

Direct Dual Fuel Stratification, a Path to Combine the Benefits of RCCI and PPC

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ABSTRACT

Control of the timing and magnitude of heat release is one of the biggest challenges for premixed compression ignition, especially when attempting to operate at high load. Single-fuel strategies such as partially premixed combustion (PPC) use direct injection of gasoline to stratify equivalence ratio and retard heat release, thereby reducing pressure rise rate and enabling high load operation. However, retarding the heat release also reduces the maximum work extraction, effectively creating a tradeoff between efficiency and noise. Dual-fuel strategies such as reactivity controlled compression ignition (RCCI) use premixed gasoline and direct injection of diesel to stratify both equivalence ratio and fuel reactivity, which allows for greater control over the timing and duration of heat release. This enables combustion phasing closer to top dead center (TDC), which is thermodynamically favorable. However, the main control mechanism in RCCI is the ratio of the two fuels, and the diesel fraction typically reaches zero before full load is achieved. We propose a new strategy that effectively combines the benefits of RCCI and PPC by injecting both gasoline and diesel directly, enabling control over the in-cylinder distribution of both fuels. We present a comparison of RCCI, PPC, and our new strategy, direct dual fuel stratification (DDFS) at a nominal gross mean effective pressure of 0.9 MPa. DDFS allowed for combustion phasing near TDC with reduced combustion noise. Cyclic combustion instability was reduced significantly with the new strategy and approached levels typical of conventional diesel combustion. Compared to RCCI, there was a reduction in noise and required exhaust gas recirculation (EGR) while maintaining similar efficiency. Compared to PPC, there was a reduction in noise and an increase in efficiency. The new strategy therefore combines the efficiency advantage of RCCI with the load advantage of PPC, while reducing EGR and combustion instability.

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INTRODUCTION

Conventional diesel combustion (CDC) engines combust high-reactivity fuels (e.g., diesel) via compression ignition in a high-temperature, mixing-controlled, process. This often results in high levels of particulate matter (PM or soot) and mono-nitrogen oxides (NO_x) [1]. In order to meet aggressive modern emissions regulations, exhaust aftertreatment is required to reduce tailpipe levels of PM and NO_x , but this typically comes at the expense of fuel economy and/or secondary reducing agents [2,3].

In an effort to bypass the need for heavy aftertreatment, many strategies have been developed to reduce the PM and NO_x emissions in-cylinder while maintaining high thermal efficiency. Most of these fall under a regime termed low-temperature combustion (LTC), which uses long ignition delays to mitigate the formation pathways for PM and NO_x , while simultaneously reducing heat transfer [4,5,6]. The main drawbacks to the use of long ignition delays are that combustion phasing control becomes more difficult, confining operation to a smaller range than CDC. Therefore, much of the research on LTC strategies is focused on improving their control and operating range [7,8].

Homogeneous charge compression ignition (HCCI), one of the earliest LTC strategies, utilizes a fully premixed mixture of fuel and air, with a kinetically-controlled ignition event which is dependent on temperature, equivalence ratio, fuel properties, and time [9,10]. Many researchers have explored the benefits of HCCI, which include high efficiency and effective reduction of PM and NO_x [7,8,9,10,11]. Despite the promising results of HCCI, there are some significant drawbacks. These include a trade-off between efficiency and peak pressure rise rate (PPRR), which limits the peak practical load and reduces efficiency through the use of later combustion phasing [8,12]. Moreover, HCCI suffers from a lack of fast-response combustion phasing control variables that are available in more traditional combustion strategies, such as fuel injection and spark timing.

To introduce more direct control of the combustion event, premixed charge compression ignition (PCCI) and partially premixed combustion (PPC) have been developed. These are derivatives of HCCI that seek to increase the operating range and stability by direct-injecting either a portion or all of the fuel to create stratification of the fuel charge, and therefore equivalence ratio, within the cylinder. This stratification causes the local ignition delay to vary across the cylinder, which has the overall effect of increasing combustion duration and decreasing pressure rise rates.

Examples of PCCI research are studies by Hardy et al. [13] with diesel fuel, and Kalghatgi et al. [4], and Hanson et al. [14] with gasoline. Research by Manente et al. [15,16] and Shen et al. [17] has demonstrated that operation of PPC over a wide range of loads requires increasing stratification and optimizing fuel selection, boost level, exhaust gas recirculation (EGR), intake temperature, and other variables for each combination of load and speed. While PPC operation has been demonstrated at high loads, this typically comes with high boost and EGR requirements. It is also necessary to retard combustion phasing in order to meet noise constraints, which limits the potential thermal efficiency.

Reactivity controlled compression ignition (RCCI) extends the concept of control by stratification to include reactivity. By injecting two different fuels, both the reactivity and equivalence ratio are stratified throughout the cylinder. This adds an additional fast-response control parameter in the form of fuel chemistry, enabling the global fuel properties to change across the operating map. Simulations and experiments of RCCI with port-injection of low-reactivity fuel and direct-injection of high-reactivity fuel have been performed by Inagaki et al. [18], Kokjohn et al. [19,20,21], Hanson et al. [22,23], and Splitter et al. [24,25,26,27,28], who demonstrated gross indicated thermal efficiencies in excess of 56% and in-cylinder levels of PM and NO_x below the EPA 2010 Heavy Duty emissions regulations for specific points in the operating map. Similar to other LTC strategies, RCCI suffers from high levels of unburned hydrocarbons (HC) and carbon monoxide (CO) which are considerably higher than in CDC. As shown by Splitter et al. [28], this is due mainly to lower wall temperatures and the inability to completely oxidize fuel that has entered crevice regions. While the ratio of high-to-low reactivity fuel is very useful for controlling heat release to maintain favorable combustion phasing and noise levels, the amount of high-reactivity fuel needed tends to decrease to zero as load increases. For operation with gasoline and diesel, this point is reached well before full load is achieved, although it can be extended by decreasing compression ratio (CR) at the expense of efficiency.

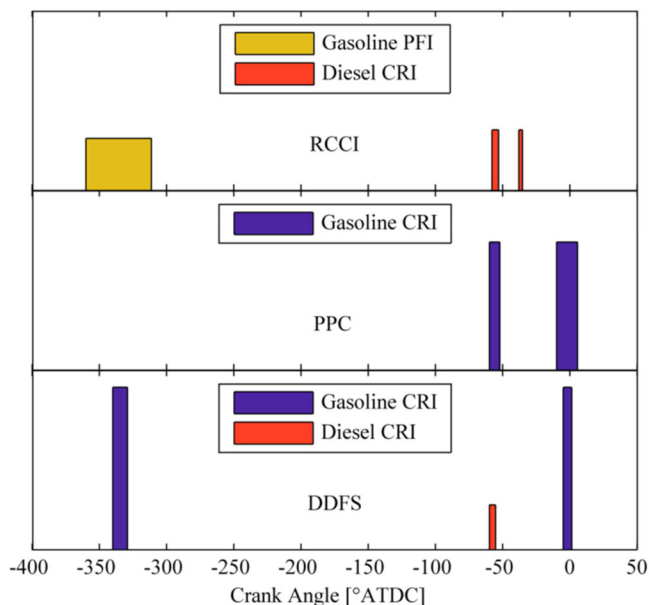


Figure 1. Conceptual illustration of injection strategies. The area of each box represents the relative quantity of each injection.

We propose a new strategy that combines the benefits of RCCI and PPC by injecting both gasoline and diesel directly, enabling control over the in-cylinder distribution of both fuels. Gasoline is injected during the intake stroke and is effectively premixed. Similar to RCCI, diesel is injected 40–60° before top dead center (BTDC) and establishes a reactivity gradient that allows for precise control over the low-temperature heat release (LTHR), which begins at roughly 20° BTDC. After LTHR is complete, and just as the main heat release is beginning (0–10° BTDC), gasoline is injected a second time and burns in a diffusion-controlled manner, allowing for considerable increase in load without increasing peak heat release rate (PHRR) and combustion noise. This strategy, which we will refer to as direct dual fuel stratification (DDFS), is illustrated along with typical implementations of RCCI and PPC in Figure 1.

EXPERIMENTAL SETUP

Engine

A 2.44 L Caterpillar 3401E Single Cylinder Oil Test Engine (SCOTE) was used for all experiments. The relevant engine parameters are listed in Table 1, and the laboratory setup is illustrated in Figure 2. The piston profile, shown in Figure 3, was optimized for RCCI by Splitter et al. [28].

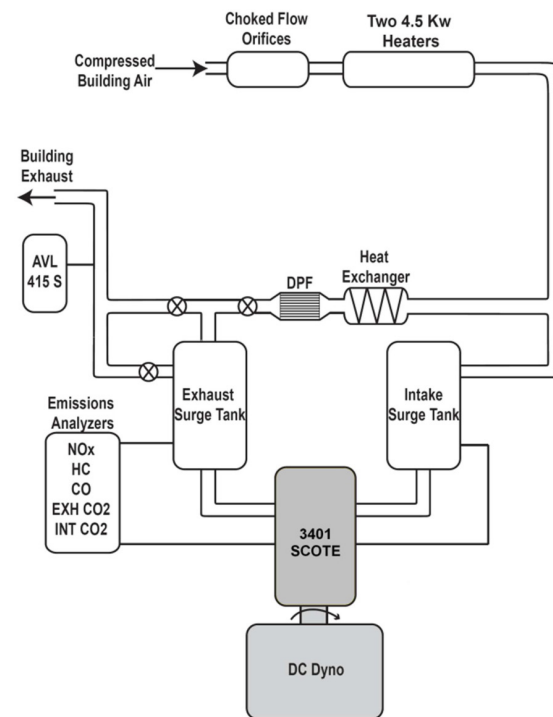


Figure 2. Diagram of experimental apparatus

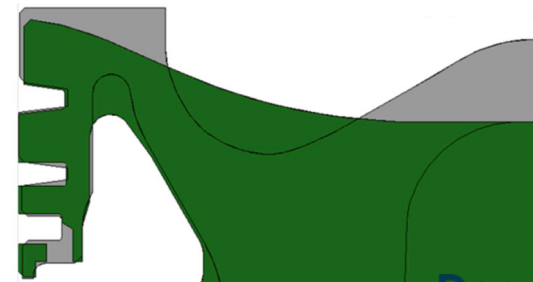


Figure 3. Profile of RCCI piston (green, CR = 14.88) overlaid on stock piston (gray, CR = 16.1)

Table 1. 3401E SCOTE specifications

Displacement [L]	2.44
Bore [mm]	137.2
Stroke [mm]	165.1
Connecting Rod Length [mm]	261.6
Squish Height [mm]	1.57
Number of Valves	4
IVO (°ATDC)	335
IVC (°ATDC)	-143
EVO (°ATDC)	130
EVC (°ATDC)	-355
Swirl Ratio (stock)	0.7
Piston Type	Articulated
Compression Ratio	14.88:1

Dual Direct Injector Head

A stock engine head was modified to allow the mounting of two common rail injectors (CRI). The stock injector sleeve was removed, and the injector bore was enlarged to accept a larger sleeve capable of holding two injectors. Due to geometrical constraints, the longitudinal axes of the two injectors are located in a plane 2 mm from the original injector axis, and are each titled 1° away from the original axis. A cross sectional view of the injector assembly model is shown in Figure 4, and pictures of the new injector bore and final injector assembly are shown in Figure 5. The injector specifications are listed in Table 2. Note that while CRI1&2 share the same body part number, CRI2 has been fitted with a 10-hole nozzle. Preliminary work has shown that using more holes of smaller diameter resulted in a significant reduction of the PM generated by the near-TDC gasoline injection with the DDFS strategy, and future investigations will likely involve optimization of nozzle geometry. The engine is also equipped with a port fuel injector (PFI), with specifications listed in Table 3.

Table 2. Common rail fuel injector specifications

Injector Name	CRI1	CRI2
Body Style	Bosch CRI 2 Series	
Body Part Number	0986435088	
Nozzle Part Number	DLLA148P1347	DLLA143PV3201426
Nozzle Angle [°]	148	143
Hole Diameter [μm]	141	117
Number of Holes	7	10
Hydraulic Flow (cc per 30 sec @ 10 MPa)	440	440

Table 3. Port fuel injector specifications

Manufacturer	RC Engineering
Steady Flow Rate [cc/min @ 3 bar]	750
Spray Angle [°]	15

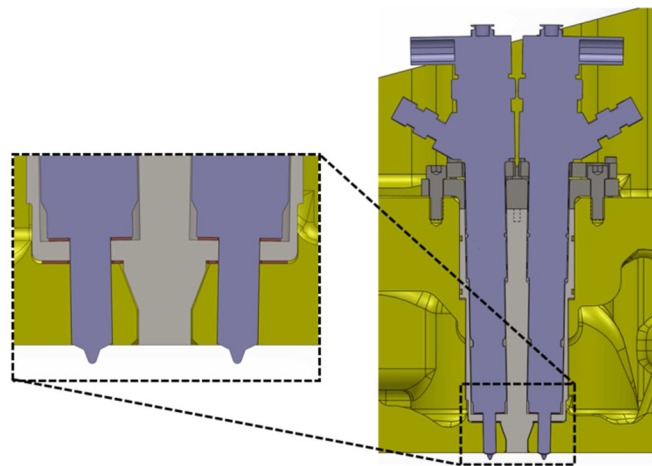


Figure 4. Cross-section of dual direct injector model

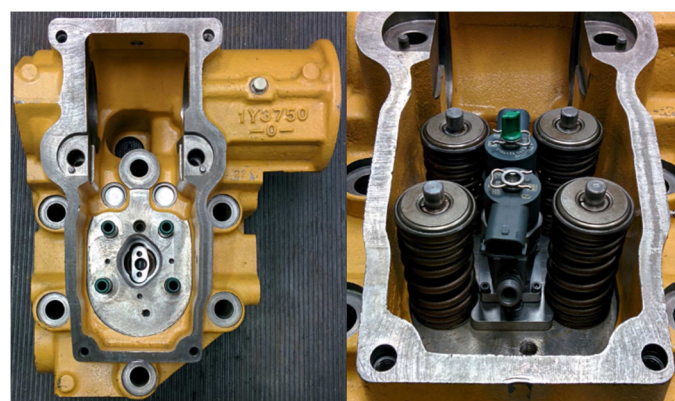


Figure 5. Pictures of new injector bore and final injector assembly

Measurements and Methodology

Cylinder Pressure

Cylinder pressure was measured with a Kistler 6125C21U20 piezoelectric pressure transducer in conjunction with a Kistler 510 charge amplifier. Cylinder pressure traces were acquired for 250 cycles, and a zero-phase, low-pass Butterworth filter with a cutoff frequency at 2200 Hz was applied to prepare the pressure traces for use in heat release calculations. The derivative of each pressure trace was obtained by the central difference method with a three-point stencil. The heat release rate (HRR) calculation implemented a variable ratio of specific heats, γ , in which the γ estimated from the early compression stroke was used until near top dead center (TDC), and the γ estimated from the late expansion stroke was used after TDC. The transition between compression and expansion γ was approximated by a logistic function centered at the PHRR and with a width proportional to duration of heat release (an iterative procedure). The HRR was then numerically integrated to find the cumulative heat release, which was used to calculate the crank angle at 50% of total heat release (CA50) for each cycle. PPRR and ringing intensity (RI) were calculated from individual filtered cycles, and sound pressure level (SPL) was calculated from individual unfiltered cycles [29], thereby generating a statistical distribution for each variable.

Temperature and Pressure

Intake airflow was measured using a manifold of six different-sized choked flow orifices, which allows for a wide range of engine operating conditions. Intake air was heated with two immersion style heaters in series, with proportional-integral-derivative (PID) temperature control to $\pm 1^\circ\text{C}$. Intake and exhaust gas temperatures were measured in the surge tanks with type K thermocouples, with a secondary exhaust gas temperature measurement directly at the cylinder head exhaust outlet. Both the intake and exhaust surge tanks were equipped with PID pressure control to ± 0.7 kPa.

Table 4. Selected properties of test fuels

Test	Method	Fuel Haltermann Product Code	
		EPA Tier II EEE HF0437	2007 Cert. Diesel HF0582
Density [kg/L]	ASTM D4052	0.742	0.858
H/C Ratio [mol/mol]	ASTM E191	1.845	
	ASTM D5291		1.771
Oxygen [weight %]	ASTM D4815	None Detected	
Sulfur [ppm]	ASTM D5453	5	9
Net Heating Value [kJ/g]	ASTM D240	42.715	42.145
Research Octane Number	ASTM D2669	96.8	
Motor Octane Number	ASTM D2700	88.6	
Cetane Number	ASTM D613		45.0
Cetane Index	ASTM D4737		44.8
Distillation Curve [$^\circ\text{C}$]			
Initial Boiling Point	ASTM D86	31	189
10%		52	220
50%		103	271
90%		156	322
End Boiling Point		208	345

Emissions

PM measurements of filter smoke number (FSN) and mass per volume (mg/m^3) were performed with an AVL 415s smoke meter and averaged among 5 samples of 2 L volume each, taken with paper saving mode off. The mass density reported by the smoke meter, which is based on the AVL correlation for CDC, was used to calculate PM in $\text{g}/\text{kW}\cdot\text{hr}$. Investigations of the PM generated by RCCI have shown that unlike CDC, it often has a very high fraction of organic carbon (>95%) and a significantly different size distribution, which is bimodal under certain conditions [30,31,32]. Thus, we do not expect that the mass density reported by the smoke meter will be accurate when using RCCI or similar combustion strategies, and the results presented from this instrument are for reference only. All gaseous emissions measurements were performed with a 5-gas, 200-series Horiba emissions bench. Hydrocarbon (HC) emissions were measured on a wet basis using a heated sampling system. All other emissions were sampled on a dry basis. EGR rate was determined by the ratio of intake to exhaust CO_2 concentrations. Gaseous emissions were averaged for 120 seconds after attaining steady-state for several minutes.

Engine Operation

The engine was operated at 1300 rev/min and a nominal load of 0.9 MPa gross indicated mean effective pressure (IMEPg) for all tests. The fuels used were EPA Tier II EEE Gasoline and 2007 Certification Diesel, both of which were supplied by Haltermann Solutions. Selected properties of each fuel are listed in Table 4. In the interest of safety, a conservative peak cylinder pressure of 15 MPa and PPRR of 1.5 MPa per crank angle degree (CAD) were imposed as upper limits for the duration of the experiment. The location of TDC was fixed at the beginning of each experiment by setting peak motored cylinder pressure to 0.4° before top dead center (BTDC). This loss angle is based on hardware determinations performed by Caterpillar for this engine platform, and agrees well with hot and cold motored traces taken on this particular engine setup. Data was acquired at approximately steady-state conditions, as determined by monitoring the pressure and temperature of the surge tanks as well as the power output and emissions levels over a span of several minutes.

RESULTS AND DISCUSSION

Comparison of RCCI, PPC, and DDFS

In order to make a direct comparison between the three strategies, it was important to operate at conditions that were as similar as possible while also being mindful of practical considerations. We set a target load of 0.9 MPa IMEPg, with a charge mass equivalence ratio of $\Phi' = 0.35$, as defined by

$$\phi' = \phi * \left(1 - \frac{\text{EGR}\%}{100}\right) \quad (1)$$

$$\phi = \frac{\text{AFR}_{\text{stoich}}}{\text{AFR}_{\text{meas}}} \quad (2)$$

As mentioned previously, combustion phasing tends to advance and become less stable as load increases with LTC strategies. As a result, relatively high levels of EGR and low intake temperatures are often used to maintain control over heat release. Unfortunately, this combination can result in condensation of water vapor, HC, and sulfur species in the EGR cooler and intake manifold, leading to fouling and corrosion, which in turn significantly reduce the performance and lifetime of components in the flow path [33,34,35,36]. During the experiments presented here, condensation in the EGR cooler was monitored via a condensate drain line, and it was found that no measurable condensate was collected at the selected load and Φ' , provided that the post-cooler EGR temperature T_{EGR} was maintained at or above 90°C . Depending on the ambient humidity, the saturation temperature after mixing EGR and air in the intake $T_{\text{sat,in}}$ may be lower than that of EGR alone, which could permit a lower intake temperature T_{in} to be used without risking condensation. This relationship is plotted in Figure 6, which shows how $T_{\text{sat,in}}$ increases with ambient relative humidity, ambient temperature, and EGR at our desired charge mass and intake pressure. If condensation is to be avoided, it is evident that EGR in excess of 60% cannot be combined with T_{in} below 50°C regardless of ambient temperature or humidity. However, if the ambient conditions are