

Use of Low-Pressure Direct-Injection for Reactivity Controlled Compression Ignition (RCCI) Light-Duty Engine Operation

N. Ryan Walker, Adam B. Dempsey, Michael J. Andrie and Rolf D. Reitz
University of Wisconsin-Madison

ABSTRACT

Reactivity controlled compression ignition (RCCI) has been shown to be capable of providing improved engine efficiencies coupled with the benefit of low emissions via in-cylinder fuel blending. Much of the previous body of work has studied the benefits of RCCI operation using high injection pressures (e.g., 500 bar or greater) with common rail injection (CRI) hardware. However, low-pressure fueling technology is capable of providing significant cost savings. Due to the broad market adoption of gasoline direct injection (GDI) fueling systems, a market-type prototype GDI injector was selected for this study. Single-cylinder light-duty engine experiments were undertaken to examine the performance and emissions characteristics of the RCCI combustion strategy with low-pressure GDI technology and compared against high injection pressure RCCI operation. Gasoline and diesel were used as the low-reactivity and high-reactivity fuels, respectively. GDI injection pressures range from 150 to 200 bar, while the CRI pressures range from 250 to 500 bar. Start of injection (SOI) timings ranged from -35° aTDC and -115° aTDC. The experimental results show comparable engine performance and emissions output, but with slight reductions in overall combustion efficiency when using low-pressure fueling with the stock re-entrant piston. CFD simulations were also performed to aid in visualization of the in-cylinder fuel distributions, which are controlling factors for RCCI combustion. By utilizing an optimized RCCI piston geometry, equivalent RCCI combustion performance can be achieved under low-pressure fueling, at moderate and high loads. The optimized geometry also allows for significant increases in thermal efficiency, with peak efficiencies over 47% observed.

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INTRODUCTION

The internal combustion engine (ICE) is ubiquitous as an economical and reliable energy conversion device for both stationary and mobile applications. However, stringent limits on exhaust emissions have driven the development of advanced combustion technologies for improved engine efficiency and reduced emissions.

Historically, the most efficient ICEs have been compression ignition (CI), namely diesel engines. Employing unthrottled air induction and near-top dead center (TDC) high-pressure common rail injection (CRI¹), the lifted diffusion flame provides a pathway for the formation of particulate matter (PM) and thermal oxides of nitrogen (NO_x), as proposed by Dec *et al.* [1]. As a result of the near-TDC injection timings the combustion process is mixing controlled, with Pickett and Siebers [2] describing that, by

increasing the air entrainment of the fuel jet, the increased free oxygen works towards reducing fuel-rich mixtures where soot formation is likely to occur.

More recent efforts to improve engine efficiency and reduce pollutant emissions have led to the development of combustion strategies that are controlled less by mixing and more by chemical kinetics [3]. Premixed compression ignition (PCI) processes, such as premixed charge compression ignition (PCCI) and homogenous charge compression ignition (HCCI), achieve low temperature combustion (LTC) characteristics by preparing a well-mixed air-fuel charge. When this charge ignites, the absence of fuel-rich regions allows for sootless and low-temperature combustion. However, because these methods are governed by chemical kinetics, the operating range for these systems is generally confined to a relatively narrow window [4]. Figure 1, as originally presented by Neely *et al.* [5] and modified by

¹CRI is used in reference to high-injection pressure technology such as direct-injection systems found on modern diesel engine platforms.

Kokjohn [6], shows the combustion pathway regions for various high-efficiency CI combustion strategies presented in the Φ -T space as first introduced by Kamimoto *et al.* [7].

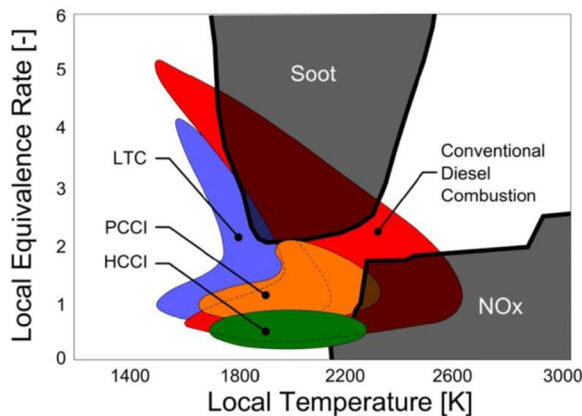


Figure 1. Combustion pathway regions for various high-efficiency CI strategies in Φ -T space [5, 6].

Inagaki *et al.* [8] experimented with a novel dual-fuel PCI strategy as a means for reducing the amount of exhaust gas recirculation (EGR) required for clean and efficient combustion. As a method of extending the load-speed range of the dual-fuel PCI combustion regime Kokjohn *et al.* [9] introduced the reactivity controlled compression ignition (RCCI) combustion strategy. RCCI utilizes two fuels with different auto-ignition characteristics to achieve an in-cylinder air-fuel blend with a fuel reactivity gradient [10, 11]. By adjusting the fuel reactivity stratification, control over heat release rate and combustion phasing is gained, as described by Kokjohn *et al.* [12].

In recent years, gasoline direct injection (GDI²) engines have become widely available in the light-duty automotive market. Compared to CRI fueling systems, GDI fueling systems cost and weigh less, and also have the ability to minimize parasitic engine load due to reduced fuel pumping requirements.

Due to the large-volume production runs of light-duty automotive engines, there is a strong economic case for the manufacturer to minimize costs in any area possible. One such area is the fuel injection system, which can consist of many parts such as injectors, fuel rails, lines, fuel pumps and associated control hardware. In 2008, the MARTEC Group performed a market research study which investigated the variable costs of advanced fuel economy technologies [13]. Taking a 4-cylinder gasoline port fuel injected platform as the baseline (with a total fuel injection system cost of \$32) the following fuel injection system upgrade costs were estimated as shown in Table 1:

Table 1. 4-cylinder direct-injection fuel system incremental upgrade costs over PFI baseline, from Ref. [13].

4-Cylinder Side-Mount GDI.....	+\$227
4-Cylinder Center-Mount GDI	+\$365
4-Cylinder Center-Mount CRI.....	+\$707

For this consideration the PFI system will not be replaced, but rather kept on the light-duty vehicle as part of the conversion to an RCCI-capable engine. From the costs given above, it is seen that an upgrade to a high-pressure center-mount CRI system results in a cost premium of 94% over an upgrade to a low-pressure center-mount GDI system. Based on the fuel injection system costs alone, there is strong incentive to work towards achieving clean and efficient RCCI engine operation with low-pressure injection hardware such as a center-mounted GDI fuel injection system.

In addition, much of the previous body of work on RCCI combustion has used high-pressure common rail injection at 500 bar or greater [14, 15, 16, 17, 18, 19, 20]. The objective of the current study is to investigate the feasibility of low-pressure fueling under RCCI operation and to develop an understanding of the underlying processes controlling the in-cylinder fuel mixing and preparation required to achieve clean and efficient RCCI combustion.

EXPERIMENTAL SETUP

Engine Specifications

A single-cylinder four-stroke diesel engine based on the GM/Fiat JTD 1.9L platform was used for this investigation. With a bore of 82.0 mm and stroke of 90.4 mm this engine is representative of a light-duty automotive diesel engine with four valves per cylinder with a dual overhead cam valve actuation arrangement. A complete description of the engine is available in Refs. [21, 22]. The pressure-volume-based effective compression ratio is 16.7:1; the geometric compression ratio is 17.5:1. Figure 2 provides a schematic of the experimental engine layout and Table 2 lists the engine specifications.

Table 2. Engine specifications.

Base Engine	GM/Fiat JTD 1.9L DI Diesel
Number of cylinders	1
Effective Compression Ratio.....	16.7:1
Displacement.....	0.477 L
Stroke	90.4 mm
Bore	82.0 mm
Connecting Rod Length.....	145.54 mm
Number of intake valves.....	2
Number of exhaust valves	2
Intake Valve Closing	-132° aTDC
Exhaust Valve Opening.....	112° aTDC
Swirl Ratio.....	2.2

²GDI is used in reference to low-injection pressure technology such as direct-injection systems found on modern gasoline engine platforms.

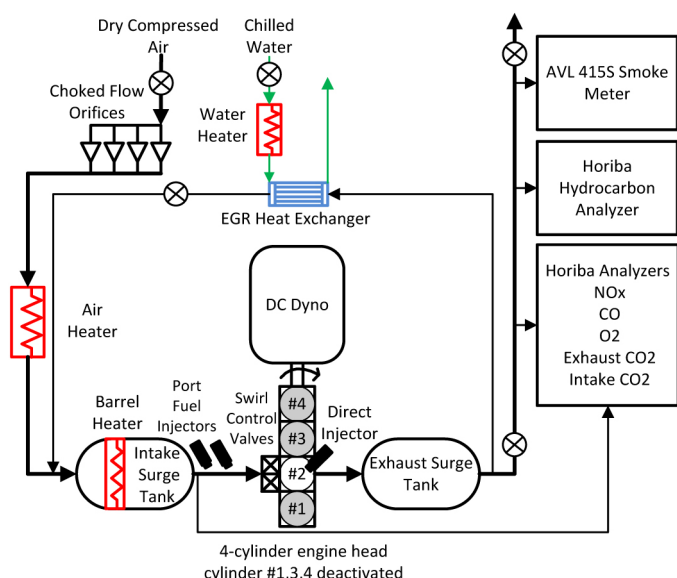


Figure 2. Experimental engine layout.

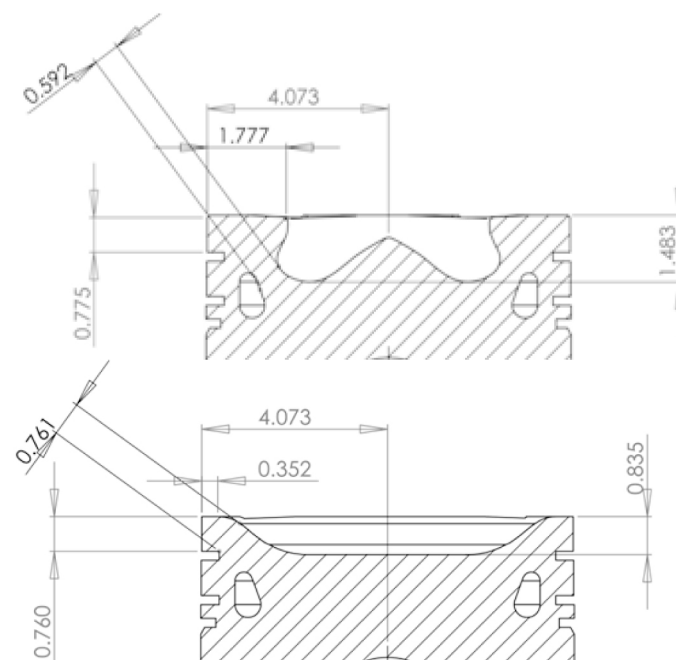


Figure 3. Stock CDC piston geometry (top) and optimized RCCI piston geometry (bottom), from Ref. [20].

Data Acquisition

System controls and data acquisition are handled through National Instruments LabView software. A Kistler 6125A pressure transducer in series with a Kistler 4010 charge amplifier measures and transmits the in-cylinder pressure measurements. A 1440 pulse per revolution shaft encoder allows for a resolution of 0.25 crank angle (CA). 300 consecutive cycles of cylinder pressure traces are acquired at each experimental point. For emissions sampling, Horiba gas benches were used to measure emissions species that include NO_x (CLA-22A), HC (FIA-236), O₂ (MPA-21A), CO (AIA-23), CO₂ (AIA-23), and EGR CO₂ (AIA-23), all with $\pm 1\%$ instrument error. All species were measured on a dry basis, except for HC which was measured on a wet basis. (HC is subsequently converted to a dry basis; all species are reported as such.) An AVL 415S smoke meter measured PM emissions.

Piston Geometry

In this study, two piston geometries are used: a stock re-entrant piston bowl designed for conventional diesel combustion (CDC), and an optimized RCCI piston geometry that has been shown effective in reducing heat transfer losses and improving cycle thermal efficiency under RCCI engine operation [19, 20] (in a heavy-duty application, indicated thermal efficiencies over 59% were experimentally demonstrated in Ref. [23] with a similar optimized RCCI bowl profile). Both pistons have a matched compression ratio. Figure 3 highlights the variations in bowl shape between the two pistons. The stock CDC piston has a TDC combustion chamber surface area of 174.01 cm² and the optimized RCCI piston has a TDC combustion chamber surface area of 152.5 cm².

Fuel Injection System

In this work, a premixed charge of low-reactivity fuel (i.e., gasoline) was delivered using an automotive-type port fuel injection (PFI) system and the high-reactivity fuel (i.e., diesel) was delivered using a common rail type direct-injection system through either a CRI or GDI injector. The PFI fuel was injected at -200° aTDC during the intake stroke. NI Drivven software drives both the direct injection and port fuel injectors; Mototune software controls the fuel rail pressure. The gasoline had an anti-knock index (AKI) of 91 and the diesel fuel had a cetane number of 42. The specifications for the GDI and CRI injectors are provided in Table 3.

Table 3. Injector specifications

CRI Injector	Bosch CRIN2
Number of Holes	6
Hole Diameter	154 μ m
Spray Angle.....	144°/90°
GDI Injector	Magneti Marelli IHP3
Number of Holes	6
Hole Diameter	150 μ m
Spray Angle.....	142°/120°

RESULTS AND DISCUSSION

Part 1: Evaluation of Low-Pressure Direct-Injection RCCI Engine Operation with CDC Piston

Engine operating conditions

To study the effects of fuel mixing and mixture preparation, a start of injection (SOI) timing sweep was performed for each injector at two different injection pressures. Boosted RCCI combustion was performed at a nominal 6 bar IMEP with a fuel mass split of 85% gasoline and 15% diesel. A single injection strategy was used for this study. Each SOI timing sweep was bounded by a stability limit of 10% COV of IMEP at the earliest injection timings and a peak pressure rise rate (PPRR) limit of 10 bar/deg at the latest injection timings. Table 4 shows the experimental engine operating conditions for the work in Part 1.

Table 4. Engine operating conditions for Part 1 experiments.

Engine Speed	1900 rev/min
Nominal Engine Load.....	6 bar IMEP _g
Total Fuel Energy	640 J/cycle
GDI Injection Pressure.....	150/200 bar
GDI Spray Angle	142°
CRI Injection Pressure	250/500 bar
CRI Spray Angle.....	144°
SOI Timings	-35° to -115° aTDC
PFI Mass Fraction.....	0.85
DI Mass Fraction	0.15
Intake Pressure.....	1.5 bar
Intake Temperature	45 °C
Global Equivalence Ratio, Φ	0.3
Piston Geometry	Stock CDC

For the work presented in Part 1, four SOI timing sweeps were performed. For each injector, SOI timing sweeps were completed at two different injection pressures. Adhering to the operating constraints described above, each sweep began at a SOI timing of -35° aTDC and was advanced in increments of 10° CA until the load stability operating constraint was surpassed. All experimental points presented satisfy the imposed operating constraints. EGR is reported only when used.

Performance and emissions

Figure 4 displays the CA50 combustion phasing as a function of SOI timing. It is seen that the GDI injector, operating at 150 bar, has a broader SOI range compared to the CRI injector while meeting both the load stability and PPRR operating limits defined above. For each sweep the achievable phasing range is approximately 9° CA. Considering that more efficient engine operation occurs with combustion phasing after TDC, the CRI injector offers a

longer achievable combustion phasing range at a fixed gasoline percentage.

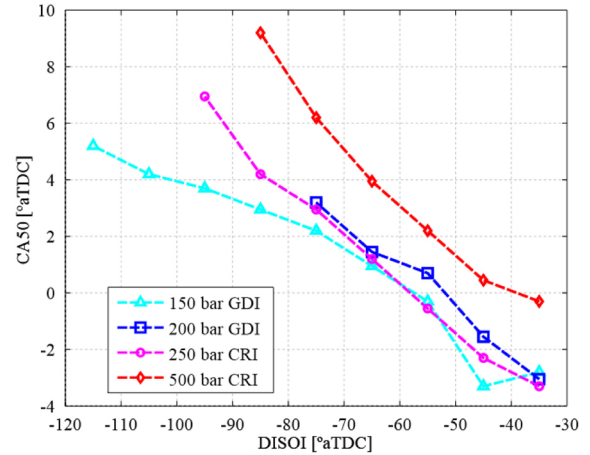


Figure 4. CA50 vs. SOI timing. Note that CRI offers increased control over combustion phasing compared to GDI.

Establishing a linear fit to each SOI sweep, a measure of combustion phasing control by SOI timing can be determined. For the 500 bar and 250 bar CRI sweeps, the phasing control is 0.19 and 0.17 °phasing/°SOI, respectively. The 200 bar and 150 bar GDI sweeps have a phasing control of 0.16 and 0.11 °phasing/°SOI, respectively. Therefore it can be noted that by increasing the injection pressure more control of combustion phasing by SOI timing can be obtained. To provide perspective on this level of control, Kokjohn *et al.* [10] showed that 2 °phasing/%gasoline was achieved by changing the total mass fraction of gasoline at 1900 rev/min. It is also worth noting that the latest injection timings give short mixing durations, which result in locally-rich regions of diesel fuel providing strong ignition sources with short ignition delays, thus advancing combustion. The earliest injection timings provide relatively long mixing durations, which result in reduced amounts of locally-rich pockets of diesel fuel, thus weakening the ignition sources along with increasing the ignition delay, allowing for combustion to retard.

From the measured HC and CO emissions, the cycle combustion efficiency can be calculated. It was assumed that the measured HCs have the same heating value as the fuel, and that CO has a heating value of 10.1 MJ/kg [24]. The fuel energy is calculated using the measured DI and PFI fuel flow rates as:

$$\dot{E}_{fuel} = \dot{m}_{gasoline} * LHV_{gasoline} + \dot{m}_{diesel} * LHV_{diesel} \quad (1)$$

The measured HC and CO emissions are used to calculate the combustion efficiency as:

$$\eta_{comb} = \frac{\dot{E}_{fuel} - \dot{m}_{HC} * LHV_{fuel} - \dot{m}_{CO} * LHV_{CO}}{\dot{E}_{fuel}} \quad (2)$$

Figure 5 shows the calculated combustion efficiency for each experimental point. It is seen that, for a proper combustion phasing after TDC, the CRI injector provides a 1% increase in combustion efficiency over the GDI injector. Overall combustion performance results show cycle combustion efficiencies in the 90-93% range. It is thought that unburned premixed gasoline residing in the squish/crevice regions of the cylinder is the major contributor to this relatively low combustion efficiency. Further discussion and experiments towards improving the overall cycle efficiency via optimized piston geometry are presented in Parts 3 and 4.

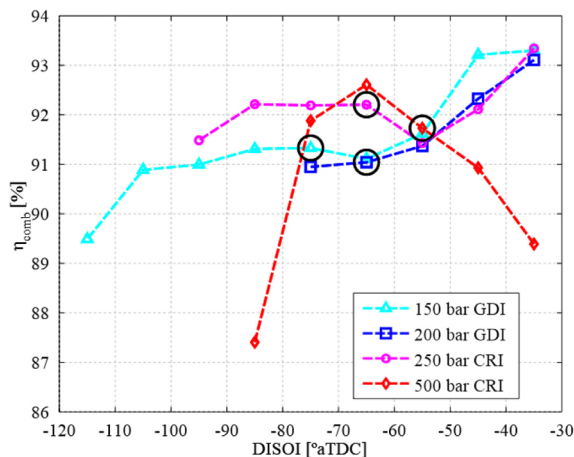


Figure 5. Combustion efficiency vs. SOI timing. In general, for proper RCCI combustion phasing (2-3° aTDC, circled), GDI shows 1% reduced combustion efficiency compared to CRI, with the stock CDC piston.

Measured NO_x emissions vs. SOI timing are displayed in Figure 6. The US EPA 2010 On-Highway NO_x standard of 0.27 g/kW-hr [25] is shown for reference. At both -35° and -45° SOI timings the only points that meet the NO_x limit are the 500 bar CRI conditions. All other test points for both injectors meet the limit. Due to the later injection timings of -35° and -45° aTDC, insufficient mixing is achieved before ignition occurs. As a result, near-stoichiometric diesel regions exist, leading to increased NO_x formation. For each injection pressure, SOI timings earlier than -45° aTDC provide sufficient mixing time to reduce near-stoichiometric regions for the diesel fuel.

Figure 7 shows measured soot levels for each sweep, and compares them to the US EPA 2010 On-Highway soot standard of 0.013 g/kW-hr [25]. Note that all test cases show acceptable soot levels. It is also observed that the GDI injector shows elevated soot levels at the later SOI timings compared to the CRI injector. It is possible that for later SOI timings, with the increase in in-cylinder density, the lower injection pressures develop a small amount of locally-rich regions which provide a pathway for soot formation.

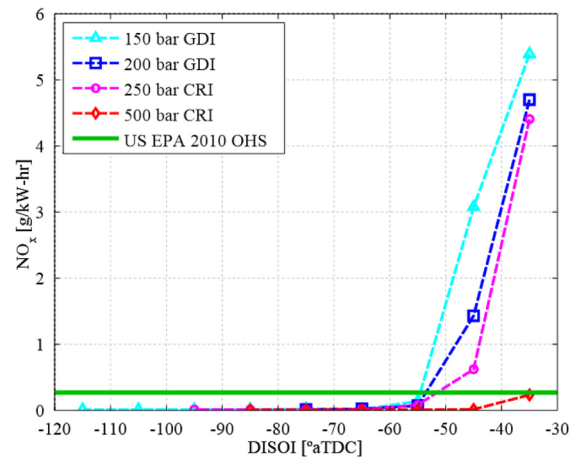


Figure 6. Oxides of nitrogen emissions vs. SOI timing. The US EPA 2010 On-Highway standard of 0.27 g/kW-hr is shown.

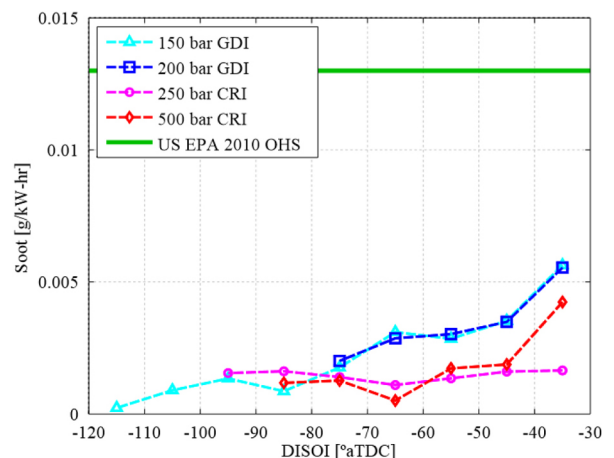


Figure 7. Soot emissions vs. SOI timing. The US EPA 2010 On-Highway standard of 0.013 g/kW-hr is shown.

Measured CO emissions are shown in Figure 8 as a function of SOI timing. It can be seen that, in general, CO levels are similar for each injector. For the 500 bar CRI sweep, the earliest and latest SOI timings (-85° and -35° aTDC, respectively) display significantly elevated levels of CO. This is caused by a slight decrease in combustion stability resulting from operating on the boundaries of proper mixture preparation, thus leading to more incomplete combustion and elevated CO output. It is interesting to note that, in general, CO and HC production is lowest at the latest injection timings, which can be attributed to the pre-TDC hotter-burning and more violent heat release rate.

Figure 9 shows measured HC emissions for each sweep. It can be seen that the GDI injector generates more HC. It is not yet understood why use of the GDI injector results in elevated HC emissions, but it is conceivable that the targeting of the individual spray plumes does not sufficiently disperse diesel fuel throughout the cylinder, leading to gasoline-only regions