

Piston Bowl Optimization for RCCI Combustion in a Light-Duty Multi-Cylinder Engine

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ABSTRACT

Reactivity Controlled Compression Ignition (RCCI) is an engine combustion strategy that produces low NO_x and PM emissions with high thermal efficiency. Previous RCCI research has been investigated in single-cylinder heavy-duty engines [1,2,3,4,5,6]. The current study investigates RCCI operation in a light-duty multi-cylinder engine at 3 operating points. These operating points were chosen to cover a range of conditions seen in the US EPA light-duty FTP test. The operating points were chosen by the Ad Hoc working group to simulate operation in the FTP test [7-8]. The fueling strategy for the engine experiments consisted of in-cylinder fuel blending using port fuel-injection (PFI) of gasoline and early-cycle, direct-injection (DI) of diesel fuel. At these 3 points, the stock engine configuration is compared to operation with both the original equipment manufacturer (OEM) and custom machined pistons designed for RCCI operation. The pistons were designed with assistance from the KIVA 3V computational fluid dynamics (CFD) code. By using a genetic algorithm optimization, in conjunction with KIVA, the piston bowl profile was optimized for dedicated RCCI operation to reduce unburned fuel emissions and piston bowl surface area. By reducing these parameters, the thermal efficiency of the engine was improved while maintaining low NO_x and PM emissions. Results show that with the new piston bowl profile and an optimized injection schedule, RCCI brake thermal efficiency was increased from 37%, with the stock EURO IV configuration, to 40% at the 2,600 rev/min, 6.9 bar BMEP condition, and NO_x and PM emissions targets were met without the need for exhaust after-treatment.

CITATION: Hanson, R., Curran, S., Wagner, R., Kokjohn, S. et al., "Piston Bowl Optimization for RCCI Combustion in a Light-Duty Multi-Cylinder Engine," *SAE Int. J. Engines* 5(2):2012, doi:10.4271/2012-01-0380.

INTRODUCTION

Rising fuel costs and a focus on the reduction of greenhouse gases has driven the need for increased efficiency from the internal combustion engine. This need for increased efficiency has placed increased attention on the diesel engine due to its higher efficiency compared to other engines. Furthermore, current and future legislation place stringent standards on the exhaust emissions and vehicle fuel efficiency. These demands require the development of advanced engine technologies, which can simultaneously yield high efficiency and low emissions. Many combustion 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 combustion (LTC). LTC is desired because low combustion temperatures result in NO_x reductions, due to the high

activation energy of NO formation reactions [8]. In addition, low temperature combustion strategies typically utilize injection timings that allow for increased air/fuel mixing prior to the start of combustion; thus, rich regions are reduced and soot formation is inhibited.

Many researchers have shown that LTC strategies such as Homogeneous Charge Compression Ignition (HCCI) and Premixed Charge Compression Ignition (PCCI) are promising techniques for simultaneous NO_x and soot reduction [10,11,12,13,14,15]. 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 LTC. For instance, gasoline has a high volatility; thus, evaporation is rapid and a premixed charge can be obtained using port fuel-injection. However, because gasoline resists auto-ignition, it becomes difficult to achieve combustion at low-



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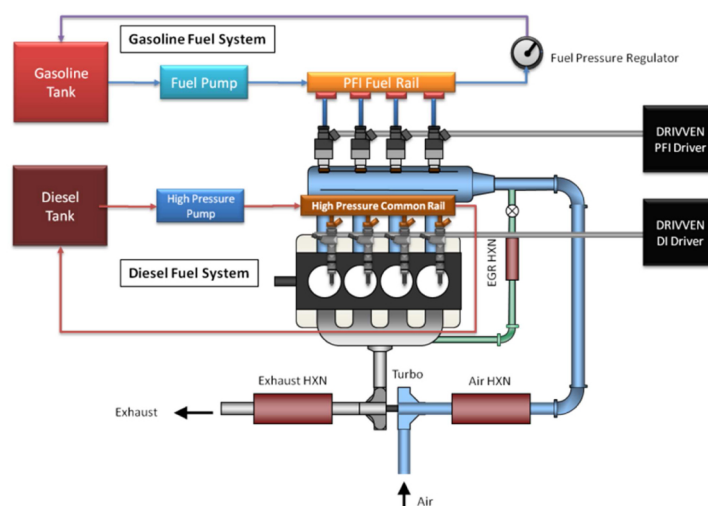


Figure 1. Diagram of the engine lab setup

Table 1. GM 1.9L engine geometry

Displacement (L)	1.9
Number of Cylinders	4
OEM Geometric Compression Ratio (-)	17.5:1
Modified Compression Ratio (-)	15.1:1
Bore x Stroke (mm)	82.0 x 90.4
Valve Timings (°ATDC firing, measured at 0.2 mm valve lift)	EVO = 128 EVC = 356 IVO = 361 IVC = 568

load conditions [16,17]. Conversely, diesel fuel has superior auto-ignition qualities; however, this can result in difficulty controlling the combustion phasing as engine load is increased.

To find the optimum fuel for HCCI over a wide engine speed/load range, experiments were performed by Bessonette et al. [18]. They suggested that the best fuel for HCCI operation may have auto-ignition qualities between those of diesel fuel and gasoline, depending on the engine speed and load. Based on these results and experiments by Inagaki et al. [19], the RCCI strategy was developed to allow the fuel reactivity to be optimized at all speeds and loads, providing an increased engine operating range for premixed LTC. Additional information on RCCI can be found in the References [20,21,22,23,24,25].

EXPERIMENTAL SETUP

The engine experiments were performed at Oak Ridge National Laboratory (ORNL), and used a 2007 GM 1.9L 4 cylinder light-duty diesel engine. The engine utilized all stock hardware, except for the high pressure EGR heat exchanger, gasoline PFI system and the custom machined pistons for RCCI operation. Also of note, is that the OEM ECU was replaced by a Drivven engine controller to allow full access

for control of all engine parameters and sub systems, including the additional PFI system. The lab setup is shown in [Figure 1](#) and the engine geometry is summarized in [Table 1](#).

EMISSIONS AND PRESSURE MEASUREMENTS

The exhaust gas emissions measurements were taken with a conventional 5 gas bench, measuring intake O_2 and CO_2 , exhaust O_2 and CO_2 , HC, CO, and NO_x . Emissions measurements were taken after the engine had reached steady state for 30 seconds and were then averaged for 180 seconds. Smoke readings were taken with an AVL 415s Smoke Meter with an average of 5 measurements. High speed combustion data were acquired using Kistler model 6058A pressure sensors installed in the glowplug ports of all 4 cylinders. Individual Kistler type 5010 Dual-Mode Amplifiers were used to process the pressure signals. The built-in combustion package from Drivven and a custom MATLAB code were used to post process the data. Cylinder pressure measurements were taken for 300 cycles. The ensemble average of the 300 cycles was then averaged over all 4 cylinders for all the pressure based calculations.

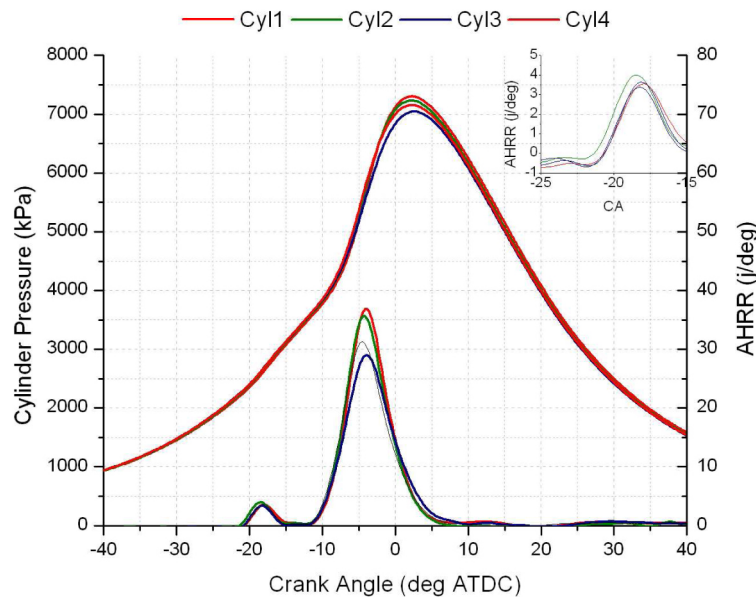


Figure 2. Example engine operating condition showing the balanced combustion phasing and load over all 4 cylinders

CYLINDER BALANCING

As can be expected with a multi-cylinder engine, there are significant variations in the initial conditions, i.e., cylinder wall temperature, EGR, trapped mass, fuel rail pressure, etc. of each cylinder. With a kinetically controlled combustion strategy like RCCI, these imbalances can lead to decreased performance due to overly advanced or delayed combustion in each cylinder. Exhaust emissions can also vary significantly from each cylinder with these variations in combustion phasing. To adjust for these variations, the total fueling and ratio of gasoline to diesel fuel for each cylinder was varied to match IMEP and combustion phasing over all 4 cylinders. The balancing was accomplished in the Drivven code by applying an “adjustment factor” multiplier to the PFI and DI duration command on each cylinder. First, for each cylinder, the DI and PFI adjustment factors were tuned to vary the ratio of gasoline:diesel fuel to match CA50 over all 4 cylinders. Next, once the combustion phasing was balanced, the PFI and DI adjustment factors were then raised or lowered in equal amounts to match IMEP over all 4 cylinders. Typically, the PFI and DI adjustment factors needed to be varied by $\pm 10\%$ from cylinder to cylinder. Sample results of the cylinder balancing for an arbitrary operating condition are shown in [Figure 2](#).

FUEL SYSTEMS

The RCCI engine experiments were performed using port fuel-injection of gasoline, and direct-injection of ultra-low sulfur diesel fuel (ULSD). The OEM common-rail injector was used for the DI fuel system and conventional spark ignition gasoline injectors were used for the additional PFI system. The specifications for both injectors are given in [Table 2](#) and [Table 3](#), respectively.

Table 2. Port fuel injector specifications

Injection Pressure (Bar)	3.8
Included Spray Angle	15°
Number of Holes	2
Steady Flowrate (g/s)	2.74

Table 3. Common rail injector specifications

Maximum Injection Pressure (Bar)	1,500
Included Spray Angle	143°
Number of Holes	7
Steady Flowrate (cc/30 sec)	440

The gasoline used in the study was a certification grade fuel (UTG 96) from CP Chem. with properties shown in [Table 4](#). Similarly, the diesel fuel chosen was also a certification grade ULSD fuel from CP Chem., shown in [Table 5](#).

Table 4. UTG 96 fuel specifications

Initial Boiling Point (°C)	33.9
Temperature 10% evaporated (°C)	50
Temperature 50% evaporated (°C)	107.2
Temperature 90% evaporated (°C)	150
Final Boiling Point (°C)	162.8
Lower Heating Value (MJ/kg)	43.12
MON	88.2
RON	96.1
Anti-Knock Index ((RON+MON)/2)	92.15
Ethanol (%)	0
Specific Gravity (@ 15.6 °C)	0.739
Sulfur (ASTM D5453) (PPM)	29.6

Table 5. ULSD fuel specifications

Initial Boiling Point (°C)	38.9
Temperature 10% evaporated (°C)	69.4
Temperature 50% evaporated (°C)	105
Temperature 90% evaporated (°C)	160.6
Final Boiling Point (°C)	215.6
Lower Heating Value (MJ/kg)	42.91
Specific Gravity (@ 15.6 °C)	0.845
Cetane Number (-)	46
Sulfur (ASTM D5453) (PPM)	9.9

RCCI PISTON DESIGN

Typically, conventional diesel engines are designed for non-premixed fuel burning in a mixing controlled combustion mode. This type of combustion requires precise piston geometry to assist fuel/air mixing and ensure complete combustion. Additional mixing from the piston bowl shape is not needed in RCCI due to the early fuel injections, which allow additional time for sufficient mixing before combustion. Therefore, a dedicated RCCI engine requires a bowl shape that is optimized for RCCI operation to fully realize the benefits of this premixed combustion strategy.

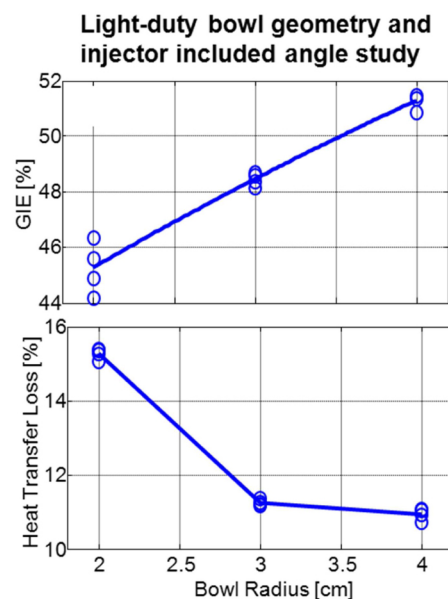
Previous, and ongoing, research has shown RCCI combustion can benefit from a different piston bowl than conventional diesel combustion (CDC) [27, 28]. The goal for this work was to use an optimized RCCI piston in a multi-cylinder engine (MCE). The light-duty RCCI piston bowl design was based on a 2nd generation heavy-duty RCCI piston with a compression ratio of 14.88:1, currently being used in the UW-Madison heavy-duty single-cylinder engine. Based on this initial design, CFD engine simulations using the KIVA code were used to examine bowl shape parameters to increase thermal efficiency and reduce emissions of RCCI operation in a light-duty engine. The two parameters varied in the piston design were the bowl diameter and compression ratio. Because the model can explore the limits of the design parameters, the bore radius was varied from a narrow, diesel-

type bowl to a nearly flat piston. The bowl radius varied from 49 to 98% of the piston radius. The initial engine operating conditions used in the model to test the piston bowl design were chosen as representative of RCCI cases previously tested. The specific conditions for the CFD simulations are shown in Table 6.

Table 6. KIVA CFD Simulation Parameters

Engine speed (rev/min)	1,900
Compression Ratio (-)	16.1
IMEP (bar)	9
Intake Pressure (bar)	1.86
EGR (-)	0.41
Intake Temperature (°C)	40
Bowl radius (cm)	2 to 4
Compression Ratio (-)	15.25 to 18.1

As seen in Figure 3, for a piston bowl radius from 2 to 4 cm as the bowl diameter increased, the gross indicated thermal efficiency (GIE) also increased. This increase in GIE was caused by a reduction in heat transfer losses due to the reduced surface area of the wider bowl.

**Figure 3. Simulated RCCI piston bowl radius vs. GIE**

Based on these results, the bowl radius was chosen to be as close to 4cm as could be safely machined into the piston blank. This resulted in a wide bowl that covered most of the piston. Next, the compression ratio needed to be chosen. Again, the CFD simulations were used to investigate efficiency and emissions performance, at a constant radius of 4cm, with various compression ratios. As seen in Figure 4, for a constant intake temperature of 40°C, the compression ratio was increased from 15.25 to 18.1. To minimize combustion phasing effects on the piston design, the combustion phasing was kept constant by changing the gasoline:diesel fuel ratio to account for changes in

compression ratio, increasing from 80% to 91%. Small changes to the gasoline:diesel ratio are not considered significant to the piston design as they have only minor impact to the engine performance and emissions, seen in Refs [1,2,3,4,5,6]. As expected, the higher compression ratio led to higher gross indicated efficiency. Roughly half of the efficiency increase was caused by the decreased unburned hydrocarbon emissions with increased compression ratio, as seen in Figure 5. Increased compression ratio leads to higher in-cylinder temperatures at TDC, which decrease HC emissions via increased fuel oxidation rates. The other half of the increased gross thermal efficiency was caused by the higher Otto cycle efficiency from the increased compression ratio. This higher compression ratio leads to a higher expansion ratio, which allows for increased work extraction, increasing the indicated efficiency.

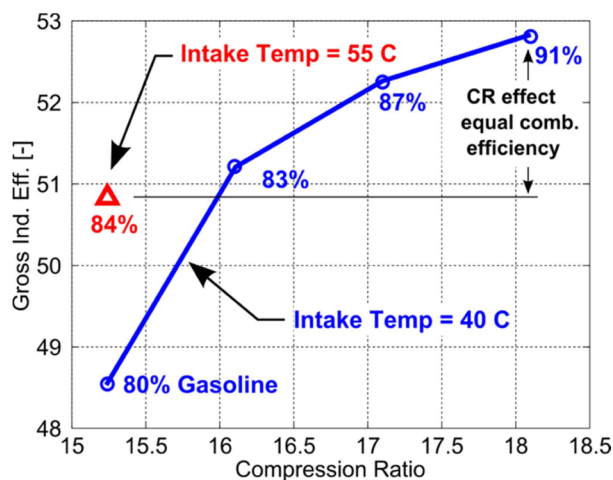


Figure 4. Gross indicated efficiency vs. compression ratio at a constant intake temperature. The gasoline fraction is the percent shown at each compression ratio.

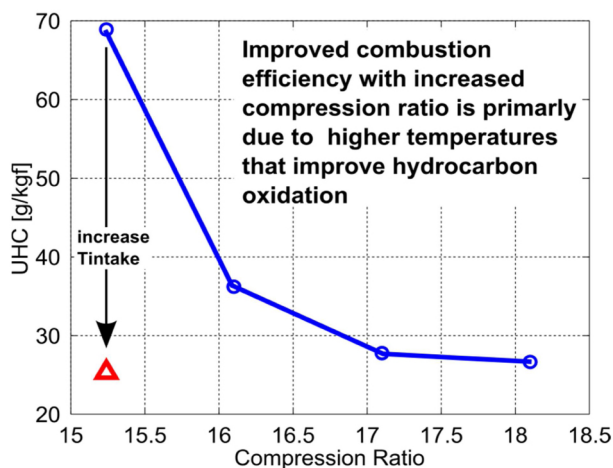


Figure 5. HC as a function of compression ratio for a constant intake temperature.

HC emissions can be decreased at lower compression ratio by increasing the intake temperature from 40 to 55°C, seen in Figure 5. As previously stated, this increase in combustion efficiency can make up about half of the efficiency gain from increasing the compression ratio alone. Based on these results, a geometric compression ratio of 16.1:1 was chosen to allow for improved low load operation with acceptable high load operation due to the light duty vehicle operating more exclusively at low loads. This compression ratio also allows reasonable intake temperatures of 40-60°C to be utilized.

Based on the selected compression ratio and bowl diameter, the general bowl profile from the 14.88:1 compression ratio heavy-duty RCCI piston, as shown in Figure 6, was directly scaled to fit the light-duty piston blank with a compression ratio of 16.1:1, shown in Figure 7 Figure 8. The bowl radius was initially made as close to the 4cm target as possible, but due to structural concerns over the bowl to ring-land clearance; it was reduced to 3.721cm, as shown in Figure 8. This distance was chosen to maintain 0.761cm between the bowl and the top ring land. The modified piston profile can be compared to the narrow, deep re-entrant bowl of the OEM piston, shown in Figure 9.

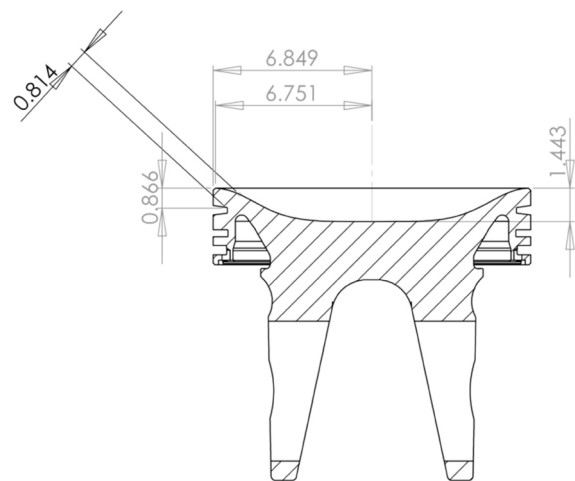


Figure 6. 14.88 compression ratio UW-Madison Heavy-duty RCCI piston