

An Experimental Investigation of Low-Soot and Soot-Free Combustion Strategies in a Heavy-Duty, Single-Cylinder, Direct-Injection, Optical Diesel Engine

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ABSTRACT

High-efficiency, clean-combustion strategies for heavy-duty diesel engines are critical for meeting stringent emissions regulations and reducing the costs of aftertreatment systems that are currently required to meet these regulations. Results from previous constant-volume combustion-vessel experiments using a single jet of fuel under quiescent conditions have shown that mixing-controlled soot-free combustion (*i.e.*, combustion where soot is not produced) is possible with #2 diesel fuel. These experiments employed small injector-orifice diameters ($< 150 \mu\text{m}$) and high fuel-injection pressures ($> 200 \text{ MPa}$) at top-dead-center (TDC) temperatures and densities that could be achievable in modern heavy-duty diesel engines. The present study examines the feasibility of mixing-controlled low-soot and soot-free combustion strategies using multi-hole injector tips in a 1.72-liter single-cylinder version of a heavy-duty diesel engine that has been modified to provide extensive optical access to the combustion chamber. Indicated-specific emissions, high-speed images of in-cylinder natural luminosity and OH^* -radical chemiluminescence, and indicated efficiency measurements were acquired over parametric sweeps of injection pressure, number of injector-tip orifices, quantity of injected fuel, start of combustion, TDC density, TDC temperature, and mixture composition. These measurements indicate that sustaining soot-free combustion over the entire duration of combustion is limited by jet-jet interactions and the re-entrainment of hot combustion products into each fuel jet near its flame lift-off length. Nevertheless, significant reductions in engine-out smoke levels are possible even if soot-free combustion is not sustained.

INTRODUCTION

To meet present and future regulations, diesel-engine manufacturers are currently relying on increasingly

complex exhaust-gas aftertreatment strategies to lower the engine-out emissions of nitrogen oxides (NO_x), hydrocarbons (HC), carbon monoxide (CO), and particulate matter (PM). Aftertreatment is necessary because conventional diesel-combustion strategies have been incapable of simultaneously lowering all of these emissions to levels compliant with continually tightening regulations. The current outlook for heavy-duty diesel aftertreatment includes a diesel particulate filter (DPF) to capture PM, a selective catalytic reduction (SCR) or lean- NO_x trap (LNT) system to control NO_x , and a diesel oxidation catalyst (DOC) to oxidize HC and CO emissions [1]. The potential for cost, weight, package-volume, complexity, and fuel-consumption penalties associated with these aftertreatment devices makes them an unwanted necessity for many engine manufacturers and operators [2].

The potential for advanced-combustion strategies to avoid the formation of regulated emissions is attractive because it could make one or more of the above aftertreatment devices unnecessary. Soot-free leaner lifted-flame combustion (LLFC) is one such advanced-combustion strategy.

LLFC is defined as mixing-controlled combustion that does not produce soot because it occurs at equivalence ratios less than or equal to approximately 2.

Numerous researchers have shown that this strategy is achievable when enough charge-gas is entrained into a diesel jet to produce equivalence ratios ≤ 2 at the downstream location where high-temperature ignition occurs. Work by Pickett *et al.* [3,4] has shown that LLFC can be sustained with diesel fuel in a constant-volume combustion vessel (CVCV) at ambient temperatures and densities that fall below and to the left of the lines shown in Fig. 1, where each line corresponds to a different orifice diameter (d_o) [4]. As shown in Fig. 1, decreasing d_o expands the soot-free region to include higher ambient temperatures and densities [4,5].

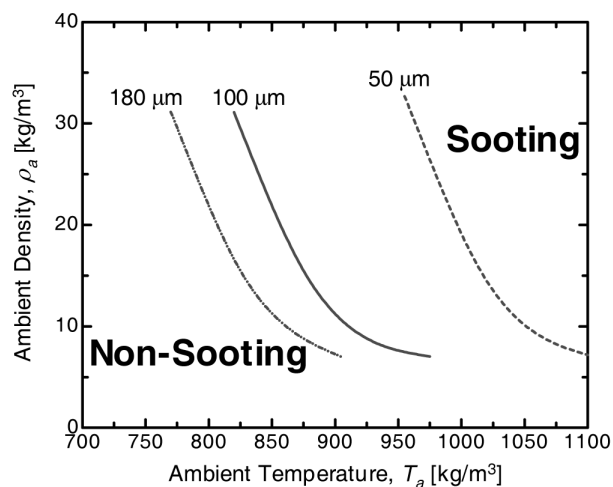


Figure 1. Non-sooting and sooting combustion regimes as a function of orifice diameter for a single, free jet of diesel fuel in a constant-volume combustion vessel [4].

The flame lift-off length (H) is defined as the axial distance between the fuel-injector orifice exit and the position where the standing premixed autoignition zone stabilizes during mixing-controlled combustion [4,6,7]. Work by Siebers and coworkers has shown that H for a single, free jet produced in a CVCV is described by Eq. 1 [7]:

$$H = C \cdot T_a^{-3.74} \cdot \rho_a^{-0.85} \cdot d_o^{0.34} \cdot U^1 \cdot Z_{st}^{-1} \quad (1)$$

where C is a constant, T_a [K] is the ambient-gas temperature, ρ_a [kg/m³] is the ambient-gas density, d_o [μm] is the orifice diameter, U [m/s] is the injection velocity, and Z_{st} is the stoichiometric mixture fraction.

As discussed in [8], either decreasing d_o or increasing the injection pressure, P_{inj} , can facilitate LLFC by increasing the extent of mixing upstream of H and thereby lowering the equivalence ratio at H , denoted $\phi(H)$. Because reducing d_o has a stronger effect on $\phi(H)$ than raising P_{inj} [8], it has been a primary tactic used in attempts to achieve mixing-controlled low-soot combustion. In heavy-duty engines, peak load capability is critical, so if d_o is reduced, other changes such as increasing the number of orifices and/or increasing P_{inj} must be implemented to compensate for the decrease in injector flow area.

Multiple researchers have attempted to achieve LLFC in both all-metal and optically accessible engines, with varying degrees of success [9-13]. In most cases, decreasing d_o and increasing P_{inj} leads to significant reductions in engine-out smoke or soot [9,13]. Others have observed a negligible impact on soot or have found that additional parameters (oxygen concentration, number of orifices, combustion phasing) strongly affect engine-out soot emissions [10,12,14]. Chartier *et al.* [9] achieved zero engine-out smoke in a heavy-duty engine with a 3-hole injector tip with 100-μm d_o , even though these conditions produced measureable amounts of natural luminosity from hot soot. Aronsson *et al.* [15] used the same setup to measure $\phi(H)$ using Raman spectroscopy at 8° and 12° after TDC to determine whether it was correlated with the observed PM

emissions. The results did not show a statistically significant correlation. Nevertheless, H and $\phi(H)$ measurements were limited to a single fuel jet at two crank angles. Detailed measurements of H and $\phi(H)$ over the entire mixing-controlled burn (MCB) phase of the combustion event are rare, and the relationship of H and $\phi(H)$ to engine-out emissions is not yet fully understood.

The charge temperature, pressure, and density within an engine can change due to piston motion and combustion during the compression and expansion strokes. This is in contrast to the relatively steady-state ambient conditions experienced in CVCV experiments. If the response of H to different ambient thermodynamic conditions observed in CVCV experiments with a single, free jet is directly applicable to the changing in-cylinder environment of an engine with multiple injector orifices, then H should change in a manner consistent with Eq. 1 as the in-cylinder temperature and density change. All of the other parameters in Eq. 1 (e.g., orifice diameter, injection velocity, and mixture fraction) remain constant or have a negligible impact on H over the MCB. In-cylinder measurements of H from [9,15-17] in optically accessible diesel engines have shown that H indeed changes throughout the MCB. The present study examines in detail how changes in the ambient in-cylinder conditions due to combustion and piston motion, coupled with changes in the number of orifices and P_{inj} , affect H and the propensity of lifted diesel flames to produce soot.

EXPERIMENTAL SETUP

ENGINE AND FUEL SYSTEM

The Sandia compression-ignition optical research engine (SCORE) is a single-cylinder version of a Cat® heavy-duty diesel engine that has been modified to provide extensive optical access to the combustion chamber. A schematic of the SCORE showing the optical access is provided in Fig. 2, and Table 1 contains the SCORE specifications. As shown in Fig. 2, the bowl in the SCORE piston has vertical side walls and a flat bottom to minimize optical distortions in imaging experiments.

As discussed in [18], the SCORE is skip-fired to reduce the risk of window failure due to thermal and mechanical stresses generated by combustion as well as to reduce the rate of window fouling. Skip-firing removes nearly all of the trapped exhaust-gas residuals from the previous fired cycle and dilutes the fired-cycle exhaust gases with fresh intake mixture (i.e., air, N₂, and/or CO₂) from motored cycles. Moreover, skip-firing reduces the heat rejection from the combustion chamber to the coolant and requires the SCORE to use a heater with closed-loop control to maintain a coolant temperature of 80°C. For an engine speed of 1500 rpm, the fuel injector typically fires once every 15 cycles in skip-fired mode. As described below, this firing frequency was increased to once every 5 cycles for the injection-pressure, TDC-temperature, and EGR studies with the 2-hole injector tip in order to increase the concentration of soot in the exhaust-gas samples acquired by the smoke meter.

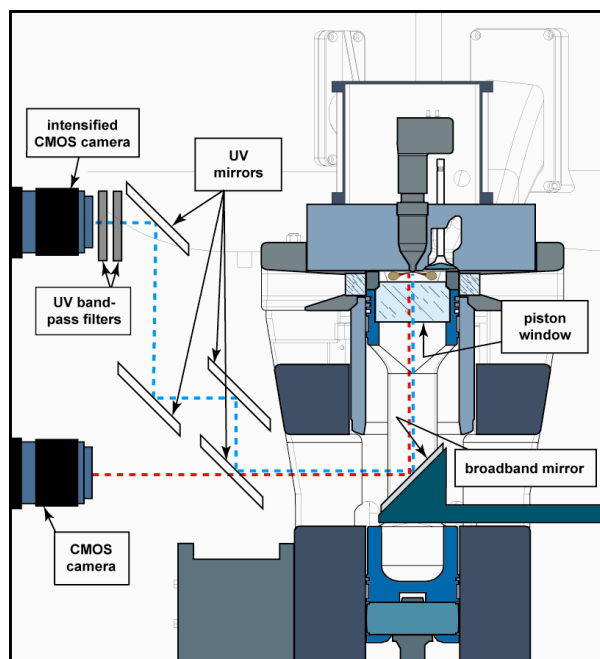


Figure 2. Schematic of SCORE showing how in-cylinder processes are imaged through the piston window in this work.

Table 1. SCORE Specifications

Research engine type	Single-cylinder
Cycle	4-stroke CIDI
Valves per cylinder	4
Bore	125 mm
Stroke	140 mm
Intake valve open ^a	32° BTDC exhaust
Intake valve close ^a	153° BTDC comp.
Exhaust valve open ^a	116° ATDC comp.
Exhaust valve close ^a	11° ATDC exhaust
Connecting-rod length	225 mm
Connecting-rod offset	None
Piston-bowl diameter	90 mm
Piston-bowl depth	16.4 mm
Squish height	1.5 mm
Swirl ratio ^b	0.59
Displacement per cyl.	1.72 liters
Geom. compression ratio	12.5:1

^aAll valve timings are for 0.03 mm lift

^bMeasured at the Caterpillar Technical Center using an AVL swirl meter

High-Pressure Common-Rail Fuel-Injection System – The high-pressure common-rail fuel-injection system consists of a diaphragm pump (manufactured by Newport Scientific Inc., Jessup, MD, USA, Model 46-16060), and an electronically actuated solenoid-operated common-rail fuel injector. The fuel-wetted components of the diaphragm pump are stainless steel, Kalrez, and Teflon, so the pump is compatible with a wide range of conventional and unconventional fuels. The reported fuel-injection pressure was

measured downstream of the pump, after which the fuel flowed through a 0.5- μ m sintered-metal filter, a 250-mL accumulator, and an ~350-mm long by 2.1-mm diameter jumper tube to the injector.

Properties of the common-rail fuel injector and the 2-, 6-, and 10-hole injector tips used in this work are provided in Table 2. Each injector tip was characterized using x-ray tomography to confirm the values in Table 2. As shown in Fig. 3, the injector as configured with the 6-hole tip produced a “top hat” injection profile, and the jet-to-jet variation in injection rate was less than 2% during the steady-state portion of the injection. The injection rate was quantified using the spray-impingement method described in [19]. While not shown, the injection-rate profiles from the 2- and 10-hole tips were nearly identical to those with the 6-hole tip, and the jet-to-jet variations in injection rate were less than 3% during the steady-state portions of the injections.

Fuel – A 2007 ultra-low-sulfur #2 diesel certification fuel (from Chevron-Phillips Chemical Company, The Woodlands, TX, USA) was used for all of the experiments described in this work. Selected fuel properties are provided in Table 3.

DIAGNOSTICS

Cylinder Pressure – Cylinder-pressure data were acquired every 0.5 CAD using a flush-mounted, water-cooled pressure transducer (made by AVL List GmbH, Graz, Austria, Model QC32C). The data were conditioned (absolute referencing, filtering, and

Table 2. Common-Rail Fuel-Injector Specifications

Injector type	Cat CR350
Injector tip style	Mini-sac
Injector tips	2-, 6-, and 10-hole
Orifice diameter (nom.)	110 μ m
Orifice L/D	8
Hydro-erosion	20 $\pm$ 2%
Included spray angle	140°

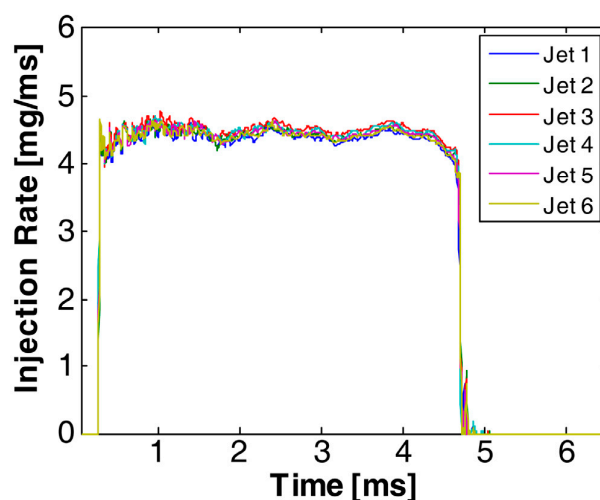


Figure 3. 50-injection-average injection-rate profiles for 6-hole tip at an injection pressure of ~24 MPa and an indicated duration of injection (DOI) of 4040 μ s.

Table 3. Ultra-Low-Sulfur #2 Diesel Fuel Specifications

Parameter	ASTM Test Method	Value
Specific Gravity	D 4052	0.8480
Cetane Number (CN)	D 613	43.3
Derived CN	D 6890	43.1
Distillation	D 86	
Initial		178°C
10%		211°C
20%		226°C
30%		235°C
40%		243°C
50%		253°C
60%		261°C
70%		272°C
80%		286°C
90%		310°C
End		342°C
Composition	D 1319	
Aromatics		25.3 vol%
Olefins		4.1 vol%
Saturates		70.6 vol%
Aromatics (by SFC)	D 5186	
1-ring		20.7 wt%
2 or more rings		9.0 wt%
Total		29.7 wt%
Sulfur	D 5453	9.2 mg/kg (typ.)
Hydrogen	D 5291	13.03 wt%
Carbon	D 5291	87.04 wt%
Kin. Viscosity (40°C)	D 445	2.3 mm ² /s (typ.)
Net Heat of Combust.	D 3338	42.9 MJ/kg (typ.)

averaging) using the procedure described in [20]. Apparent heat release rate (AHRR) and gross indicated mean effective pressure (gIMEP) were derived from cylinder-pressure data [20,21] acquired and averaged over 60 or more fired-engine cycles. Gross indicated fuel-conversion efficiency ($\eta_{f,ig}$) was determined using the procedure described in [20].

In-Cylinder Temperature – The majority of previous work on LLFC has been conducted in CVCVs. These vessels typically utilize mixing fans to create a spatially uniform ambient-temperature field into which fuel is injected [6]. In contrast, the temperature field within an engine is stratified due to turbulent charge motion and heat transfer between the charge-gas, piston, cylinder liner, and cylinder head [22]. The spatial and temporal temperature variations throughout the ambient in-cylinder gases in an engine make it difficult to select a gas temperature that is analogous to the ambient-gas temperature used in CVCV experiments. Since measuring the spatially and temporally resolved distribution of temperature throughout the trapped charge was beyond the scope of this study, spatially averaged measurements of temperature based on cylinder-pressure data are used in the present work.

The bulk-gas-average trapped-charge temperature (T_{bulk}) was derived from measured cylinder-pressure data using the ideal gas relation, slider-crank relation, and known charge mass [21]. While T_{bulk} is a crank-angle-resolved measurement, the extent to which T_{bulk} is quantitatively representative of the temperature of the charge gases entrained by the fuel jets at any given crank angle is unknown.

An alternative mixture temperature based on an assumption of isentropic compression of gases within the combustion chamber was derived from the measured cylinder pressure and is denoted the isentropic core temperature (T_{core}) [16]. T_{core} is representative of regions in the charge mixture that: (1) have not lost or gained energy from heat transfer, and (2) have yet to mix with fuel or combustion products. T_{core} should be more representative of the temperature of the gases that are entrained by the fuel jets near the center of the combustion chamber and far from the reaction zones or boundaries of the combustion chamber where the highest rates of heat transfer occur. Nevertheless, as is the case with T_{bulk} , the extent to which T_{core} is quantitatively representative of the temperature of the charge gases entrained by the fuel jets at any given crank angle is unknown.

Start of Combustion – A common definition for the start of combustion (SOC) is the crank angle where the AHRR first becomes positive after the actual start of injection (*i.e.*, after fuel begins to enter the combustion chamber) [20]. Under conventional diesel operating conditions, this method provides a repeatable estimate of the start of high-temperature heat-release (HTHR); however, some of the conditions used in this work produced measureable amounts of low-temperature heat-release (LTHR) prior to the HTHR. As a result, some of the SOC measurements correspond to the start of LTHR rather than the start of HTHR, as shown in Fig. 4. Over the course of a 60-injection experiment (*i.e.*, 60 fired cycles), a bi-modal distribution of SOC's with a spread of 2-5 CAD was sometimes obtained, which made it difficult to maintain consistent combustion phasing between operating conditions. To address this problem, a new metric, SOC_{fit}, was defined to quantify the start of HTHR. As shown in Fig. 4, SOC_{fit} is defined as the zero crossing of a line that intersects the points where the AHRR trace first attains 20% and 60% of its peak value.

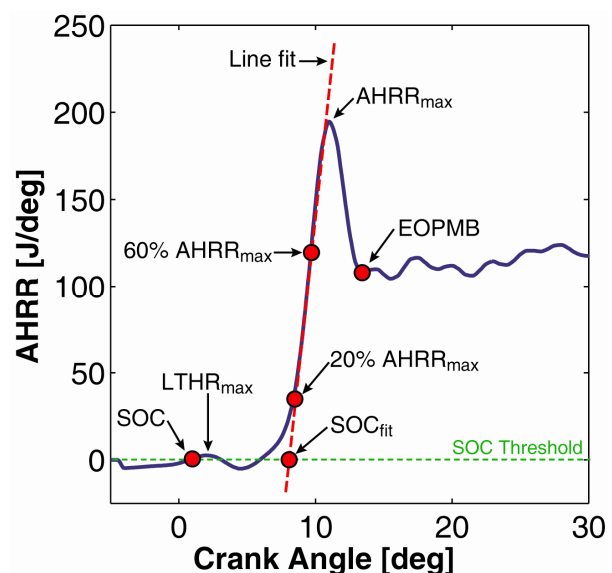


Figure 4. AHRR as a function of CAD for 6-hole tip to illustrate the measurements of SOC and SOC_{fit} from AHRR.

Engine-Out Emissions – The exhaust gases produced by skip-firing the SCORE were sampled downstream of a 125-liter exhaust-gas surge-tank as described in [18]. An AVL Model 415S smoke-meter measured engine-out smoke, and a heated chemiluminescence detector (made by California Analytical Instruments (CAI), Orange, CA, Model 600 HCLD) measured the mole fraction of nitrogen oxides (NO_x) in the exhaust-gas stream. A heated flame-ionization detector (CAI Model 600 HFID) measured the mole fraction of hydrocarbons (HC), and a non-dispersive infra-red absorption and paramagnetic detector (CAI Model 602-P NDIR) measured the mole fractions of CO , CO_2 , and O_2 . Procedures described in [20] were used to convert the exhaust-gas measurements from the skip-fired experiments to values that would be measured if the engine were operated in a continuous-fired mode.

The SCORE was fired once every 15 engine cycles for 72 seconds in all of the 6- and 10-hole experiments, as well as in the 2-hole SOC and duration-of-injection (DOI) sweeps. The firing frequency was increased to once every 5 engine cycles for 72 s for the other 2-hole experiments. This was necessary because the smoke-meter paper-blackening measurements for these operating conditions were below the stated detection limit of the instrument. Increasing the firing frequency increased the concentration of soot in the exhaust-gas stream to levels that were measurable by the smoke-meter.

Natural Luminosity – A high-speed CMOS camera (Vision Research, Wayne, NJ, USA, Model Phantom v7.3) fitted with a 35-mm lens (Nikon Corp., Tokyo, Japan) captured the natural luminosity (NL) every 0.5 CAD during fired operation. The NL signal included wavelengths in the range of 380-1000 nm, and is primarily a measure of incandescence from hot soot [23]. As shown in Fig. 2, the camera was set up to view the combustion chamber through the bottom of the fused-silica piston window via a 45° mirror. Each NL image is 208 pixels x 192 pixels, which corresponds to a field of view of 104 mm x 96 mm at a spatial resolution of 0.5 mm/pixel.

The spatially integrated natural luminosity (SINL) was measured from each NL image using Eq. 2 after the intensities of all pixels located outside of the piston bowl were zeroed.

$$\text{SINL}(\text{CAD}) = \left[\sum_{y=1}^m \sum_{x=1}^n \frac{f_s^2 \cdot \text{NL}(x, y, \text{CAD})}{t_{\text{exp}}} \right] / \text{SINL}_{\text{max}} \quad (2)$$

In Eq. 2, n and m are the number of pixels in the x - and y -directions, respectively; f_s is the lens f-stop (e.g., 16); and t_{exp} is the image exposure duration. SINL_{max} is the signal that would be measured if all pixels within the piston bowl were saturated when the lens aperture and exposure time were set to their minimum values (i.e., $f_s = 16$ and $t_{\text{exp}} = 1.0 \mu\text{s}$). The maximum extent of saturation allowed in the NL images was 10% of the total pixels in the image.

As discussed in detail in [20] and [23], the magnitude of the SINL signal is typically a complicated function of the concentration, temperature, and properties of

the soot particles inside the combustion chamber. Nevertheless, because the NL signal produced by hot soot is 3-5 orders of magnitude stronger than the NL from chemiluminescence [24], the absence of significant NL coupled with the presence of substantial AHRR indicates soot-free combustion. As a result, the SINL measurements in this work are used to distinguish between sooting and soot-free combustion.

OH* Chemiluminescence – A Phantom v7.1 intensified CMOS camera equipped with a 105-mm ultra-violet (UV) lens (Nikon Corp., Tokyo, Japan) captured the chemiluminescence (CL) of electronically excited hydroxyl radicals (OH^*) at $308 \pm 10 \text{ nm}$ during fired operation of the SCORE. The repetition rate for the intensified CMOS camera was one image/CAD at an image size of 384 pixels x 384 pixels. The image size corresponds to a 96 mm x 96 mm field of view and a spatial resolution of 0.25 mm/pixel. As shown in Fig. 2, light exiting the combustion chamber via the piston window was reflected from the piston mirror into a system containing four 308-nm high-reflector mirrors that directed the CL into the intensified CMOS camera. As the 308-nm light was reflected by each UV mirror, visible and infra-red (IR) wavelengths were mostly transmitted rather than reflected, resulting in the strong attenuation of those wavelengths. Prior to entering the intensified CMOS camera, the remaining light was directed through a 3-mm-thick piece of UG 11 glass and a 310-nm optical band-pass filter to further suppress visible and IR wavelengths.

Flame Lift-Off Length – Previous work by Higgins *et al.* [6] used the average OH^* CL at 310 nm along the axis of a reacting diesel-fuel jet to determine H . Higgins identified H as the position along the jet axis where the CL intensity rapidly increases by an order of magnitude. Mueller *et al.* [20] employed a similar approach for measuring H -values produced by a multi-hole fuel-injector tip using images of OH^* CL at 308 nm.

In the current study, the following procedure (based on that employed in [20]) was used to measure H for each fuel jet from CL images such as that shown in Fig. 5:

1. Isolate pixels within ~2 mm of the jet axis from the rest of the CL image pixels.
2. Accumulate pixel intensities perpendicular to the jet axis to create a CL intensity profile proportional to the average CL along the jet axis.
3. Attenuate atypical features in the CL intensity profile that are not indicative of H using a 10th-order quadratic filter.
4. Find the position along the jet axis where the gradient of the base-10 logarithm of the CL intensity profile reaches its maximum value. This distance is converted from units of pixels to mm using a scale factor based on the spatial resolution of the intensified CMOS camera (i.e., 0.25 mm/pixel). This measurement (H_{2d}) is a two-dimensional representation of H .
5. Use Eq. 3 to apply correction factors to H_{2d} for the included spray angle (θ) and radial offset of the orifice from the center of the injector tip (r_{inj}). The result is H .



$$H = \frac{H_{2d} - r_{inj}}{\sin(\theta_i/2)} \quad (3)$$

As shown in Fig. 5, the measured H -values (denoted by yellow lines) correspond to the upstream edges of the visibly significant regions of CL. The method described above was used to measure H for each jet between the end of premixed burn (EOPMB) and the end of injection (EOI) for all experiments. EOPMB is defined as the crank angle corresponding to the first local minimum in the apparent heat-release rate after the peak associated with the premixed burn (see Fig. 4). EOI is defined as the crank angle at which fuel stops entering the combustion chamber. Measurements of H outside of the range from EOPMB to EOI were possible, but the physical relevance of these measurements is less clear due to the transient nature of the combustion during these periods.

Equivalence Ratio at H – The model shown in Eq. 4 was developed by Naber and Siebers [25] to quantify the cross-sectional average equivalence ratio along the axis of a non-reacting, isothermal jet.

$$\phi(x) = \frac{2(A/F)_{st}}{\sqrt{1 + 16 \cdot \left(\frac{x}{x^+}\right)^2} - 1} \quad (4a)$$

$$x^+ = \sqrt{\frac{\rho_F}{\rho_A}} \frac{d_o \sqrt{C_a}}{c \left[\left(\frac{\rho_A}{\rho_F} \right)^{0.19} - 0.0043 \sqrt{\frac{\rho_F}{\rho_A}} \right]} \quad (4b)$$

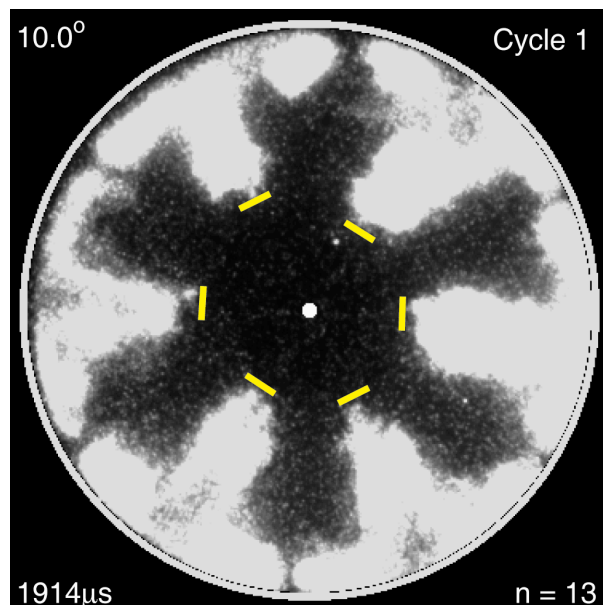


Figure 5. Single-cycle CL image for 6-hole injector tip. Each yellow line represents a measured H . Piston bowl-rim is denoted by white circular border, and injector tip is denoted by white dot at center of image. CL image was acquired using an exposure duration of 15 μ s at f/4.5 with an intensifier gain of 50,000. Number in top-left corner of image is CAD after TDC when image was taken. Also shown are: time after actual start of injection (bottom-left), image number (bottom-right), and fired cycle number (top-right).

In Eq. 4, $(A/F)_{st}$ is the stoichiometric mass ratio of ambient-gas to fuel, x^+ is a characteristic length scale, ρ_F is the fuel density, ρ_A is the ambient-gas density, d_o is the injector-tip orifice diameter, and values for the coefficients c and C_a were taken from [26]. Recent work by Pickett and co-workers [27] has shown that the equivalence ratio calculated using Eq. 4 is actually more representative of the jet-centerline equivalence ratio than the cross-sectional average equivalence ratio.

In this work, the $\phi(x)$ notation is used to represent an equivalence ratio estimated using Eq. 4. These estimates use measured values of H and values of bulk-gas-average density (ρ) calculated based on trapped-charge mass and the slider-crank relation for the SCORE. This approach implicitly assumes that $\phi(H)$ changes on the same time scale as ρ and H , where ρ and H are used in place of ρ_A and x , respectively, in Eq. 4. The use of Eq. 4 in this work also implicitly assumes that the ϕ -field for a multi-hole injector tip is the same as that for the single, free jet configuration used to develop the equation. The assumption that Eq. 4 can be used to accurately quantify $\phi(H)$ on a CAD basis is supported by SINL data (Fig. 7) showing that NL only appears in significant non-zero quantities when $\phi(H)$ exceeds a value of 2, a result that matches observations made by Pickett *et al.* [4] in the CVCV where Eq. 4 was developed.

OPERATING CONDITIONS

The experimental test matrix shown in Table 4 was designed to elucidate the effects that various engine operating parameters have on achieving LLFC. The test matrix includes varying: SOC timing; injected fuel mass (m_f); injection pressure (P_{inj}); bulk-gas-average charge-gas temperature at TDC under motored conditions (T_{TDC}); bulk-gas-average charge-gas density at TDC under motored conditions (ρ_{TDC}) coupled with the trapped-charge O_2 mole fraction (X_{O_2}) or global in-cylinder equivalence ratio (ϕ_g); simulated EGR level; specific heat at TDC under motored conditions (C_p); and the number of injector orifices.

As noted in Table 4, for some of the sweeps a secondary variable changed in addition the primary variable of interest. Where this occurred, every effort was made to minimize and/or isolate the effects of the secondary-variable changes. For instance, two density sweeps were conducted, one at constant X_{O_2} and one at constant ϕ_g , to better separate the secondary-variable effects.

Each operating condition was repeated at least 3 times to ensure a good statistical sample. The in-cylinder characteristics of each operating condition were based on CVCV experiments performed by Pickett and Siebers [4] that yielded soot-free combustion. The operating conditions were further constrained to those that are realistically achievable in modern heavy-duty diesel engines. The baseline operating condition with the 6-hole injector tip was designed to represent a mid-speed, 50%-load point of a metal engine with a displacement per cylinder comparable to that of the SCORE. EGR effects were simulated by adding N_2 and CO_2 to the dry intake air

Table 4. Experimental Test Matrix

	# of Inj. Tip Orifices	SOC [°ATDC]	T_{TDC} [K]	P_{inj} [MPa]	ρ_{TDC} [kg/m ³]	m_f [mg/inj]	ϕ_g [-]	X_{O_2} [mol %]	X_{CO_2} [mol %]	$C_{p(TDC)}$ [J/mol-K]
Baseline	2	0	765	240	35.0	36.5	0.115	19.1	1.54	31.8
	6					114.0	0.341			
	10					114.0	0.341			
SOC	2	-2 2 4 6	765	240	35.0	36.5	0.115	19.1	1.54	31.8
	6					114.0	0.341			
	10					114.0	0.341			
T_{TDC}	2	0	845	240	35.0	36.5	0.115	19.1	1.54	32.4
	6					114.0	0.341			
	10					114.0	0.341			
T_{TDC} & P_{inj}	2	0	845	140	35.0	36.5	0.115	19.1	1.54	32.4
	6					114.0	0.341			
	10					114.0	0.341			
P_{inj}	2	0	765	140	35.0	36.5	0.115	19.1	1.54	31.8
	6					114.0	0.341			
	10					114.0	0.341			
ρ & X_{O_2}	6	0	765	240	32.8 40.8	114.0	0.341	20.4 16.3	1.40 1.79	31.8
ρ & ϕ_g	6	0	765	240	32.8 40.8	114.0	0.364 0.293	19.1	1.54	31.8
m_f	2	0	765	240	35.0	20.3 28.5 44.5 -	0.063 0.089 0.139 -	19.1	1.54	31.8
	6					60.0 88.0 139.0 -	0.201 0.268 0.424 -			
	10					60.0 88.0 139.0 180.0	0.201 0.268 0.424 0.550			
EGR	2	0	765	240	35.0	36.5	0.135	16.4	3.87	32.2
	6					114.0	0.402			
	10					114.0	0.402			
C_p	6	0	765	240	35.0	114.0	0.345	19.1	3.60	32.3

Primary Variable
Secondary Variable

supplied to the engine. The resulting mixture was designed to match the X_{O_2} and C_p of the charge-gas mixture at 10° BTDC in a metal engine with comparable displacement and load operating at 17.5% EGR (baseline).

Fuel Injection Considerations – Except for the m_f sweep, a constant-fueling approach was used instead of a constant-load approach to minimize variations in the actual duration of injection (DOI_a), ϕ_g , and the location of EOI. As shown below, DOI_a , ϕ_g , and the conditions at EOI are potentially important factors affecting engine-out smoke. Allowing these variables to change in order to maintain constant load across different operating conditions would have made it difficult to isolate the primary variable effects being studied.

The fueling rates for the 2-, 6-, and 10-hole tip experiments were based on the injected fuel mass required by the 6-hole tip to produce ~12 bar gIMEP at the baseline condition. The 6-hole baseline condition required an indicated duration of injection (DOI_i) of 4040 μ s (~36.5 CAD) to inject 114.0 mg of fuel and produce 12 bar gIMEP. At the same injection pressure and orifice diameter, the 10-hole experiments required shorter DOI_i -values to match the total injected fuel mass of the 6-hole experiments (e.g., 2625 μ s for the baseline) because the total flow area is larger for the 10-hole injector tip. Since the 2-hole experiments would have required an excessive DOI_i exceeding 12120 μ s (~110 CAD) to match the injected quantity of the 6-hole tip experiments at the baseline condition, the DOI_i -values for the 2-hole injector tip experiments match those of the 6-hole injector tip experiments. As a result, the injected quantity per jet for the 2-hole experiments matched the injected quantity per jet for the 6-hole experiments.

Number of Injector Orifices Sweep – This sweep shows the effect that increasing the number of injector-tip orifices from 2 to 6 to 10 (and,

correspondingly, decreasing jet spacing from 180° to 60° to 36°) has on indicated-specific emissions, load, and the ability to achieve LLFC. Increasing the number of orifices may be required to compensate for the loss of nozzle flow (and hence peak engine load capability) resulting from the use of smaller orifices to achieve LLFC.

Injection Pressure and TDC Temperature Study – Another way to compensate for the loss of nozzle flow resulting from using smaller orifice diameters is to increase P_{inj} . This study shows the impact that increasing P_{inj} from 140 MPa to 240 MPa and decreasing T_{TDC} from 845 K to 765 K has on H , $\phi(H)$, and the ability to sustain soot-free combustion. Note that increasing T_{TDC} produces a secondary increase in C_p from 31.8 J/mol-K to 32.4 J/mol-K.

Start-of-Combustion Timing Sweep – In this sweep, SOC was retarded from -2° ATDC to +6° ATDC in increments of 2 CAD (SOC_{fit} from -1.5° ATDC to +8.0° ATDC). Retarding SOC shifts the combustion to cooler and less-dense in-cylinder conditions that could be more conducive to LLFC.

Injected-Fuel-Mass Sweep – This sweep shows the impact that m_f has on combustion through changes to EOI, ϕ_g , and the extent of jet-jet interactions.

In-Cylinder Density Study – This study shows the effects that density has on soot-free combustion using conditions at a high ρ_{TDC} of 40.8 kg/m³ and a lower ρ_{TDC} of 32.8 kg/m³. At a fixed T_{TDC} of 765 K, an increase in ρ_{TDC} required an increase in the trapped charge mass, which was accomplished by increasing the intake manifold absolute pressure. This required either: (1) a decrease in ϕ_g if the intake-mixture composition (including X_{O_2}) was maintained, because the net mass of oxygen trapped in the cylinder would increase without a change in m_f ; or (2) a change in intake-mixture composition including a decrease in X_{O_2} if ϕ_g was maintained, because the mass of oxygen

trapped in the cylinder would not change with respect to the baseline operating condition.

EGR Study – This study shows how increasing the EGR rate from 17.5% to 35% affects LLFC. Increasing EGR suppresses the formation of NO_x by reducing peak in-cylinder and flame temperatures. Nevertheless, these same factors typically lead to higher engine-out smoke emissions [28], presumably because lower in-cylinder temperatures and X_{O_2} inhibit soot oxidation.

Specific-Heat Study – This study shows how increasing C_p from 31.8 J/mol-K to 32.3 J/mol-K affects LLFC. The increased C_p value matches that of the high-EGR condition without the reduction in X_{O_2} , and provides insight into the individual impacts of C_p and X_{O_2} on engine-out emissions.

RESULTS

Figure 6 shows images of natural luminosity (NL) and chemiluminescence (CL) during combustion at the baseline conditions with the 2-, 6-, and 10-hole injector-tip configurations. Substantial differences in H as well as the distributions and levels of the NL and CL are evident, as discussed below.

STATISTICAL VARIATION OF H AND $\phi(H)$

Previous studies in optically accessible engines have shown significant cycle-to-cycle variations in H that are on the order of 30% of the mean value (e.g., [16,20]). As shown in Fig. 7a, the minimum and maximum values of H in the current work can vary by > 30% of the mean value at a given CAD when measurements from multiple jets, cycles, and

replicate experiments are included. Moreover, statistically significant jet-jet differences in H on the order of 2 mm to 5 mm were consistently observed over all of the operating conditions. Some potential sources of the observed variability in these measurements are: (1) natural variations of H due to spatial variations in ambient-gas properties (e.g., temperature, density, composition) arising from systematic and/or cyclic variability in charge motion, thermal/mixture stratification, and heat release; (2) jet-to-jet differences in orifice characteristics (e.g., diameter, discharge coefficients, taper); (3) difficulty determining a single H value for the irregular base of a turbulent three-dimensional flame; and (4) the sensitivity of the measurement method to the rate of change in intensity along the jet axis [6,20].

Typically, variations arising from (1) and (3) are minimized using a long exposure (>100 μs) to average out transient fluctuations [6,20]. This method works well for CVCV experiments conducted at quiescent conditions because the ambient-gas thermodynamic properties upstream of H are effectively constant during the injection. On the other hand, increasing the exposure duration often does not work well for measurements of H inside an engine because H can change significantly on timescales >100 μs due to the combined effects of piston motion, charge motion, thermal stratification from heat transfer, and compressive heating from combustion. Such a change in H during a long exposure disperses the CL along the jet axis in the CL image, reducing the signal intensity gradients and making it more difficult to determine H . In the current work, exposure times in the range of 10 μs to 40 μs were used to balance these tradeoffs.

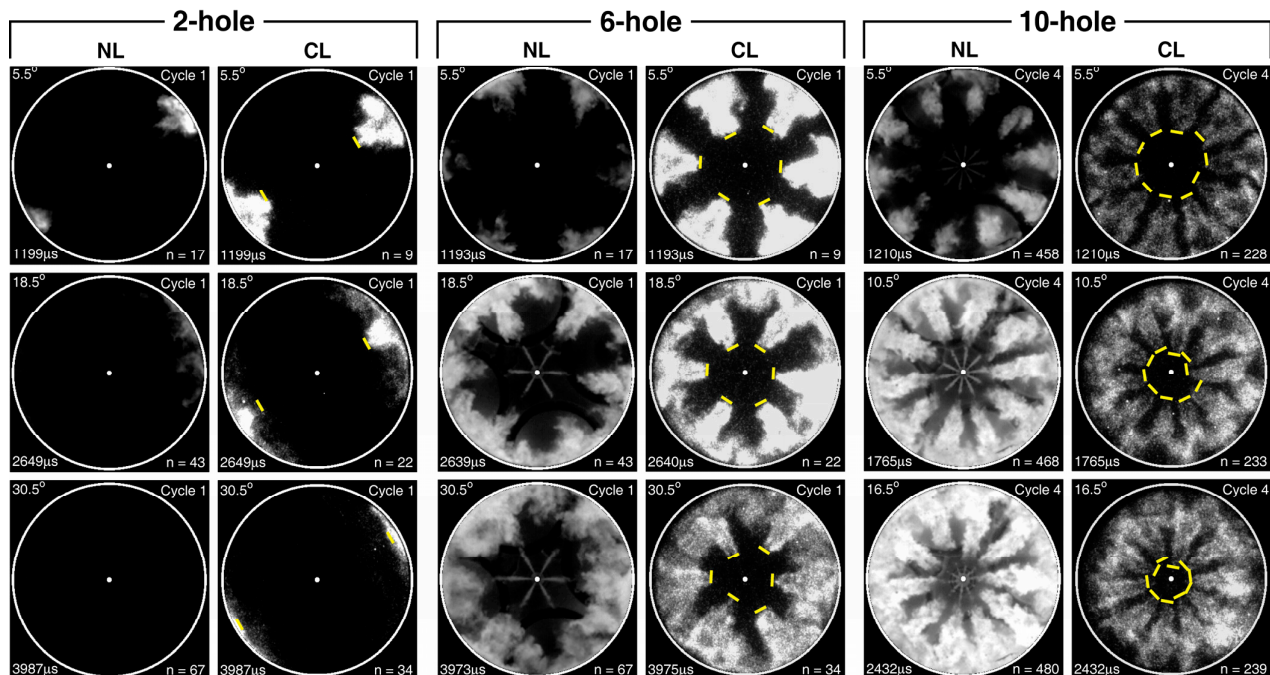


Figure 6. NL and CL images acquired at EOPMB (top row), midpoint between EOPMB and EOI (middle row), and EOI (bottom row) for the 2-, 6-, and 10-hole injector-tip baseline conditions. NL images were acquired using an exposure of 3 μs at f/11 for the 2-hole condition, and an exposure of 1 μs at f/16 for the 6- and 10-hole conditions. CL images were acquired using an exposure of 15 μs at f/4.5 and an intensifier gain of 50,000 for the 2- and 6-hole conditions, and an exposure of 7 μs at f/8 and an intensifier gain of 50,000 for the 10-hole condition.

As shown in Fig. 7a, the inherent jet-to-jet and cycle-to-cycle variation of H presents difficulties for the designation of statistically significant trends of H with respect to CAD. To address this problem, it is helpful to define the mean value of H as a function of CAD, denoted $\langle H(\text{CAD}) \rangle$, as follows:

$$\langle H(\text{CAD}) \rangle = \sum_{c=1}^{n_{\text{cycle}}} \left[\sum_{j=1}^{n_{\text{jet}}} H(j, c, \text{CAD}) \right] / (n_{\text{cycle}} \cdot n_{\text{jet}}) \quad (5)$$

In Eq. 5, H includes the lift-off length measurements from a single experiment, n_{jet} is the number of injector-tip orifices, and n_{cycle} is the number of engine cycles imaged. Using this definition, the replicate-to-replicate variation of $\langle H \rangle$ is less than 1%, as shown in Fig. 7b.

The error bars on $\langle H \rangle$ in Fig. 7b, which represent the minimum and maximum observations of $\langle H \rangle$, are generally smaller than their associated data points, suggesting that $\langle H \rangle$ can be measured with high confidence from as few as 20 samples in a single experiment. For example, each 2-hole experiment has only 2 jets and 10 cycles imaged, but the corresponding error bars in Fig. 7b are negligible. As a result, differences observed between operating conditions can be assigned statistical significance on the basis of $\langle H \rangle$.

Using $\langle H \rangle$ solved a similar problem that arose from the computation of $\phi(H)$. As shown in Fig. 7c, the direct use of individual H measurements in the computation of $\phi(H)$ leads to cycle-to-cycle variations in $\phi(H)$ that are 1.5 to 2 times their corresponding mean values. This represents a significant increase in the relative cycle-to-cycle variation with respect to the measurement of H at a given CAD, which is typically only 20-30% of the mean value. The increase in variation is a result of the non-linear relationship between H and $\phi(H)$ (see Eq. 4). As evident from comparisons between the minimum and maximum values of $\phi(H)$ for the 2- and 10-hole experiments (Fig. 7c), $\phi(H)$ is more sensitive to cycle-to-cycle variations when H is short. As a result, the mean value of $\phi(H)$ becomes skewed toward the measurements with the shortest H -values. For example, the mean $\phi(H)$ at EOI for the 6-hole data in Fig. 7c is ~0.5 higher than the corresponding $\phi(\langle H \rangle)$ at EOI shown in Fig. 7d. It was decided to characterize ϕ at H using $\phi(\langle H \rangle)$ rather than $\phi(H)$ to avoid this issue and to reduce the variability of the estimate. The extent to which $\phi(\langle H \rangle)$ is quantitatively representative of the true mean ϕ at H is unknown because direct ϕ -measurements were not made in this work.

Assuming the observed jet-to-jet and cycle-to-cycle variations of H are physically real (not an artifact of the measurement process), the question arises as to whether the steady-state engine-out emissions are

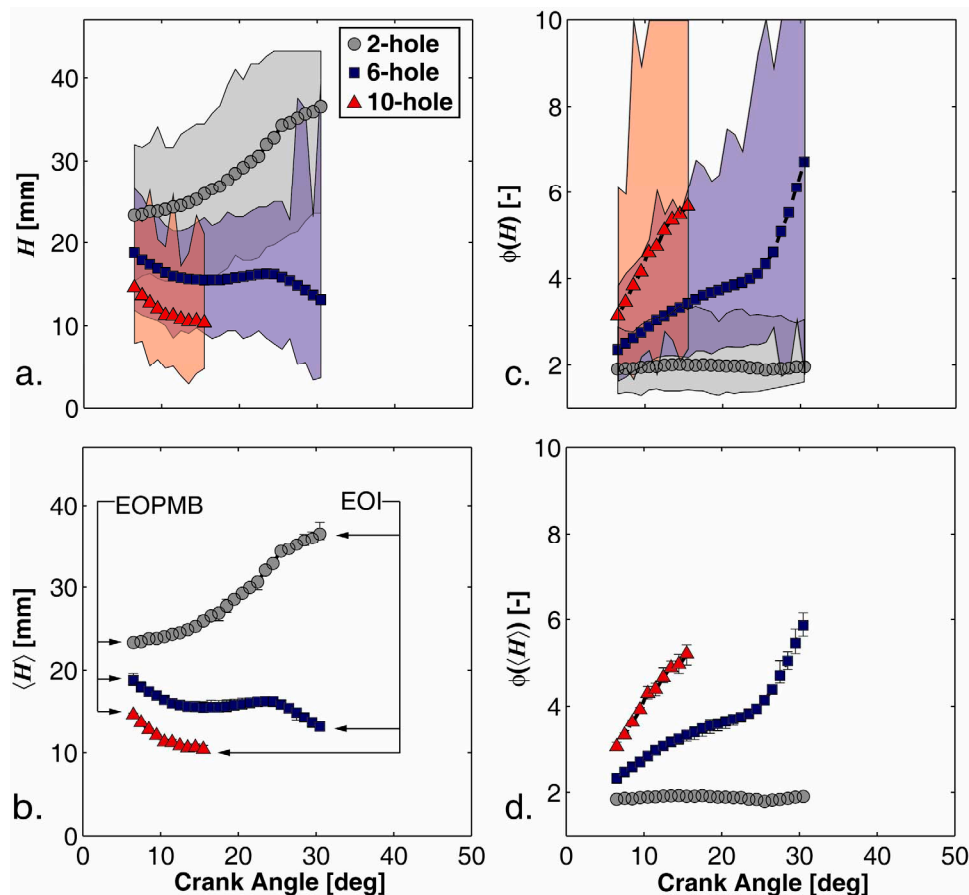


Figure 7. a. H , b. $\langle H \rangle$, c. $\phi(H)$, and d. $\phi(\langle H \rangle)$ as functions of crank angle at 2-, 6-, and 10-hole baseline conditions. Markers represent average values using data from each jet, cycle, and replicate experiment, and color-shaded regions represent minimum and maximum values. First and last values on each curve represent measurements at EOPMB and EOI respectively.