

Investigation of the Combustion Instability-NO_x Tradeoff in a Dual Fuel Reactivity Controlled Compression Ignition (RCCI) Engine

David Klos, Daniel Janecek, and Sage Kokjohn
University of Wisconsin

ABSTRACT

The tradeoff between NO_x emissions and combustion instability in an engine operating in the dual-fuel Reactivity Controlled Compression Ignition (RCCI) combustion mode was investigated using a combination of engine experiments and detailed CFD modeling. Experiments were performed on a single cylinder version of a General Motors/Fiat JTD 1.9L four-cylinder diesel engine. Gasoline was injected far upstream of the intake valve using an air assisted injector and fuel vaporization system and diesel was injected directly into the cylinder using a common rail injector. The timing of the diesel injection was swept from -70° ATDC to -20° ATDC while the gasoline percentage was adjusted to hold the average combustion phasing (CA50) and load (IMEPg) constant at 0.5° ATDC and 7 bar, respectively. At each operating point the variation in IMEP, peak PRR, and CA50 was calculated from the measured cylinder pressure trace and NO_x, CO, soot and UHC were recorded. It was observed that a late injection strategy with start-of-injection (SOI) timings ranging from -30° ATDC to -20° ATDC could significantly reduce cycle-to-cycle variation with only a marginal increase in NO_x emissions. To explain the sources of increased combustion stability, detailed computational fluid dynamics (CFD) simulations were used. The CFD simulations confirmed that the late injection timing produced a higher equivalence ratio ignition site that is less sensitive to fluctuations. By optimizing injection timing large improvements in combustion instability can be made while still maintaining low temperature combustion-like emissions.

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INTRODUCTION

In the search for more efficient, cleaner engines, premixed compression ignition (PCI) engines have emerged with the potential to simultaneously increase engine efficiency while reducing NO_x and soot emissions [1, 2, 3, 4, 5]. Although promising, PCI engines have not been widely implemented due to difficulties controlling the combustion phasing, the combustion duration, and cycle-to-cycle variation. Recent work has shown that blending two fuels with different auto-ignition characteristics inside of the combustion chamber can provide control over both the combustion phasing and heat release rate [1]. In this dual-fuel combustion process, the combustion phasing and duration are controlled by the auto-ignition characteristics, or reactivity of the charge; thus, it was named Reactivity Controlled Compression Ignition (RCCI) combustion [6]. Figure 1 shows the typical setup for RCCI combustion using port-fuel-injection of a low reactivity fuel (e.g. gasoline) and direct-injection of a high reactivity fuel (e.g., diesel fuel).

Providing control over combustion phasing and duration represented a significant advance in PCI combustion; however, research has shown that even when the mean combustion phasing is well

controlled, the cycle-to-cycle fluctuations in the combustion phasing are still significant. Figure 2 shows the measured [7] peak pressure rise rate for 300 cycles of RCCI, gasoline partially premixed combustion (PPC), and conventional diesel combustion (CDC) operating at similar conditions. Compared to CDC, it is clear that both RCCI and PPC have significantly higher cycle-to-cycle variation in peak PRR variation. This variation, coupled with a higher average peak PRR, limits the upper load achievable by the engine. Further, Kolodziej and Ciatti [8] showed in their gasoline compression ignition (GCI) work that the lower load limit can also be limited by combustion instability. In their work, they defined stable combustion as having a COV of IMEP less than 3%, which adequately limited the amount of cycles resulting in misfire. By sweeping injection timing, they varied the amount of fuel stratification in the cylinder and determined that there was a tradeoff between local equivalence ratio stratification and fuel in-cylinder residence time. To maximize the impact of advanced combustion modes (e.g., RCCI), control strategies that simultaneously reduce combustion instability and maintain LTC like emissions must be determined.

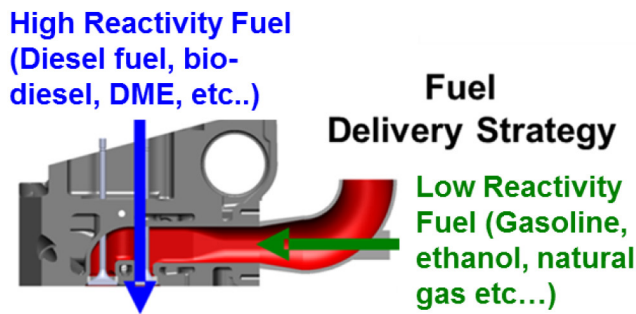


Figure 1. RCCI combustion using port-fuel-injection (PFI) of a low reactivity fuel (e.g., gasoline) and direct-injection (DI) of a high reactivity fuel (e.g., diesel fuel).

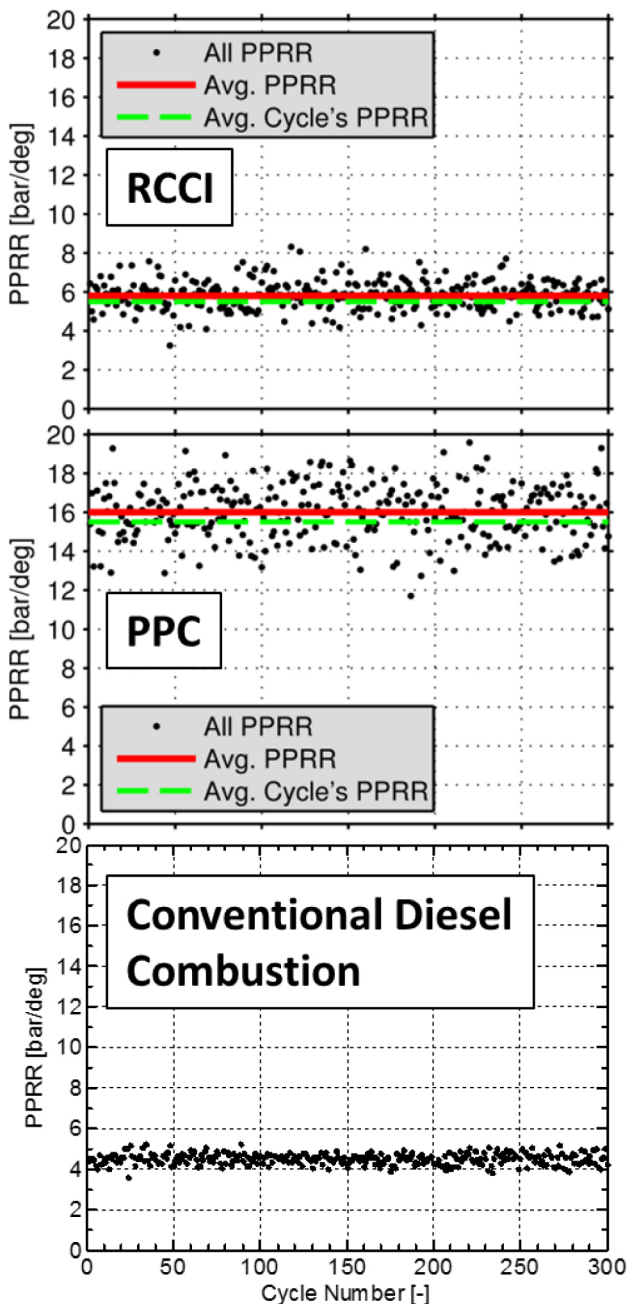


Figure 2. Measured peak pressure rise rate for 300 cycles of RCCI, gasoline partially premixed compression ignition combustion (PPC), and conventional diesel combustion (CDC) operating at similar conditions [7].

TESTING PROCEDURE

Engine and Setup

Engine experiments are performed on a light-duty diesel single-cylinder research engine based on the General Motors/Fiat JTD 1.9L four-cylinder diesel engine. The stock head and valve train are modified to use only the second cylinder and mounted atop a single cylinder block. Table 1 shows the specifications for the engine geometry and fuel injection systems.

Engine airflow is supplied by the buildings compressed air and is metered by choked flow orifices. The air is preheated by an electric heater before reaching an air assisted upstream injector. The injector mixes fuel and compressed air in a pre-chamber prior to injecting it into the air stream to aid in fuel atomization. The air then passes through a surge tank several times larger than the displacement of the engine where it mixes with cooled EGR. The surge tank is equipped with a Pulse Width Modulated (PWM) tape heater to maintain a steady intake temperature. Two Port Fuel Injectors (PFI) are positioned just after the surge tank in the intake runner. The head is equipped with two swirl-control vanes that provide an adjustable swirl ratio from 1.5 to 4.8. The minimum amount of swirl (1.5) is used for this testing. The stock common-rail fuel injection system is used for direct-injection of diesel fuel. After leaving the cylinder, the exhaust passes through an exhaust surge tank to dampen wave dynamics. A portion of the exhaust gas then passes through an exhaust gas recirculation (EGR) cooler and is recirculated into intake surge tank. Small samples of the exhaust gas are acquired by an AVL 415S smoke meter and Horiba gas analyzers that measure UHC, NO_x, CO, O₂, exhaust CO₂ and intake CO₂. Figure 3 shows the engine laboratory setup.

Table 1. Engine specifications.

Engine Geometry	
Base Engine	GM 1.9L Diesel
CR	16.3
Displacement [L]	0.477
Bore x Stroke [mm]	82 x 90.4
IVC [°ATDC]	-132
EVO [°ATDC]	112
Swirl Ratio	1.5
Piston Bowl Type	Stock (Re-entrant)
Upstream Injector	
Injection Pressure [psi]	20
Rated Flow [cc/min]	~25
Port Fuel Injectors	
Included Spray Angle	20
Injection Pressure [bar]	3
Flow Rate [cc/min]	578
Bosch Common Rail Injector	
Number of Holes	7
Hole Diameter [μm]	140
Included Spray Angle	155°
Injection Pressure [bar]	500

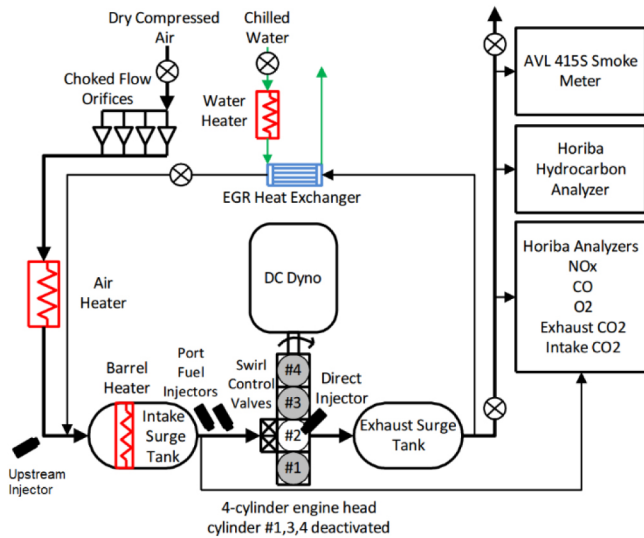


Figure 3. Single-cylinder research engine layout.

The lab is controlled with a National Instruments LabView system and the fuel injection systems are controlled by Driven PXI controller cards. The system controls and records all necessary operating parameters (e.g., airflow, pressures, temperatures, emissions data). Coriolis-type mass flow meters (Endress-Hausser Proline Promass 80A01) measure the gasoline and diesel fuel flowrates. A shaft encoder with a resolution of 0.25 crank angle degrees records crank position and triggers the high speed data acquisition system and for fuel injection systems. In-cylinder pressure is measured with a Kistler 6125A pressure transducer in series with a Kistler 5010 charge amplifier. The apparent heat release rate (AHRR) is calculated from the measured cylinder pressure using the standard first-law analysis [9]. Prior to calculating the AHRR, the cylinder pressure data are smoothed using a Fourier series low-pass filter with a Gaussian roll-off function having a transmission of 100% from 0 to 800 Hz and dropping to 1% at 3360 Hz.

Computational Model

To aid in understanding the physical phenomenon effecting combustion stability, detailed computational fluid dynamics (CFD) simulations are performed. The results of the CFD modeling are processed using zero-dimensional chemical kinetics modeling. Multi-dimensional simulations were performed with the KIVA-3v release 2 code [10] with improvements to many physical and chemistry models developed at the Engine Research Center (ERC) [11, 12, 13, 14, 15]. This section provides an overview of the physical models important to the present study. The chemistry of gasoline and diesel fuel are represented by that of iso-octane and n-heptane, respectively. A reduced reaction mechanism made up of 45 species and 142 reactions [14] describes the combined oxidation of n-heptane and iso-octane. This approach has been shown to yield good agreement for blends of gasoline and diesel fuel (e.g., Kokjohn et al. [1]). Furthermore, Puduppakkam et al. [16] compared the primary reference fuel (PRF) mechanism used in the present study to a detailed gasoline mechanism (425 species and 3128 reactions) and found similar bulk combustion characteristics. Details of other important sub-models are given in Table 2.

Table 2. Important ERC KIVA sub-models used in the present work.

Model	Description
Combustion	Direct Integration of Reduced PRF Mechanism [14]
Turbulence	RNG k- ϵ [13]
Soot	Two-step [17]
NOx	Reduced GRI NO Mechanism [15]
Spray Model	Lagrangian-Drop and Eulerian-Fluid (LDEF) with Gasjet Model [11]
Droplet breakup	Kelvin Helmholtz (KH) – Rayleigh Taylor (RT) [12]
Droplet Collision	O'Rourke [18] with Radius of Influence (ROI) Collision Model [11]

To reduce computing time, only a 51.4-degree sector mesh representing a single hole of the seven hole common-rail injector was simulated. The computational grid consists of 22,608 cells at bottom dead center (BDC) and 11,376 cells at top dead center (TDC). The average grid spacing in the axial, azimuthal, and vertical direction are 1.7 mm, 1.5 deg, and 1.0 mm, respectively. Figure 4 shows the computational grid at TDC.

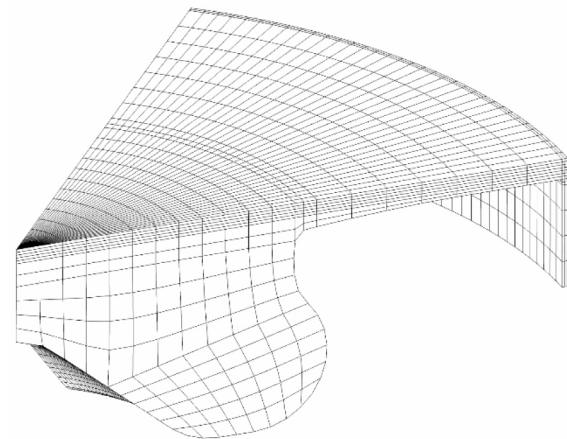


Figure 4. Computational grid at TDC.

Constant volume ignition delay calculations were performed using Cantera [19] and the same reduced reaction mechanism used for the CFD simulations [14].

RESULTS AND DISCUSSION

The engine is operated at 7 bar IMEP and 2300 rev/min using premixed gasoline and direct-injected diesel fuel. Gasoline and diesel fuel properties are given in Table 4. The air-assisted fuel vaporizer delivers the gasoline upstream of the intake. This ensures a homogenous fuel air mixture, removes any charge cooling effects, and minimizes cycle-to-cycle fueling variation due to injector shot-to-shot variation. A Bosch CRIN-II solenoid-type, common-rail injection system delivers the diesel fuel. A single injection strategy is used for the diesel fuel. An EGR/fueling combination allowing stable operation with an SOI timing at -55° ATDC is selected as the baseline condition. Table 3 shows the conditions of the baseline case. The diesel fuel injection timing is swept from -55° to -15° ATDC to vary the mixing time prior to ignition. To remove the effect of