

Fuel Reactivity Controlled Compression Ignition (RCCI) Combustion in Light- and Heavy-Duty Engines

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ABSTRACT

Single cylinder engine experiments were used to investigate a fuel reactivity controlled compression ignition (RCCI) concept in both light- and heavy-duty engines and comparisons were made between the two engine classes. It was found that with only small changes in the injection parameters, the combustion characteristics of the heavy-duty engine could be adequately reproduced in the light-duty engine. Comparisons of the emissions and performance showed that both engines can simultaneously achieve NOx below 0.05 g/kW-hr, soot below 0.01 g/kW-hr, ringing intensity below 4 MW/m², and gross indicated efficiencies above 50 per cent. However, it was found that the peak gross indicated efficiency of the baseline light-duty engine was approximately 7 per cent lower than the heavy-duty engine. The energy balances of the two engines were compared and it was found that the largest factor contributing to the lower efficiency of the light-duty engine was increased heat transfer losses. Detailed CFD modeling was used to explore options to reduce the heat transfer losses of the light-duty engine. It was found that by reducing the swirl ratio from 2.2 to 0.7, increasing the engine speed from 1900 to 2239 rev/min, and improving the combustion chamber geometry, the heat transfer losses in the light-duty engine could be reduced by the equivalent of 2 per cent of the fuel energy. The modeling showed that light duty engine could achieve 53 per cent gross indicated efficiency, while maintaining near zero NOx and soot, and an acceptable ringing intensity.

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 the diesel or compression ignition (CI) engine in the spotlight due to its superior fuel efficiency compared to spark-ignited (SI) engines. However, under conventional diesel operation regions of the combustion chamber pass through both rich- and lean-high temperature regimes, forming soot and NOx, respectively. In recent years many aftertreatment devices have been developed to eliminate soot and NOx emissions. However, these devices generally result in increased fuel consumption or consumption of a secondary reducing agent (e.g., urea) [1,2]. Therefore, to maximize overall engine efficiency, the need for aftertreatment-type emissions reduction should be minimized. Accordingly, reduction of NOx and soot in-cylinder has been investigated by many researchers. Most of the current strategies can be lumped into the category of premixed, low temperature combustion (LTC). Lower combustion temperatures result in NOx reduction due to the high activation energy of NO formation reactions [3]. In addition, utilizing long ignition delay times allows adequate time for mixing prior to the start of combustion; thus, rich regions are reduced and soot formation is inhibited. Many researchers have shown that Homogeneous Charge Compression Ignition (HCCI) and Premixed Charge Compression Ignition (PCCI) concepts are promising techniques for simultaneous NOx and soot reduction [4, 5, 6, 7, 8, 9, 10].

In addition to significant NOx and soot reductions, with appropriate control over combustion phasing, premixed LTC operation can provide fuel efficiency advantages compared to conventional mixing or diffusion controlled strategies. This fuel efficiency advantage is primarily due to two factors. First, premixed LTC often operates highly dilute, either by using high levels of recirculated exhaust gas (EGR) or high intake pressure. By operating highly dilute, flame

temperatures are reduced and heat transfer losses can be minimized. Secondly, efficiency can be increased due to improved start- and end-of-combustion (EOC) timings. The combustion process in premixed LTC is nearly entirely dominated by chemical kinetics [11], thus the duration of combustion can be significantly shorter than mixing or diffusion limited processes.

The possibility to simultaneously achieve high-efficiency, low-NO_x and soot combustion makes premixed compression ignition (PCI) strategies attractive. However, the need to create a very well mixed charge requires a significant amount of pre-combustion mixing, resulting in a de-coupling of the injection and combustion events. That is, in contrast to conventional mixing controlled diesel combustion, the injection event can no longer be used to control the start-of-heat release. Further, because PCI combustion is a kinetically controlled process, the combustion phasing is very sensitive to charge gas temperature. In the laboratory, where intake temperature and EGR can be precisely controlled, single fuel PCI combustion can demonstrate very promising results. However, in practice the charge gas temperature depends on many factors and varies with speed, load, and operating environment. To understand the motivation for investigating in-cylinder fuel blending, Figure 1 shows constant volume ignition delay calculations using a primary reference fuel (PRF) mechanism developed by Ra et al. [12] for neat iso-octane, neat n-heptane, and a 50-50 blend of iso-octane and n-heptane (i.e., PRF 50) [13]. For optimum thermal efficiency the combustion phasing must be precisely controlled and the engine must be able to accommodate changes in the operating conditions (e.g., charge temperature) while maintaining the optimum combustion phasing. Figure 1 shows that since each fuel has a unique ignition delay - temperature curve, each fuel will be limited to a narrow operating range where peak efficiency can be achieved. For example, if a representative temperature of 750 K is selected and the ignition delay required to achieve optimum combustion phasing is ~40° CA, neat diesel fuel would provide the optimum fuel reactivity to achieve peak efficiency. However, if the operating conditions are changed such that the representative temperature increased to 800 K, neat diesel fuel would result in overly advanced combustion phasing, increased heat transfer losses, and reduced thermal efficiency. However, by using in-cylinder fuel blending, peak efficiency could be maintained by moving from neat diesel fuel to a 50-50 blend of diesel and gasoline.

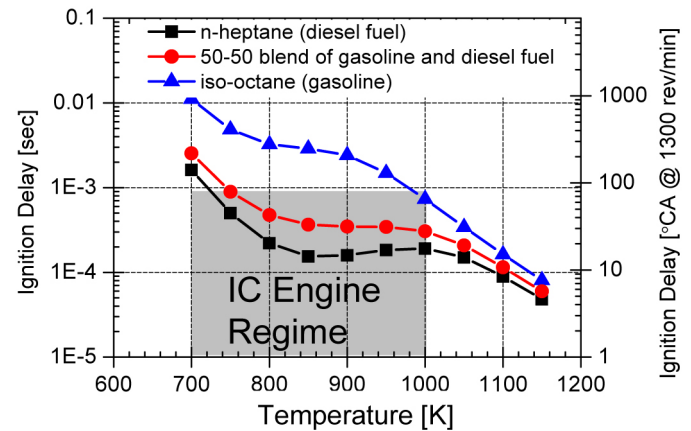


Figure 1. Constant volume ignition delay times for neat n-heptane, neat iso-octane, and a 50-50 blend of iso-octane and n-heptane calculated using the SENKIN [14] code and a reduced PRF mechanism [12]. This set of simulations was performed at an initial pressure of 70 bar and an equivalence ratio of 0.5. The shaded area shows temperatures and ignition delay times representative of those seen near TDC in a typical internal combustion (IC) engine.

Based on the previous discussion, the authors' have proposed a fuel reactivity controlled compression ignition (RCCI) combustion concept using in-cylinder blending of two fuels with different reactivities [7, 15]. Previous research has demonstrated the RCCI strategy in a heavy-duty single-cylinder research engine [7, 16, 17]. The results have demonstrated a high-level of control over the combustion process and an extension of the high-efficiency clean combustion regime¹. In this work, the RCCI concept is applied to a light-duty engine and comparisons are made between the light- and heavy-duty engine platforms.

HEAVY-DUTY RCCI BACKGROUND

Prior to applying RCCI combustion to the light-duty engine it is informative to review the RCCI results in the heavy-duty engine. Figure 2 shows the emissions and performance results for RCCI combustion in a heavy-duty Caterpillar Single Cylinder Oil Test Engine® (SCOTE) over a range of loads² from 4.6 to 14.6 bar IMEP [13]. It can be seen that RCCI combustion is able to achieve NO_x levels nearly an order of magnitude below the 2010 heavy-duty limits while achieving gross indicated efficiencies up to 56 per cent. It should be noted that the most optimization has been conducted at the 9.3 bar IMEP operating point, which may be the reason for the peak in the fuel efficiency at this point [13]. The very low

¹In this work the high-efficiency, clean combustion regime is defined as operation that results in efficiency equal to or greater than a conventional diesel engine, while meeting the EPA 2010 heavy-duty NO_x and soot emissions targets.

²For information regarding RCCI operation at loads below 4.6 and above 14.6 bar IMEP, the reader is referred to the authors' other works. Low-load operation is discussed in Hanson et al. [38] and high-load operation is discussed in Splitter et al [39].

NO_x levels are achieved through well mixed, highly dilute operation. Figure 2 shows that soot is below the 2010 heavy-duty limit for all but the 6 bar IMEP operating point. However, it is expected that with slight changes in the operating conditions, the soot levels could be held below the 2010 limits over the entire range of loads. Furthermore, even as load is increased up to 14.6 bar IMEP, the ringing intensity can be held below 3 MW/m². The results of Figure 2 show that RCCI combustion is a promising method to simultaneously achieve high-efficiency and low emissions.

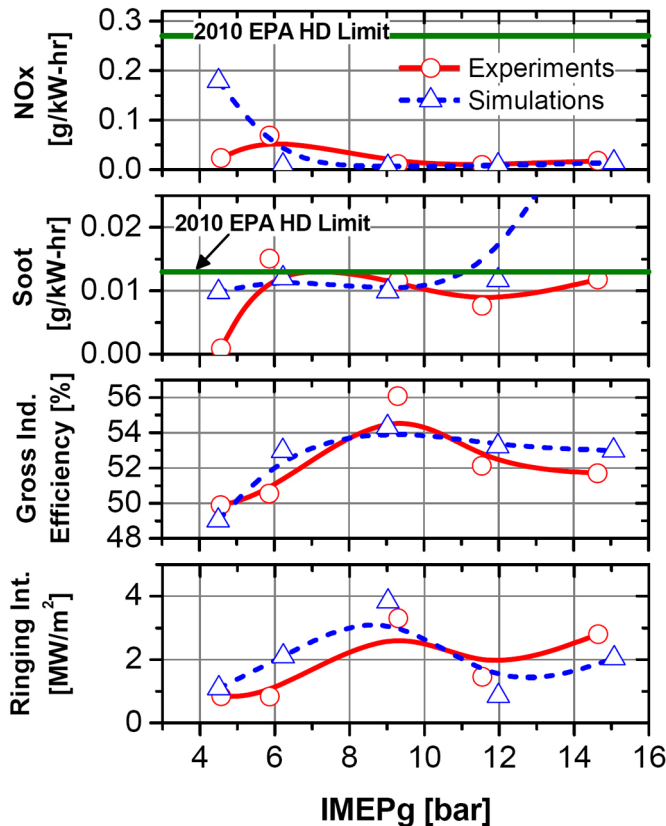


Figure 2. Emissions and performance of heavy-duty RCCI combustion over a range of loads [13]. The solid lines are results from the engine experiments and the dashed lines are CFD simulation results. The green horizontal lines on the NO_x and soot plots show the US EPA 2010 on-road heavy-duty truck limits, 0.268 g/kW-hr and 0.0134 g/kW-hr for NO_x and soot, respectively.

EXPERIMENTAL SETUP

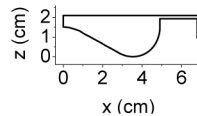
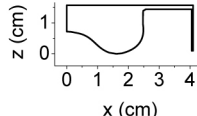
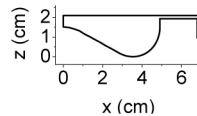
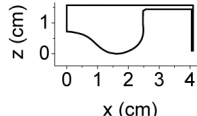
ENGINE SPECIFICATIONS

This work considers comparisons between a large-bore heavy-duty engine and a small-bore automotive-type light-duty engine. The specifications for both engines are given in Table 1. The heavy-duty engine is a Caterpillar® Single Cylinder Oil Test (SCOTE) engine and is typical of a line-haul application. This engine has a 13.72 cm bore, is

undersquare with a bore-to-stroke ratio of 0.83, and has a displacement of 2.44 L per cylinder. The heavy-duty engine has a geometric compression ratio of 16.1:1 and intake valve closure (IVC) and exhaust valve opening (EVO) timings of -143° and 130° ATDC, respectively. The heavy-duty engine has an open crater piston bowl geometry and a relatively quiescent combustion chamber with a swirl ratio of 0.7.

The light-duty engine is a single cylinder version of a GM 1.9 L and is typical of an automotive application. The light-duty engine is ~5 times smaller than the heavy-duty engine and has an 8.2 cm bore, a bore-to-stroke ratio of 0.91, and a displacement of 0.48 L per cylinder. The engine has an open crater-type piston bowl and a geometric compression ratio of 15.24:1. In contrast to the quiescent chamber of the heavy-duty engine, the stock light-duty engine has a much higher level of charge motion with a swirl ratio of 2.2. The IVC and EVO events occur at -132 and 112° ATDC, respectively. Notice that the TDC surface area-to-volume ratio of the light-duty engine is more than two times that of the heavy-duty engine.

Table 1. Engine specifications for the light- and heavy-duty engines.

	Heavy Duty	Light Duty
Engine	CAT SCOTE	GM 1.9 L
Displacement (L/cyl)	2.44	0.48
Bore (cm)	13.72	8.2
Stroke (cm)	16.51	9.04
Squish (cm)	0.157	0.133
CR	16.1:1	15.24:1
Swirl ratio	0.7	2.2
IVO (°ATDC)	335	344
IVC (°ATDC)	-143	-132
EVO (°ATDC)	130	112
EVC (°ATDC)	-355	-332
Number of valves	4	
TDC surface area-to-volume ratio (1/cm)	2.7	5.6
Piston bowl geometry	 heavy-duty  open crater	
Piston profile	 heavy-duty  open crater	

INJECTION SYSTEMS

This work uses in-cylinder fuel blending of gasoline and diesel fuel to control the combustion characteristics. In both engines the gasoline was delivered using automotive type

port-fuel-injection (PFI) systems and the diesel fuel was delivered using common rail type direct-injection (DI) systems. [Table 2](#) gives the specifications for the PFI system used in both engines and [Tables 3](#) and [4](#) give the specifications for the common rail systems.

Table 2. Port-fuel-injector specifications. Note that identical port-fuel-injectors were used for the light- and heavy-duty engines.

Steady flow rate @ 3 bar	12.5 cc/sec
Included spray angle	15°
Fuel pressure.....	5.17 bar

Table 3. Heavy-duty engine common rail injector specifications

Manufacturer	Bosch
Number of holes	6
Steady flow rate @100 bar	1000 cc/30sec
Hole diameter	250 µm
Included spray angle.....	145°
Injection pressure.....	800 bar

Table 4. Light-duty engine common rail injector specifications

Manufacturer	Denso
Number of holes	8
Steady flow rate @100 bar	440 cc/30sec
Hole diameter	128 µm
Included spray angle.....	130°
Injection pressure.....	500 bar

DATA ACQUISITION AND LABORATORY SETUP

The light- and heavy-duty engine laboratories are described in detail in Refs. [18] and [7], respectively. Thus, only brief descriptions of the laboratories are presented here. In the light-duty laboratory cylinder pressure was measured with a Kistler model 6125A piezoelectric cylinder pressure transducer and in the heavy-duty laboratory a Kistler model 6043 was used. Both laboratories used Kistler model 510 charge amplifiers and acquired cylinder pressure traces were averaged for 500 cycles.

Intake air was heated with two immersion style heaters and maintained to $\pm 1^\circ\text{C}$ of the set point using PID controllers. The intake air flow was measured using choked flow orifices and both the intake and exhaust system surge tank pressures were equipped with PID control to ± 0.7 kPa. PM measurements were performed with an AVL model 415s smoke meter. PM measurements of filter smoke number (FSN), mass per volume (mg/m^3) and specific emissions ($\text{g}/\text{kW}\cdot\text{hr}$) were related with the factory AVL calibration and averaged between 5, 2 L samples with paper saving mode off.

All gaseous emissions measurements were performed with a 5 gas emissions bench. The EGR rate was determined through the ratio of intake CO_2 to exhaust CO_2 levels.

FUELS

Commercially available 91.6 antiknock index ((RON+MON)/2) gasoline and ultra-low sulfur diesel fuel (ULSD) with a cetane number of 46 were used for all engine tests. [Tables 5](#) and [6](#) show an independent lab analysis of the fuel properties.

Table 5. Gasoline fuel properties

Distillation curve method	ASTM D86
Initial boiling point	38.9 °C
Temperature 10% evaporated	69.4 °C
Temperature 50% evaporated	105.0 °C
Temperature 90% evaporated	160.6 °C
Final boiling point	215.6 °C
Lower heating value	43.2 MJ/kg
MON.....	87.8
RON.....	95.6
(R+M)/2	91.6
Ethanol.....	0 %
H/C ratio	1.88
Density.....	756 kg/m^3

Table 6. Diesel fuel properties

Viscosity (@ 40 °C)	2.71 cSt
Surface tension (@ 25 °C)	30.0 dyne/cm
Lower heating value	42.5 MJ/kg
Cetane number.....	46.1
H/C ratio	1.74
Density.....	856 kg/m^3

COMPUTATIONAL MODEL

Computations were performed using the KIVA-3v release 2 code [19] with improvements to many physical and chemistry models developed at the Engine Research Center (ERC) [20, 21, 22, 23]. The KIVA-3v code is coupled with the CHEMKIN II solver for detailed chemistry calculations. In this study the chemistry of gasoline and diesel fuel were represented by that of iso-octane and n-heptane, respectively. A 45 species and 142 reaction mechanism for primary reference fuels [12] was used to simulate gasoline and diesel fuel chemistry. Note that many studies (e.g., Ra et al. [24]) have shown that the combustion characteristics of gasoline and diesel are represented well by iso-octane (i.e., PRF 100) and n-heptane (i.e., PRF 0), respectively. This approach has also been shown to yield acceptable agreement for blends of gasoline and diesel fuel (e.g., Kokjohn et al. [7, 16, 25]). The physical properties (for spray and mixing processes) of diesel fuel were represented by tetradecane. This modeling approach has been shown to yield acceptable results in numerous studies (e.g., Kong et al. [22]). Soot was predicted

using a phenomenological soot model [22] based on the approach of Hiroyasu [26]. The soot model used in the present study uses acetylene as an inception species, which allows the soot model to be coupled to the chemistry solver through the addition of 13 reactions involving acetylene. NO_x emissions were predicted using a reduced NO mechanism [27] consisting of 4 additional species and 12 reactions.

The spray model employed in this study uses the Lagrangian-Drop and Eulerian-Fluid (LDEF) approach. In order to reduce the grid size dependency of the LDEF spray model and to allow accurate spray simulation on a relatively coarse grid, the Gasjet model of Abani et al. [21] was used to model the relative velocity between the droplets and gas phase in the near nozzle region. Their approach assumes that the relative velocity between a droplet and the gas phase is equal to that between the droplet and a turbulent gas-jet with the same mass and momentum of that of the injected fuel.

Droplet breakup was modeled using the hybrid Kelvin Helmholtz (KH) - Rayleigh Taylor (RT) model described by Beale and Reitz [23]. The droplet collision model is based on O'Rourke's model; however, a radius of influence method is used to determine the possible collision partners to further reduce mesh dependency [28]. In addition, the collision model was expanded by Munnannur [28] to include a more comprehensive range of collision outcomes. The current implementation of the droplet collision model considers bounce, coalescence, and fragmenting and non-fragmenting separations. Droplet interactions with the wall were considered through a wall film submodel [29, 30], which includes effects associated with splash, film spreading, and motion due to inertia.

Simulations were performed on the computational grids shown in Figure 3. Each grid represents a single hole of the direct injectors described in Tables 3 and 4. Notice that only the region above the top piston ring is resolved. That is, the simulations do not consider ring motion or flow in the ring-pack.

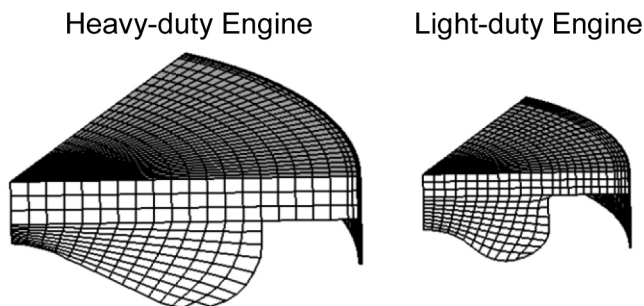


Figure 3. Computational grids used for the CFD modeling studies.

INJECTION STRATEGY

To control the overall fuel reactivity and therefore combustion phasing, RCCI combustion uses in-cylinder blending of two fuels with differing auto-ignition characteristics. In this work, gasoline is delivered using a port-fuel-injector and diesel fuel is delivered using the common rail type direct-injector. In addition to controlling the overall fuel-reactivity, in-cylinder fuel blending allows the generation of fuel reactivity gradients which result in control over the combustion duration [31].

In previous work RCCI operation was demonstrated at 6, 9, and 11 bar IMEP [16] using an injection and in-cylinder fuel blending strategy developed with detailed CFD modeling [15]. The injection strategy is highlighted in Figure 4. A first injection, shown in Figure 4 as the squish conditioning pulse, is delivered near 60° BTDC targeting the squish region of the combustion chamber. The purpose of this injection is to control the fuel reactivity in the squish region ensuring complete combustion of the premixed gasoline in the outer portion of the combustion chamber. The second injection occurs near 35° BTDC and targets the bowl region of the combustion chamber. This injection event generates a relatively high reactivity region which acts as an ignition source.

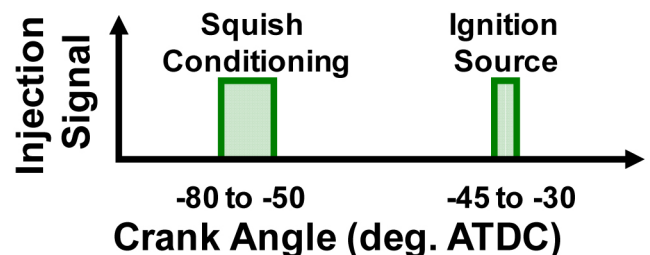


Figure 4. Injection strategy used for split injection dual-fuel RCCI combustion. The injection timing ranges are shown for illustration only. Details on the selection of the injection strategy can be found in Kokjohn et al. [15].

RESULTS AND DISCUSSION

EXTENSION OF RCCI COMBUSTION TO THE LIGHT-DUTY ENGINE

As previously stated, the most optimization work in the heavy-duty engine has been performed at the 9.3 bar IMEP operating point; thus, the 9.3 bar IMEP operating point is the focus of the comparisons between the light- and heavy-duty engines. The operating conditions for the two engines are presented in Table 7. The current heavy-duty engine was operated at 1300 rev/min. Engine size scaling laws (e.g., Shi et al. [32]) do not provide a scaling parameter for engine speed; however, two arguments can be made regarding the engine speed at which to operate the light-duty engine in