

Fuel Effects on Reactivity Controlled Compression Ignition (RCCI) Combustion at Low Load

2011-01-0361

Published
04/12/2011Reed Hanson, Sage Kokjohn, Derek Splitter and Rolf Reitz
Univ. of Wisconsin

Copyright © 2011 SAE International

doi:10.4271/2011-01-0361

ABSTRACT

Reactivity Controlled Compression Ignition combustion (RCCI) has been demonstrated at mid to high loads [1, 2, 3, 4, 5, 6] as a method to operate an internal combustion engine that produces low NO_x and low PM emissions with high thermal efficiency. The current study investigates RCCI engine operation at loads of 2 and 4.5 bar gross IMEP at engine speeds between 800 and 1700 rev/min. This load range was selected to cover the range from the previous work of 6 bar gIMEP down to an off-idle load at 2 bar. The fueling strategy for the low load investigation consisted of in-cylinder fuel blending using port-fuel-injection of gasoline and early cycle, direct-injection of either diesel fuel or gasoline doped with 3.5% by volume 2-EHN (2-ethylhexyl nitrate). At these loads, engine operating conditions such as inlet air temperature, port fuel percentage, and engine speed were varied to investigate their effect on combustion. Results show that at the 4.5 bar gIMEP operating condition it was possible to maintain 54% gross indicated thermal efficiency with NO_x and PM emissions below US EPA 2010 limits. The results also show that it is possible to operate at a near idle load of 2 bar gross IMEP load with a gross indicated thermal efficiency of 49% at 1300 rev/min and 44% at 800 rev/min.

INTRODUCTION

Rising fuel costs and a focus on the reduction of greenhouse gases has driven a need for increased efficiency from the internal combustion engine. This need for increased efficiency has placed the diesel engine in the spotlight due to its higher efficiency compared to other engines. Furthermore, current and future legislation place stringent standards on the exhaust emissions. These demands require the development of advanced engine technology, which can simultaneously yield high efficiency and low emissions. Many strategies

have been investigated to meet the future requirements of clean, efficient diesel engines. Most of the current strategies fall into the category of low temperature premixed combustion. Lower combustion temperatures result in NO_x reductions, due to the high activation energy of NO formation reactions [7]. In addition, low temperature combustion utilizes longer ignition delays, allowing additional time for air/fuel mixing prior to the start of combustion; thus, rich regions are reduced and soot formation is inhibited.

Many researchers have shown Homogeneous Charge Compression Ignition (HCCI) and Premixed Charge Compression Ignition (PCCI) concepts to be promising techniques for simultaneous NO_x and soot reduction [8, 9, 10, 11, 12]. Due to the existing fuel infrastructure, most HCCI and PCCI research has been conducted using either gasoline or diesel fuel. However, in their neat forms, each fuel has specific advantages and shortcomings for premixed combustion. Gasoline has a high volatility; thus, evaporation is rapid and a premixed charge can be obtained using port-fuel-injection. However, because the autoignition qualities of gasoline are poor, it becomes difficult to achieve combustion at low-load conditions (e.g., [13,14]). Conversely, diesel fuel has superior auto-ignition qualities; however, this can result in difficulty controlling the combustion phasing as engine load is increased.

Experiments performed by Bessonette et al. [15] suggested that the best fuel for HCCI operation may have autoignition qualities between those of diesel fuel and gasoline. Based on these results, a computational study was performed by the authors to find the ideal primary reference fuel (PRF) blend and exhaust gas recirculation (EGR) level for an HCCI engine operating at 6 and 11 bar IMEP [1,16]. The computational study confirmed that the ideal fuel blend is located between that of gasoline and diesel fuel and is

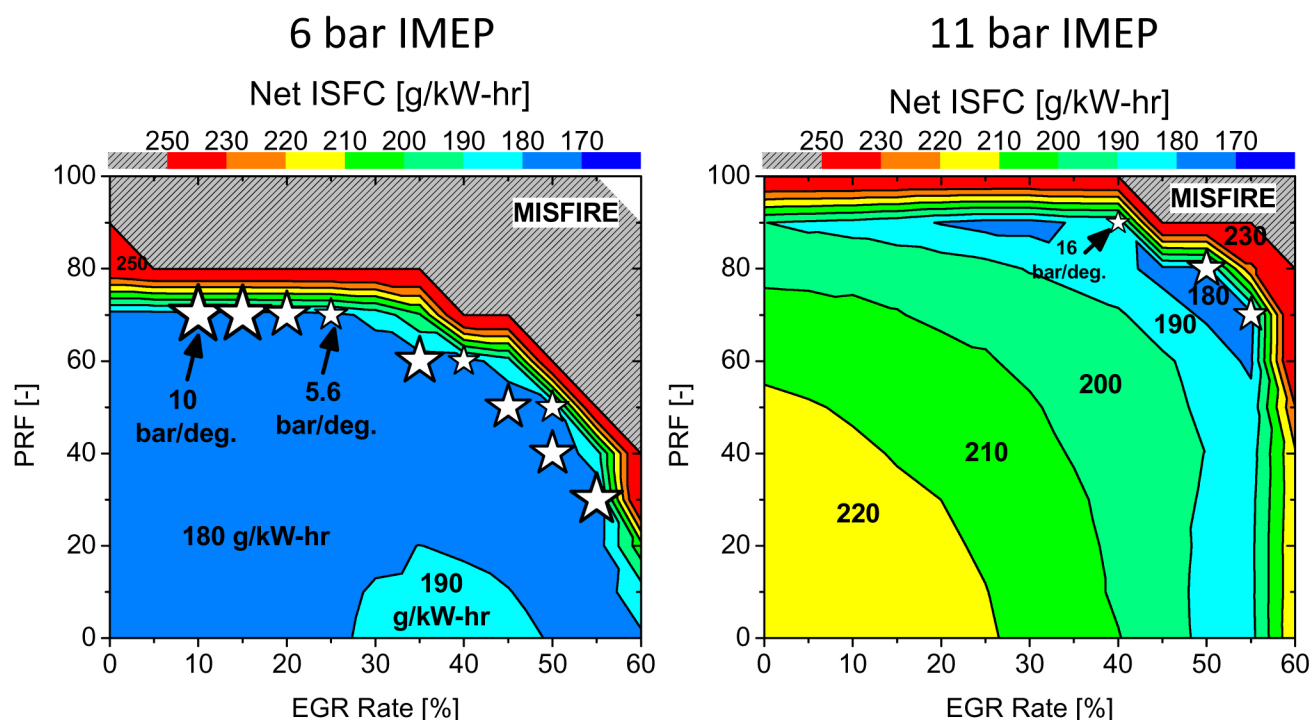


Figure 1. ISFC contour for different PRF (surrogate gasoline/diesel fuel ratio) and EGR levels at 6 and 11 bar IMEP [1]

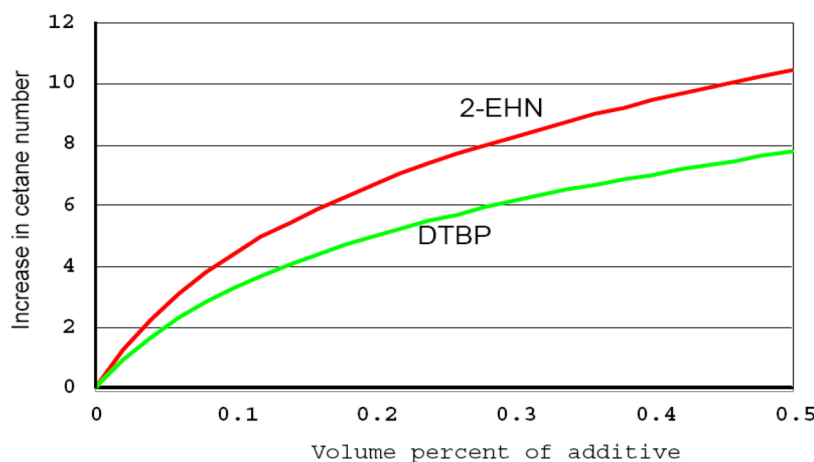


Figure 2. EPA study of cetane improvement to diesel fuel [18] using the correlations from [19]

dependent on engine load and operating conditions, as shown in Figure 1. These results then led to engines experiments that were the basis of RCCI strategies. With success in implementing RCCI at engine loads from 6 to 16 bar IMEP, it is of interest to explore the operational window of loads between 2 to 4 bar gIMEP.

Previous studies [1, 2, 3, 4, 5, 6] have demonstrated successful dual-fuel RCCI engine operation either using blends of commercially available gasoline and diesel fuels, or using a single-fuel strategy [17] which utilizes port-fuel-injection of gasoline and direct-injection of the same gasoline doped with the cetane number (CN) improver Di-tertiary

butyl peroxide (DTBP). Although DTBP is an effective CN improver, it is not widely used, due to cost and chemical stability issues. A cheaper and more common CN improver is 2-ethylhexyl nitrate (2-EHN). One specific study on the use of adding DTBP and 2-EHN to diesel-like fuels was published by the EPA [18] using correlations developed by Thompson et al. [19]. The results of that study are shown in Figure 2, where for a given addition percentage of the CN improvers 2-EHN and DTBP, 2-EHN displays greater increase in CN.

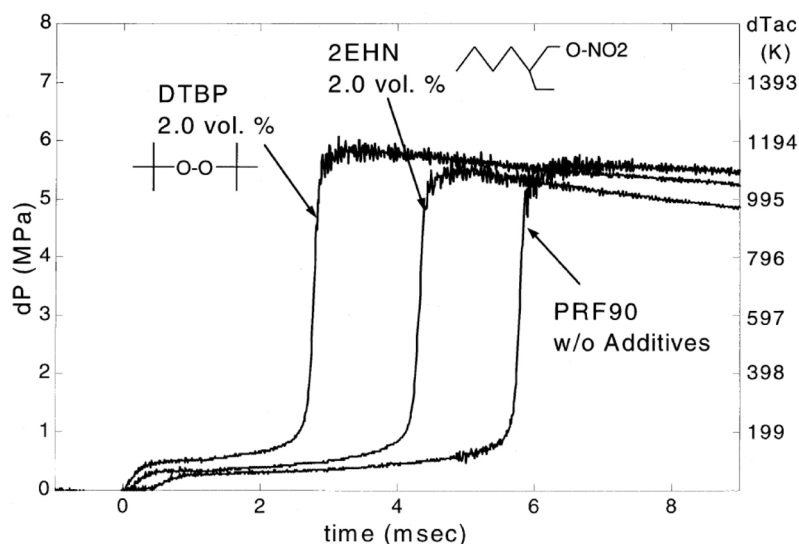


Figure 3. Rapid Compression Machine ignition characteristics of PRF blends with 2% by volume of 2-EHN and DTBP. Note that the DTBP is more effective with a PRF representative of gasoline than 2-EHN, opposite to the behavior with diesel fuels [23]

Oxley et. al. [20] suggested that the initial decomposition step of 2-EHN is the cleavage of a NO_2 group bound to the end of the 2-EHN molecule and could potentially lead to increased NO_x emissions. The emissions results of this NO_2 group are conflicting, and seem to be combustion regime dependent. For instance, Thompson [19] has demonstrated that the NO_x emissions of 2-EHN doped fuels in conventional diesel operation are identical or lower than non-doped diesel fuels. However, low temperature combustion (LTC) results by Ikes [21] and Bunting et. al. [22] have shown that 2-EHN doped diesel fuels lead to increased engine out NO_x emissions. Specifically, Ikes demonstrated that in LTC combustion NO_x emissions increase proportional to 2-EHN amount, with approximately 50% of the NO_x present in the fuel leading to NO_x emissions.

Although 2-EHN is a more effective than DTBP when used in diesel fuel, opposite trends have been demonstrated with rapid compression machine (RCM) measurements with gasoline-like fuels [23]. This trend, where DTBP is more effective than 2-EHN, is explained by the hypothesis that both 2-EHN and DTBP decomposition initiates with the production of alkoxy radicals, and the initial dissociation energies for each respective CN improving additive are similar. The decreased ignition delay with DTBP and gasoline-like fuels results from the dissociation of DTBP forming twice the number of alkoxy radicals as 2-EHN. The nearly doubled radical density results in approximately double the ignition delay improvement over the base fuel. Figure 3 presents ignition delay results from RCM experiments with and without CN improved PRF90 [23].

Based on the fuel sensitivity of cetane improver effectiveness, and the LTC NO_x emissions trends of 2-EHN, it was desired to investigate the results of replacing DTBP with 2-EHN to explore whether a fuel NO_x penalty exists during RCCI operation. The present investigation examines and compares RCCI operation using a single-fuel strategy consisting of using gasoline port-fuel-injection and direct injection of gasoline doped with 3.5% 2-EHN by volume to dual-fuel operation with port-fuel-injection of gasoline and direct-injection of diesel fuel.

EXPERIMENTAL SETUP

A heavy-duty 2.44 L Caterpillar 3401 Single Cylinder Oil Test Engine (SCOTE) was used for all the engine tests. The engine geometry is summarized in Table 1 and the lab setup is shown in Figure 4.

The RCCI engine experiments were performed using port fuel injection of a lower reactivity fuel and early direct injection of a higher reactivity fuel with a conventional solenoid operated common-rail injector. A lubricity additive was added at 500 ppm to the common rail fuel system when using gasoline doped with 2-EHN to provide adequate lubrication. The specifications for the port-fuel-injector and common rail injector are given in Table 2 and Table 3, respectively.

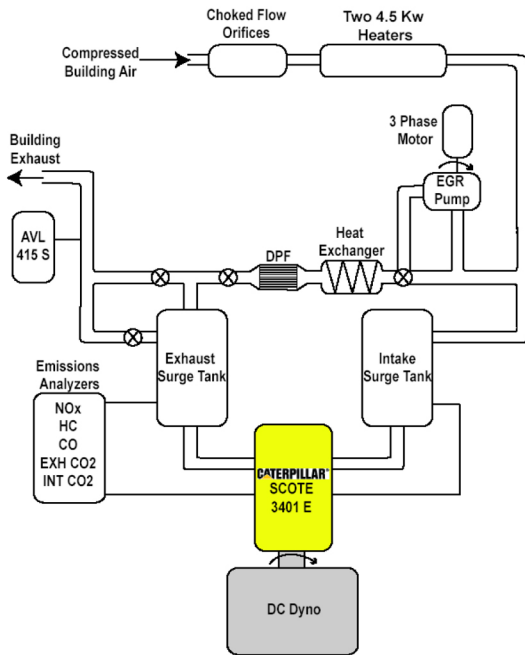


Figure 4. Diagram of the engine lab setup

Table 1. Caterpillar 3401E SCOTE engine geometry

Displacement (L)	2.44
Geometric Compression Ratio (-)	16.1:1
Connecting Rod Length (mm)	261.6
Bore x Stroke (mm)	137.2 x 165.1
Piston	Standard SCOTE Open Crater
Valvetrain	EVC = -355 deg (°ATDC) IVC = -143 deg (°ATDC) EVO = 130 deg (°ATDC) IVO = 335 deg (°ATDC)
Swirl Ratio (-)	0.7 (stock)

Table 2. Port fuel injector specifications

Injection Pressure (MPa)	0.5
Included Spray Angle	15°
Number of Holes	3
Steady Flowrate (cc/min)	750

Table 3. Common rail injector specifications

Injection Pressure (MPa)	40
Included Spray Angle	143°
Number of Holes	7
Hole Diameter (µm)	182
Steady Flowrate (cc/30 sec) at 100 bar	1000

INJECTION STRATEGY

The injection strategy for the present low load RCCI investigation was similar to previous RCCI work in that gasoline was injected very early in the cycle and the reactive fuel was injected using a double injection strategy later in the cycle. The injection timing windows of the DI event are shown in Figure 5. The DI strategy utilized one early injection to raise the reactivity of the fuel in the squish region near the cylinder liner and a second injection closer to top dead center (TDC) to increase the reactivity in the center of the cylinder. The timings for this work were fixed for all cases, with the squish conditioning pulse at -62 °ATDC and the ignition pulse at -37 °ATDC for all cases.

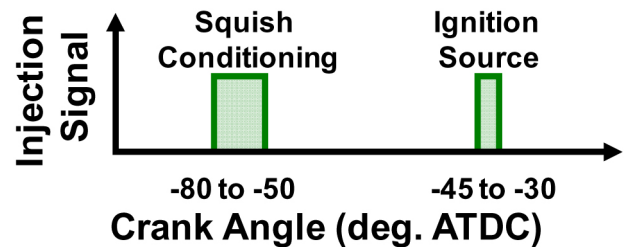


Figure 5. Injection strategy used for RCCI combustion [6].

Since this work explores both a dual-fuel and a single-fuel plus an additive approach for RCCI combustion, Figure 6 graphically shows both fuel injection strategies. As can be seen, the dual-fuel strategy used both gasoline and diesel fuels from their own respective supply tanks, while the single-fuel plus an additive approach used the same main fuel (i.e., gasoline) in both streams with the CN improver (i.e., 2-EHN) added to the direct injected stream. Physical properties for the gasoline and diesel fuels are shown in Table 4 and Table 5, respectively.

Table 4. Diesel fuel properties

Specific Gravity (@ 15.5 °C) (-)	0.856
Viscosity (@ 40 °C) (cSt)	2.71
Surface Tension (@ 25 °C) (dyne/cm)	30
Lower Heating Value (MJ/kg)	42.526
Cetane Number	46.1
H/C ratio (-)	1.74

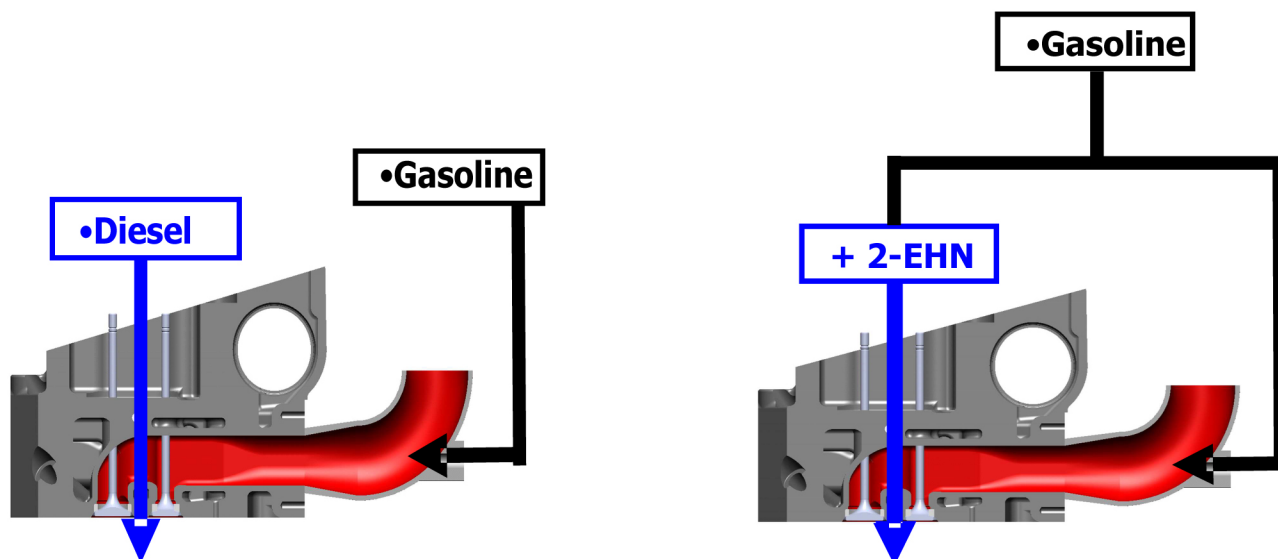


Figure 6. Examples of the dual-fuel and single-fuel injection strategies

Table 5. Gasoline fuel properties

Distillation Curve	ASTM D86
Initial Boiling Point (°C)	38.89
Temperature 10% evaporated (°C)	69.4
Temperature 50% evaporated (°C)	105
Temperature 90% evaporated (°C)	160.56
Final Boiling Point (°C)	215.56
Lower Heating Value (MJ/kg)	43.22
MON	87.8
RON	95.6
(RON+MON)/2	91.6
Ethanol (%)	0
H/C ratio (-)	1.88
Specific Gravity (@ 15.6 °C) (-)	0.737
Sulfur (ASTM D5453) (PPM)	4.6

Table 6. Engine Operating conditions

Nominal gross IMEP (bar)	2.0	4.5
Fuel type (less reactive/more reactive)	(G/G+2EHN)	(G/G+2EHN) (G/D)
Engine Speed (rev/min)	800 and 1300	800, 1300 and 1700
IVC timing (°ATDC)	-143	
EGR rate (%)	0	
Intake temperature (°C)	52 and 60	40 to 70
Absolute intake pressure (bar)	0.69	1
Total fuel (mg/cycle)	30	55
Percent port fuel by mass (%)	20 to 40	50 to 94
Nominal equivalence ratio (-)	0.3	
Direct injection fuel pressure (MPa)	40	
SOI1 (°ATDC) (first DI event)	-62	
SOI2 (°ATDC) (second DI event)	-37	
Fract. of DI fuel in SOI1 (-)	0.62	

EXPERIMENTAL CONDITIONS

The primary objective of the present study was to investigate low load RCCI operation. Two load points were investigated. First, operation is considered at a load point of 4.5 bar gIMEP over a range of engine speeds. In this study, the effects of port fuel fraction and intake temperature are investigated for the dual-fuel and single-fuel plus additive cases and are shown in Table 6. Next, operation at a load point of 2.0 bar gIMEP (i.e., near break-even power) was also conducted at a near idle and rated torque speed.

GASOLINE/DIESEL 4.5 BAR IMEP TESTS

The initial tests of the study used gasoline as the low reactivity fuel and diesel as the high reactivity fuel. In these tests, different combustion phasings were investigated at engine speeds of 800, 1300 and 1700 rev/min, respectively. Figure 7 shows measured cylinder pressure and apparent heat release rate (AHRR) for engine speeds of 1300 and 1700 rev/min. As can be seen, the higher engine speed shows lower peak heat release rates, as well as longer combustion durations. This is to be expected, as the fuel chemical time scale is independent of engine time scale, resulting in fewer crank angle degrees for the same chemical reaction rates to take place. Also, interesting to note is the low temperature heat release (LTHR) for both speeds is roughly the same for the 1300 and 1700 rev/min cases, suggesting that the required low temperature reaction temperature history is similar for

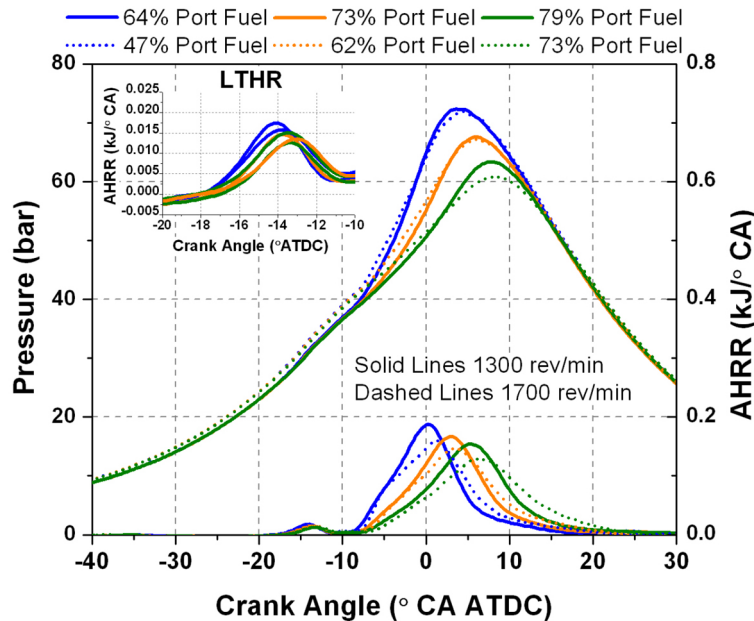


Figure 7. Port fuel fraction sweep at 1300 and 1700 rev/min at 4.5 bar IMEP.

these two engine speeds. While the LTHR is similar for the 3 engine speeds, as seen in Figure 8, the HTHR becomes significantly faster when engine speed is reduced from 1300 to 800 rev/min.

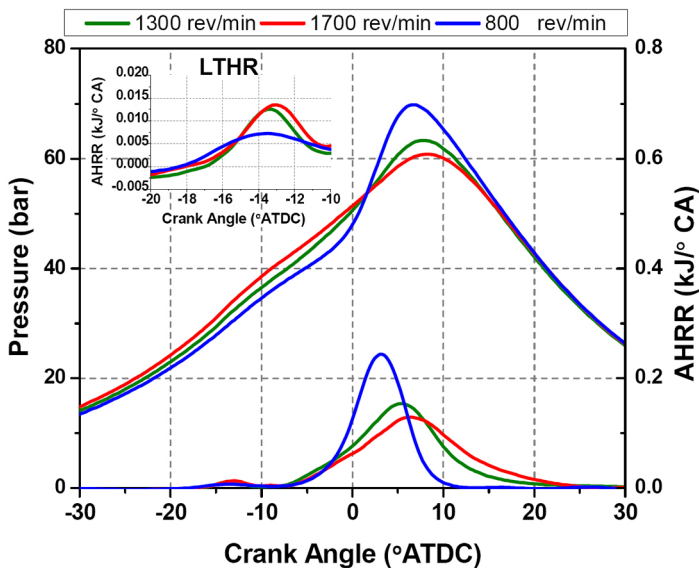


Figure 8. Engine speed sweep at 4.5 bar IMEP

In order to investigate the ability to control combustion phasing at different engine speeds, as seen in Figure 8, the ratio of the more reactive to less reactive fuels was varied in order to maintain a CA50 timing of 5 deg ATDC. Altering the reactivity ratio with changes in engine speed was required because as engine speed is increased, the ignition delay must also decrease to maintain the same CA50 timing. This is

accomplished by increasing the amount of more reactive direct injection (DI) fuel relative to the port fuel injection (PFI) fuel. As shown in Figure 9, as the ratio of DI fuel was increased, the emissions trends changed. For instance, as the DI fuel fraction was increased NO_x emissions increased, while CO and HC decreased. This is thought to be an effect of the additional DI fuel forming higher local equivalence ratios, which increase the local combustion temperatures. These locally hotter regions tend to increase HC and CO oxidation rates and also increase NO_x formation rates. Since the combustion is highly premixed, the effect of the locally richer regions on PM formation is small given the local equivalence ratio is approximately the global equivalence ratio.

Combustion phasing also was found to have a significant impact on thermal efficiency. As combustion phasing was retarded, thermal efficiency dropped. Also interesting to note is the reduced efficiency at higher engine speeds. Normally, at higher engine speeds there is less time available for heat transfer to occur, thus raising the thermal efficiency. This trend was not observed in the tests at 4.5 bar. As seen in Figure 10, there were less heat transfer losses at 1700 rev/min vs. 800 rev/min, but there were also increases in combustion and pumping losses. These increased combustion losses are due to the increased engine speed lowering the equivalence ratio. This decrease in equivalence ratio showed to cause large decreases in combustion efficiency. Finally, the net effect of increased combustion and pumping losses overcome the reduced heat transfer losses to bring the thermal efficiency below the 800 rev/min case. However, it is still interesting to note that the thermal efficiency is still quite high for all engine speeds, near 50% for most cases.