

Effect of Ignition Delay on In-Cylinder Soot Characteristics of a Heavy Duty Diesel Engine Operating at Low Temperature Conditions

Mohan K. Bobba, Caroline L. Genzale and Mark P. B. Musculus

Sandia National Laboratories

ABSTRACT

Low temperature combustion (LTC) strategies, which can mitigate emissions of particulate matter (PM) and nitrogen oxides (NO_x) from diesel engines, typically have longer ignition delays compared to conventional diesel operation. With extended ignition delays, more time is available for premixing, which reduces PM formation. The effect of varying ignition delay on the spatial and temporal evolution of soot in LTC diesel jets is studied by imaging the natural soot luminosity, while the in-cylinder soot mass and temperature are measured using two-color soot thermometry. Ignition delay in the engine is controlled by adjusting the intake air temperature while keeping the same charge density at TDC. This allowed us to study sooting characteristics at various ignition delays while keeping the same diesel jet penetration for all the cases.

Results show a 95% decrease in the total in-cylinder soot mass as ignition delay increases from 3 to 15 crank angle degrees (CAD) at an engine speed of 1200 RPM. Furthermore, the structure of the sooty regions in the jet is strongly affected by the ignition delay. For a short ignition delay of 3 CAD, soot formation originates downstream in the jet, 25 mm from the injector. After the end of injection, the sooty region first spreads back to the injector and then it is rapidly oxidized in the near-injector region within a few crank angle degrees. This suggests that rapid mixing occurs in the near injector mixtures just after the end of injection, which promotes soot oxidation. For a longer ignition delay of 15 CAD, soot first appears farther downstream in the jet, and it does not spread back to the injector. Indeed, soot never forms in the jet near the injector when the ignition delay is long, indicating that those regions do not promote soot

formation, likely because they become too fuel-lean during the ignition delay.

INTRODUCTION

In recent years, stringent pollutant emissions regulations for on-road heavy-duty diesel engines have required significant reductions of nitrogen oxides (NO_x) and particulate matter (PM) emissions [1]. All current U.S. 2007-compliant heavy-duty diesel engines use exhaust particulate filters to control PM and some form of exhaust-gas recirculation (EGR) to control NO_x [2]. Regulations already in place require even greater reductions in nitrogen oxides in 2010, so improved understanding of emissions formation processes with EGR is needed.

Dilution by EGR is effective to reduce combustion temperatures and thereby reduce NO_x formation, but it has two opposing effects on formation of soot, a major component of PM. With lower intake oxygen concentrations, *overall* in-cylinder mixtures are more fuel rich with EGR, which promotes soot production [3]. On the other hand, both the ignition delay and the flame lift-off length are increased with EGR, so that *local* pre-combustion mixing is increased, which reduces soot formation [4]. Regardless of the net effect on soot formation, increasing EGR reduces soot oxidation considerably, so that soot emissions generally increase with EGR, at least initially [5-7]. At very high EGR rates (typically 60% or higher), with very low intake oxygen concentrations of 10% or less, soot emissions eventually decrease because of very low soot formation, even with poor soot oxidation [5-8].

Such high EGR levels required to significantly reduce soot formation can decrease fuel efficiency and cause other emissions problems. Furthermore, such high EGR levels far exceed those needed to satisfy NO_x limits. Rather than using excessive EGR, soot formation can also be reduced by increasing pre-combustion mixing by increasing the ignition delay, but with more moderate EGR levels. Indeed, many low-temperature combustion (LTC) strategies use long ignition delays, which serve to increase pre-combustion mixing [9-11]. Previous optical measurements have shown that soot formation is much different for LTC strategies with long ignition delay [12] than for conventional diesel combustion [13]. However, the evolution of the quantity and spatial distribution of in-cylinder soot with increasing ignition delay at LTC conditions has not yet been carefully explored.

The objective of this study is to examine spatial and temporal evolution of soot in a direct injection (DI) heavy-duty diesel engine operating at LTC conditions while systematically adjusting the ignition delay. Here, the ignition delay is adjusted by controlling the intake temperature while holding the charge density constant. In this way, the ignition delay may be adjusted using temperature, while the constant charge density keeps the jet penetration characteristics unchanged. This approach greatly aids interpretation of results for the conditions with widely varying ignition delays explored here. High-speed soot luminosity movies reveal the spatial distribution of soot, while two-color optical thermometry quantifies the evolution of in-cylinder soot mass and temperature.

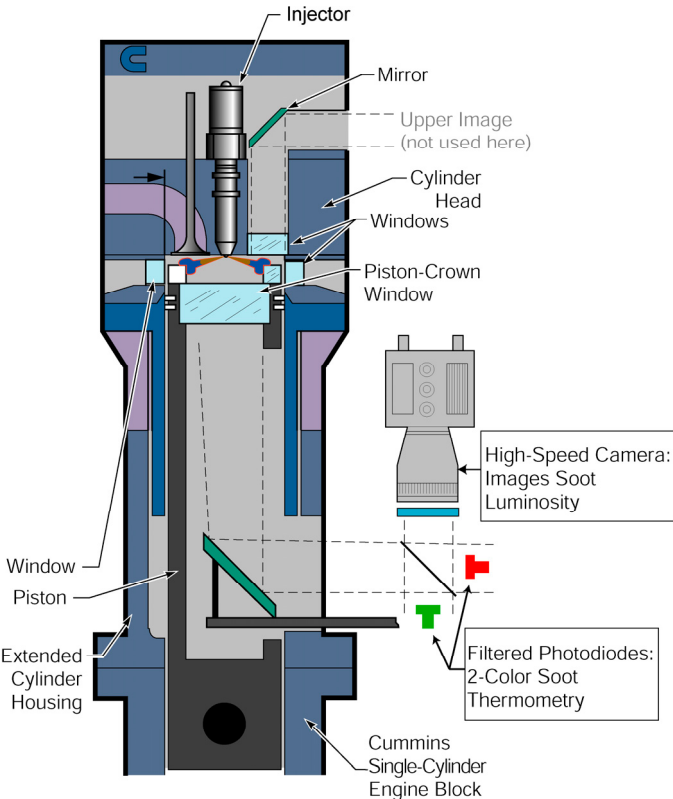


Figure 1. Schematic of the heavy-duty DI diesel engine and layout of the two-color soot thermometry and high-speed imaging diagnostics

EXPERIMENTAL SETUP AND DIAGNOSTICS

OPTICAL ENGINE

A single-cylinder, direct-injection (DI), 4-stroke diesel engine based on a Cummins N-series production engine was used in this investigation. A schematic of the engine is shown in Fig. 1 and its specifications are summarized in Table 1. A brief description of the engine is provided here, while a complete account is available in Refs. [13,14].

The Cummins N-series engine is typical of heavy-duty size-class diesel engines, with a bore of 140 mm and a stroke of 152 mm, yielding a displacement of 2.34 liters per cylinder. The cylinder head produces a nearly quiescent charge, with a swirl ratio of approximately 0.5. The optical engine is designed so that the upper cylinder liner separates and drops down from the head without engine disassembly, allowing rapid cleaning of in-

Table 1. Engine and injector 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*
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
Simulated compression ratio	16:1
Fuel injector type	Common-rail, solenoid actuated
Cup (tip) type	mini-sac
Number of holes & arrangement	8, equally-spaced
Spray pattern included angle	152°
Rail pressure	1200 bar
Orifice treatment	None (square-edged)
Nominal orifice diameter	0.196 mm
Orifice L/D	5

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

cylinder surfaces. To provide optical access, the engine is equipped with an extended piston and a flat piston-crown window. A window replaces one of the two exhaust valves in the cylinder head to provide imaging access to the squish region, although it is not used in the current study. Several windows are also located around the top of the cylinder-wall for introduction of laser light for other diagnostics (also not used in the current study).

FUEL INJECTOR

The engine is equipped with a non-production, high-pressure, electronically-controlled, common-rail fuel injector. Specifications for the fuel injector are included in Table 1. This injector uses a solenoid-actuated pilot valve and a pressure-balanced needle to control fuel delivery. It is capable of multiple injections at up to 2000 bar fuel-rail pressure. For the experiments presented here, an eight-hole, mini-sac injector cup (tip) was employed, having an included angle of 152° (14° down-angle from the firedeck). The eight fuel orifices are equally spaced and have nominal diameter of 0.196 mm, and have no edge treatment (they are sharp-edged). The mass rate of injection was measured using a spray impingement meter [15].

ENGINE PERFORMANCE DIAGNOSTICS

For all operating conditions, cylinder pressure and fuel rail pressure were digitized and recorded at quarter crank angle degree increments, simultaneously with the acquisition of the optical data. The apparent heat release rate (AHRR) was calculated from ensemble averaged pressure data using an air-standard first-law analysis (e.g., Heywood [16]). Prior to calculating the AHRR, the 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. These cut-off frequencies were selected to remove acoustic ringing in the cylinder pressure data, while retaining the general features of the AHRR. Using these filter parameters, the apparent start of combustion (SOC), indicated by the first zero-crossing of AHRR, was advanced by about 0.5 crank-angle degrees (CAD) from that for the unfiltered data. Additionally, the peak of the premixed burn spike in the filtered AHRR was reduced by about a factor of 2, while its width was increased by a factor of 2. The apparent energy released during the premixed burn (area under the AHRR curve), however, was virtually unchanged by this filtering technique.

ENGINE OPERATING CONDITIONS

A single distillate diesel fuel has been used in this study and its properties are listed in Table 2. The operating conditions employed here to investigate soot formation are characterized by the nominal charge conditions at top dead center (TDC) for a motored cycle, as shown in Table 3. These nominal TDC conditions were calculated using a classical polytropic compression from the intake manifold conditions, using the geometric compression

Table 2. Fuel specifications

Phillips Petroleum Ultra-Low Sulfur 2007 Emissions
Certification Number 2 Diesel Fuel

Total aromatics by volume	27 %
Olefins	0.5 %
Saturates	72.5 %
Sulfur (by weight)	9.1 ppm
Distillation Temperatures	
Initial Boiling Point	190 °C
10% Distillation Temperature	212 °C
50% Distillation Temperature	254 °C
90% Distillation Temperature	315 °C
End Point	350 °C
Cetane number	46
Specific gravity @ 20 °C	0.8426
C/H by weight	6.5
Net Heat of Combustion	43 MJ/kg
Viscosity @40°C	2.35 cS

ratio and a polytropic exponent of 1.36. The initial bottom dead center (BDC) pressure for the polytropic calculation was taken as the intake pressure, while the effects of heat transfer during induction on the BDC temperature (T_{BDC}) were estimated using the following equation, as described in [17]:

$$T_{BDC} = 0.68(T_{Intake} - T_{Coolant}) + T_{Coolant} \quad (1)$$

In Eq. 1, T_{Intake} is the intake gas temperature, measured in the manifold near the head, and $T_{Coolant}$ is the temperature of the heated “coolant” water flowing through the engine.

To understand the effect of increasing ignition delay on soot formation, TDC temperature is decreased from 1000 to 775 K in 10 steps covering conditions typical of hot to very cold intake conditions. For each case tested, the intake conditions were set to yield a required nominal TDC (motored) temperature. Then the intake pressure was adjusted to achieve a constant motored TDC charge density of 22.1 kg/m³. A constant TDC density ensures a similar jet penetration, making it easier to compare across various cases.

Note that the intake temperatures and pressures in Table 3 are significantly higher than typical production diesel engines because of the reduced compression ratio of the optical engine. Due to compromises necessary to implement optical access in the engine, the geometric compression ratio of the optical engine is about 11.2:1, compared to 16:1 in the production engine. As a result, elevated intake air temperatures and pressures are required to yield charge conditions at TDC

Table 3. Operating Conditions

Speed [RPM]	1200										
O ₂ Conc. [Vol. %]	12.7										
Rail Pressure [bar]	1200										
Actual SOI [CAD]	-5 ATDC										
DOI [CAD]	9.6										
TDC Motored Density [kg/m ³]	22.1 (low boost for R _c =16)										
TDC Motored Temp. [K]	1000	975	950	925	900	875	850	825	800	788	775
Intake Temp. [°C]	170	154	138	123	109	94	78	63	48	40	33
Intake Press. [kPa]	241	235	229	223	217	211	205	199	193	190	187
GIMEP [kPa]	599	593	600	595	599	606	616	610	602	591	558
Intake Temp. for 16:1 Engine [°C]	119	105	90	76	62	48	34	20	6	-1	-9
Intake Press. for 16:1 Engine [kPa]	155	151	147	143	139	135	132	128	124	122	120

typical of those in a production diesel engine. The intake pressure and temperatures required for a 16:1 compression ratio engine to achieve the same TDC charge conditions are also listed in the bottom two rows of Table 3 for reference.

Although the thermodynamic conditions at TDC are representative of production hardware, the temperature and density of the air charge in the optical engine are slightly higher away from TDC than in the production engine, because of the lower geometric compression ratio. The total mass of the air charge is also about 50% greater than that of the production engine for similar thermodynamic conditions at TDC.

All the tests are performed at a high simulated EGR rate (intake oxygen concentration of 12.7% equivalent to 50-60% EGR) and the injection duration was held constant with start of injection at -5 CAD after TDC. To avoid thermal loading problems in the optical engine, the fuel injector was skip-fired, so that fuel injection occurred in every 10th engine cycle. As a result, the use of real EGR is not a practical option to dilute the oxygen in the intake stream to the low levels used in this study. Rather, the intake stream was diluted with a metered flow of nitrogen to achieve the desired intake oxygen concentrations to simulate dilution by EGR. A regular schedule is maintained while performing these experiments where the engine is motored for 2 minutes before the start of fuel injection to minimize variability of results due to transients.

SOOT OPTICAL PYROMETRY

During the diesel combustion event, light is emitted over a broad range, from ultraviolet to infrared wavelengths. This naturally-occurring luminosity is primarily composed of soot incandescence, with some chemiluminescence, as discussed in detail in Ref. [18]. The temperature and optical density of the soot may be discerned from the spectral measurements of the combustion luminosity, as

long as soot incandescence dominates the luminosity. Under typical diesel engine operating conditions, soot luminosity is the primary source of emission, being at least 4-5 orders of magnitude stronger than chemiluminescence for typical diesel fuels [18]. As will be shown in the results section, only for the lowest temperature operating conditions of the current study is soot luminosity weak enough that measurement error from chemiluminescence may become significant.

The light emitted by glowing soot is captured by two photodiodes and imaged by an IDT-XS-4 high-speed digital CMOS camera, manufactured by Integrated Design Tools, Inc. (Fig. 1). The two photodiodes equipped with band pass filters (BPF) record light intensity in 10-nm-wide bands located about the center wavelengths (580 nm, and 900 nm). A dichroic beam-splitter directs about half of the collected soot luminosity towards each of the photodiodes as shown in Fig. 1. A portion of soot luminosity reflects back from both band-pass filters, passes through the beam-splitter again, and is captured using the high speed camera.

The mathematics of the two-color thermometry has been described in numerous sources, and a complete treatment is not included here. The complete details of the soot thermometry technique may be found in Ref. [19]. A brief summary follows.

The signal from an optical detector viewing a radiating soot cloud may be described as [19]

$$S_{\lambda} = C_{\lambda} \int_A \varepsilon_{\text{soot}}(f_v L, \lambda) I_{\lambda, \Omega, bb}(T_{\text{soot}}, \lambda) dA \quad (2)$$

where S_{λ} is the signal from the detector at the BPF wavelength λ , C_{λ} is a calibration constant, A is the projected area of the soot cloud within the field of view of the detector, $\varepsilon_{\text{soot}}$ is the soot emissivity, $I_{\lambda, \Omega, bb}$ is the blackbody spectral intensity (radiative power per source area per solid angle per wavelength), T_{soot} is soot

temperature, and $f_v L$ is the product of the mean soot volume fraction and the line-of-sight path length, or “ $f_v L$ factor”. The blackbody spectral intensity $I_{\lambda, \Omega, bb}$ was taken from Planck’s equation [20], and the soot emissivity ε_{Soot} in Eq. 2 was determined by an empirical relation taken from Ref. [21]

$$\varepsilon_{Soot}(f_v L, \lambda) = 1 - \exp\left(-\frac{g f_v L}{\lambda^\alpha}\right) \quad (3)$$

For consistency of units in this empirical relation, the wavelength term λ is treated as unitless, but its value is the light wavelength, in microns. The constant g was assigned the value of $6.3 \mu\text{m}^{-1}$, taken from [21]. The variable α is given as [20]

$$\alpha = 1.22 - 0.245 \ln(\lambda [\mu\text{m}]) \quad (4)$$

Treating $I_{\lambda, \Omega, bb}$ and ε_{Soot} as constant over A , the integral in Eq. 2 may be replaced by the product of A with the mean soot emissivity $\bar{\varepsilon}_{Soot}$ and the mean blackbody spectral intensity $\bar{I}_{\lambda, \Omega, bb}$, both of which are effectively averaged over A .

$$S_\lambda = C_\lambda A \bar{\varepsilon}_{Soot}(f_v L, \lambda) \bar{I}_{\lambda, \Omega, bb}(T_{Soot}, \lambda) \quad (5)$$

The calibration constant, C_λ , was determined using a calibrated integrating sphere as an absolute light source. The integrating sphere was placed atop the engine cylinder in place of the cylinder head. It provided a known source of spectral intensity for the two-color-thermometry optical train, which was left intact. By using Eqs. 3, 4, and 5, along with Planck’s equation, the detector signal at any wavelength can be modeled as a function of A , $f_v L$, and T_{Soot} , with C_λ as a calibration constant.

Solution of Eq. 2 requires measurement of the projected area of the soot cloud, A . In many previous applications of two-color thermometry, A was small and unchanging due to the finite field of view of the detection system, so it could be folded into the calibration constant C_λ . But here, A is large and dynamic, so it was measured from images taken by the high-speed digital camera. In a particular image, the area of the soot cloud is equal to the sum of physical area of all the pixels having intensities over a threshold level (10% of maximum intensity). Intensity gradient at the edges of the soot cloud is relatively sharp, so projected area of the soot cloud is fairly insensitive to the intensity threshold level, changing by only 12% even when the threshold is doubled to 20% of full scale.

The T_{Soot} and $f_v L$ solutions were determined from the measured signals through a least-squares fit to soot model predictions (Eq. 5) at the two colors, using the built-in “fminsearch” function in Matlab®, a commercial mathematical software package [22]. The least square fit approach used here is found to be very robust and

hence it is favored over the use of usually employed analytical or an iterative solution to Eq 5.

Soot deposits that accumulate on the surface of the imaging window with each fired cycle can significantly attenuate the light reaching the detectors. Previous soot deposit attenuation measurements through the cylinder-wall windows using a stable laser source under conditions similar to that of the current study show attenuation of 5-15% per cycle [23,24]. Simple ensemble averaging of the data over multiple cycles, as is common to help to narrow the uncertainty in the mean values, is undesirable because of increased attenuation effects for later cycles. To better correct for the effects of attenuation, the crank-angle-resolved $f_v L$ and T_{Soot} data for the first 10 fired cycles are fit to a linear curve with respect to the fired cycle number. That is, the $f_v L$ and T_{Soot} data acquired at each crank angle are plotted versus the fired cycle number, and the data are fit to a linear function by minimizing the sum of the squared errors. Then, using the curve-fit parameters, the data are extrapolated back to the “zeroth” cycle, which would have no attenuation from soot deposits. (Even the first fired cycle has some attenuation from soot deposited during that cycle.) In this way, the $f_v L$ and T_{Soot} data presented in the results section represent cycle-averaged data for the first 10 fired cycles, corrected to zero attenuation. Finally, in-cylinder soot mass was estimated from $f_v L$ and the projected area of the soot cloud as follows:

$$m_{soot} = f_v L A \rho_{soot} \frac{V_{Cylinder}}{V_{Bowl}} \quad (6)$$

In Eq. 6, the product of $f_v L$ and A is equal to the soot volume within and directly above the piston bowl, and multiplication by an assumed soot density, ρ_{Soot} (1.8 g/cm^3), gives the soot mass within the bowl. Note that the optical diagnostic only measures soot within the bowl (see Fig. 1). To account for soot residing outside of the visible area (mainly the squish volume), the soot mass is corrected according to the ratio of the total cylinder volume ($V_{Cylinder}$) to the volume within and above the piston bowl (V_{Bowl}), which is visible to the optical system. This correction assumes that the soot volume fraction in the regions not visible to the optical system is the same as that visible within the bowl.

RESULTS AND DISCUSSION

APPARENT HEAT RELEASE RATE ANALYSIS

Shown in Fig. 2 are the AHRR data acquired over a range of TDC temperatures with a constant TDC density. Also overlaid in the figure is the rate of injection profile (duration ~10 CAD) which was held constant for all the conditions tested. The solenoid-actuated injector used in this study has a moderate injection ramp-down lasting ~0.35 ms. AHRR data presented in the figure is ensemble averaged over 20 fired cycles for all the cases presented, while the injection profile measured using a