

Investigation of Fuel Reactivity Stratification for Controlling PCI Heat-Release Rates Using High-Speed Chemiluminescence Imaging and Fuel Tracer Fluorescence

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ABSTRACT

Premixed charge compression ignition (PCI) strategies offer the potential for simultaneously low NO_x and soot emissions with diesel-like efficiency. However, these strategies are generally confined to low loads due to inadequate control of combustion phasing and heat-release rate. One PCI strategy, dual-fuel reactivity-controlled compression ignition (RCCI), has been developed to control combustion phasing and rate of heat release. The RCCI concept uses in-cylinder blending of two fuels with different auto-ignition characteristics to achieve controlled high-efficiency clean combustion.

This study explores fuel reactivity stratification as a method to control the rate of heat release for PCI combustion. To introduce fuel reactivity stratification, the research engine is equipped with two fuel systems. A low-pressure (100 bar) gasoline direct injector (GDI) delivers iso-octane, and a higher-pressure (600 bar) common-rail diesel direct-injector delivers n-heptane. A sweep of the common-rail injection timing creates a range of fuel reactivity stratification. A high-speed digital camera provides images of ignition and combustion luminosity, composed primarily of chemiluminescence. A quantitative laser-induced fuel-tracer fluorescence diagnostic also provides two-dimensional measurements of the mixture distribution prior to ignition. The injection timing sweep showed that the peak heat-release rate is highest for either early or late common-rail injections of n-heptane, and displays a minimum at mid-range injection timings near 50° BTDC. At very early injection timings, the optical data show that the charge is well-mixed and overall fuel lean, so that it ignites volumetrically, resulting in rapid energy release. Conversely, when the injection timing is late in the cycle (near TDC), the mixing time is relatively short and much of the fuel-air mixture in the n-heptane jet is fuel-rich. Such mixtures that are near stoichiometric or richer have similar ignition delays, so that the charge ignites nearly instantaneously throughout the n-heptane jets. For the mid-range injection timings, at the minimum in the peak energy release rate, ignition occurs in the downstream portion of the n-heptane jet in localized auto-ignition pockets generated by the common-rail injection of n-heptane. The subsequent combustion process then progresses upstream toward the centrally mounted common-rail injector at a slower rate than either the early or late injection timings. In agreement with the observed combustion zone progression from the bowl-wall toward the injector, the fuel concentration measurements show that the fuel reactivity generally decreases from the bowl-wall toward the common-rail injector.

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INTRODUCTION

Highly premixed compression-ignition strategies (e.g., homogenous-charge compression-ignition (HCCI)) offer attractive emissions and performance characteristics (i.e., high efficiency and low NO_x and soot emissions) [1,2,3,4,5,6,7]. However, these operating strategies are

generally confined to low engine loads due to inadequate control of both the heat-release rate and combustion phasing.

To address the issues with HCCI combustion phasing control, Bessonette et al. [8] performed experiments using fuels with a range of auto-ignition characteristics. Their findings suggest that the best fuel for HCCI operation may have an autoignition quality between that of diesel fuel and



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gasoline. Using a compression ratio of 12:1 and a fuel with a derived cetane number of ~ 27 (i.e., a gasoline boiling range fuel with an octane number of 80.7), they were able to extend the HCCI operating range to 16 bar BMEP - a 60% increase in the maximum load they achieved compared to operation using traditional diesel fuel. Furthermore, their results showed that low-load operation, below 2 bar BMEP, required a derived cetane number of ~ 45 (i.e., traditional diesel fuel). Thus, their results suggest that the fuel blending might allow HCCI to be optimized for specific operating conditions. Inagaki et al. [9] investigated dual-fuel PCI operation as a means for reducing the EGR requirements of PCI strategies. Using premixed iso-octane and direct injected diesel fuel, they were able to operate in PCI mode at 12 bar IMEP. Based on the work of Bessonette et al. [8] and Inagaki et al. [9], it is likely that different fuel blends will be optimal at different operating conditions (e.g., a high cetane fuel at light load and a low cetane fuel at high load). Thus, it may be desirable to have the capability to operate with fuel blends covering the spectrum from neat gasoline to neat diesel fuel depending on the operating regime.

Our previous work [3, 10] further explored the dual-fuel PCI approach of Inagaki et al. [9] (i.e., blending two fuels with different auto-ignition characteristics in-cylinder). Results from metal engine experiments [2, 3, 11, 12, 13, 14, 15] suggest that stratifying the fuel reactivity via in-cylinder fuel blending improves control over the heat-release rate allowing an extension of the PCI combustion regime to higher engine loads (up to 16.5 bar IMEP [2]). CFD [3, 10] modeling predicts that the combustion phasing is controlled by the overall fuel reactivity and the combustion duration is controlled by spatial gradients in the fuel reactivity. Consequently, we termed this alternative combustion mode "reactivity-controlled compression-ignition," or RCCI. Further, the metal engine experiments have shown that RCCI combustion can achieve gross indicated efficiencies over 50 percent for a wide range of operating conditions while meeting current, heavy-duty, on-highway NO_x and soot emissions limits in-cylinder. The high-efficiency is primarily due to reductions in heat transfer losses by avoiding high temperature regions [15]. Although they are useful to understand emissions and performance tradeoffs, details of the dominant in-cylinder processes controlling RCCI combustion cannot be observed in metal engine experiments.

In an attempt to improve the fundamental understanding of RCCI combustion, Splitter et al. [14] used in-cylinder Fourier-transform infrared (FTIR) spectroscopy to investigate the evolution of the RCCI combustion process. FTIR spectra were acquired at two locations, to provide a degree of spatial resolution, and were indexed to engine crank-angle to give cycle-averaged, crank-angle-resolved in-cylinder spectroscopy. Their results suggest that the RCCI combustion process proceeds at different rates in different locations of the cylinder. However, their measurements were restricted by the limited optical access of their engine. Additionally, several studies have used detailed CFD modeling (e.g., Kokjohn et

al. [15]) with reduced kinetics mechanisms to investigate RCCI combustion. The CFD modeling predicts that in-cylinder fuel blending creates non-uniformities in the auto-ignition characteristics of the charge (i.e., fuel reactivity) and thereby controls the heat-release rate. To further the understanding of the mechanisms controlling RCCI heat release, we used optical engine experiments to investigate ignition and reaction zone progression in an engine operating in the dual-fuel RCCI combustion mode [16]. High-speed chemiluminescence imaging showed that ignition generally occurs in the downstream portion of the jet - ignition sites were found both near the piston bowl rim and in the squish region. These ignition sites grew for several crank angle degrees and merged into larger reaction zones. In the same study, the fuel distribution prior to ignition was imaged with a fuel tracer fluorescence diagnostic. In agreement with the chemiluminescence study, the fuel distribution showed that the region of highest fuel reactivity was located above the piston bowl-rim. This study expands our previous work [16] and explores fuel reactivity stratification as a method to control the rate of heat release for PCI combustion using a heavy-duty, single cylinder, optically accessible research engine using a combination of high-speed chemiluminescence imaging and fuel-tracer planar laser-induced fluorescence (PLIF).

EXPERIMENTAL SETUP

ENGINE SPECIFICATIONS

A single-cylinder, direct-injection (DI), 4-stroke diesel engine based on a Cummins N14 production engine was used for this investigation. A schematic of the engine and optical setups are shown in [Figure 1](#) and specifications are given in [Table 1](#). A complete description of the engine is available in Refs. [17, 18]. The research engine is typical of a heavy-duty diesel engine with a 13.97 cm bore and 15.24 cm stroke giving a displacement of 2.34 L per cylinder. The intake port geometry of the production engine, which has a steady-state (i.e., measured on a flow-bench) swirl ratio of 0.5 [19], is preserved in the research engine.

To allow optical access, the engine is equipped with an extended piston and the stock metal piston bowl has been replaced with a flat fused silica piston crown window. Further, one of the two exhaust valves has been replaced with a window, and a periscope mirror in the rocker box gives a view of the squish region (i.e., the region above the piston bowl-rim). Four windows are also located around the upper portion of the cylinder wall to allow cross-optical access for laser based diagnostics. Finally, to allow the laser sheet to penetrate into the combustion chamber when the piston was near TDC, a 37 mm wide section of the piston-bowl rim was removed as indicated in [Figure 2](#).

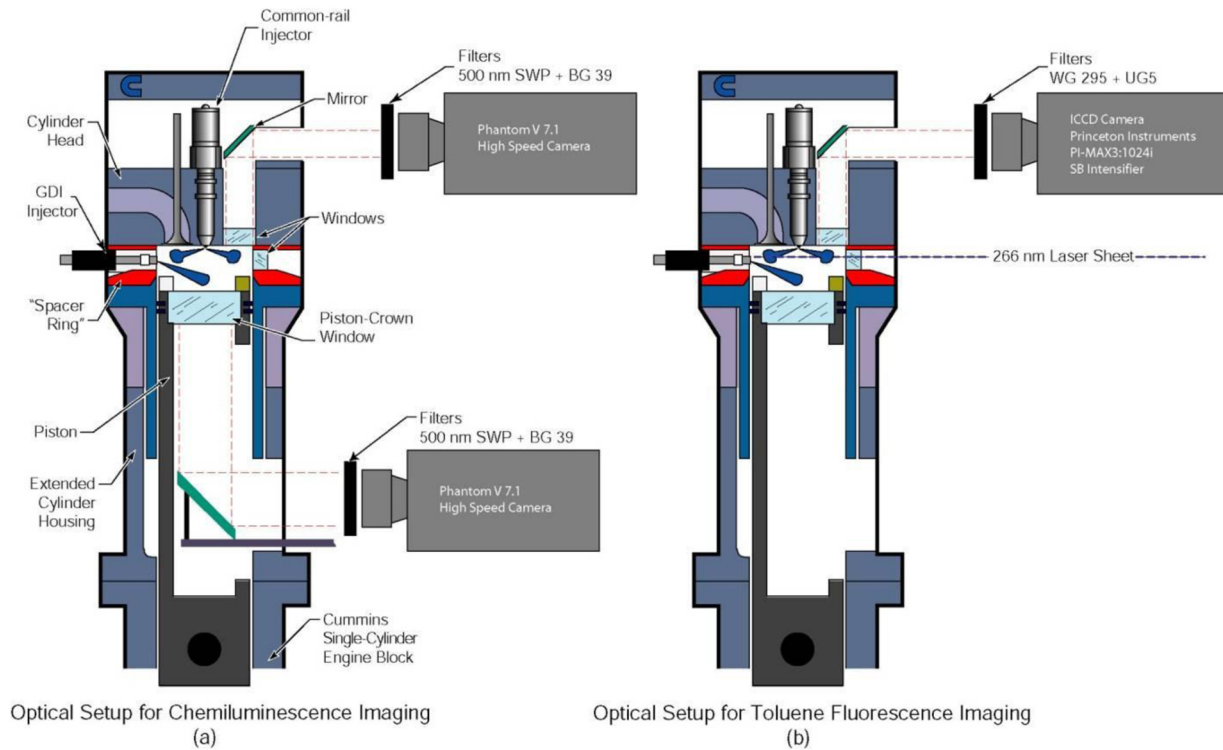


Figure 1. Schematic of the optically accessible research engine, (a) shows the camera setup for the high-speed chemiluminescence imaging study and (b) shows the camera and laser sheet setup for the fuel tracer PLIF study. Note that the actual GDI injector location is perpendicular to the laser sheet (see Figure 2).

Table 1. Engine 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 ¹
Combustion chamber	Quiescent, flat bowl
Swirl Ratio	0.5 (approximate)
Bore	13.97 cm [5.5 in]
Stroke	15.24 cm [6.0 in]
Bowl width	9.78 cm [3.58 in]
Displacement	2.34 liters [142 in ³]
Connecting rod length	30.48 cm [12.0 in]
Piston pin offset	None
Geometric compression ratio	10.75:1

¹In this optically accessible diesel engine, one of the two exhaust valves of the production cylinder head has been replaced by a window and periscope

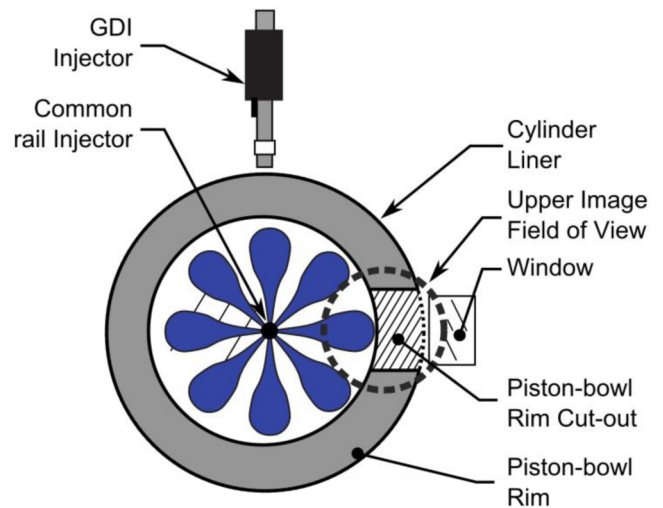


Figure 2. Top view of piston showing piston-bowl rim cut-out (hatched region) and the location of the cylinder head window field-of-view.

INJECTION SYSTEMS

RCCI combustion is achieved by using in-cylinder blending of two fuels with different auto-ignition characteristics, such as gasoline and diesel fuels. Multi-component gasoline and diesel fuels contain molecules that fluoresce when illuminated by ultraviolet light, which would interfere with the toluene fuel-tracer fluorescence diagnostic (described later). Hence, to facilitate the fuel-tracer measurements, purified gasoline primary reference fuel (PRF) components (i.e., n-heptane and iso-octane) were used because they display insignificant fluorescence interference. A charge of high-purity (99% or better) iso-octane was created using a gasoline direct injection (GDI) system. The GDI injector is mounted in a metal blank in place of one of the optical access windows formerly located in the cylinder liner (see Figure 1). The specifications of the GDI injector are provided in Table 2. The GDI has a 7-hole, asymmetric tip, with all seven holes located on one hemisphere of the injector. This hole layout prevented fuel impingement on the cylinder head when mounted in the cylinder wall window (i.e., perpendicular to the axis of the piston). HPLC-grade n-heptane was pressurized to 600 bar by a Newport Scientific high-pressure diaphragm pump and delivered through a Cummins XPI high-pressure, electronically-controlled, common-rail (CR) fuel injector with specifications given in Table 3. The injector is capable of multiple injections at up to 2000 bar injection pressure. For the present study, an 8-hole mini-sac tip was installed. Each hole of this tip, has a nominal orifice diameter of 140 microns with no hydrogrounding (i.e., the edges are sharp). The nozzle has an included angle of 152° (14° downward from the fire deck, i.e., the bottom surface of the cylinder head exposed to the combustion chamber). Finally, to minimize thermal loading, the engine was only fired 1 out of 10 cycles.

Table 2. Gasoline direct injector (GDI) specifications.

Injector type.....	Bosch GDI
Fuel	high-purity (99%+) iso-octane
Number of holes.....	7, asymmetric
Supply pressure	100 bar
Nominal orifice diameter	150 micron

Table 3. XPI common-rail (CR) fuel injector specs.

Injector type.....	common-rail, solenoid actuated
Fuel	HPLC-grade n-heptane
Cup (tip) type.....	mini-sac
Number of holes.....	8, equally spaced
Included angle.....	152°
Rail pressure	600 bar
Orifice treatment.....	none (square-edged)
Nominal orifice diameter	140 micron

OPERATING CONDITIONS

The engine operating conditions for the chemiluminescence imaging study are shown in Table 4. The engine was operated at a speed of 1200 rev/min and a gross indicated mean effective pressure (IMEP) of 4.2 bar. This light-load condition was chosen to allow exploration of a wide range of conditions without damaging the large piston crown window.

With the intent of creating a well-mixed charge, 41 mg of iso-octane was delivered by the GDI injector at -240° ATDC (during the intake stroke) with a 36° crank angle (CA) duration of injection (DOI) at an injection pressure of 100 bar. The equivalence ratio of the premixed fuel alone was 0.27. The n-heptane was delivered through the common-rail injector at a pressure of 600 bar. The common-rail injection pressure was selected to be similar to the authors' previous RCCI research [3]. A sweep of common-rail start of injection timings was used to generate a range of fuel distributions from well mixed (i.e., early common-rail injection) to stratified (i.e., late common-rail injection)¹. The total injected mass of n-heptane was 23 mg/cycle for all experiments in this work. The engine was operated without dilution (i.e., the inlet oxygen concentration was 21% by volume) and the intake temperature was adjusted at each injection timing to hold combustion phasing (CA50) fixed at 2° ATDC in order to allow direct comparisons of the combustion rate. This value of CA50 was selected to be comparable to previous RCCI research (e.g., Hanson et al. [12]) and was found to provide a good balance between combustion noise and combustion stability. The air flowrate was held constant at 23 g/s for each case, resulting in an intake pressure near 1.1 bar absolute and a relatively low TDC density of 11.1 kg/m³ for the low-load conditions of this study.

¹The purpose of this work is not to optimize the injection timing, but rather show the influence of fuel reactivity stratification on the heat-release rate. Liquid impingement on the cylinder liner was observed for injection timings earlier than -90° ATDC; however, since n-heptane was used as the common-rail fuel and the intake and coolant temperatures were set near the atmospheric boiling point of n-heptane, it is expected that any liquid impinging on the liner will quickly vaporize. The high-speed imaging showed very little evidence of the "pool fires" characteristic of liquid films. Of course, if diesel fuel were used as the common-rail fuel, it is expected that liquid impingement would be problematic; however, it is likely that, even using diesel fuel, a multiple injection strategy could be used to generate a range of fuel reactivity stratification [3]. Such an investigation, however, is beyond the scope of this study.

Table 4. Engine operating conditions for the chemiluminescence imaging study and reference operating condition for the toluene fuel tracer fluorescence study. The toluene fluorescence study operated under oxygen-free conditions by supplying the intake with 100% nitrogen.

Engine speed	1200 rpm
Gross IMEP	4.2 bar
Intake temperature	73°C to 100°C
Intake pressure	1.1 bar abs.
Intake flowrate	23.0 g/s
Inlet oxygen concentration	21 vol. %
GDI injection pressure	100 bar
CR injection pressure	600 bar
GDI SOI (command)	-240° ATDC
CR SOI (actual)	-155° to -15° ATDC
Actual GDI DOI	36° CA
Actual CR DOI	9° CA
Total injected fuel mass	64 mg
iso-octane mass (GDI)	64%
n-heptane mass (CR)	36%
Premixed (iso-octane) equivalence ratio	0.27
Overall equivalence ratio	0.42
Overall PRF	64
Nom. motored TDC density	11.1 kg/m ³

The toluene fuel-tracer fluorescence study was conducted at the same thermodynamic conditions as the chemiluminescence imaging study; however, to avoid complications from oxygen quenching of the toluene fluorescence, the intake was supplied with 100% nitrogen. Further, HPLC n-heptane doped with 1% toluene² was used in both the GDI and common-rail fuel systems for quantitative fuel distribution measurements. The same toluene-doped n-heptane was used in both fuel systems (non-combusting conditions) to avoid uncertainties in the toluene fluorescence yield in different solvents.

CONVENTIONAL DATA ACQUISITION

Cylinder pressure was measured with an AVL QC43D piezoelectric transducer at ¼° crank angle increments and the measurements were set to match (pegged to) the intake pressure near bottom dead center (BDC). The apparent heat-release rate (AHRR) was calculated from the measured pressure data using the standard first-law analysis (e.g., Heywood [20]) with constant gas properties. Prior to calculating the AHRR, the cylinder pressure data were 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.

COMBUSTION LUMINOSITY IMAGING

Crank-angle resolved combustion luminosity was recorded with an unintensified Phantom 7.1 complementary metal oxide semiconductor (CMOS) high-speed camera (HSC) using a 50-mm glass Nikkor lens with the aperture fully open (f/1.2). The unintensified CMOS camera is not sensitive in the UV range; accordingly, OH* does not contribute to the recorded combustion luminosity. Furthermore, a combination of a 500 nm short-wave pass (SWP) filter and a BG39 colored glass filter were used to reject long-wavelength (green through IR) soot luminosity (if present). The recorded luminosity is likely chemiluminescence from CH₂O*, HCO*, CH*, and CO₂* and broadband emission from the CO continuum [21, 22]. The HSC images were acquired with a resolution of 512 × 512 pixels with an imaging resolution of 3.9 pixel per mm, and the exposure time for each frame was 125 μs. The long exposure time was required to collect reasonable light levels with the non-intensified HSC used in this study. Although the long exposure time results in a small amount of smearing for cases showing rapid energy release, the smearing does not influence the conclusions of the study.

FUEL-TRACER FLUORESCENCE IMAGING

Quantitative fuel-vapor concentrations were measured using toluene fuel-tracer fluorescence. As previously discussed, the n-heptane used for the fuel-tracer fluorescence imaging study was doped with toluene at a concentration of 1% by volume. Previous work (e.g., Genzale et al. [19]) showed that this toluene concentration provided a good balance between signal strength and attenuation of the laser light. The toluene fluorescence was excited by the 266-nm output of a frequency-quadrupled Nd:YAG laser. The laser beam was formed into a thin sheet (less than 1 mm thick) using a combination of a negative cylindrical lens (f = -50 mm) and positive plano-spherical lens (f = 500 mm). The low-intensity tails on either end of the sheet were clipped to leave the center of the beam, with relatively uniform intensity. After being clipped, the sheet passing through the window was approximately 31 mm wide with a remaining energy of 26 mJ per pulse. All of the images presented in this study were acquired with the sheet positioned 13 mm below the firedeck³ (i.e., the flat bottom surface of the cylinder-head).

A blue-optimized, intensified CCD camera (Princeton Instruments PI-MAX3:1024i SB) with a 55 mm f/3.5 UV lens with the aperture fully open imaged the resulting fluorescence viewing downward through the cylinder head window (see Figure 1). The intensifier gate was set to the minimum

²To avoid minor changes in the fuel reactivity due to the addition of 1% toluene, only the non-reacting experiments used the toluene-doped fuel.

³At TDC the laser sheet is 8.1 mm above the bottom of the piston-bowl surface.

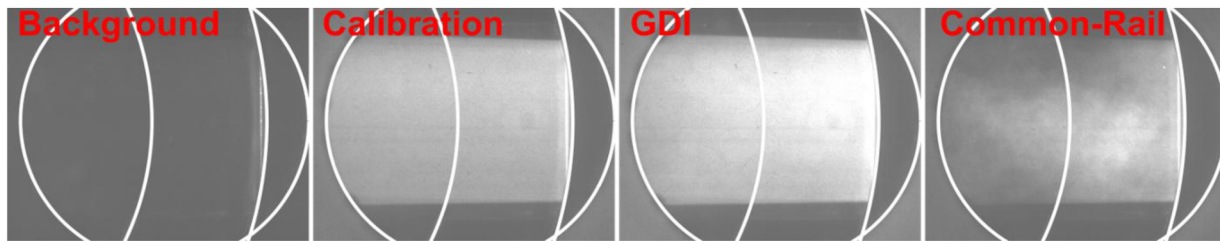


Figure 3. Typical image acquisition sequence for the toluene fuel-tracer PLIF diagnostic. The images were acquired at -5° ATDC for a case with a CR SOI timing of -50° ATDC.

practically achievable (approximately 200 ns). A WG295 long-wave pass (LWP) filter helped to isolate the toluene fluorescence (265-330 nm) from scattered laser light. Additionally, a UG5 UV-bandpass filter (passband in the range of 220 to 400 nm) blocked red-shifted fluorescence from motor oil and other sources [22].

Quantitative fuel-vapor concentration measurements require calibration of the fluorescence signals with a known fuel/tracer concentration. In this work, calibration images at a known fuel/tracer concentration were obtained by injecting toluene-doped n-heptane, using the common-rail injector, near TDC of the intake stroke. In similar studies (e.g., Musculus et al. [23]) early-cycle injections of low boiling-range fuels have been shown to yield acceptably homogenous mixtures during the compression stroke. Indeed, the calibration images from this study (e.g., Fig. 3) appear reasonably uniform. Although the calibration images appear nearly homogeneous, it is possible that non-uniformities exist outside of the camera field of view. These non-uniformities, if present, introduce uncertainty into the vapor fuel concentration measurements of this study, though it is not possible to quantify this level of uncertainty in the present work. Further, with the early-cycle injection used in this study, some liquid may impinge on the combustion chamber surfaces; however, the intake temperature and coolant temperature were set to near the atmospheric boiling point of n-heptane. Thus, it is likely that any liquid impinging on a combustion chamber surface would rapidly vaporize. However, should liquid fuel remain at the imaging time, the fuel vapor concentration measurements would be biased towards higher fuel concentrations.

Fuel-vapor concentrations were measured for separate injections from both the common-rail and GDI fuel systems. At each crank angle of interest, images were acquired in sets of four. Between each image, the engine was motored (without supplying fuel) for 9 cycles to scavenge residual fuel from the combustion chamber. The first image had no fuel injection and recorded background fluorescence interference - likely due to residual lubrication oil excited by scattered laser light [22, 23]. The second image provided a flat-field calibration at a known toluene/fuel concentration as previously discussed. The third and fourth images recorded fluorescence from the fuel injected through the GDI only and common-rail injector only, respectively. This four-image

sequence was repeated 40 times at each crank angle of interest. After the image acquisition sequence, the optics were cleaned and the procedure was repeated for the next operating condition. Figure 3 shows a representative image sequence.

To extract quantitative information using LIF, the photophysics of the LIF signal must be understood. The engine was operated using 100% N_2 for the fluorescence measurements to avoid the collisional quenching by oxygen, but two primary concerns remain: temperature dependence of the fluorescence quantum yield and absorption cross section, and potential pressure dependence due to collisional quenching with N_2 and fuel [24].

In a review paper, Schulz et al. [24] showed that, under 266 nm excitation, the absorption cross section increases by more than a factor of three when the temperature is increased from 300 to 1000 K. Further, the fluorescence quantum yield over the same temperature range decreases by two orders of magnitude. At the image acquisition time, the temperatures of the calibration and raw data images are slightly different due to differences in the fuel concentrations (i.e., differences in the evaporative cooling and in the compression heating). Therefore, it is important to correct the images for temperature prior to conversion to a fuel concentration. The temperature dependence of toluene fluorescence was evaluated under well-mixed conditions (i.e., common-rail fuel injection at -345° ATDC) over a range of temperatures. Figure 4 shows the relative toluene fluorescence yield evaluated over a range of in-cylinder temperatures from 550 K to 940 K during the compression stroke, corrected for changing charge density. Following the work of Musculus et al. [23], an offset exponential was fit to the data and used as a calibration curve.

Similar to the temperature dependence, the pressure dependence was evaluated over a range of conditions. Consistent with the findings of Musculus et al. [23], the toluene LIF signal was shown to be independent of pressure after temperature corrections were applied.

With the pressure and temperature dependence known, the images were processed with the iterative procedure outlined in Figure 5. Initially, the fuel distribution was calculated by assuming a uniform temperature distribution at the image acquisition time. The temperature fields of the calibration and data images (i.e., GDI or CR images) are different due to evaporative cooling and changes in the

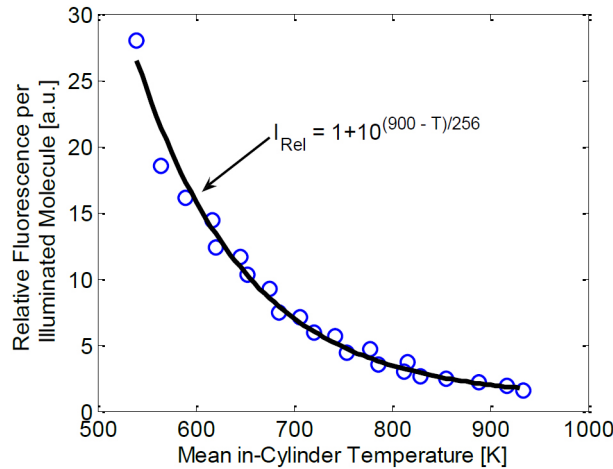


Figure 4. Relative toluene fluorescence as a function of mean in-cylinder temperature. The fueling was held constant at 39 mg/inj.

specific heats of the mixture due to the differences in fuel concentrations. To account for the temperature dependence of toluene fluorescence, the fluorescence intensity of each data image was corrected for the local temperature difference from the calibration image by

$$I_{corrected_pixel} = I_{data_raw_pixel} \frac{I_{rel} T_{cal}}{I_{rel} T_{data_pixel}} \quad (1)$$

where $I_{corrected_pixel}$ is the corrected fluorescence intensity of the data image, $I_{data_raw_pixel}$ is the local fluorescence intensity of the GDI or common-rail image, $I_{rel}(T_{cal})$ was calculated from the temperature calibration curve given in Fig. 4 at the average temperature of the calibration image, and $I_{rel}(T_{data_pixel})$ was calculated from the temperature calibration curve at the local temperature of the data image. Then, the fuel distribution was calculated by normalizing the corrected fluorescence intensity by the flat field calibration image as shown in Eq. 2

$$n_{data} = n_{calibration} \frac{I_{corrected_pixel}}{I_{calibration_pixel}} \quad (2)$$

In Eq. 2, n_{data} is the local fuel vapor molar density and $n_{calibration}$ is fuel vapor molar density of the calibration image.

The above procedure provides an initial estimate of the fuel distribution. The final fuel distribution was calculated iteratively as follows. First, the temperature of the N_2 charge at BDC was adjusted from the intake stream temperature to account for heat transfer during induction according to the procedure of Sjöberg et al. [25]. Second, the rise in temperature of the charge of N_2 due to compression from

BDC to the start of injection (SOI) timing was calculated by numerical integration assuming adiabatic compression according to the measured pressure rise. The compression calculation accounted for variable specific heats. In this way, the resulting temperature represents the adiabatic core of the charge. Third, the fuel was assumed to be injected and mixed at the SOI timing. The charge of N_2 and fuel was then compressed from the SOI timing to the image timing assuming that the fuel distribution remains frozen. Although the real fuel distribution is not actually frozen, this simplification in the calculation provides a method to account for the temperature distribution resulting from fuel vaporization and the different heat capacities of N_2 and n-heptane. Hwang et al. [22] showed the importance of considering the change in temperature due to evaporative cooling and differences in compression heating due to the different specific heats of different equivalence ratio zones. In their work, they showed that neglecting these temperature corrections caused the peak equivalence ratio to be over-predicted by 38%, thus limiting the usefulness of the diagnostic. This iterative procedure was looped until convergence was obtained in the fuel distribution.

To improve the signal-to-noise ratio (SNR) of the images, the fuel distributions of the GDI and common-rail injections were ensemble averaged over 40 images. The average fuel-distribution maps were generated by combining the GDI and common-rail fuel concentration measurements. Of particular interest to this study are the relative concentrations of fuel from the GDI and common-rail fuel streams. Thus, a PRF number is defined as

$$PRF = 100 \frac{m_{GDI} / \rho_{iso-octane}}{m_{GDI} / \rho_{iso-octane} + m_{CR} / \rho_{n-heptane}} \quad (3)$$

where the m_{GDI} is the measured vapor-fuel mass from the GDI images corrected for the difference in molecular weight