

Effect of Piston Bowl Geometry on Dual Fuel Reactivity Controlled Compression Ignition (RCCI) in a Light-Duty Engine Operated with Gasoline/Diesel and Methanol/Diesel

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ABSTRACT

A single-cylinder light-duty diesel engine was used to investigate dual fuel reactivity controlled compression ignition (RCCI) operated with two different fuel combinations: gasoline/diesel fuel and methanol/diesel fuel. The engine was operated over a range of conditions, from 1500 to 2300 rpm and 3.5 to 17 bar gross IMEP. Using the stock re-entrant piston bowl geometry, both fuel combinations were able to achieve low NO_x and PM emissions with a peak gross indicated efficiency of 48%. However, at light load conditions both gasoline and methanol yielded poorer combustion efficiencies. Previous studies have shown that the high-levels of piston induced mixing that are created by the stock piston are not required, and in fact are detrimental due to increased heat transfer losses, for premixed combustion. Thus a modified piston featuring a shallow, flat piston bowl with nearly no squish land was also investigated. Using the modified piston, the gross indicated efficiency of RCCI combustion was significantly improved at light loads due to increases in combustion efficiency and decreases in heat transfer losses. At higher loads the modified piston also performed better than the stock piston, but the improvements were not as significant. Over the entire load and speed range, the modified piston yielded low NO_x and PM emissions with a peak gross indicated efficiency of nearly 51%.

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INTRODUCTION

RCCI Background & Motivation for Current Study

In an effort to reduce NO_x and soot emissions while maintaining high thermal efficiency many new compression ignition combustion strategies have been proposed. Homogeneous Charge Compression Ignition (HCCI) is a concept that has been researched for many years and was initially studied in four-stroke engines by Najt and Foster [1]. HCCI combustion has been shown to have superior characteristics in terms of thermal efficiency, which results from lower heat transfer losses and shorter combustion durations. In addition, the lean, low temperature combustion process yields near zero NO_x and soot emissions.

Although HCCI combustion appears thermodynamically attractive, it is entirely controlled by chemical kinetics, which means the in-cylinder temperature, pressure, equivalence ratio, and fuel reactivity (i.e., octane number) will govern when ignition occurs. This presents a controllability

challenge because there is no direct means to control the timing of combustion. In addition, due to its fully premixed nature, the combustion process is very rapid, which leads to excessive pressure rise rates and possible engine damage. Finally, Bessonette et al. [2] demonstrated experimentally that the best fuel for HCCI operation has autoignition qualities between that of diesel fuel and gasoline, depending on the operating condition. For instance, they demonstrated that at 16 bar BMEP the most suitable fuel had an octane number of ~81. However, at 2 bar BMEP a cetane number of ~45 (i.e., a traditional diesel fuel) was required for stable operation. Thus, it seems that exploring fuel blends for specific operating conditions would be beneficial. These advantages and challenges of HCCI combustion have motivated the investigation of Partially Premixed Combustion (PPC) concepts.

Reactivity Controlled Compression Ignition (RCCI) is a dual fuel partially premixed combustion concept. In this strategy, a low reactivity fuel, such as gasoline, is premixed via port fuel injection (PFI) and a high reactivity fuel, such as diesel fuel, is direct injected (DI) during the compression stroke. This process of in-cylinder fuel blending develops fuel reactivity gradients in the combustion chamber. The stratification of fuel reactivity results in a broad combustion event and reduced pressure rise rates compared to fully premixed HCCI combustion. In addition, in-cylinder fuel blending of two fuels with very different fuel reactivity (i.e., octane number), allows the global octane number to be varied on a cycle-to-cycle basis. This allows a wide range of loads to be achieved by varying the ratio between the two fuels.

The majority of RCCI combustion research conducted to date has used port injected gasoline and direct injected diesel fuel on both heavy and light-duty engines. Kokjohn et al. [3] used a single cylinder heavy-duty Caterpillar engine in its stock form (compression ratio of 16.1) to develop the RCCI combustion concept. Over a load sweep from 4.6 bar to 14.6 bar IMEP, they were able to achieve NO_x emissions below 0.1 g/kW-hr, soot emission below 0.013 g/kW-hr, and a peak gross indicated efficiency of 56%. This was accomplished by using relatively fixed diesel injection timings of -60° and -40° ATDC and varying the gasoline to diesel ratio from 50% to 90% port injected gasoline. The 14.6 bar IMEP operating condition utilized only 10% of the total fuel as the high reactivity diesel, 57% EGR, and a global equivalence ratio of 0.95. Thus, achieving higher loads with the stock compression ratio of 16.1 proved difficult.

In an effort to extend the load range of clean and efficient RCCI combustion, a computational study with an ultra-low compression ratio was conducted on the heavy-duty Caterpillar engine by Dempsey and Reitz [4]. They found that by using a compression ratio of 11.7, RCCI combustion could be achieved with gasoline and diesel fuel from 4 to 23 bar IMEP. Using the operational strategy of Dempsey and Reitz and a compression ratio of 12, Splitter et al. [5] were able to successfully operate a heavy-duty engine experimentally using RCCI combustion up to 20 bar BMEP. However, both of the aforementioned studies, which utilized low compression ratios showed that the indicated thermal efficiency is reduced compared to the stock compression ratio of 16.1.

Another approach to achieving high load RCCI operation is through modification of the fuels. Dempsey et al. [6] computationally investigated the use of port injected hydrated ethanol and direct injected diesel fuel using the stock Caterpillar piston with a compression ratio of 16.1. They found that due to the large amount of charge cooling from the port injected hydrated ethanol and the high octane number of ethanol, 17 bar IMEP without the need for EGR could be achieved. By utilizing the stock compression ratio, the peak indicated thermal efficiency was restored to the level observed with gasoline/diesel operation without EGR (i.e., $\sim 55\%$ gross indicated efficiency). This concept was

experimentally demonstrated on a light-duty engine up to ~ 11 bar IMEP without the need for EGR [7]. As will be discussed, ethanol and methanol have very similar physical and chemical properties. It is expected that methanol will perform similarly to ethanol as the port injected low reactivity fuel for RCCI combustion. One of the focuses of this study is to compare an alcohol fuel (i.e., methanol) to a conventional hydrocarbon (i.e., gasoline) as the low reactivity fuel.

The main reason for investigating alcohol fuels is to reduce petroleum consumption. On a global basis, petroleum based fuels are the predominant source of energy in the transportation sector. In the United States, over 90% of the energy consumed for transportation comes from petroleum (i.e., crude oil). Petroleum products are a natural resource and thus the supply is finite. Alcohols currently are and will continue to play a pivotal role in the search for alternative fuels, mainly because they can be made from a variety of feedstocks and are liquid, which fits well with the current fuel supply infrastructure. Methanol can be produced from natural gas and coal, both of which are abundant in the world and much more uniformly distributed compared to crude oil, making them less prone to supply disruptions [8]. Methanol has similar physical and chemical properties to hydrated ethanol, which have been shown to be beneficial for achieving high load RCCI operation while maintaining high compression ratio [6].

In addition to the need to investigate the use of alternative fuels for advanced combustion strategies, it is vital that the engine hardware be optimized for partially premixed combustion. Hanson et al. [9], using detailed CFD modeling demonstrated that as the piston bowl radius increases while the compression ratio is held constant (i.e., the piston bowl depth decreases), the gross indicated efficiency of RCCI increases. This increase in efficiency was caused by a reduction in heat transfer losses due to the reduced surface area of the wider bowl. In addition, the high-levels of piston induced mixing that are created by narrow/deep piston bowls increase the heat transfer coefficient and further add to the heat transfer losses. Thus, the more quiescent combustion chamber created by the wide/shallow bowl is beneficial for premixed combustion strategies. Based on these findings, Hanson et al. custom machined an aluminum piston with a wide/shallow bowl and a compression of 15.1. The same piston profile used by Hanson et al. [9] was utilized in the current study and comparisons were made to the stock, re-entrant piston bowl geometry. However, in this study the compression ratios of the two piston bowl geometries were matched at ~ 17.3 .

The main objective of the current study is to investigate the potential of methanol to be used as the low reactivity fuel for RCCI combustion in a light-duty engine. Extensive research has already been conducted using gasoline for RCCI combustion [10, 11], thus comparisons will be made to operation with gasoline/diesel. In addition to the fuels comparison, two different piston bowl geometries will be

investigated: the stock piston and a modified RCCI piston. The stock piston features a deep/narrow bowl typically found in light-duty diesel engines today. The modified RCCI piston has a shallow/wide bowl. Each piston has a geometric compression ratio of ~ 17.3 , so the piston comparison is a direct comparison of the bowl shapes.

EXPERIMENTAL SETUP

Single-Cylinder Light-Duty Diesel Engine Laboratory

The experiments were carried out on a light-duty, four-stroke diesel engine based on the General Motors/Fiat JTD 1.9L platform. Table 1 shows the specifications for the engine and Figure 1 provides a schematic of the single-cylinder engine laboratory. Two piston bowl geometries were used in this study: the stock GM piston with a re-entrant bowl and the modified piston with a wide/shallow bowl. These bowls will be referred to throughout this study as the stock and RCCI pistons, respectively. With a bore of 82.0 mm and a stroke of 90.4 mm this engine is representative of a light-duty automotive diesel engine with four valves per cylinder driven by dual overhead cams. A stock four cylinder head is installed atop a single cylinder block with only the valves for cylinder #2 activated by the cams.

Table 1. GM 1.9L Single Cylinder Engine Specifications

<u>Engine Geometry</u>		
Base Engine	GM 1.9L Diesel	
Geometric Compression Ratio	17.4	17.3
Piston Bowl Type	Re-entrant (Stock-GM)	Wide/Shallow (RCCI)
Displacement (Liters)	0.477	
Stroke (mm)	90.4	
Bore (mm)	82.0	
Intake Valve Closing	-132° aTDC	
Exhaust Valve Opening	112° aTDC	
Swirl Ratio	1.5	
<u>Port Fuel Injector</u>		
Model Number	TFS-89055-1	
Included Spray Angle	20°	
Injection Pressure (bar)	2.5 to 3.5	
Rated Flow (lbs./hr.)	55	
<u>Common Rail Injector</u>		
Model	Bosch CRI2.2	
Number of Holes	7	
Hole Diameter (mm)	0.14	
Included Spray Angle	148°	
Injection Pressure (bar)	250 to 1000 bar	

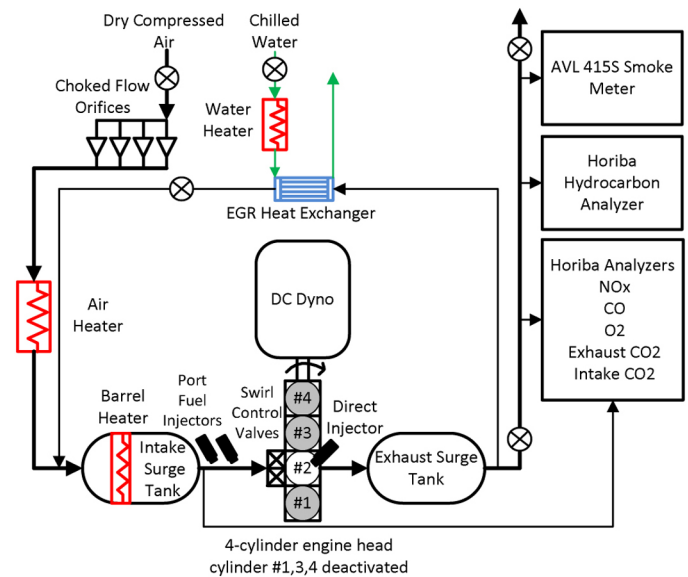


Figure 1. GM 1.9L Single Cylinder Engine Laboratory

Engine controls and data acquisition were handled through National Instruments LabView software. The fuel mass flow rates for the port injected fuel and direct injected fuel were measured separately by two Endress+Hauser Promass 80A coriolis flow meters. A Kistler 6125A pressure transducer in series with a Kistler 4010 charge amplifier was used to measure the in-cylinder pressure. A shaft encoder with 1440 pulses per revolution was used, which supplies a resolution of 0.25 crank angle degrees. For each operating point 300 consecutive cycles of cylinder pressure data were acquired. Each individual cycle's pressure data was smoothed using a Fourier series low-pass filter with a Gaussian roll-off function having a transmission of 100% from 0 to 1600 Hz and dropping to 1% at 6720 Hz. Once filtered, the collected cycles were ensemble averaged to yield a representative cylinder pressure trace, which was used to calculate the apparent heat release rate (AHRR) and, in turn, determine combustion phasing. The AHRR is essentially the chemical heat release rate minus the heat loss rate to the combustion chamber walls and is given by

$$AHRR = \frac{dQ}{d\Theta} \Big|_{chem} - \frac{dQ}{d\Theta} \Big|_{wall} = \frac{1}{\gamma-1} V \frac{dP}{d\Theta} + \frac{\gamma}{\gamma-1} P \frac{dV}{d\Theta} \quad (1)$$

where P is the average in-cylinder pressure, V is the cylinder volume, and γ is the ratio of specific heats. From the AHRR, the accumulated heat release can be determined. The combustion phasing was calculated to be the crank angle at which the accumulated heat release had reached 50% of its maximum, denoted as CA50. For each operating condition the fuel energy input per cycle is calculated using

$$E_{fuel} = m_{PFI} LHV_{PFI} + m_{DI} LHV_{DI}$$

where m is the mass of fuel per cycle from the port injected fuel or the direct injected fuel. The indicated gross work is calculated using

$$W_{cycle} = \oint_{cycle} P dV \quad (3)$$

where the cycle is defined as -180° ATDC to 180° ATDC. Note 0° ATDC is used to denote top dead center of the compression stroke. The indicated net work is calculated for the entire four-stroke cycle (-360° ATDC to 360° ATDC). The indicated pumping work is the difference between the gross and the net work. The cycle's indicated efficiency is defined as the ratio of the work produced over that cycle to the fuel energy input. The combustion efficiency is a measure of the completeness of the combustion process and is calculated as

$$\eta_{comb.} = \frac{E_{fuel} - E_{comb, loss}}{E_{fuel}} = \frac{E_{fuel} - m_{UHC}^{EVO} LHV_{fuel} - m_{CO}^{EVO} LHV_{CO}}{E_{fuel}} \quad (4)$$

The LHV_{fuel} is calculated based on the relative amounts of port injected fuel and direct injected fuel. The LHV_{CO} is taken to be 10 MJ/kg.

Other parameters that will be used to calculate the fuel energy distribution over the net cycle are the thermal exhaust loss and heat transfer loss. The thermal exhaust loss is a measure of the sensible enthalpy remaining in the exhaust gas

$$E_{exh-loss} = m_{exh} [C_{p,exh} T_{exh} - C_{p,in} T_{in}] \quad (5)$$

In this study the exhaust temperature, T_{exh} , is measured in the exhaust runner. The heat transfer loss is taken to be the remainder of a first law energy balance around the single cylinder engine

$$E_{heat\ transfer\ loss} = E_{fuel} - W_{gross} - W_{pump} - E_{comb, loss} - E_{exh, loss} \quad (6)$$

where the combustion losses are a measure of the chemical energy remaining in the exhaust gas and is calculated using one of the terms in Equation (4).

Piston Bowl Geometries

Figure 2 shows a comparison of the piston bowl profiles used in this study. The stock GM piston (solid line) features a deep re-entrant bowl. This bowl shape is typical of light-duty diesel engines that benefit from strong in-cylinder flows to aid in fuel/air mixing. The modified RCCI piston (dashed

line) features a very wide and shallow bowl. Both pistons were cut from the same aluminum piston blank and therefore have the same top ring crevice volume. Note this is important when comparing pistons for premixed combustion strategies because the top ring crevice is a major source of unburned hydrocarbon emissions.

As shown in Table 1 these pistons have very similar geometric compression ratios, despite the fact that they have different bowl volumes. This was accomplished by using four stacked head gaskets for the stock piston and only one for the RCCI piston, thus the RCCI piston has a slightly smaller squish height (i.e., distance between the piston top and the firedeck at TDC). Figure 3 shows the results from motored traces with matched boundary conditions for the two pistons. As can be seen, both pistons yield very similar peak motor pressures, with the RCCI piston just below the stock piston, due to its slightly lower compression ratio.

The marked difference between the two pistons is the bowl shape, in which the modified RCCI piston has reduced surface area. At TDC the total combustion chamber surface area for the stock piston is 174 cm^2 and for the modified RCCI piston is 152 cm^2 , a reduction of 13%. This reduction in surface area in combination with the more quiescent combustion chamber created by the modified RCCI piston will be shown to yield a significant reduction in the heat transfer losses.

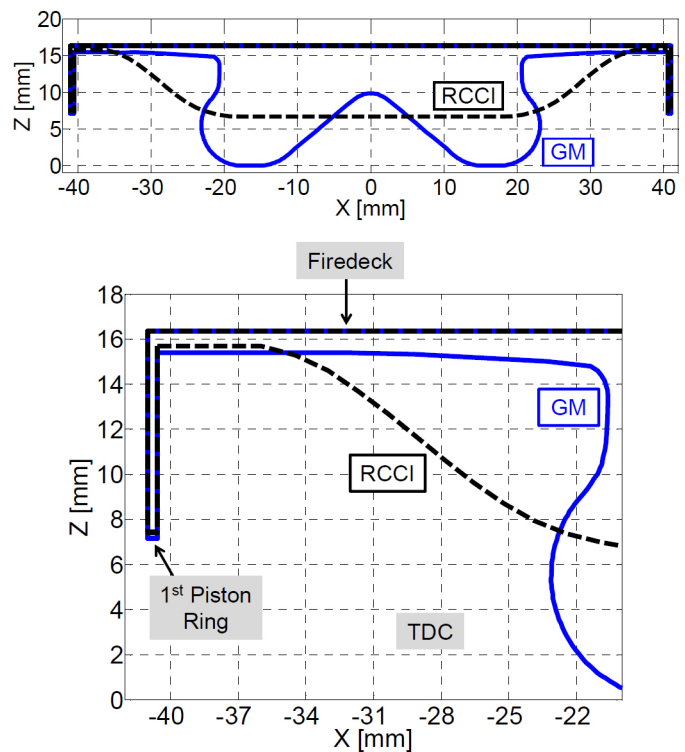


Figure 2. Comparison of the Stock GM piston (solid line) and the modified RCCI piston bowls (dashed line) at TDC

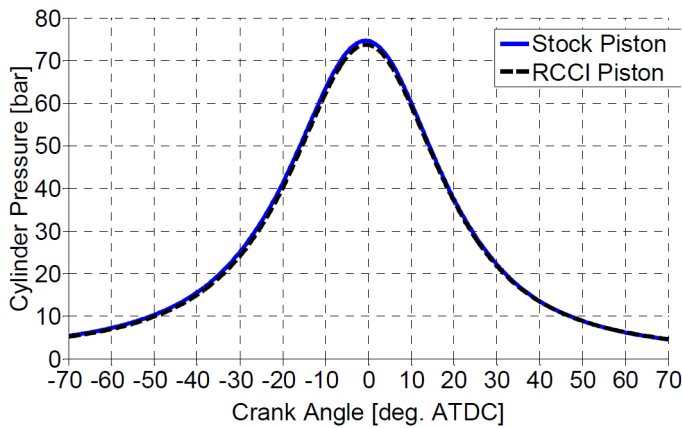


Figure 3. Motored trace comparison between the stock and modified RCCI pistons. Operating conditions: 1900 rpm, 1.65 bar intake pressure, 40°C intake temperature.

Fuel Properties

The physical and chemical properties of the fuels used in this study are shown in Table 2. As can be seen in the table, the gasoline and methanol are extremely volatile fuels, which ensured that a homogeneous premixed charge was created in the combustion chamber. A marked difference between the gasoline and methanol is the enthalpy of vaporization and the lower heating value (LHV). Methanol has significantly higher charge cooling potential (i.e., higher enthalpy of vaporization) and a smaller LHV compared to gasoline. Thus to achieve a given engine load, the methanol fueling rate must be significantly increased compared to gasoline. This serves to amplify the charge cooling effect of the port injected methanol.

Using one dimensional cycle simulations which accounted for the charge cooling from port injected fuels, the bulk gas temperature in the combustion chamber at intake valve closure (IVC) could be estimated [13]. Figure 4 illustrates the results from these simulations, which shows a plot of the estimated bulk gas temperature at IVC as a function of the intake surge tank temperature. Here the fuel energy from the port injected fuel is being held constant at ~375 J/cycle. This requires 8.8 mg of gasoline and 18.4 mg of methanol. As can be seen, with no port fuel injection at this operating condition, there is a nominal increase in the gas temperature from the intake surge tank to IVC of 25°C. If the gasoline is port injected, it is estimated that a slight charge cooling effect (~ 5°C) will be experienced. However, if methanol is port injected the charge cooling impact is much more significant (~ 25°C).

Table 2. Physical and chemical properties of the fuels tested under RCCI combustion

	Diesel Fuel	Gasoline	Methanol*
Chemical Formula	CH _{1.76}	CH _{2.13}	CH ₃ OH
Lower Heating Value [MJ/kg]	42.61	44.30	20.0
Stoichiometric Air-Fuel Ratio [Mass]	14.4	14.93	6.46
Research Octane Number (RON)	-	94.7	106
Motored Octane Number (MON)	-	87	92
Cetane Number	42	-	-
Liquid Density @ 25°C [g/cc]	0.859	0.701	0.792
Enthalpy of Vaporization** [KJ/kg]	~270	~300	~1100
Initial Boiling Point [°C]	174	26	65
Temperature 50% Evaporated [°C]	262	98	65
Final Boiling Point [°C]	350	213	65

*VP Racing Fuels (Product M1). Properties from [12].

**Properties from [12].

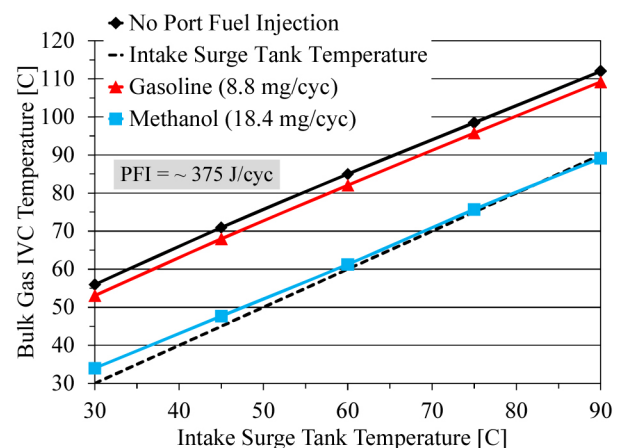


Figure 4. Cycle simulation results used to estimate the in-cylinder bulk gas temperature at IVC as a function of intake temperature and port injected fuel composition.

Operating Conditions

The engine operating conditions (i.e., speed and load) investigated in this study are shown in Figure 5. Several of these operating conditions were taken from the Ad Hoc diesel fuel test program [14, 15]. The Ad Hoc fuels group identified several engine speed and load conditions that are representative of the U.S. FTP-75 light-duty drive cycle. These operating points are shown in Figure 5 as modes 1, 2, 4, 7, and 8. (Note that the engine speed of mode 8 has been reduced from 2600 rpm to 2300 rpm due to limitations of the current laboratory setup).

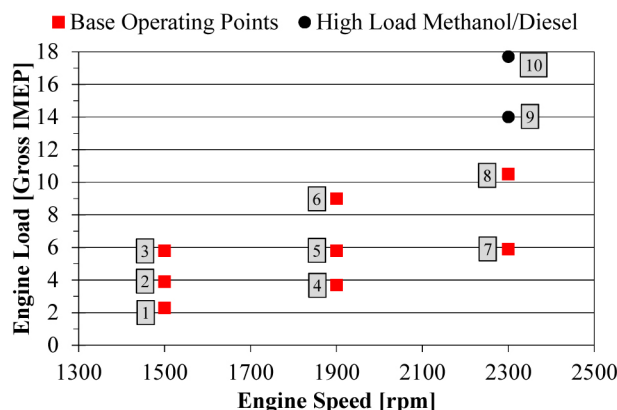


Figure 5. Engine operating points investigated in this study

These operating points are confined to relatively lighter loads. In order to investigate a wider operating map, modes 3, 5, and 6 were added to the set of base operating points from the Ad Hoc fuels group. While operating with gasoline/diesel, the maximum achievable load was 10.5 bar gross IMEP, mode 8. However, when methanol was used as the low reactivity port injected fuel, much higher loads were achievable, as indicated by modes 9 and 10.

While studying the differences between the two different piston bowl geometries, the boundary conditions were held fixed for gasoline and methanol, respectively. Figure 6 shows the nominal absolute intake pressure that was used for all the RCCI operating modes, which ranges from naturally aspirated at low load to 2.2 bar absolute at high load. These intake pressures were chosen somewhat arbitrarily, but are representative of what a turbocharged light-duty diesel engine can achieve [9]. Figure 7 shows the nominal intake temperatures. In general, the intake temperatures used for methanol were higher than that for gasoline, but still reasonable at 90°C or less. Figure 8 shows the nominal EGR level used for the RCCI operating conditions. In this study EGR was used to phase the combustion later to avoid excessive pressure rise rates during RCCI operation. This was only needed when the PFI fueling percentage could no longer be increased due to the finite minimum amount of diesel fuel that can be delivered through the common rail injector (i.e., ~1.5 mg/cyc). As can be seen, gasoline/diesel operation

required EGR to be used at ~7 bar IMEP and above, but methanol/diesel operation did not require EGR until ~12 bar IMEP. In addition, the EGR levels needed for methanol/diesel operation are lower.

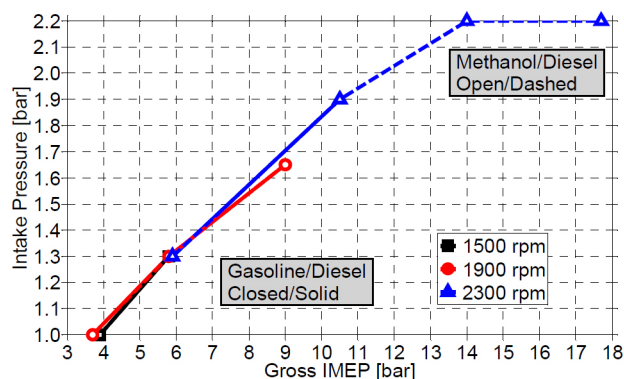


Figure 6. Nominal absolute intake pressures [bar] for all the RCCI operating modes: both stock and RCCI pistons

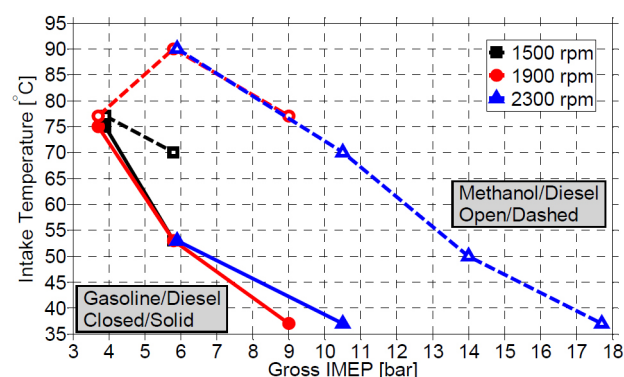


Figure 7. Nominal intake temperatures [°C] for all the RCCI operating modes: both stock and RCCI pistons

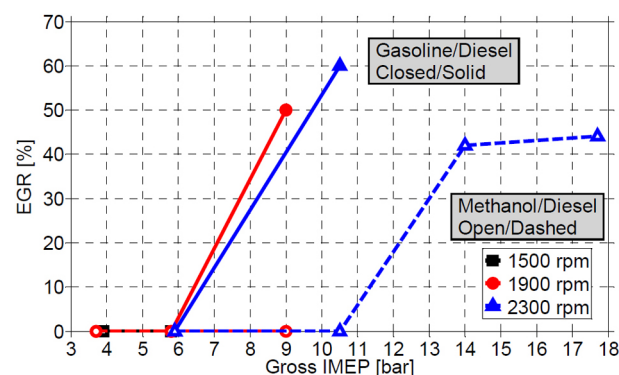


Figure 8. Nominal EGR levels [%] for all the RCCI operating modes: both stock and RCCI pistons

In this study the EGR level is calculated as

$$EGR[\%] = \frac{[CO_2]_{intake} - [CO_2]_{ambient}}{[CO_2]_{exhaust} - [CO_2]_{ambient}} * 100 \quad (7)$$

where $[CO_2]_x$ is the dry volumetric concentration of CO_2 at location x. The ambient CO_2 concentration was taken to be 383 ppm.

STOCK RE-ENTRANT PISTON BOWL RESULTS

Results with the stock re-entrant piston bowl will be shown first. The focus of this section will be on the selection of the RCCI operating strategy (i.e., diesel injection strategy and gasoline-to-diesel ratio). In addition, a comparison between methanol and gasoline as the port injected fuel will be made. At mode 1 - 1500 rpm and 2.3 bar IMEP, HCCI calculations showed that the optimum combustion phasing is achieved without using gasoline [16]. In addition, premixing the gasoline to such a low equivalence ratio (< 0.2) results in significant unburned hydrocarbon emissions. Therefore, the RCCI strategy reverts to a diesel low temperature combustion (LTC) strategy at this condition. Note that because the conventional diesel injection hardware is being used, the RCCI engine has the ability to revert to a diesel strategy in any portion of the operating map where CDC or LTC are more desirable than RCCI. The results at this mode will be discussed in the next section: the piston bowl comparison.

Previous RCCI research has shown that the combustion phasing is predominantly determined by the low-to-high reactivity fuel ratio and not the diesel injection timing. Operating on gasoline and diesel fuel, Kokjohn et al. [11] demonstrated in a light-duty engine that the combustion phasing (CA50) is changed by 2° CA per percent change in the gasoline-to-diesel ratio. On the contrary, Walker et al. [17] demonstrated that for a single diesel injection timing, the level of combustion phasing control is only 0.2° CA per degree change in the start of injection. This level of combustion phasing control is an order of magnitude less. Thus, for a given set of boundary conditions, (intake pressure and temperature, EGR, and engine speed) the combustion phasing is mainly set by the low-to-high reactivity fuel ratio and the high reactivity fuel injection strategy is a secondary effect. Therefore, in the current study the ratio of low-to-high reactivity fuel was predominantly used to set the combustion phasing (i.e., CA50). The targets for combustion phasing are loosely defined as 0° to 3° ATDC at lower loads and are progressively retarded as the engine load is increased to avoid excessive pressure rise rates (i.e., 10 bar/deg. or greater).

Once the combustion phasing was set through ratio of low-to-high reactivity fuel, the effect the diesel injection strategy (i.e., number and timing of injections) has on engine performance and emissions has been studied. Thus, before making comparisons between gasoline and methanol as the low reactivity fuel, a favorable diesel injection strategy needed to be determined for each port injected fuel. For the sake of brevity, not all operating modes will be discussed, but a select number of modes will be discussed in detail in order to explain why the final diesel injection strategy for the entire operating map was chosen. The final operating strategies and results for both methanol/diesel and gasoline/diesel can be seen in the [Appendix](#).

RCCI (Modes 2 - 8) Low Reactivity Fuel: Gasoline High Reactivity Fuel: Diesel

[Figure 9](#) shows a comparison of three different injection strategies for gasoline/diesel RCCI at mode 2 - 1500 rpm and ~ 3.9 bar IMEP. As can be seen from the figure, the combustion phasing is essentially unchanged by the different injection strategies because the gasoline/diesel ratio was fixed at $\sim 83\%$ percent gasoline. The gross indicated efficiency and combustion efficiency are relatively unchanged, with incremental improvements experienced with the triple injection. The small improvements in engine performance observed with the triple injection come at a cost: the NOx emissions are increased by an order of magnitude. For steady state engine operation, NOx targets have been established by Cooper et al. [18] to meet U.S. Tier 2 Bin 5 emissions mandates for light-duty applications. Considering the majority of the light-duty regulatory cycle is focused on low load operation, Cooper et al. have shown that very low NOx emissions are needed at low load (i.e., less than 0.05 g/kW-hr). Thus, for this operating condition a single diesel injection has been chosen. Note that all three injection strategies yield soot emissions that are very low, < 0.008 g/kW-hr (FSN < 0.06).

[Figure 10](#) shows a similar comparison at mode 4 - 1900 rpm and ~ 3.7 bar IMEP. At this operating condition a single injection yields the most favorable results, with a 5% absolute improvement in combustion efficiency over the double and triple injection cases. Again, the soot emissions are very low and therefore not shown in the figure.

Finally, [Figure 11](#) shows a diesel injection strategy comparison for mode 5 - 1900 rpm and ~ 5.8 bar IMEP. At the higher load condition (> 5 bar IMEP) the triple diesel injection results in the combustion phasing being too advanced and thus only a single and double injection were investigated. As can be seen from the figure, the single and double injections are nearly identical in every aspect. This was the case at all operating conditions above 5 bar IMEP and typically the double injection was selected, as can be seen in the [Appendix](#).