

Equivalence Ratio Distributions in a Light-Duty Diesel Engine Operating under Partially Premixed Conditions

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ABSTRACT

The performance of Partially Premixed Compression Ignition (PPCI) combustion relies heavily on the proper mixing between the injected fuel and the in-cylinder gas mixture. In fact, the mixture distribution has direct control over the engine-out emissions as well as the rate of heat release during combustion. The current study focuses on investigating the pre-combustion equivalence ratio distribution in a light-duty diesel engine operating at a low-load (3 bar IMEP), highly-dilute (10% O₂), slightly boosted (P_{in} = 1.5 bar) PPCI condition. A tracer-based planar laser-induced fluorescence (PLIF) technique was used to acquire two-dimensional equivalence ratio measurements in an optically-accessible diesel engine that has a production-like combustion chamber geometry including a reentrant piston bowl. Equivalence ratio distributions are presented at a single vertical plane and three different horizontal planes within the combustion chamber, including two planes within the bowl of the piston. The quantitative measurements provide a detailed picture of the mixture formation process for an early injection PPCI combustion regime. Based on the measured equivalence ratio distributions, the mass of engine-out UHC and CO were approximated and the results confirm that the main source of UHC and CO emissions is from over-mixed, overly-lean fuel/air mixtures. The measured equivalence ratio distributions were also combined with previous in-cylinder velocity and CO measurements to describe the mechanism by which these overly-lean mixtures are formed and how they are transported to the squish region late in the engine cycle.

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INTRODUCTION

Partially Premixed Compression Ignition (PPCI) and other low-temperature combustion (LTC) strategies are currently being pursued in an effort to reduce the need for costly exhaust aftertreatment systems on light-duty diesel engines. Diesel PPCI strategies achieve low-temperature operation by using relatively low global equivalence ratios, low compression ratios, and high levels of exhaust gas recirculation (EGR). The resulting lower in-cylinder gas temperatures and O₂ concentrations extend the ignition delay period, allowing an early or late injection of fuel to form a largely premixed, but stratified, distribution of equivalence ratios within the combustion chamber. This distribution of equivalence ratios is critical for tailoring the heat-release rate and is an important factor influencing the engine-out emissions. PPCI strategies have been shown to significantly reduce the regulated emissions of nitrogen oxides (NO_x) and particulate matter (PM), but the emissions of unburned hydrocarbons (UHC) and carbon monoxide (CO) have been observed to increase under these types of combustion regimes

[1, 2, 3, 4, 5, 6, 7]. The purpose of this study is to measure the in-cylinder equivalence ratio distribution in a light-duty diesel engine operating in a PPCI regime to investigate how the mixture formation process impacts the development of UHC and CO emissions.

The fundamental dependence of the fuel oxidation process on mixture stoichiometry was previously studied by Ekoto *et al.* [8,9] using homogeneous, constant volume CHEMKIN simulations. The results are repurposed here to highlight the importance of the in-cylinder equivalence ratio distribution on the emissions of UHC and CO at highly dilute conditions. The conditions used for the simulations are similar to the low-load (3 bar IMEP), highly-dilute (10% O₂), slightly boosted (P_{in} = 1.5 bar) PPCI condition that was investigated in the current study. Figure 1 shows that for a range of equivalence ratios spanning from 0.5 to 1.0, the UHC and CO emissions are negligible and complete combustion is achieved. The combustion process is kinetically-limited for overly-lean mixtures ($\phi < 0.5$) and transitions to being oxygen-limited for overly-rich mixtures ($\phi > 1.0$). Note that

the NO_x and PM levels are also negligible at these conditions.

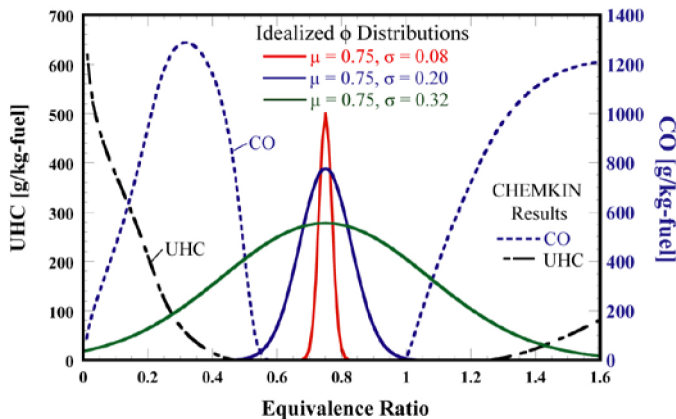


Figure 1. UHC and CO concentrations determined from homogeneous, constant volume CHEMKIN simulations under highly dilute conditions (10% O_2). Note that the idealized equivalence ratio distributions are not associated with either vertical axis and are only intended to represent the probability distributions of idealized fuel/air mixtures.

Using the simulation results shown in Fig. 1 as a general guide for engine combustion, one can assess the ability of different mixture formation strategies to prevent UHC and CO emissions. Several theoretical ϕ distributions were added to Fig. 1 to demonstrate the effect that mixture stratification has on the engine-out emissions. As the standard deviation of the ϕ distribution increases, i.e. as the level of fuel stratification is increased, the distribution widens and encompasses equivalence ratios that are unable to sustain complete combustion. As the standard deviation in ϕ decreases, the distribution narrows within the UHC oxidation limits ($0.5 < \phi < 1.0$) and the UHC and CO emissions are minimized. This simple analysis suggests that in order to avoid excessive UHC and CO emissions, it is advantageous for the mixture formation process to create at least a moderately homogeneous fuel/air mixture.

Unfortunately, other constraints, such as maximum pressure rise rate, place a limit on the level of homogeneity that can be achieved for operating conditions utilizing high levels of premixing. For truly homogeneous ϕ distributions, all of the fuel ignites simultaneously, creating very high pressure-rise rates. PPCI combustion strategies lengthen the high-temperature heat-release by producing a range of equivalence ratios. The stratified in-cylinder mixtures ignite in progression according to their local equivalence ratio, thereby providing more control over the heat-release than with LTC strategies that incorporate homogeneous mixtures. The pressure rise rate and combustion duration can be modified by adjusting the in-cylinder distribution of equivalence ratios. The need to produce an equivalence ratio distribution that meets the requirements for complete

combustion while maintaining acceptable pressure rise rates, places a high demand on the fuel injection system and has inspired many investigators to study a number of different injection strategies to improve PPCI combustion [10, 11, 12].

This study was designed to directly investigate the mixture formation process of a single, early injection, low-load PPCI operating condition. Tracer-based planar laser-induced fluorescence (PLIF) was used to acquire quantitative ϕ distributions at a single vertical (r - z) plane and three horizontal (r - θ) planes, including two within the piston-bowl of an optically accessible light-duty diesel engine. The quantitative ϕ measurements are combined with previous in-cylinder measurements of the velocity and CO fields to provide insights into where UHC and CO emissions are likely to be formed due to improper mixing. Overall, the measurements give a detailed description of the mixture formation process and provide a data set that is useful for validating engine combustion simulations.

EXPERIMENTS

ENGINE

The engine used in this study was modified from a 1.9 L GM light-duty diesel engine to provide extensive optical access to the combustion chamber. The main geometric specifications are provided in Table 1 and a schematic of the engine is shown in Fig. 1. The main feature of this engine is a Bowditch-style piston that has a fused silica piston-top that includes the re-entrant bowl and valve cutout geometries of typical production engines. Optical access to the side of the combustion chamber is achieved through fused silica windows (50 mm wide by 25 mm tall) located at the top of the cylinder liner. These windows are positioned to allow unobstructed optical access up to the bottom surface of the firedeck. The height and width of the top ring-land were increased slightly to allow imaging within the bowl portion of the piston. The resulting crevice volume is roughly 3-4 times larger than a production diesel engine. Two gapless compression rings were used to ensure blowby was minimized. Despite the larger crevice volume and increased wall temperatures of the fused silica surfaces, the optical engine has been shown to reproduce the emissions and combustion behavior of a similar all-metal engine [13].

The fuel injector used in this study was a Bosch CRI2.2 high pressure, electronically controlled, common-rail injector. The injector was equipped with a 7-hole, mini-sac injector tip with an included angle of 149° . The 7 fuel orifices are equally spaced and have a nominal diameter of 0.14 mm. The nozzles are specified to have 10% hydro-grinding. Additional specifications for the fuel injector are included in Table 1.

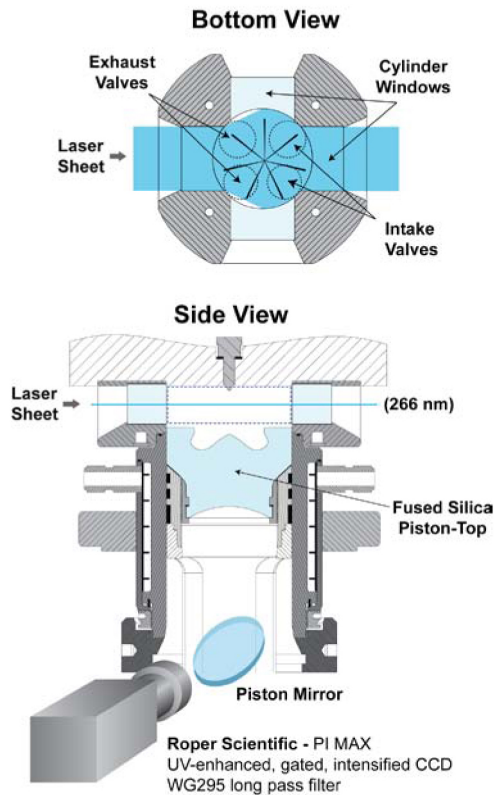


Figure 2. Schematic of the optically accessible diesel engine depicting the location of the laser sheet and camera viewing direction for the horizontal plane measurements. The laser sheet was rotated and the camera was moved towards the top of the cylinder for the vertical plane measurements.

Table 1. Engine and fuel injector specifications.

Engine Geometry	
Bore	82.0 mm
Stroke	90.4 mm
Displ. Volume	0.477 L
Geometric CR	16.7
Squish Height	0.88 mm
Injector Parameters	
Injector	Bosch CRI2.2
Sac Type	Mini Sac
Sac Volume	0.23 mm ³
Holes	7
Nozzle Diameter	0.14 mm
Included Angle	149°
Rail Pressure	860 bar

DIAGNOSTIC TECHNIQUE

A tracer-based PLIF technique was selected to acquire quantitative in-cylinder measurements of the fuel/air equivalence ratio. To avoid interferences by species typically found in diesel fuel, toluene was added (0.5% by weight) as a fluorescent tracer to a non-fluorescing mixture of n-heptane and iso-octane. A 75% by volume mixture of HPLC-grade n-heptane (C_7H_{16}) and 25% by volume iso-octane (C_8H_{18} , 2,2,4 trimethylpentane) was selected to match the heat-release of a 2007 certification diesel fuel with a cetane number of 47. Figure 3 shows that the characteristics of both the low- and high-temperature heat-release from the certification diesel fuel are well captured by the n-heptane/iso-octane blend, referred to hereafter as PRF25 [14,15]. Previous studies [14,15] have also shown that the engine-out emissions of UHC and CO at this condition are similar for both fuels and follow the same trends with respect to injection timing. This provides confidence that, although the fuel properties of the n-heptane/iso-octane mixture are different than typical diesel fuel, using the PRF25 fuel blend results in the same general combustion and emissions formation processes that would be observed with diesel fuel.

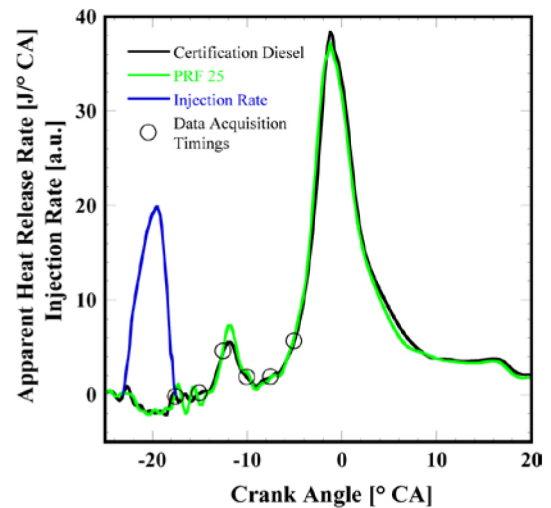


Figure 3. Comparison of the apparent heat release rate (AHRR) for a US 2007 certification diesel fuel and the PRF25 fuel mixture employed in the present study. The fuel injection rate was measured with diesel fuel. Open circles represent the crank angles at which equivalence ratio measurements were made.

The toluene fluorescence was excited by the 266-nm (UV) output of a 10-Hz pulsed Nd:YAG laser, with a pulse energy of 40 mJ. The shot-to-shot relative laser energy was measured by directing a portion of the excitation beam to a Coherent model J-25MB-LE energy sensor. The laser beam was focused at the center of the cylinder with a spherical lens ($f = 1000$ mm) and spread horizontally with a negative cylindrical lens ($f = -250$ mm). The resulting laser sheet had a thickness of approximately 500 μ m across the cylinder bore.

A UV-enhanced, gated, intensified charge-coupled device (ICCD) camera (Roper Scientific Pi-Max), equipped with a 105-mm focal length f/4.5 UV Nikkor lens, captured the toluene fluorescence, which occurs in the wavelength range of 260–400 nm [16]. The intensifier gate width was set to 250 ns and a fixed intensifier gain was used for all crank angles. A WG295 long-wave-pass filter was placed in front of the camera to reject elastically scattered 266-nm laser light.

The fluorescence data were converted to quantitative equivalence ratio values using a method similar to the process used in [17]. A comprehensive description of the data processing routine is provided in [18] and only a brief overview of the image processing is included here. A set of 100 images, acquired in 100 separate cycles, was collected for each plane at each measurement location. The raw fluorescence images were first corrected by subtracting an average background image acquired with the engine and laser running, but without any fluorescence tracer present. The resulting image was calibrated against an average flatfield image acquired with a homogeneous in-cylinder mixture of toluene at a known concentration. Pressure and temperature corrections were performed on the data to account for the differences in thermodynamic conditions between the raw data and the calibration images. Finally, the images were corrected for the optical distortion generated by the highly-curved surfaces of the optical piston. This was done by acquiring an image of a calibration target and mapping the coordinates of the grid points in the distorted target image to the corresponding undistorted target coordinates. Note that the final equivalence ratio values are ensemble averaged and were calculated based on a fuel/charge mixture containing 10% O₂.

OPERATING CONDITION

To avoid oxygen quenching of the toluene fluorescence, the engine was operated with a pure nitrogen intake stream and thus under non-combusting conditions. The intake temperature was adjusted in order to match the near-TDC temperature and density of a baseline operating condition (10% [O₂], 3.0 bar IMEP, and 1.5 bar intake pressure) that has been studied extensively [8,9,14,15,18,19]. This requires the intake temperature to be reduced in order to compensate for the lower molecular weight and lower heat capacity of the pure nitrogen intake flow. An engine model was developed using GT Power software to estimate the required intake conditions to match the baseline operation. The model included the heat transferred to the intake charge within the cylinder and intake runner system and accounted for the gas dynamics in the intake and exhaust manifolds. As shown in Table 2, the estimated motored TDC temperature and density of the baseline condition were 909 K and 20.9 kg/m³, respectively. To achieve this with a pure nitrogen intake flow, the intake temperature was maintained at 27°C at a pressure of 1.5 bar. Although the engine was operated under non-reacting conditions, the measured equivalence ratio

distributions are representative of fired engine operation until the onset of significant heat release - approximately 5° bTDC (see Fig. 3).

Table 2. Engine operating conditions.

	Baseline Condition	Current Condition
Engine Speed [RPM]	1500	1500
Intake Charge [mole fractions]	O ₂ : 0.10 N ₂ : 0.81 CO ₂ : 0.09	N ₂ : 1.0
Intake Temperature [K]	372	300
Intake Pressure [bar]	1.5	1.5
Load (IMEP) [bar]	3.0	N.A.
Injected Fuel [mg]	8.8	8.8
Rail Pressure [bar]	860	860
(Air/Fuel) _{stoich} [mol/mol]	113.4	113.4
Global Equivalence Ratio	0.3	0.3
SOI _{command} [°CA]	-26.6	-26.6
SOI _{actual} [°CA]	-23.1	-23.1
Inj. Duration (act.) [°CA]	5.4	5.4
Swirl Ratio	2.2	2.2
Motored TDC Density* [kg/m ³]	20.9	21.1
Motored TDC Temperature* [K]	909	908

**Predicted values based on measured intake pressures and temperatures*

A consistent thermal state between experimental runs was maintained by following a strict eight-minute measurement schedule. At the beginning of the schedule, the engine was motored for 90 seconds to stabilize the intake plenum pressure. Fuel was then injected once every fifth cycle for 90 seconds. The image acquisition began 15 seconds after the start of the injection sequence. After the images were collected, the engine was stopped and allowed to cool for the rest of the eight-minute schedule. During this period, any accumulation of fuel films or other deposits were cleaned from the optical and cylinder head surfaces before beginning the next run.

MEASUREMENT LOCATIONS

Images of the fuel tracer fluorescence were acquired from 17.5° to 5° bTDC in 2.5° increments at a single vertical (*r-z*) plane and three different horizontal (*r-θ*) planes within the combustion chamber. The location of each of the planes is shown in Fig 4a. This figure shows a side profile of half of the combustion chamber that corresponds to the vertical plane measured in this study and includes an illustration of a typical spray. The upper horizontal plane, referred to here as Plane 1, was located at the mid-plane of the squish volume for each crank angle. Therefore, the location of Plane 1 moves upward as the piston travels toward TDC. Plane 2 is located at the rim