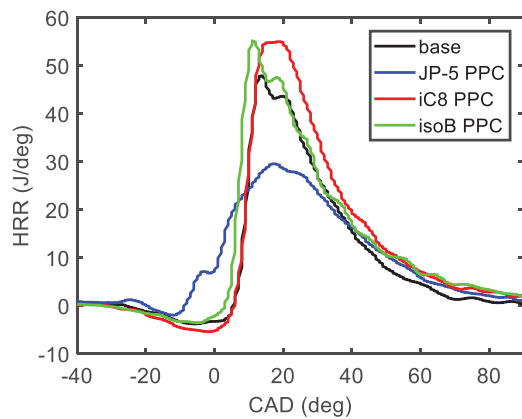


Figure 58. Instantaneous Heat Release Rate (HRR) as a function of piston position (0 CAD is TDC)

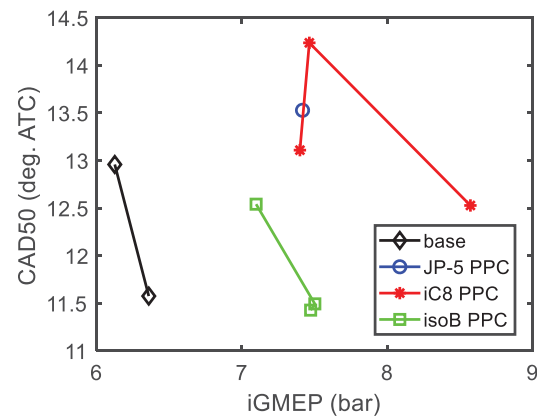


Figure 59. 50% Heat Release Location as a function of engine load and combustion mode

Ignition Delay (IGD), shown in Figure 60, was defined as the difference between SOI and start of combustion (SOC). SOC was defined as the engine crank angle degrees where 10% of the heat release has occurred (CAD10). Burn Duration was defined as the difference between CAD10 and CAD90 (the engine crank angle degrees where 90% of the heat release has occurred). CAD90 was relatively constant near 28 degrees ATC, except for in the case of JP-5 PPC, where it was 17 degrees ATC. This was again due to the early autoignition nature of premixed JP-5, which is also the cause of the short IGD in Figure 60. Another paper by the authors outlines this auto-ignition nature of JP-5 at various compression ratios [16]. Otherwise, the Conv. DI, iC8 PPC, and isoB PPC have similar IGDs across the load range. At fixed SOI, burn duration strongly correlates with IGD, since CAD50 (shown in Figure 59) is similar for all cases.

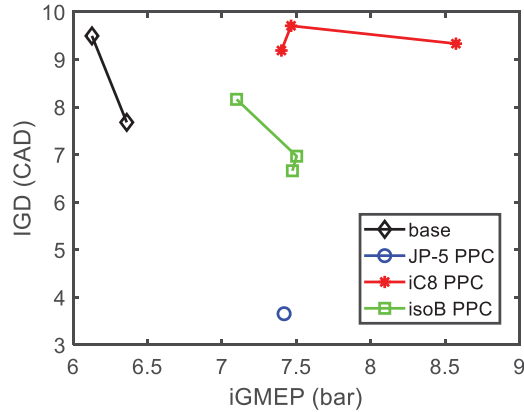


Figure 60. Ignition Delay as a function of engine load and combustion mode

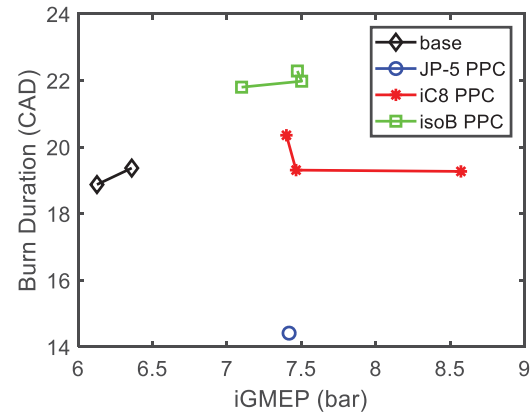


Figure 61. Burn Duration at varying engine loads

Figure 62 shows the overall fuel-air equivalence ratio (ϕ , ϕ) as a function of load. Figure 62 shows that ϕ increases linearly with load. The baseline case was fuel-load limited by a maximum fueling emission control device. With PPC, ϕ could be increased with a resulting increase in engine load. In Figure 63, exhaust opacity is shown as a function of load. Notably, for iC8 PPC and isoB PPC, the exhaust opacity remains near 0.5%, similar to Conv. DI despite the increase in fuel mass and resulting engine load. Thus for these cases, combustion is not soot-limited. The ability of PPC to extend the high load limit of Conv. DI by maintaining low exhaust soot, as measured by opacity, is the critical factor in power improvement for military engines. Load could be extended considerably, up to 8.5 bar for iC8 PPC and 7.5 bar for isoB PPC, compared to 6.5 bar for Conv. DI. Thus, PPC shows the potential to delay the soot limit for this engine. At fixed engine speed, this torque improvement translated into a moderate power improvement over base conventional IDI operation. Likely due to the unique nature of JP-5 combustion shown previously, exhaust opacity was considerably higher at 2%. However, in an emissions context, 2% opacity is still relatively low compared to previous studies by the authors [27].

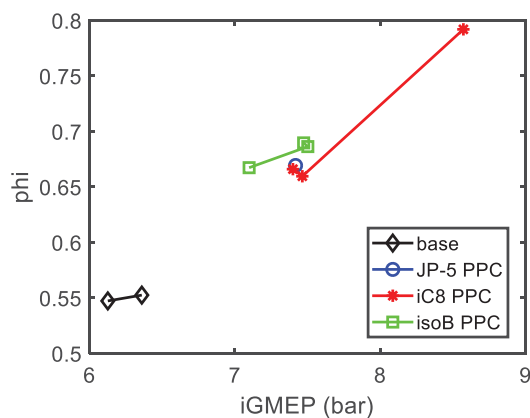


Figure 62. Fuel-Air Equivalence Ratio as a function of engine load

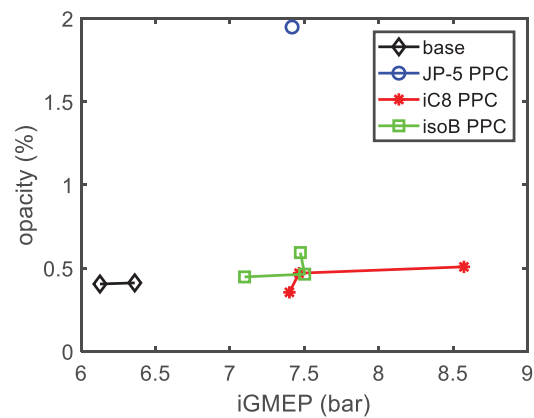


Figure 63. Exhaust Opacity at various engine loads and combustion modes

Figures 64 and 65 illustrate the key finding of this study, relative torque/power improvements with JP-5 PPC, iC8 PPC, and isoB PPC. At constant engine speed of 3600 RPM, indicated power was directly proportional to iGMEP, shown in Figure 64. More fuel could be added using

PPC leading to increased torque levels without increased soot. Figure 65 shows the comparison of the highest load case of the PPC mode compared to the highest load case of Conv. DI. The greatest improvements can be seen with iso-octane at nearly a 35% improvement. JP-5 and isobutanol show similar 15% improvements, although soot levels for JP-5 were significantly higher. Overall, iso-octane performed better in increasing power than JP-5, primarily due to the early heat release of JP-5.

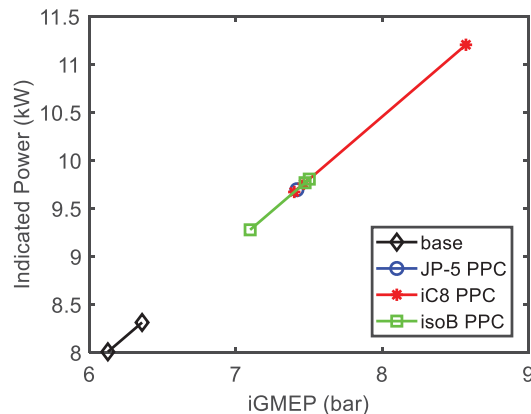


Figure 64. Indicated Power at increasing load, illustrating PPC high load extension at fixed engine speed

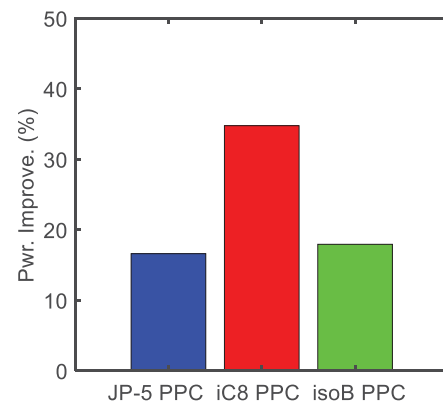


Figure 65. Power Improvement over baseline Conv. DI for each combustion mode at the high load limit

Conclusions

This study applied a PPC advanced combustion approach using port fuel injection in a Yanmar L100V6 single cylinder engine-generator in order to extend the diesel engine high load limit based on exhaust soot opacity evaluation. Isobutanol, iso-octane, and jet JP-5 fuel were used for port injection, while jet JP-5 fuel was the only fuel used for in-cylinder injection. It was seen that the Conv. DI soot limit occurred at leaner overall maximum fuel-air equivalence ratio, which afforded a greater opportunity to increase port injected fuel levels. Thus, power levels were then able to increase over conventional diesel combustion. Iso-octane provided the best improvements. Results show that engine Indicated Gross Mean Effective Pressure (iGMEP) from 7.5 to 8.5 bar could be attained with PPC combustion as compared to only 7 bar with conventional operation. This led to power levels of 9.3 kW to 11.3 kW compared to 8.5 kW at conventional operation without a soot penalty. In general, JP-5 PPC led to the lowest power increase due to early combustion initiation. JP-5/iC8 provided for the highest power improvement due to the low reactivity port injected iC8 fuel, which resulted in later start and bulk combustion as compared to conventional operation and the other PPC modes. Iso-butanol PPC provided moderate power improvements.

Part 4: Power Improvement Comparison in the HMMWV ‘Humvee’ Engine

Summary

An advanced combustion mode, Partially Premixed Combustion (PPC), approach was applied in the 6.5L turbocharged V8 HMMWV “Humvee” engine under high operating loads in an effort to increase maximum power. As the HMMWV has increased in weight significantly over the past two decades with armor and technology upgrades, there is a distinct need to increase engine power to maintain performance. PPC (both single and dual fuel) modes are alternative advanced

combustion approaches that generally result in lower engine-out soot. Because the military limits high load fueling based on exhaust soot, a PPC approach was applied to extend the high load limit of this engine. In this study, these modes were accomplished with port fuel injection through the intake runner while the majority fraction of JP-5 fuel was delivered directly to the combustion chamber near the start of combustion (SOC). Direct in-cylinder fueling was conventional JP-5, a Navy jet fuel with moderate reactivity and high flash point. Port injected fuel consisted also of JP-5, however, an additional low reactivity fuel, neat isooctane (iC8), was also explored. Fast in-cylinder pressure based combustion metrics were acquired along with exhaust soot opacity. Results show that engine Indicated Gross Mean Effective Pressure (iGMEP) of 12.9 and 13.0 bar could be attained with JP-5 and iC8 PPC, respectively, as compared to only 12.1 bar with conventional operation, without a soot penalty. This led to indicated power improvements over conventional operation of 7% and 8%, with JP-5 and iC8 fuel, respectively. Ignition delay and burn duration, measured from the in-cylinder injection event, decreased with PPC, and start of combustion advanced. In general, JP-5/JP-5 (in-cylinder/port fuel) and JP-5/iC8 PPC exhibited early heat release behavior, thus creating longer ‘true’ burn durations compared to conventional operation, leading to diffusion-controlled flames and less soot-reduction benefit than typically seen with PPC. This was likely a result of port injection occurring upstream of the turbocharger and a lack of intercooling that effectively preheated the premixed charge. While combustion phasing was advanced, other metrics, like peak pressure and peak pressure location, remained constant and within acceptable ranges.

Background

The Humvee is the High Mobility Multipurpose Wheeled Vehicle. There are at least 16 variants. It is operated by the US military as well as 49 allied nations and has been in service since 1984 with 281,000 built [49]. The unit cost is \$220,000 (2011) (up-armored) [49]. In that time, it has increased in weight from weight 5,200–5,900 lb. (2,359–2,676 kg) curb weight. At the start of the Operations Iraqi/Enduring Freedom, armor was also added, which adds another 1,000 lb. Therefore, the Humvee is much heavier than the engine was initially designed for. At max gross weight, the top speed is just over 60 mph [49]. The Humvee is partly being replaced by the Joint Light Tactical Vehicle (JLTV), but that replacement is not a one-for-one replacement [49]. Both the Army and Marine Corps will still maintain large Humvee fleets, comprising more than one-half of their light vehicle fleets. Therefore, the Humvee is expected to be in military service until 2050. Humvees will still serve as ambulances, while another approach is turning the Humvee into an autonomous ground vehicle.



Figure 66. AM General HMMWV ‘Humvee’

Engine Details

The Humvee engine is a 6.5L V8 turbocharged diesel. It is a robustly built engine; it has a compression ratio of 21.5 and can run on many different types of fuels. However, it has a fairly low power output for its size at 190 hp. Its combustion chamber is an indirect injection prechamber with counterclockwise swirl. It is mechanically fuel injected with a cam-driven injection pump. There is no exhaust gas recirculation (EGR). It can be turbocharged up to near 2 bar. However, the intake is not intercooled, which, as will be seen, created problems for control over combustion phasing.

Table 8. Humvee Engine details

Engine type	V8 Turbo Diesel
Bore × Stroke	86 mm × 75 mm
Displacement	6500 cc (6.5 L)
Compression ratio	21.5
Max power rating	142 kW (190 hp) @ 3400 RPM
Max torque rating	515 N-m (380 ft-lbf) @ 1700 RPM

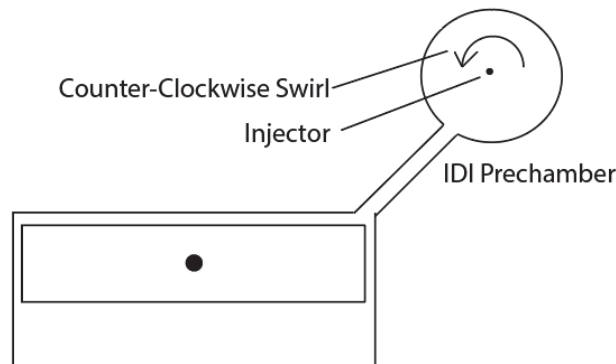


Figure 67. Humvee Combustion Chamber geometry

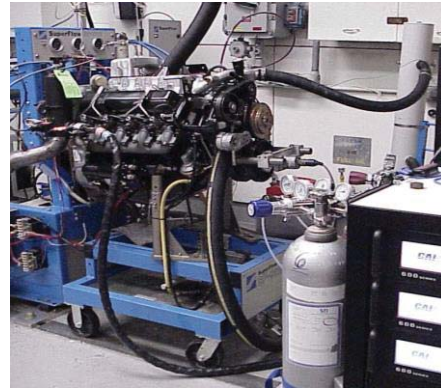


Figure 68. Humvee Engine test cell at the U.S. Naval Academy



Figure 69. Humvee Engine

Experimental Setup

Instantaneous in-cylinder pressure was measured with a Kistler/BEI piezoelectric pressure transducer in one cylinder, which allowed for the calculation heat release and rate of heat release using a single zone model. Real time data acquisition was achieved with NI hardware and MATLAB software. All measurements are shown as indicated measurements. The other key metric that was recorded was exhaust opacity as a measurement for soot, measured with a Wager Opacity meter.

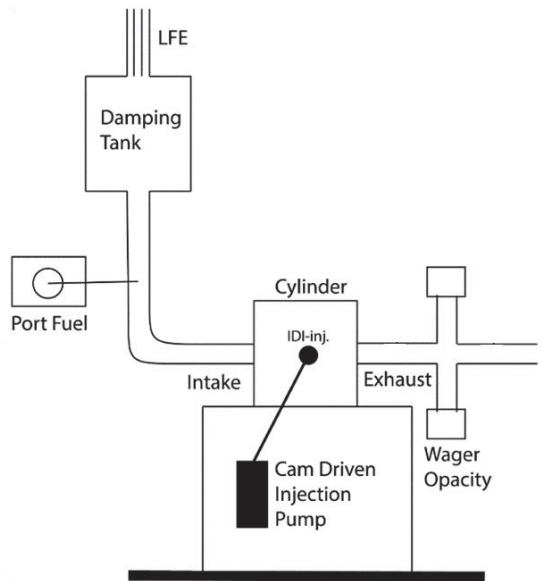


Figure 70. Experimental Setup

Table 9. Experimental matrix

Intake Port Fuel	IDI Pre-Chamber Fuel	Speed (RPM)
(0%)	JP-5 (100%)	2500, 3400
JP-5 (50%)	JP-5 (50%)	2500, 3400
iC8 (50%)	JP-5 (50%)	2500, 3400

PPC was achieved with an electronic fuel injector in the intake runner 25 cm upstream of the turbocharger. A controller, shown in Figures 71 and 72 was developed to control port fuel injection. An optical sensor was used to read the flywheel speed which went into an Arduino microcontroller, which then controlled fuel injection using script “interrupts,” so that the injector could be open a certain amount of time per rotation.

Maximum load fueling levels are shown in Table 9, based on 6% exhaust opacity. The goal of this study was to show that maximum power could be increased without a soot penalty by applying fuel to the engines intake air flow. This was achieved by setting a nominal maximum soot level at 6% opacity. Speeds of 2500 and 3400 RPM were tested. Baseline conventional operation was first recorded, then the port fuel percentage was increased with both JP-5 and iso-octane until the 6% opacity threshold was then reached.

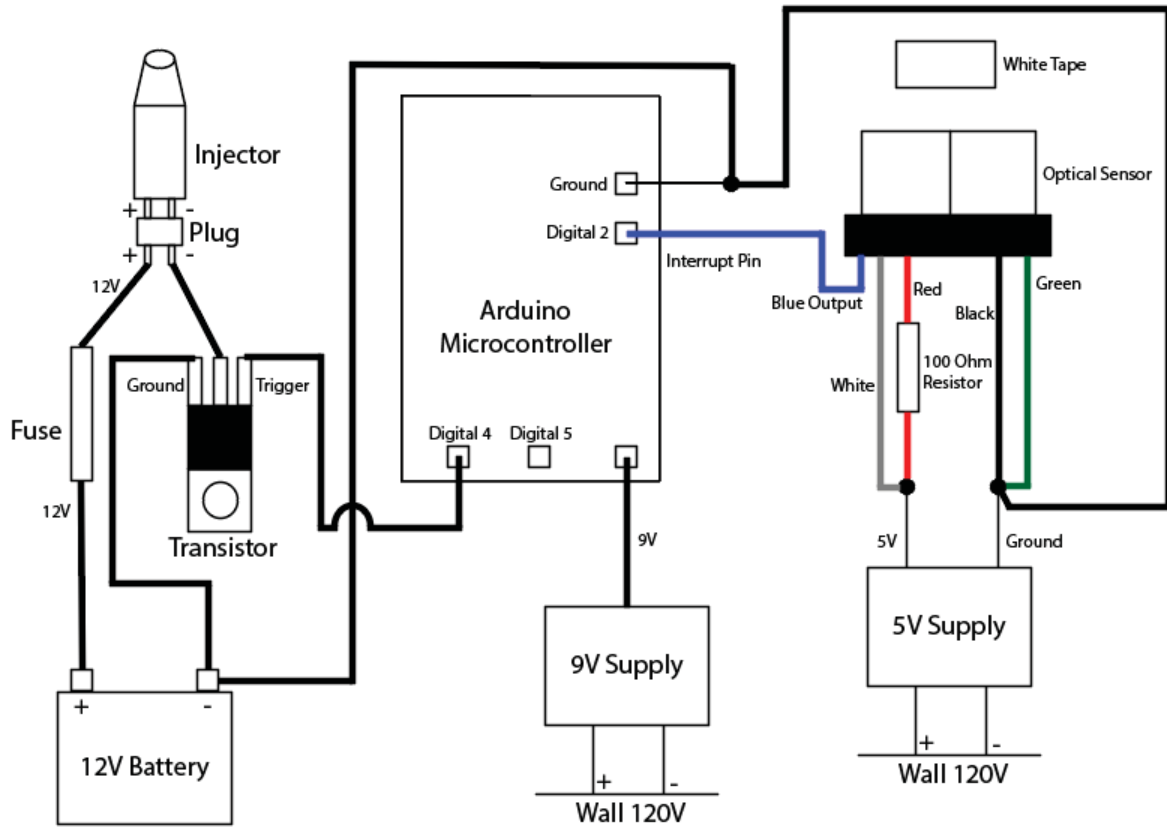


Figure 71. Controller setup for port fuel injection

```

int interruptPin=2; //needs to be D2 or 3
int outPin1=4;
int outPin2=5;
int trig;

void setup() {
  // put your setup code here, to run once:
  pinMode(interruptPin, INPUT);
  pinMode(outPin1,OUTPUT);
  pinMode(outPin2,OUTPUT);
  attachInterrupt(digitalPinToInterrupt(interruptPin),goFast,FALLING);
  trig=0;
  digitalWrite(outPin1,LOW);
  digitalWrite(outPin2,LOW);
  Serial.begin(9600);
}

void loop() {

  if (trig==1)
  {
    Serial.print("inject ");
    digitalWrite(outPin1,HIGH);
    digitalWrite(outPin2,HIGH);
    delay(10);
    digitalWrite(outPin1,LOW);
    delay(5);
    digitalWrite(outPin2,LOW);
    trig=0;
  }
}

void goFast(){
  trig=1;
}

```

Figure 72. Microcontroller script

Results and Analysis

Figures 73 and 74 show the pressure comparison to baseline using JP-5 in the port. Figure 75 shows these pressure traces overlayed. There are some differences in combustion phasing. Peak pressure and peak pressure location were similar and within an acceptable range. Start of combustion was defined as the 10% accumulated heat release location and burn duration as the difference in 10% and 90% heat release location. Ignition delay was the difference in start of injection of the IDI injection event and start of combustion. It was observed that ignition delay and burn duration decreased and SOC advanced. Initially a big power improvement from 330 ft lb to 390 ft lb was seen at 3400 RPM, but there was a soot penalty of 18%.

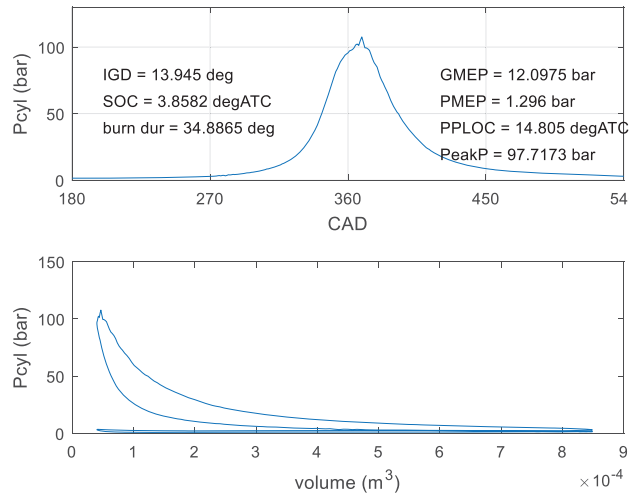


Figure 73. Maximum Baseline Power with JP-5

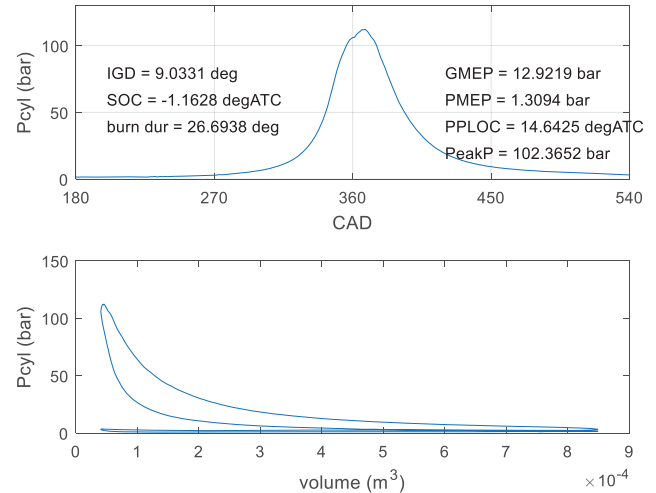


Figure 74. Maximum Baseline Power with PPC with JP-5

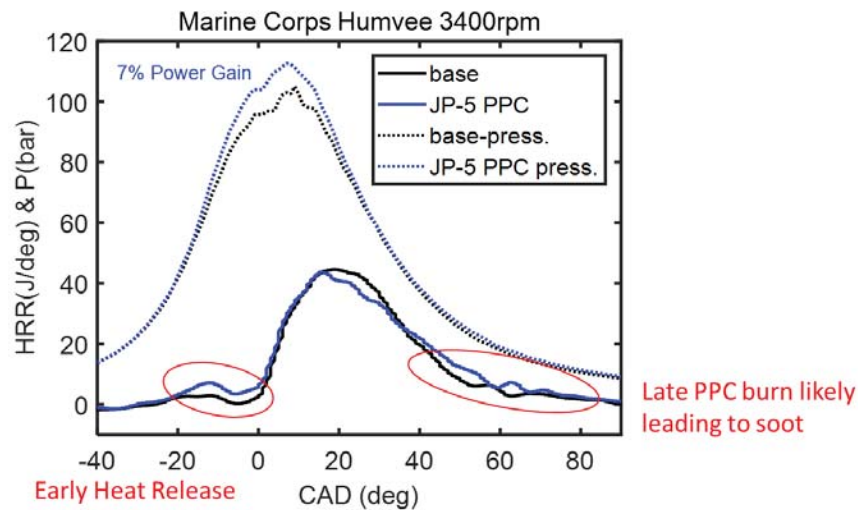


Figure 75. PPC and Conventional IDI Comparison with JP-5

These trends were a function of the fact that early heat release was seen with JP-5. Because the combustion phasing metrics were defined based on the second injection event, that is why SOC advanced and ignition delay actually goes negative. Burn duration was actually longer, but the burn actually started before SOI. This early heat release lead to a more diffusion controlled flame, which likely led to more soot than usually seen with PPC. The early heat release can likely be attributed to the fact that the port injection was upstream of the turbocharger, but not intercooled, so it was essentially heated before entering the combustion chamber. Overall there was still a 7% indicated power increase based on gIMEP at 3400 RPM.

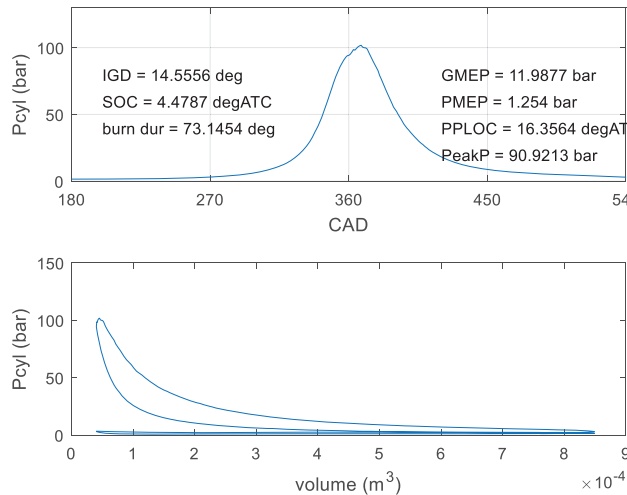


Figure 76. Maximum Baseline Power with iC8

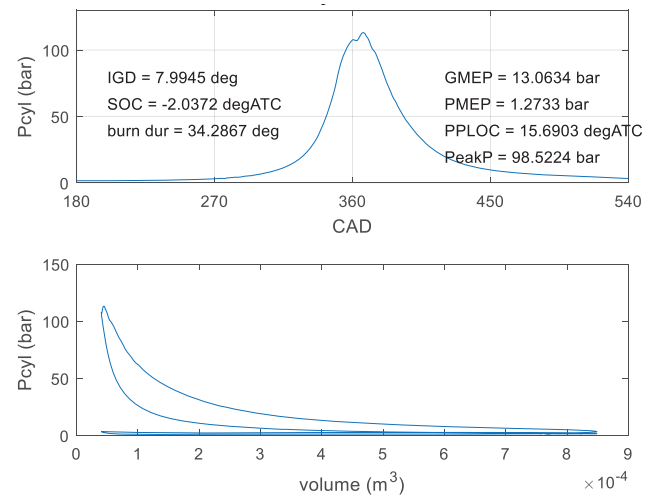


Figure 77. Maximum Baseline Power with PPC with iC8

Shown in Figures 76 and 77 is the maximum baseline power comparison of PPC and Conventional IDI with iso-octane. Because iso-octane is much lower cetane number and low reactivity, it was anticipated that there would be less early heat release. However, similar results to JP-5 were observed. Peak pressure and peak pressure location were similar and within an acceptable range. Ignition delay and burn duration decreased and SOC advanced.

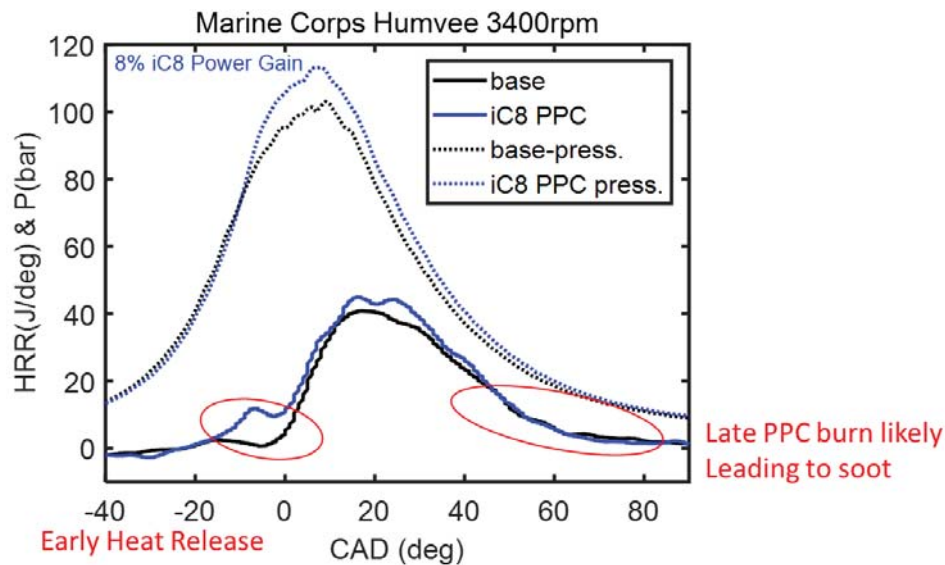


Figure 78. PPC and Conventional IDI Comparison with iC8

Figure 78 shows the early heat release with iso-octane with iso-octane. Overall an 8% indicated power increase based on gIMEP at 3400 RPM was seen.

Conclusions

Results show that engine Indicated Gross Mean Effective Pressure (iGMEP) of 12.9 and 13.0 bar could be attained with JP-5 and iC8 PPC, respectively, as compared to only 12.1 bar with conventional operation, without a soot penalty. This led to indicated power improvements over conventional operation of 7% and 8%, with JP-5 and iC8 fuel, respectively. Ignition delay and

burn duration, measured from the in-cylinder injection event, decreased with PPC, and start of combustion advanced. In general, JP-5/JP-5 (in-cylinder/port fuel) and JP-5/iC8 PPC exhibited early heat release behavior, thus creating longer ‘true’ burn durations compared to conventional operation, leading to diffusion-controlled flames and less soot-reduction benefit than typically seen with PPC. This was likely a result of port injection occurring upstream of the turbocharger and a lack of intercooling that effectively preheated the premixed charge. While combustion phasing was advanced, other metrics, like peak pressure and peak pressure location, remained constant and within acceptable ranges. Future work will include adjustments to reduce early heat release.

Overall Conclusions

In Part 1, this study applied a PPC approach with a minority fraction port fuel injection in order to extend the high load limit based on exhaust soot opacity in the Waukesha Diesel CFR engine. Jet fuel was used as the same fuel for both port and in-cylinder injection. It was seen that the Conventional IDI soot limit occurred at leaner overall maximum fuel-air equivalence ratios as CR was increased. This afforded a greater opportunity to increase port injected fuel levels with increased CR. It was thus seen that power levels were then able to increase from 5 to 23% (at CR 21.5) over conventional diesel combustion by applying port injected companion fuel (PPC) without a soot opacity penalty.

Part 2 of this study applied a PPC advanced combustion approach using a minority fraction of port fuel injection in order to extend the diesel engine high load limit based on exhaust soot opacity in the Waukesha Diesel CFR engine. Isobutanol, iso-octane, and jet JP-5 fuel were used for port injection, while jet JP-5 fuel was the only fuel used for in-cylinder injection. It was seen that the conventional IDI soot limit occurred at leaner overall maximum fuel-air equivalence ratios as CR was increased. This afforded a greater opportunity to increase port injected fuel levels with increased CR. Thus, power levels were then able to increase from -2 to 27% (at CR 21.5) over conventional diesel combustion by applying port injected companion fuel without a soot opacity penalty. Iso-octane provided the best improvements. PPC allowed for these power increases (defined by a threshold opacity level of 3%) due to smaller particles (and lower overall number of particles) in the exhaust that influence measured opacity less significantly than larger and more numerous conventional diesel combustion exhaust particulates. Carbon monoxide levels at the higher driven power levels were only modestly higher for the advanced combustion modes, although NO_x was generally lower due to the overall enriched operation.

Part 3 of this study applied a PPC advanced combustion approach using port fuel injection in a Yanmar L100V6 single cylinder engine-generator in order to extend the diesel engine high load limit based on exhaust soot opacity evaluation. Isobutanol, iso-octane, and jet JP-5 fuel were used for port injection, while jet JP-5 fuel was the only fuel used for in-cylinder injection. It was seen that the Conv. DI soot limit occurred at leaner overall maximum fuel-air equivalence ratio, which afforded a greater opportunity to increase port injected fuel levels. Thus, power levels were then able to increase over conventional diesel combustion. Iso-octane provided the best improvements. Results show that engine Indicated Gross Mean Effective Pressure (iGMEP) from 7.5 to 8.5 bar could be attained with PPC combustion as compared to only 7 bar with conventional operation. This led to power levels of 9.3 kW to 11.3 kW compared to 8.5 kW at conventional operation without a soot penalty. In general, JP-5 PPC led to the lowest power increase due to early combustion initiation. JP-5/iC8 provided for the highest power improvement due to the low reactivity port injected iC8 fuel, which resulted in later start and

bulk combustion as compared to conventional operation and the other PPC modes. Iso-butanol PPC provided moderate power improvements.

In Part 4 of this study, results show that engine Indicated Gross Mean Effective Pressure (iGMEP) of 12.9 and 13.0 bar could be attained with JP-5 and iC8 PPC, respectively, as compared to only 12.1 bar with conventional operation, without a soot penalty in the AM General ‘Humvee’ engine. This led to indicated power improvements over conventional operation of 7% and 8%, with JP-5 and iC8 fuel, respectively. Ignition delay and burn duration, measured from the in-cylinder injection event, decreased with PPC, and start of combustion advanced. In general, JP-5/JP-5 (in-cylinder/port fuel) and JP-5/iC8 PPC exhibited early heat release behavior, thus creating longer ‘true’ burn durations compared to conventional operation, leading to diffusion-controlled flames and less soot-reduction benefit than typically seen with PPC. This was likely a result of port injection occurring upstream of the turbocharger and a lack of intercooling that effectively preheated the premixed charge. While combustion phasing was advanced, other metrics, like peak pressure and peak pressure location, remained constant and within acceptable ranges. Future work will include adjustments to reduce early heat release.

Glossary

Acronym	Phrase	Explanation
ASTM	American Society of Testing and Materials	
ATC	After Top Dead Center	A reference to piston position after the top of the stroke when expressing piston location in Crank Angle Degrees
BDC	Bottom Dead Center	Piston location at the bottom of the stroke
BTC	Before Top Dead Center	A reference to piston position before the top of the stroke when expressing piston location in Crank Angle Degrees
CAD	Crank Angle Degrees	A unit to express piston location by dividing the 4-stroke (or 2 revolution) cycle into 720 degrees
CAD10	crank angle for 10% heat release	Piston location when 10% of combustion has occurred
CAD50	crank angle for 50% heat release	Piston location when 50% of combustion has occurred
CFR	Cooperative Fuels Research engine	A fixed speed internal combustion engine that is configured for manipulation of time of fuel injection, amount of fuel injection, compression ratio, and intake charge heating.
CI	Compression Ignition	Ignition of the fuel in an internal combustion engine due to the high temperature when the charge is compressed (as opposed to spark ignition)
CN	Cetane number	A measure of the ignition properties of diesel fuel relative to cetane fuel as a standard.

CO	Carbon Monoxide	A poisonous exhaust emission that results from incomplete combustion; since CO ₂ is desired for complete combustion, CO can be considered “half burned” fuel
CO ₂	Carbon Dioxide	
Conv. IDI	Conventional Indirect Injection	Fuel injection outside of the combustion chamber, typically in the intake manifold or port
CR	Compression Ratio	The ratio of the maximum to minimum volume in the cylinder over one stroke, a measure of how much the charge is compressed
DI	Direct Injection	Fuel is injected near the top of the compression stroke into the cylinder and air and fuel mix in the cylinder (as opposed to port injection, where air and fuel are mixed before entering the cylinder)
DoD	Department of Defense	United States Department of Defense
EGR	Exhaust Gas Recirculation	Recirculating a portion of an engine's exhaust gas back to the engine cylinders in order to reduce emissions or improve combustion phasing/timing
ϕ	Equivalence Ratio	The ratio of the actual fuel-air ratio to the stoichiometric (all oxygen consumed) fuel-air ratio, where an equivalence ratio of one denotes a stoichiometric mixture, less than one denotes a fuel lean mixture, while greater than one denotes a fuel rich mixture
GC	Gas Chromatography	Chromatography used in analytical chemistry for separating and analyzing compounds that can be vaporized without decomposition
gIMEP	Gross Indicated Mean Effective Pressure	A measure of an engine's capacity to do work that is independent of engine displacement; directly proportional to engine torque; for a fixed speed engine like the Waukesha, can be thought of as how much fuel is being burned
HCCI	Homogeneous Charge Compression Ignition	A combustion mode in reciprocating compression ignition engines that mixes fuel with the intake charge rather than injecting fuel at the top of the compression stroke
HRR	Heat Release Rate	Rate at which combustion releases energy, calculated from instantaneous pressure data
IDI	Indirect Injection	Fuel injection outside of the combustion chamber, typically in the intake manifold or port
IGD	Ignition Delay	The difference (in CAD) between SOI and SOC; how long it takes the fuel to burn after it is injected
IMEP	Indicated Mean Effective Pressure	A measure of an engine's capacity to do work that is independent of engine displacement; directly proportional to engine torque; for a fixed speed engine

		like the Waukesha, can be thought of as how much fuel is being burned
JP-5	Jet Propellant 5	Common Navy jet fuel
JP-8	Jet Propellant 8	Commercial jet fuel
LLNL	Lawrence Livermore National Laboratory	Department of Energy science lab in Livermore, CA
max. ROHR	Maximum Rate of Heat Release	See ROHR.
MS	Mass Spectrometry	An analytical technique that ionizes chemical species and sorts the ions based on their mass-to-charge ratio in order to measure the masses within a sample
NIST	National Institute of Standards and Technology	
NO _x	Oxides of Nitrogen	A generic term for the mono-nitrogen oxides NO and NO ₂ ; a harmful engine emission formed during high temperature combustion
NVH	Noise, Vibration, and Harshness	Noise is unwanted sound; vibration is the oscillation that is typically felt rather than heard; harshness is the qualitative “roughness” of engine operation
PCCI	Premixed Charge Compression Ignition	See PPC.
PPC	Partially Premixed Combustion	A combustion mode in reciprocating compression ignition engines that premixes some fuel with the intake charge and injects fuel at the top of the compression stroke
PPCI	Partially Premixed Compression Ignition	See PPC.
PPLOC	Peak pressure location	Location (in CAD) in the four stroke cycle where pressure is maximum
RCCI	Reactivity Controlled Compression Ignition	A combustion mode in reciprocating compression ignition engines that premixes a low reactivity fuel with the intake charge and injects a higher reactivity fuel at the top of the compression stroke to increase air-fuel mixing but provide more control of combustion phasing
ROHR	Rate of Heat Release	Rate at which combustion releases energy, calculated from instantaneous pressure data
RPM	Revolutions per minute	A measure of engine speed, number of revolutions of the crankshaft/flywheel per minute
SOC	Start of Combustion	The time (in CAD) at which combustion of fuel in the combustion chamber begins, typically defined by 10% of the net accumulated heat release
SOI	Start of Injection	The time (in CAD) at which injection of fuel into the combustion chamber begins
TDC	Top Dead Center	Piston location at the top of the stroke
UHCs	Unburned Hydrocarbons	Fuel in the exhaust that was not burned in combustion

Endnotes

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