

Exploring the Role of Reactivity Gradients in Direct Dual Fuel Stratification

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ABSTRACT

Low-temperature combustion (LTC) strategies have been an active area of research due to their ability to achieve high thermal efficiency while avoiding the formation of NO_x and particulate matter. One of the largest challenges with LTC is the relative lack of authority over the heat release rate profile, which, depending on the particular injection strategy, either limits the maximum attainable load, or creates a tradeoff between noise and efficiency at high load conditions. We have shown previously that control over heat release can be dramatically improved through a combination of reactivity stratification in the premixed charge and a diffusion-limited injection that occurs after the conclusion of the low-temperature heat release, in a strategy called direct dual fuel stratification (DDFS). This paper will focus on the role of the reactivity gradients in the premixed charge, which are achieved by the relatively early injection of gasoline and the relatively late injection of diesel. Three regimes were identified for the diesel injection timing: premixed, reactivity-controlled, and diffusion-limited, with the reactivity-controlled regime being observed to offer superior control of combustion phasing and noise, while also having the 30% lower unburned hydrocarbons, 40% lower CO , 35% lower soot, and 4% higher gross efficiency than the premixed regime without any increase in NO_x . It was also observed that increasing the energy fraction of the diesel fuel while in the reactivity-controlled regime resulted in further improvements to efficiency and hydrocarbon and CO emissions, while at the same time decreasing peak heat release rate and noise. This was achieved by simultaneously advancing combustion phasing and increasing combustion duration, which is a method of control unique to the DDFS strategy.

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INTRODUCTION

Typical compression ignition engines operate with conventional diesel combustion (CDC), in which high-reactivity fuels such as diesel are burned in a high-temperature, mixing-controlled (i.e., diffusion-limited), process. In other words, the rate of combustion is determined by the speed at which the fuel can vaporize and mix with oxygen. When simultaneously mixing and burning fuel, there is a tendency for some areas in the reaction zone to have high equivalence ratio (Φ), resulting in high levels of particulate matter (PM) and mono-nitrogen oxides (NO_x) [1]. These are both criteria pollutants which contribute to smog formation and human respiratory illness, and are therefore subject to strict regulations. Exhaust after-treatment devices are required on modern diesel engines in order to meet emissions targets in most markets. However, this after-treatment typically comes at the expense of fuel economy, performance, and/or secondary reducing agents, adding complexity and upfront and recurring costs [2,3].

Much of the work on advanced combustion strategies has been focused on reducing the dependence on after-treatment while maintaining high thermal efficiency. One promising area of research is low-temperature combustion (LTC), which uses long ignition delays (achieved by premixing the fuel and oxygen) to mitigate the formation pathways for PM and NO_x , while simultaneously reducing heat transfer [4,5,6]. One of the biggest drawbacks to the use of long

ignition delays is that ignition becomes kinetically controlled, and therefore subject to small stochastic variations in boundary conditions. As a consequence, combustion control becomes more difficult, confining operation to a smaller range than CDC. Improving the control and operating range of LTC strategies has been an active area of research [7,8].

Homogeneous charge compression ignition (HCCI), one of the first LTC strategies to be developed, is by definition a fully premixed mixture of fuel and air, with a kinetically-controlled ignition event which is path-dependent on temperature, pressure, equivalence ratio, and fuel properties [9,10]. Many researchers have explored the benefits of HCCI, which include high efficiency and orders-of-magnitude reduction of PM and NO_x when compared to CDC [7,8,9,10,11]. While this would seem to nicely address the shortcomings of CDC, the HCCI strategy has its own significant drawbacks. The volumetric nature of HCCI heat release results in a high peak heat release rate (PHRR) and peak pressure rise rate (PPRR), leading to excessive combustion noise even at moderate loads. This can be mitigated by retarding combustion phasing and extending combustion duration, but both come at the expense of reduced efficiency. This leads to a trade-off between efficiency and noise, and limits the peak practical load [8,12]. Because ignition is kinetically controlled, HCCI is also very sensitive to boundary conditions, a problem that is exacerbated by a lack of

fast-response combustion phasing control variables typically available in more traditional combustion strategies, such as fuel injection and spark timing.

As it became evident that a fully homogenous charge was not suitable over a wide range of operating conditions, a variety of partially premixed compression ignition (PPCI) strategies were developed with the idea of maintaining the benefits of HCCI while adding control. In general, PPCI strategies 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 and temperature, within the cylinder. This causes variations in local ignition delay throughout the cylinder, which has the overall effect of increasing combustion duration and retarding combustion phasing, resulting in decreased PHRR and noise. Stratification can be increased by increasing the amount of direct-injected fuel and also by retarding its injection, and the primary difference between the various strategies is the duration and timing of the fuel injections. These strategies range from mostly-premixed, such as partial fuel stratification (PFS) [8], to direct-injecting most or all of the fuel just before combustion starts, as with partially premixed combustion (PPC) [13,14,15]. Between these extremes exists a continuum of stratification levels and methods of achieving them, with each having their relative merits [16,17,18]. Successful operation has been demonstrated over a wide range of speeds and loads, requiring optimization of stratification and fuel selection, boost level, exhaust gas recirculation (EGR), intake temperature, and other variables for each combination of load and speed. While many of the problems with HCCI have been mitigated to some extent, there still exist several issues to address:

1. High levels of unburned hydrocarbons (HC) and carbon monoxide (CO) due to low combustion temperatures and premixed fuel entering crevice regions.
2. High boost is required at high loads in order to maintain low equivalence ratio. This is compounded by low exhaust enthalpy, resulting in the requirement of high combined turbocharger efficiency. If a supercharger is used instead, there will be a significant penalty to brake thermal efficiency in order to drive the required boost.
3. High EGR is required at high loads in order to maintain sufficient ignition delay to avoid excessive combustion noise.
4. Lack of sufficiently independent control over duration and phasing of heat release with a single fuel, resulting in a tradeoff between efficiency and noise as combustion phasing is retarded with increasing load.

Reactivity controlled compression ignition (RCCI) seeks to address the issue of combustion duration control by extending the concept of control by stratification to include fuel reactivity. By injecting two fuels with different levels of resistance to auto-ignition, 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 are typically performed with port-injection of low-reactivity fuel, such as gasoline (often modeled as iso-octane), and direct-injection of high-reactivity fuel, such as diesel (often modeled as n-heptane). Examples include the work of Inagaki et al. [19], Kokjohn et al. [20,21,22], Hanson et al. [23,24], and Splitter et al. [25,26,27,28,29]. Experimental operation with RCCI has demonstrated gross indicated thermal efficiencies in excess of 56% and in-cylinder levels of soot and NO_x below the EPA 2010 Heavy Duty emissions regulations [30]. As with other LTC strategies, RCCI produces relatively high levels of HC and CO, which are primarily caused by low wall temperatures and the inability to completely oxidize fuel that has entered crevice regions [29].

By adjusting the ratio of high- and low-reactivity fuel, the RCCI strategy allows the shape and phasing of heat release to be adjusted with much greater fidelity than is possible in single-fuel strategies. For a given set of operating conditions, the combustion phasing can be controlled entirely by the fuel ratio, and to lesser degree, the timing of the high-reactivity fuel injections. However, the operating range is limited by the reactivity range of the fuels being used. As load increases, the fraction of high-reactivity fuel required to achieve a given combustion phasing decreases. If the load is increased to a high enough point, the fraction of high-reactivity trends to zero, and there is no longer any direct control over the combustion process, as the combustion strategy has essentially become HCCI. 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. Load can also be extended by decreasing the reactivity of the low-reactivity fuel, with examples such as gasoline-ethanol blends [27] and natural gas [31]. However, if operation is desired with conventional fuels at compression ratios typical of diesel engines, it is evident that some additional method of control is required. One path is to explore the possibility of stratifying both fuels.

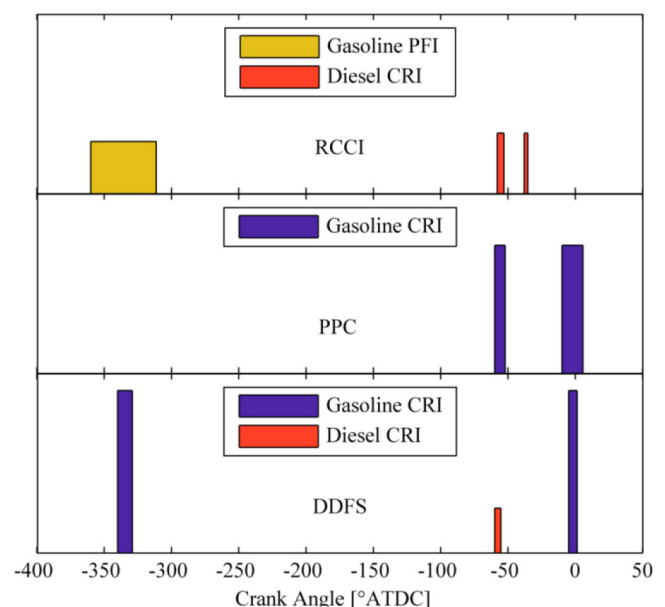


Figure 1. Conceptual illustration of RCCI, PPC, and DDFS. The area of each box represents the relative quantity of each injection.

In our previous work [32], we demonstrated that control over the heat release event could be further improved by coupling RCCI with a diffusion-limited injection of gasoline in a strategy called direct dual fuel stratification (DDFS), which is illustrated alongside RCCI and PPC in Figure 1. Like RCCI, the DDFS strategy utilizes an early injection of gasoline that is almost completely premixed, along with a direct injection of diesel to establish a reactivity gradient along which the first stage of combustion will propagate. As this paper will show, the timing and quantity of the diesel injection affords a large degree of control over the early heat release. DDFS also uses an injection of gasoline just before top dead center (TDC), which was inspired by the highly-stratified injection schedule used in PPC. However, unlike PPC, in which the late gasoline injection is still afforded some mixing time before the start of combustion, the late gasoline injection in DDFS enters after the conclusion of low-temperature heat release (LTHR) into an already-burning charge, and its burn rate is limited primarily by diffusion rather than kinetics. The present work will focus on the role of the reactivity gradients created by the diesel injection and how they enable control over the heat release profile in the DDFS strategy.

EXPERIMENTAL SETUP

Engine

The experimental setup was identical to that used in the previous work [32], and as such, only a brief description of the relevant components will be provided here. The experiments were performed on a 2.44 L Caterpillar 3401E Single Cylinder Oil Test Engine (SCOTE), with specifications listed in Table 1, laboratory setup as illustrated in Figure 2, and modified piston shown in Figure 3.

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

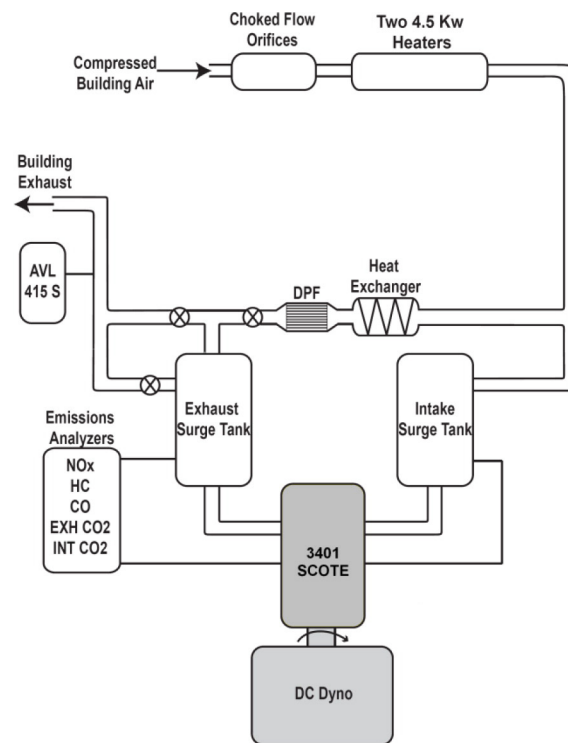


Figure 2. Diagram of experimental apparatus

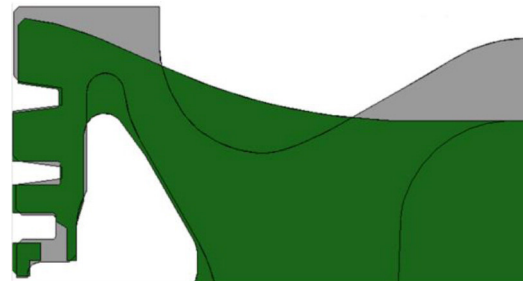


Figure 3. Profile of RCCI piston (green, CR = 14.88) overlaid on stock piston (gray, CR = 16.1)

Dual Direct Injector Head

A stock engine head was modified to allow the mounting of two common rail injectors (CRI), shown in Figure 4. Full details of the design can be found in previous work [33]. The injector specifications are listed in

Table 2. The naming convention for the injectors is that the first subscript refers to the physical location (1 = exhaust side, 2 = intake side), and the second subscript is the number of holes in the nozzle. The stock 7-hole nozzle was used for injecting diesel, but a 10-hole nozzle was used for gasoline injection, as this was seen to reduce soot levels in previous work [33].

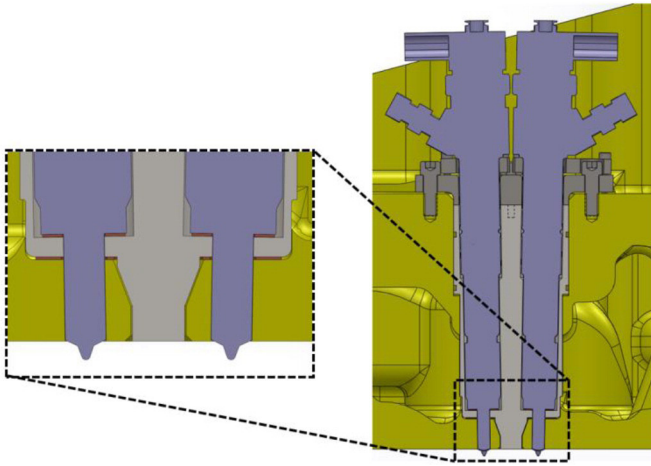


Figure 4. Cross-section of dual direct injector model [32]

Table 2. Common rail fuel injector specifications

Injector Name	CRI _{1,7}	CRI _{2,10}
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

Measurements and Methodology

Cylinder Pressure

Cylinder pressure was measured with a Kistler 6125C21U20 piezoelectric pressure transducer in conjunction with a Kistler 510 charge amplifier. The cylinder pressure traces were processed as described in the previous work [32], and quantities such as PPRR and sound pressure level (SPL) were calculated from individual cycles, thereby generating a statistical distribution for each variable.

Heat Release

Heat release calculations for experimental combustion work are often performed by using a single-zone heat-addition model that assumes the contents of the cylinder can be modeled as an ideal gas at uniform temperature and composition with fixed mass. The chemical heat addition $\dot{Q}_{ch}$ is then defined as the rate of conversion from chemical to sensible internal energy, and can be represented as

$$\dot{Q}_{ch} = mc_v \frac{dT}{dt} + p \frac{dV}{dt} - \dot{Q}_w \quad (1)$$

where m is the total mass in the cylinder and $\dot{Q}_w$ is the rate of heat transfer into the cylinder from the walls. Equation (1) can be combined with the ideal gas law and simplified into the commonly-used expression shown in Equation (2), in which γ represents the ratio of specific heats.

$$\dot{Q}_{ch} = \left(\frac{1}{\gamma - 1} \right) V \frac{dP}{dt} + \left(\frac{\gamma}{\gamma - 1} \right) p \frac{dV}{dt} - \dot{Q}_w \quad (2)$$

When used with experimental data, this approach requires models for both γ , which is dependent on temperature and composition, and $\dot{Q}_w$, which is typically dominated by turbulence. While models for both exist for the typical spark-ignited gasoline and compression-ignited diesel combustion regimes, no such models exist for the dual-fuel, multistage combustion presented in this work. As a substitute, we make an additional assumption that the process is locally polytropic, with polytropic index n . This allows for $\dot{Q}_w$ to be represented directly as

$$\dot{Q}_w = mc_n \frac{dT}{dt} \quad (3)$$

where c_n is the polytropic specific heat capacity, which is given by

$$c_n = c_v - \frac{R}{n - 1} \quad (4)$$

By substituting Equations (3) and (4) into Equation (1) and proceeding the same as before, we arrive at an expression for the apparent heat release rate (AHRR) in terms of the polytropic index n :

$$\text{AHRR} = \dot{Q}_{ch} = \left(\frac{1}{n - 1} \right) V \frac{dP}{dt} + \left(\frac{n}{n - 1} \right) p \frac{dV}{dt} \quad (5)$$

The result in Equation (5) is quite similar to that from Equation (2), with the difference being that the substitution of n for γ has the effect of including the heat transfer.

The assumption of a polytropic process is quite good during the compression and expansion strokes, but is obviously incorrect during the combustion process. However, for the purposes of calculating combustion parameters such as PHRR and crank angle at 50% of total heat release (CA50), the use of an *effective* polytropic index produces reasonable and repeatable results even during combustion mode changes. For the results shown here, n was calculated for both the early compression stroke and late expansion stroke. The value for the compression stroke was used until the start of combustion, and the value for the expansion stroke was used after combustion. The transition between the two values was modeled by a logistic function centered at the location of PHRR and with a width proportional to the