

An Optical Investigation of Fuel Composition Effects in a Reactivity Controlled HSDI Engine

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ABSTRACT

Reactivity controlled compression ignition combustion was investigated for three fuel combinations: isooctane-diesel, PRF90-diesel, and E85-diesel. Experiments were conducted at 1200 rpm, 160 kPa absolute intake pressure, and fixed total fuel energy using 'optimal' operating condition for each fuel combination that were chosen based on combustion performance from SOI timing and premixed energy fraction sweeps. The heat release duration was found to scale with the difference in reactivity between the premixed and direction injected fuel; a small difference gives rise to short heat release duration, similar to that of HCCI combustion. Conversely, as the difference increases, the heat release period lengthens. The high-speed optical data confirmed that the combustion occurred in a staged manner from the high-reactivity zones, which were located at the periphery of the chamber, to low-reactivity zones in the field of view. Further, the range of ignition timing across the imaged area was found to scale with the difference in the two fuel's reactivity. In cases where there was a small difference in reactivity, the ignition took place in a relatively narrow window within the heat release profile. In cases with large reactivity difference, the progression of the ignition front from high- to low-reactivity regions was found to occur over an extended part of the heat release duration. Despite the contrast in combustion characteristics and injection timings, similar integrated luminosity levels in the 310 nm range were observed for the tested fuels. The luminosity levels were comparable to HCCI combustion levels and in excess of two orders of magnitude lower than standard diesel combustion.

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INTRODUCTION

Rising fuel costs and stricter emission standards continue to drive the development of newer and more advanced combustion strategies. Homogenous charge compression ignition (HCCI) and premixed charge compression ignition (PCCI) combustion strategies have been proven to be capable of reducing NO_x and soot emissions [1,2] but both PCCI and HCCI engines have drawbacks that are often related to the fuel properties. Diesel fuel has low volatility, causing mixture preparation issues when early injection times are desired to create a nearly homogenous mixture, but it is highly reactive and easy to compression ignite, which gives good control over combustion phasing. Gasoline is highly volatile, allowing flexible injection timing, but it is resistant to autoignition, causing difficulties for combustion phasing control and for cold operating conditions.

Experiments performed by Bessonette *et al.* [3] suggest that the best fuel for HCCI operation may have auto-ignition quality between those of diesel fuel and gasoline. Using a compression ratio of 12:1 and a fuel with a derived cetane

number of ~27 they were able to extend the HCCI operating range to 16 bar BMEP - a 60% increase in the maximum achievable load compared to operation using traditional diesel fuel. Since such a fuel is not readily available in the current infrastructure, it may be beneficial to explore engine operation using blends of conventional fuels.

Fuel blends can be explored with an HCCI combustion strategy, but there will be inherent drawbacks. HCCI combustion phasing depends critically on the in-cylinder thermodynamic history, which is difficult to actively control. This makes on-the-fly combustion changes difficult. Also, the intrinsic heat release rate duration is short, due to the premixed combustion, which leads to high pressure rise rates across the operating range. Therefore other fuel blending techniques need to be explored.

Recently, researchers [4, 4, 5, 6, 7] have begun investigating fueling an engine with both gasoline and diesel fuel, with the fuels being mixed in-cylinder. This strategy allows for cycle-to-cycle changes to the fuel blend, essentially giving any desired fuel reactivity within the

constraints of the two fuels. Computational results [4] have shown port fuel injection of gasoline and early-cycle injection of diesel fuel can meet EPA 2010 heavy-duty emission standards for NO_x and PM without after treatment while achieving a 50% indicated thermal efficiency. The computational results showed a staged consumption of the more reactive diesel fuel followed by the less reactive gasoline. This staged consumption extended the combustion duration and allowed for low emissions and high efficiency. Experimental results [5] were able to confirm the computational results [4]. The experiments showed NO_x and PM engine-out emissions below the EPA 2010 standards and a 53% net indicated thermal efficiency, but higher CO and HC emissions were reported, which was also predicted by the model. Splitter *et al.* [6] used an in-cylinder FTIR diagnostic to measure combustion emission spectra from a limited number of locations in the combustion chamber. The results confirmed that the more reactive fuel initiates combustion. The in-cylinder blending results in a stratified fuel distribution and thus a reactivity gradient. Further FTIR results showed reaction progression from areas of the more-reactive to less-reactive fuels, as predicted by the computational results. Because the combustion progression is dictated by the reactivity of the fuels, this combustion strategy has been collectively labeled reactivity controlled compression ignition (RCCI) combustion.

Multi-cylinder RCCI experiments were successfully completed by Curran *et al.* [7] and Prikhodko *et al.* [8]. Their experiments showed low NO_x and PM emissions with an increase in indicated thermal efficiency over conventional diesel and similarly an increase in CO and HC emissions. Prikhodko *et al.* showed that an oxidizing catalyst can successfully be used with RCCI to simultaneously remove CO and HC from the exhaust gases.

Recent experiments by Splitter *et al.* [9] investigated fuel combinations such as gasoline-diesel, E85-diesel, and gasoline-gasoline+DTBP (di tert-butyl peroxide cetane improver). Combined with computational results for injection timing optimization, the experimental results showed indicated thermal efficiencies greater than 50% for all fuel combinations across a range of operating conditions, reaching a peak of 59% for E85-diesel experiments. Again, the engine-out emissions for the NO_x and PM met EPA 2010 heavy-duty standards.

The reviewed literature shows the versatility of the RCCI combustion strategy. A wide range of operating conditions can be achieved with a variety of fuels, indicated thermal efficiency in excess of 50% with low engine-out emissions for NO_x and PM is possible, and combustion control can be accomplished with both injection timing variations and adjustment of the fuel split ratio. Furthermore a diesel oxidation catalyst (DOC) can significantly reduce the CO and HC emissions incurred by the low temperature combustion.

The objective of this paper is to optically investigate the RCCI combustion process and to verify the concept of staged combustion, especially how it scales with different overall

reactivity gradients. Three fuel combinations were tested: isooctane-diesel, PRF90-diesel, and E85-diesel. Optimized conditions for each fuel combination, determined from pressure-based measurements, will be compared optically.

EXPERIMENTAL SETUP

A single-cylinder engine with Bowditch-style optical access was used for the experiments. The engine uses a commercially available GM/Fiat 1.9L high-speed, direct-injection diesel engine head. Table 1 summarizes the engine geometry. The piston used a simple flat-bottomed bowl that was 50 mm in diameter. The bottom surface of the bowl was a fused-silica window that allowed optical access. Figure 1a shows the optical engine setup with notable components identified. Figure 1b shows the camera's view of the combustion chamber through the piston window.

Table 1. Engine geometry, crank angle timing relative to TDC of compression.

Displacement (L)	0.41
Geometric compression ratio	14.0:1
Squish height (mm)	1.33
Bowl volume (mL)	20.8
Bowl diameter (mm)	50
Connecting rod length (mm)	144.8
Bore × Stroke (mm)	82.4 × 76.2
Valve timing	EVC= 373 °aTDC
	IVC= -139 °aTDC
	EVO= 123 aTDC
	IVO= 350 °aTDC
Swirl ratio	1.5

Choked-flow orifices were used to control and measure the intake air flow, which was supplied from the building's dried, compressed air line. The air passed through a 1 kW in-line air heater and into an intake surge tank; the surge tank and intake runner are sufficiently heated and insulated to maintain an intake temperature within $\pm 1^\circ\text{C}$. The engine operates on a 10 cycle skip-fire pattern. For each case, data were recorded for 89 cycles; 80 motored cycles, and 9 fired cycles. All experiments were performed at 1200 rpm with an intake tank pressure of 160 kPa, absolute.

An air-assisted injector, which fires every cycle, supplied the low-reactivity fuel, *e.g.* isooctane, far enough upstream of the engine to ensure uniform mixing with the air. A Bosch high-pressure, common-rail injector was actuated only on the fired cycles. In this study a single injection pulse was used. The specifications for the common-rail injector are given in Table 2. Ultra low sulfur diesel fuel, with a cetane number of 42, was used as the high-reactivity fuel for all experiments. Injector calibrations were performed gravimetrically. The calibrations resulted in a linear relationship between injection duration input and fuel mass; the calibrations were acquired over the operating conditions in this work.

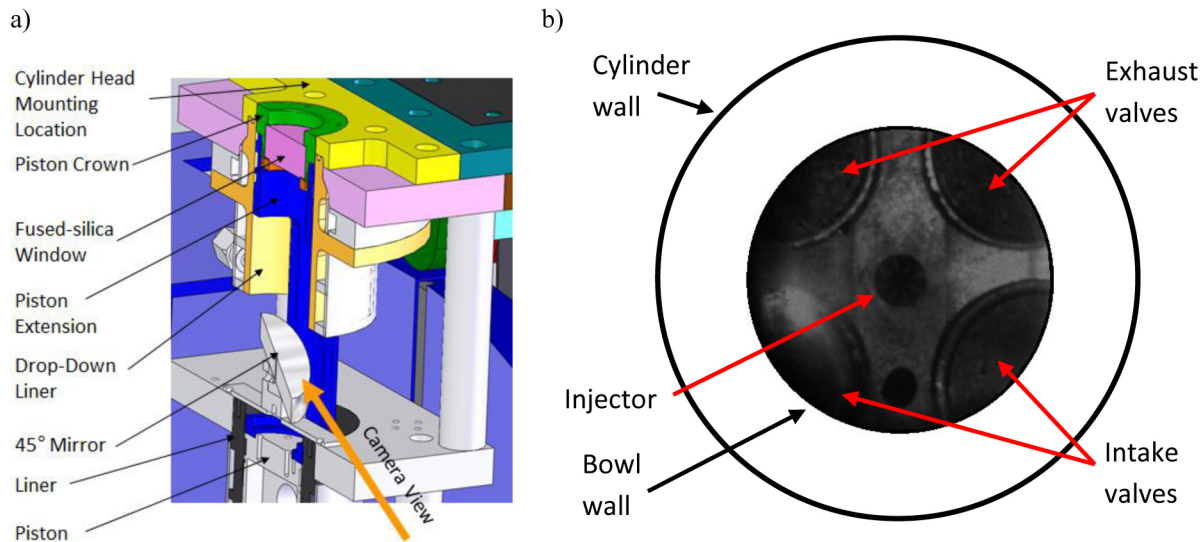


Figure 1. Bowditch optical engine setup and optical viewing area.

Table 2. Common rail injector specifications.

Injection pressure (MPa)	50
Included spray angle (deg)	154
Number of holes	8
Hole diameter (μm)	133

Combustion luminosity data were recorded using a Phantom v7.1 high speed camera. The camera was triggered every 0.72 crank angles (100 μs), commencing with the start-of-injection command (SOC) signal, and continued recording until the visible combustion event was completed.

Data were acquired using two different optical setups that highlight different chemical processes:

1. Broadband UV images were acquired using a high-repetition rate intensifier having an S20 UV-sensitive photo cathode and the combination of a Rapp 400 nm short-pass dichroic filter and a Schott UG11 glass filter. The overall transmission window was approximately 240–400 nm, which included emission from species such as HCHO^* and other precursor species, in addition to OH^* and the broadband CO^* . This filter combination was chosen to eliminate soot luminosity (when present from rogue droplets).

2. Narrowband UV images were acquired with the intensifier and a 310 nm, 11 nm FWHM dichroic filter. This filter combination transmits light from the OH^* and from the CO^* broadband emission, both of which arise from high-temperature combustion reactions.

For both setups, a UV-Nikkor f/4.5 lens was used, and the intensifier was operated at maximum gain. The camera exposure time for each data set was set to 95 μs , and the intensifier was gated to 90 μs .

Due to the design compromises and relatively low speeds of Bowditch-style optical engines, higher losses (mass and heat) are observed than for a similar metal engine. This

makes it difficult to obtain accurate, quantitative heat release rate data. In this work, the heat release was calculated using an open-system 1st law thermodynamic balance. The open-system allows mass to move between the combustion chamber and the crevices, where the temperature is much lower. Assuming ideal gas behavior, one can calculate the chemical rate of heat release by [10].

$$\frac{dQ_{ch}}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} + \frac{dQ_{ht}}{d\theta} - \left(\frac{RT}{\gamma-1} + \tilde{h} - u \right) \frac{dm}{d\theta}$$

where γ is the specific heat ratio of air, p is the cylinder pressure, V is the volume, $\frac{dQ_{ht}}{d\theta}$ is the heat transferred to the cylinder walls, $\tilde{h}$ is the enthalpy of the mass crossing the system boundary, u is the internal energy of the working fluid, and $\frac{dm}{d\theta}$ is the rate of change of the system mass. When the in-cylinder pressure is increasing, $\frac{dp}{d\theta} > 0$, mass is leaving the system and entering the crevice, thus $\tilde{h}$ is calculated using the system's temperature; when the in-cylinder pressure is decreasing, $\tilde{h}$ is calculated using the crevice temperature. $\frac{dm}{d\theta}$ is calculated using the ideal gas law, shown in the following equation:

$$\frac{dm}{d\theta} = - \frac{V_{crevice}}{RT_w} \frac{dp}{d\theta}$$

Where $V_{crevice}$ is the crevice volume and T_w is the crevice temperature. When $\frac{dp}{d\theta} > 0$, mass is leaving the cylinder and entering the crevice, thus $\frac{dm}{d\theta} < 0$. The heat transfer rate was calculated using the Woschni correlation [10], where the actual motored pressure from the skip fired cycles was used