

Optical Investigation of the Reduction of Unburned Hydrocarbons Using Close-Coupled Post Injections at LTC Conditions in a Heavy-Duty Diesel Engine

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ABSTRACT

Partially premixed low-temperature combustion (LTC) using exhaust-gas recirculation (EGR) has the potential to reduce engine-out NO_x and soot emissions, but increased unburned hydrocarbon (UHC) emissions need to be addressed. In this study, we investigate close-coupled post injections for reducing UHC emissions. By injecting small amounts of fuel soon after the end of the main injection, fuel-lean mixtures near the injector that suffer incomplete combustion can be enriched with post-injection fuel and burned to completion. The goal of this work is to understand the in-cylinder mechanisms affecting the post-injection efficacy and to quantify its sensitivity to operational parameters including post-injection duration, injection dwell, load, and ignition delay time of the post-injection mixture. Three optical diagnostics - planar laser induced fluorescence of OH radicals, planar laser induced fluorescence of formaldehyde, and high-speed imaging of natural combustion luminescence - complement measurements of engine-out UHC with parametric variations of main- and post-injection timing and duration. Across all conditions tested, each at 1200 RPM, the optimal post-injection command duration for UHC reduction was approximately 400 microseconds (2.9 °CA). Also, conditions with shorter (3.4 °CA) post-injection ignition delays were over twice as effective on a percentage basis at reducing engine-out UHC as those with longer (5.5 °CA) post-injection ignition delays. Optical data at the post-injection "sweet-spot," where UHC emissions are minimized, indicate that the post injection promotes transition to second-stage ignition in the near-injector region, most likely by enriching the overly fuel-lean mixtures in the wake of the main injection.

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INTRODUCTION

Low temperature combustion (LTC) has been proposed as a strategy for meeting market demands for fuel efficiency while also satisfying regulated limits on NO_x and soot emissions from heavy-duty diesel engines. In-cylinder strategies designed to reduce engine-out emissions of these two pollutants are of interest because they could reduce or even eliminate the burden on aftertreatment systems. Compared to conventional diesel combustion, LTC strategies generally can be characterized by high levels of intake charge dilution and increased mixing of fuel with the intake charge prior to combustion.

One method for diluting the intake charge is exhaust gas recirculation (EGR), which reduces the peak temperatures of combustion and hence the thermally produced NO_x [1]. The slowing of NO_x formation is a result of two effects: an increase in the heat capacity of the intake mixture, which

reduces peak temperatures, and reduction in intake oxygen, which slows NO_x formation kinetics.

Increased pre-combustion mixing is typically achieved by altering the fuel-injection timing. Conventional diesel operation uses injection timings that occur several crank angle degrees before top-dead-center (TDC), so that the ignition delay is short and combustion occurs near TDC. This combustion phasing is favorable for thermodynamic efficiency, but much of the charge is fuel-rich during combustion, which can also lead to significant soot formation [2]. LTC techniques achieve less soot formation by placing the injections either much earlier or much later than the conventional injection timing.

Early injection schemes - often termed partially-premixed compression ignition (PPCI), premixed-charge compression ignition (PCCI), premixed compression ignition (PCI), or in some cases in the literature, even homogeneous charge compression ignition (HCCI) - operate by providing

significant time between the injection and the thermodynamic state during compression that induces ignition [3, 4, 5]. Given the long ignition delay of a mixture at early injection timings, these schemes create a positive ignition dwell, which is the time between the end of the injection and the start of combustion. A positive ignition dwell gives all of the injected fuel and air more time to mix before combustion, reducing rich regions where soot could form, while often creating a fast, efficient premixed heat-release profile [6].

Late-injection schemes - such as "modulated kinetics" (MK) combustion [7, 8] - achieve similar ends as the early-injection schemes by injecting fuel very near TDC or even into the expansion stroke. Here, a positive ignition dwell is possible because of high rates of EGR and the cooling of the charge due to expansion. As with early-injection strategies, the positive ignition dwell provides more time for mixing and hence lower soot formation and a more premixed-type combustion event.

While these methods of LTC can lead to substantial reductions in NO_x and soot emissions, there are still issues remaining. In particular, unburned hydrocarbon (UHC) and carbon monoxide (CO) emissions can increase under these conditions [9, 10]. Going from conventional diesel combustion to LTC operation can be viewed as moving along a trade-off between soot and UHC (and CO); as the ignition delay increases and more premixing is allowed before the start of combustion, fewer regions of the reactants are fuel-rich enough to promote soot production, but UHC emissions increase, especially as the load decreases (for instance see Figure 2). For this reason, we will focus primarily on low-load, LTC conditions where UHC emissions are particularly problematic.

Previous studies have investigated the sources of UHC at LTC conditions. Our own previous work has shown that some overly-lean mixtures in the region of the injector formed after the end of injection do not achieve complete combustion and can be significant sources of hydrocarbons [11, 12]. Other UHC sources, including wall-wetting and fuel in the crevices, can also be important, especially in light-duty engines [13, 14]. The current study focuses on UHC emissions from over-leaning in the bulk charge.

Overly-lean mixtures in the injector region have been measured using a variety of techniques. In our previous work, we used a toluene fluorescence technique in a heavy-duty, single-cylinder, optical research engine as well laser-Rayleigh-scattering in an engine simulation spray facility [12]. For the tracer fluorescence measurements, fuel was doped with a known quantity of toluene and injected into a non-reacting charge to enable quantitative fuel concentration measurements, at least up to the time that combustion would normally commence. Using the fuel concentration data, an equivalence ratio map can be generated corresponding to a given intake oxygen concentration. Equivalence ratio contour maps for an intake-oxygen concentration of 12.7% are shown in Figure 1.

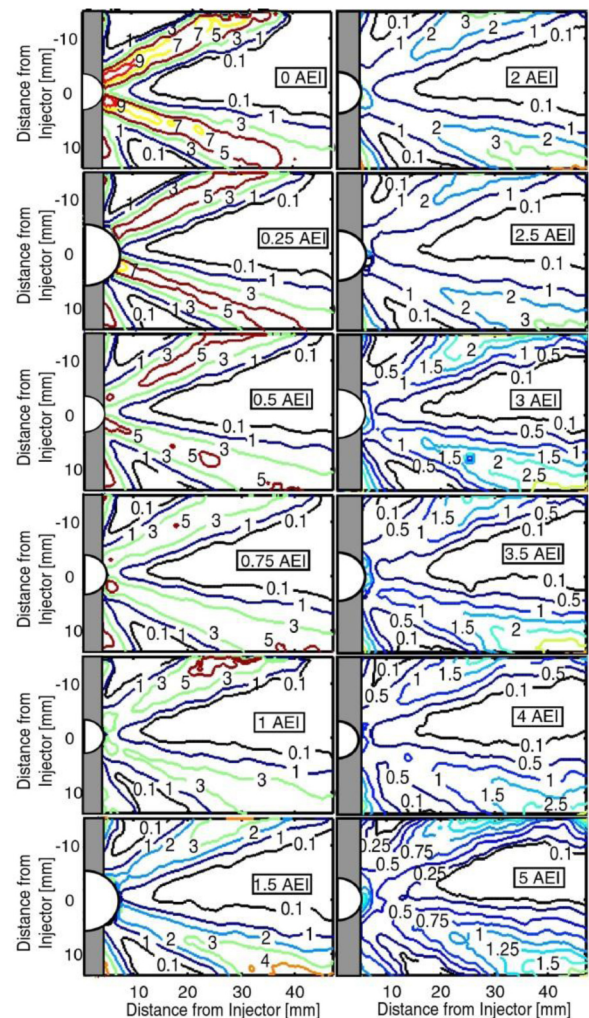


Figure 1. Equivalence ratio measurements in the optical engine after the end of injection using toluene PLIF for a 12.7% O_2 intake charge, from Ref. [12]. Crank angle degrees are displayed in units of degrees after the end of injection. The lines indicate contours of constant equivalence ratio. The injector is located on the left of the images and two jets of the 8-hole injector pass through the measurement plane - one upward and to the right (near 2 o'clock) and the other downward and to the right (near 4 o'clock). Copyright © SAE International. Reprinted with permission [12].

The equivalence ratio contours in Figure 1 show that the mixtures near the injector (left side) become leaner very quickly after the end of the injection, initially at a rate of approximately 4 equivalence ratio units per crank angle degree. By 5 °CA after the end of injection (AEI), the equivalence ratio is 0.5 or lower in the near-injector region. Other optical data showed that first-stage (cool-flame) ignition occurs throughout the jet, but second-stage ignition occurs only in the richer downstream mixtures (right side of images), near 5 °CA AEI [11]. At the conditions examined, the upstream fuel-lean mixtures never achieved second stage

ignition in the time available before significant cylinder expansion. Hence, incomplete combustion of those fuel-lean mixtures contributed to UHC emissions.

These results and others like them [15] motivated a follow-up study in the current engine, showing promising results for UHC reduction with the use of post injections [16]. The rationale for using post injections is to enrich the overly-lean mixture near the injector, essentially “kicking” it into second-stage ignition and reducing unburned hydrocarbons from the injector region. Because of limitations in the fuel injection system, only two-injection schedules were tested, one with a short, close-coupled post injection (injection dwell $\approx 1^\circ\text{CA}$) and one with a longer post-injection duration and dwell (injection dwell $\approx 3^\circ\text{CA}$). The injection schedules for these two tests are in Figure 2a.

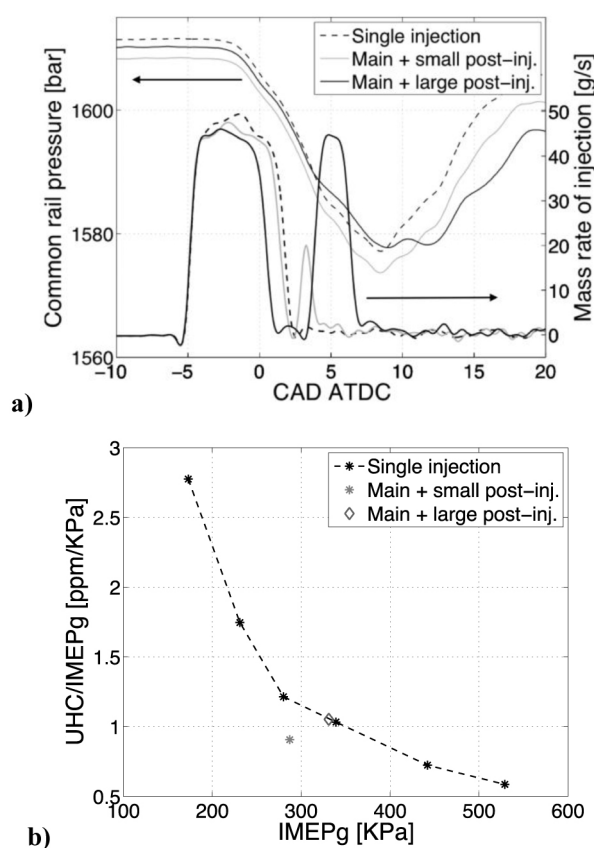


Figure 2. Results from Chartier et al. [16] that show a) injection schedule for a single injection and two post-injection schedules and b) normalized UHC measurements for these three conditions. Copyright © SAE International. Reprinted with permission [16].

The engine-out UHC results in Figure 2b illustrate the significant effect that a short, close-coupled post injection can have on UHC reduction. Whereas the longer post injection did not reduce UHC emissions relative to the single-injection baseline, the short, close-coupled post injection reduced the UHC by 27%. Optical investigations of the progression of combustion for these two conditions were consistent with

enrichment of the near-injector overly-lean mixture by the post injection, bringing the combustion to completion to reduce UHC emissions.

Despite the obvious gains that were observed by using a short, close-coupled post injection in this previous study, only one short, close-coupled post-injection timing was found to be repeatable due to limitations of the heavy-duty injector. The goal of the current study is to further explore the effect of a post injection on UHC reduction over a much wider range of injection schedules with a fast-responding, light-duty injector.

Fluid mechanical (mixing) and chemical kinetics effects of the post injections are explored in two ways. First, the command duration of injection for close-coupled post injection (DOI_{2C}) is varied from very short ($\text{DOI}_{2C}=200$ microseconds) to relatively long ($\text{DOI}_{2C}=800$ microseconds) for a given main injection duration and timing. This variation in post-injection duration not only changes the amount of fuel delivered in the post injection, but also the penetration and fluid mixing characteristics of the post jet. Short post injections ($\text{DOI}_{2C}<350$ microseconds) do not penetrate to the bowl wall of this engine within the time available for combustion, and have different injection profiles and ignition dwells than long post injections. Additionally, post injections that do not penetrate to the bowl wall do not have fuel/air mixtures that “roll up” along the wall into the large side rollers in the jet-jet interaction regions that characterize low-temperature combustion in diesel engines [17]. These possible differences in penetration, injection profile, and mixing time could significantly change how the post injection enriches the overly-lean mixture left by the main injection.

Second, the ignition delay of the mixture created with the post injection is varied by changing the combustion phasing of the entire injection schedule. Here, main- and post-injection events are moved as a unit to earlier and later timings relative to a baseline timing. In doing so, the ignition delay of the near-TDC main injection changes only slightly because ambient conditions change minimally near TDC, when the piston velocity is low. The ignition delay of the post injection, however, varies much more because the ambient gases change more significantly at the later injection timing of the post injection when the cylinder contents are starting to expand. In this case, an increase (or decrease) in the ignition delay of the post injection would allow for more (or less) time for mixing between the post-injection fuel and the overly-lean mixture from the main injection. Even though the schedule is shifted, the main- and post-injection durations and dwell between them are held constant, so the fluid mechanics of injection remain largely the same. In this way, the fluid mechanical effects of post-injection penetration and mixing can be separated from kinetics effects, to some degree.

EXPERIMENTAL SETUP

Optical Engine Experiment

Table 1. Engine and fuel system specifications.

Engine base type	Cummins N-14, DI diesel
Number of cylinders	1
Cycle	4-stroke
Number of intake valves	2
Number of exhaust valves	1*
Intake valve opening	17° BTDC Exhaust [‡]
Intake valve closing	195° ATDC Exhaust [‡]
Exhaust valve opening	235° BTDC Exhaust [‡]
Exhaust valve closing	27° ATDC Exhaust [‡]
Combustion chamber	Quiescent, direct injection
Swirl ratio	0.5 (approx.)
Bore	139.7 mm [5.5 in]
Stroke	152.4 mm [6.0 in]
Bowl width	97.8 mm [3.85 in]
Displacement	2.34 liters [142 in ³]
Connecting rod length	304.8 mm [12.0 in]
Piston pin offset	None
Geometric compression ratio	11.2:1
Replicated compression ratio	16:1
Fuel Injector	Delphi DFI-1.5 (light duty)
Fuel injector type	Common-rail, solenoid actuated
Cup (tip) type	Mini-sac
Number of holes & arrangement	8, equally-spaced
Spray pattern incl. angle	156°
Nominal orifice diameter	0.131 mm

*In this optically accessible diesel engine, one of the two exhaust valves of the production cylinder head was replaced by a window and periscope (see Fig. 1).

[‡]All valve timings correspond to the crank angle of the first detectable movement from fully closed.

The engine used in this study is an optical, single-cylinder Cummins N-series direct-injection, heavy-duty diesel engine. The engine has a bore of 139.7 mm and a stroke of 152.4 mm, and is equipped with a Bowditch piston with an open, right-cylindrical bowl and a flat fused-silica piston-crown window providing imaging access to the bowl, viewing from below. A flat, round window is also installed in place of one of the exhaust valves for imaging access to a portion of the squish region above the piston (view not used in the current study). A 30-mm wide curved window matching the contour of a portion of the bowl-wall allows laser access into the

bowl. Additional laser access is available through flat rectangular windows installed in a ring at the top of the cylinder. Because of thermal load limitations of the windows, the engine was skip-fired, with each fired cycle followed by nine motored cycles. Information about engine geometry is in Table 1 and a schematic of the experiment is in Figure 3. Further details about this engine can be found in Refs. [18, 19].

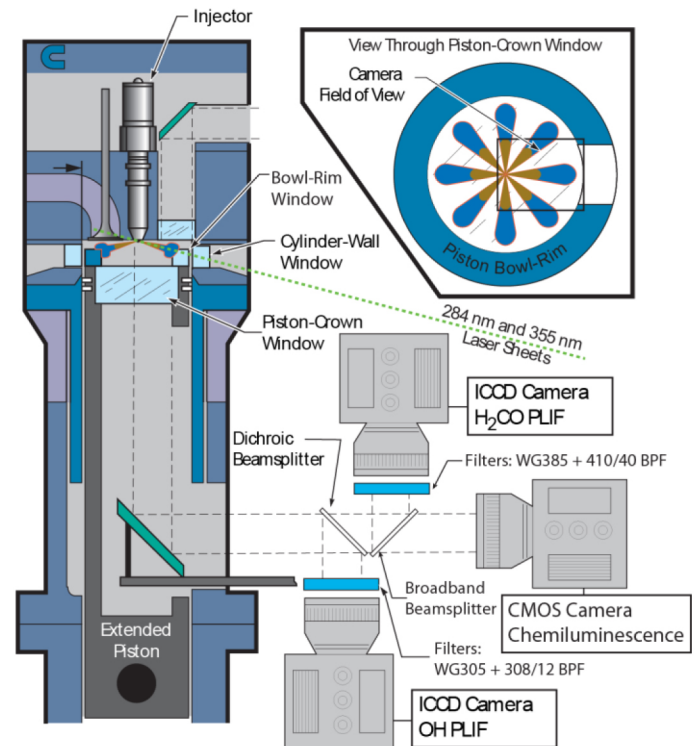


Figure 3. Experimental setup of the single-cylinder engine, laser configuration, and three-camera optical system. The camera field-of-view is shown in the upper right.

The injector used in this study is a Delphi DFI 1.5 light-duty, common-rail injector. A light-duty solenoid-driven injector with 131 micron orifices was used in place of the previous heavy-duty injector for its fast response time and its ability to deliver consistent, close-coupled, short-duration post injections over a range of injection schedules. Because of the low lubricity and low viscosity of special fuels (such as n-heptane) selected for optical research, the production common-rail fuel pump could not be utilized. Instead, a custom high-pressure diaphragm pump specially designed for low-lubricity fuels pressurizes the fuel rail at up to 2000 bar. The delivery rate of the diaphragm pump is limited, however, and as a result it could only sustainably pressurize the fuel rail to 1200 bar at the static back-leak rate of this light-duty injector.

Engine Operating Conditions

The engine is operated in a high-EGR (12.6% intake O₂) late-injection LTC mode summarized in Table 2, for which the start of main combustion begins slightly after TDC. The low intake temperature in Table 2 was intentionally selected for several reasons. First, diluting the intake with N₂ rather than real EGR yields a higher specific-heat ratio of the charge, so that lower intake temperatures are required to achieve the same compressed temperatures at TDC as with EGR. Even after accounting for differences in compression-heating effects, a relatively low TDC temperature allows fuel injection close to TDC while still achieving a long enough ignition delay for LTC conditions. Injecting close to TDC is desirable from a fundamental perspective because the in-cylinder density remains nearly constant for a relatively long time compared to injecting during the intake or expansion strokes. As a result, the ignition delay for the main injection remains nearly constant for small shifts of the injection schedule, which facilitates the study on kinetic effects described at the end of the introduction. Furthermore, studying in-cylinder processes at nearly constant density is useful for comparison to analysis and/or experiments in constant volume combustion vessels. The cool conditions with near-TDC conditions also provided combustion phasing favorable for efficiency while still achieving pre-combustion mixing times characteristic of LTC. Finally, as discussed below, for more typical U.S. diesel fuels with cetane numbers near 47 (rather than 56 for n-heptane [20]), such cool intake conditions would not be required to achieve the same ignition delays at the high EGR conditions of this study.

As discussed above, UHC emissions are a particular issue at low-load conditions; the engine load-range in this study is varied between one (1) and about six (6) bar gIMEP. More details about the engine operating conditions are in Table 2.

Table 2. Engine operating conditions.

Engine Speed	1200 RPM*
Engine Load Range	1-6 bar gIMEP†
Intake O ₂	12.6%
Fuel Pressure	1200 bar
TDC Motored Density	18 kg/m ³
TDC Motored Temperature	837 K
BDC Pressure	164 kPa
BDC Temperature	78 °C
(16:1 Compression Ratio BDC Pressure)	101 kPa (abs)
(16:1 Compression Ratio BDC Temperature)	36°C

* At this speed, each crank angle degree (°CA) is 139 microseconds in duration.

† gIMEP = Gross Indicated Mean Effective Pressure, calculated using indicated work done during compression and expansion strokes only.

In this partially-premixed combustion mode, the heat release displays a small first-stage (cool-flame) ignition peak followed by a large second-stage peak and a small mixing-controlled combustion tail. The addition of a post injection typically adds another small peak to the heat release curve after the main second-stage ignition. These features are illustrated in Figure 4.

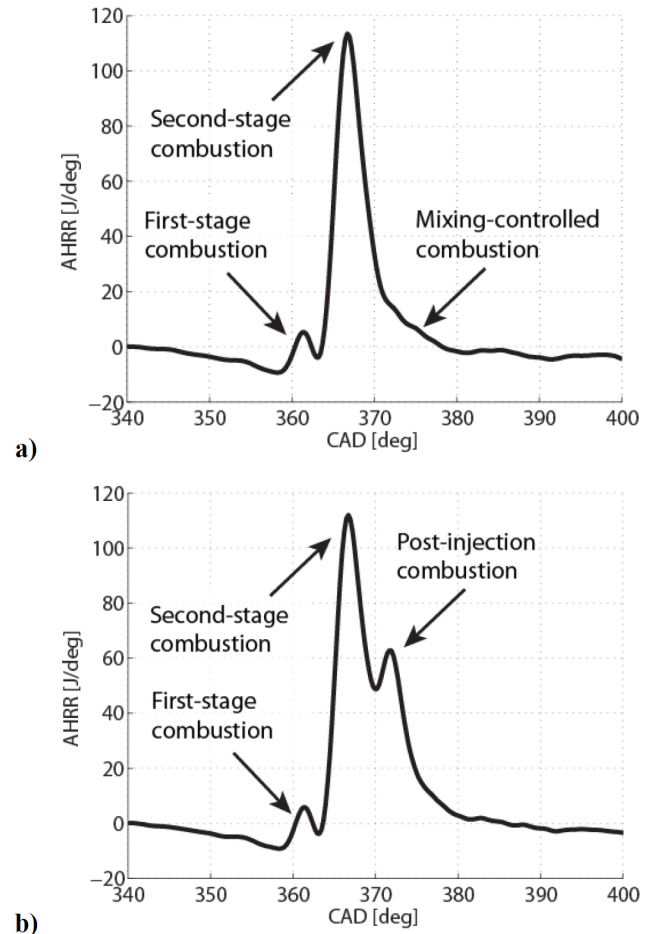


Figure 4. Apparent heat release rate (AHRR) for single injection (a) with $SOI1_C=352$ CAD and $DOI1_C=1000$ microseconds and a main plus post injection (b) with $SOI1_C=352$ CAD, $DOI1_C=1000$ microseconds, $SOI2_C=363.5$ CAD, and $DOI2_C=400$ microseconds.

These operating conditions are designed to parallel those of the previous study by Chartier *et al.* [16] as closely as possible. As described above in the Introduction, this is done purposefully to further investigate the role of short, close-coupled post injections for UHC reduction.

Two important differences between the two studies are the fuel and the fuel injector. In anticipation of fuel concentration and other optical diagnostic measurements to follow, we selected n-heptane as the fuel. While the diesel primary reference fuels of the previous study better reflect the physical characteristics (e.g., boiling point) of typical diesel fuel, as yet, a feasible and quantitative laser-induced fuel-

tracer fluorescence diagnostic has not been developed for such heavy fuels. Techniques for lighter fuels like n-heptane have been developed and successfully employed (see Figure 1 for instance), so it was chosen as the fuel. As will be shown later in the results, the characteristic post-injection UHC behavior observed in Chartier *et al.* [16] with high boiling-point fuels is retained here using n-heptane.

As shown in Table 3, however, the cetane number for n-heptane is significantly higher than for the diesel primary reference fuel blend used in Chartier *et al.*. Using the same in-cylinder thermodynamic conditions would result in much shorter ignition delays with n-heptane. To match ignition delay, either the temperature or the pressure of the intake stream could be adjusted. The chemical kinetics of ignition and the following combustion processes (including incomplete combustion) depend strongly on temperature, so we were hesitant to decrease the temperature. With the change of injectors, the fluid mechanical processes of fuel penetration and mixing would already be somewhat different, so we chose to reduce the intake pressure (boost) to extend the ignition delay. A lower intake pressure yields a lower in-cylinder density, which would yield richer mixtures at a given downstream position in the jets, but the smaller nozzle size of the light duty injector offsets this effect, so that the mixtures are similar for the two injectors. The TDC conditions required to match temperature and ignition delay are shown in Table 3, and the heat release for the two studies are in Figure 5.

Table 3. Comparison of operating conditions between current study and Chartier *et al.* [16].

	Current Study	Chartier <i>et al.</i> [16]
Intake O ₂	12.6%	12.7%
Fuel	n-heptane	32.3% cetane 67.7% heptamethylnonane
Cetane number	56	42.5
TDC Motored Density	18 kg/m ³	22.1 kg/m ³
TDC Motored Temperature	837 K	837 K
TDC Motored Pressure	43.2 bar	54.9 bar
Start of main combustion	360.1 – 366.6 CAD	362.2 CAD
Ignition delay to main combustion	9.1-9.6 CAD	7.2 CAD

The heat release profiles of the two studies in Figure 5, Ref. [16] and the current study, are qualitatively similar, though the timing is not exactly the same as a result of the 1 °CA difference in SOI_{1C} and slight changes in ignition delay. In the single injection case, a similar first-stage ignition heat release peak appears before the second-stage ignition heat release peak. In the case with a main- plus post-injection

schedule (Figure 5b), the post-injection heat release profile is different between the two cases. In the previous study, the post-injection heat release for the sole condition studied was small, so that it appeared on the main injection heat release profile as a small shoulder rather than as a distinct peak. In the current study, the optimal post injection is larger, and the main-injection duration is longer. As a result, the post-injection, while still close coupled, is phased later relative to the main-injection heat release, so that the post-injection heat release peak is later and more distinct.

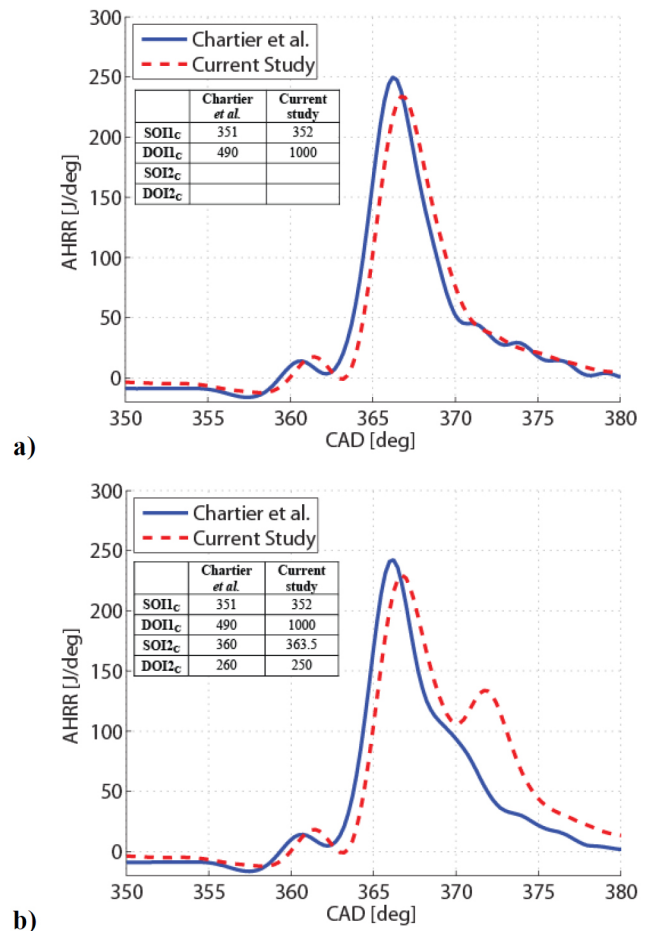


Figure 5. Apparent heat release rate comparison between the previous study (Chartier *et al.* [16]) and the current study for single (a) and main plus post injection (b). Injection schedules are listed in tables in the figures.

Engine Diagnostics

Several diagnostics are used to investigate the origin of UHC emissions at LTC conditions: cylinder pressure measurements, exhaust UHC emissions analysis, and three different optical diagnostics.

Cylinder pressure is measured with an AVL QC34D pressure transducer every quarter crank-angle degree. The apparent heat release rate is calculated from the measured

cylinder pressure using standard techniques described in Ref. [21].

Engine-out UHC emissions are measured with a California Analytical Instruments (CAI) 600 series hydrocarbon analyzer. This analyzer samples the exhaust stream and uses a flame ionization detector to measure total unburned hydrocarbons. The exhaust stream is pulled from the exhaust manifold, within 100 mm of the exhaust valve. The line from the engine to the analyzer is heated to 155 degrees Celsius to prevent condensation of water and hydrocarbons. A pump provides a continuous flow of exhaust gases to the analyzer, and the transit time between the engine and the analyzer is about 4 seconds. All UHC data are averaged over a run-time of 100 fired cycles. To account for the skip-fired operation of the engine, the reported UHC data have been multiplied by 10 to approximate what they would be in a continuously-fired, single-cylinder engine. The standard deviation in UHC measurements between runs is approximately 2 ppm (corresponding to 20 ppm after correcting for a continuously-fired engine).

Three different optical diagnostic techniques are employed to gain insight into the origins of UHC emissions. Two important radical species involved in ignition provide information about the evolution of in-cylinder unburned fuel. Formaldehyde (H_2CO) is produced during first-stage ignition and consumed during second-stage ignition. Detailed chemical kinetics simulations predict that after first-stage ignition, the evolution of formaldehyde mirrors that of the overall pool of unburned hydrocarbons [11]. Hence, formaldehyde serves as a marker of unburned fuel from mixtures that have achieved first-stage ignition but have not yet reached second-stage ignition. Formaldehyde does not serve as a good marker for all unburned fuel, such as that from regions that have not achieved first-stage ignition (e.g., wall wetting and potentially crevices), nor does it mark UHC in rich mixtures after second-stage ignition. However, previous studies [11, 12] have shown that in the near injector region, overly-lean mixtures are a primary source of UHC emissions, so that formaldehyde serves as a good marker for those UHC sources.

The second radical species, the hydroxyl radical (OH), rises in concentration by orders of magnitude at second-stage ignition in fuel-lean to stoichiometric mixtures [22], and hence provides an indication of second-stage ignition and relatively complete combustion and hence low unburned fuel concentration.

As described below, two laser-based techniques are used to measure these two radical species, similar to previous studies conducted at LTC conditions in this engine [11, 16].

Laser induced fluorescence of formaldehyde (H_2CO -PLIF): The third harmonic output of a Spectra-Physics Quanta-Ray single-cavity Nd:YAG laser at a wavelength of 355 nm is formed into a 30-mm wide sheet with a thickness of 1 mm or less. A pulse energy of 80 mJ at the engine is used laser-induced fluorescence of formaldehyde within the

engine cylinder. The sheet is oriented at a 12° angle relative to the firedeck to probe H_2CO along the approximate symmetry plane of one of the fuel jets. LIF emission is collected at wavelengths longer than 310 nm with an intensified Princeton Instruments PI-MAX 3 intensified charge-coupled device (ICCD) camera with a HQf (GEN-III) intensifier and a resolution of 1024×1024 , a gate time of 50 ns, and at maximum gain. The camera is fitted with a Nikkor glass f/2.5 lens. Two filters, a 385-nm long-wave-pass and a 40-nm wide bandpass filter centered at 408 nm, reject laser scatter and other interference outside the formaldehyde fluorescence emission band.

Laser induced fluorescence of OH (OH-PLIF): A Spectra-Physics Quanta-Ray single-cavity Nd:YAG laser pumps a Spectra-Physics optical parametric oscillator (OPO) to produce the laser excitation for the OH-PLIF technique. The OPO is tuned to near 568 nm and the light is doubled using a BBO crystal to approximately 284 nm to excite the Q1(9), Q2(8), and P1(5) transitions of OH that merge as a result of pressure broadening. The 284-nm laser beam is co-aligned with the 355-nm laser beam for H_2CO -PLIF upstream of the sheet-forming optics. As a result, the laser sheet at 284 nm is approximately the same size as that at 355 nm: 30 mm wide and up to 1 mm thick. The 284-nm laser pulse energy at the engine is 19 mJ per pulse. LIF emission is collected at wavelengths shorter than 385 nm with an intensified Princeton Instruments PI-MAX 3 ICCD camera with a Super-blue, slow-gate intensifier (Gen-II) and a resolution of 1024×1024 , a gate time of 410 ns, and at maximum gain. The camera is equipped with an ultraviolet (UV) Nikkor f/4.5 lens. Three filters, a 65-nm wide blocked bandpass filter at 310 nm, a 15-nm wide unblocked bandpass filter at 310 nm, and a color-glass SWG305, reject laser scatter and other interference outside the OH fluorescence band near 308 nm. Detuning the laser off the transition at 284 nm produces almost no signal through the filter pack, indicating that the signal shown in the OH-PLIF images is almost entirely fluorescence of OH.

For both laser-induced fluorescence techniques, the LIF data are limited to one frame per cycle, due to repetition-rate constraints of both the laser and camera systems. The OH-LIF signal is collected 1 microsecond before that of the H_2CO -LIF signal, so that they are virtually simultaneous relative to engine time scales. As shown in Figure 3, the fluorescence emission is collected through the large piston-crown window and directed to the two ICCD and one high-speed complementary metal oxide semiconductor (CMOS) camera using two beamsplitters. A long-wave-pass beamsplitter with a cut-off near 385 nm directs UV light to the OH-PLIF camera. The remaining light is split by a 50% reflectance broadband beamsplitter between the H_2CO -PLIF camera and the high-speed CMOS camera (described below).

These two LIF techniques are used together to simultaneously image intermediate species after first- and second-stage ignition. As described above, H_2CO is a marker