

Effect of Post Injections on In-Cylinder and Exhaust Soot for Low-Temperature Combustion in a Heavy-Duty Diesel Engine

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Mohan Bobba and Mark Musculus
Sandia National Laboratories

Wiley Neel
Stanford Univ.

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ABSTRACT

Multiple fuel-injections during a single engine cycle can reduce combustion noise and improve pollutant emissions tradeoffs. Various hypotheses have been proposed in the literature regarding the in-cylinder processes responsible for the pollutant emissions improvements. This paper provides a brief overview of these hypotheses along with an investigation exploring which of these mechanisms are important for post injections under low-temperature combustion (LTC) conditions in a heavy-duty diesel engine. In-cylinder soot and exhaust smoke are measured by 2-color soot thermometry and filter paper blackening, respectively. The evolution and interaction of soot regions from each of the injections is visualized using high-speed imaging of soot luminosity, both in the piston bowl and in the squish regions. Results for early main-injection (-18° after top dead center command start of injection) conditions show an abrupt reduction in exhaust soot as the post injection is delayed to timings where the post-injection jet starts to enter the squish volume. At these conditions, the post injection in the squish volume produces no soot of its own, and it does not interact with soot from the main injection, which remains in the piston bowl. Reduction in soot for this case is primarily due to fuel in the post injection burning soot-free in the squish volume. In contrast, for a late main-injection (2° after top dead center), the exhaust soot decreases to levels below “main-injection-only” values as the post injection is retarded. With late main-injection, significant soot from the main injection remains in the squish volume, so that strong interaction between the main and post jets is evident. The retarded post injection conditions have higher late-cycle soot

temperature and faster oxidation of main-injection soot remaining in the squish volume. Finally, for either main-injection timing, the optimal post-injection timing is at roughly the same crank angle position. Hence, for these LTC post-injection conditions, the absolute crank angle position of the post injection appears more important for soot reduction than the dwell between injections, though the dwell also could play a role in soot reduction, especially when using a late main injection.

INTRODUCTION

With regulations on diesel engine pollutant emissions becoming more stringent, engine designers have been prompted to explore various alternative injection strategies for compliance. Splitting the fueling into multiple injections within a single engine cycle can significantly reduce engine-out soot emissions [1]. Additionally, multiple injections also can reduce the peak cylinder pressure and combustion noise [2,3].

To comply with recently enacted and upcoming emissions regulations, modern diesel engines frequently use low-temperature combustion (LTC) strategies for nitrogen oxides (NOx) emissions reduction. Here, “LTC” is defined as an operating strategy where the peak combustion temperatures are significantly lower than for stoichiometric combustion in undiluted air. For our study, we achieve LTC by using exhaust-gas recirculation (EGR). Multiple injections can be particularly attractive for reducing soot emissions for diesel engine operational strategies that use EGR to achieve LTC. Specifically, a combination of high EGR (over 40%) and post injections has been shown to be an effective way to reduce



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soot emissions while maintaining low NO_x emissions [4,5,6]. Post injections have shown significant benefit on soot emission performance typically for conditions where mixing is difficult, e.g., low injection pressures or high EGR. Hence, these strategies seem to work best at medium to high-load and high EGR conditions where effective utilization of in-cylinder air is critical [6,7]. Ojeda et al. [8] have demonstrated that adding a post injection helps extend the load limits of clean combustion (NO_x level of 0.2g/hp-hr and an after treatment-tolerant soot level) for loads ranging from 10 bar to 16.5 bar brake mean-effective pressure (BMEP) in a medium-duty diesel engine with the use of an EGR rate of 44%. However, the improvement achieved in exhaust emission performance was reported to be very sensitive to the post-injection parameters like the delay from main injection and fraction of fuel injected in the post. The exhaust soot was reduced only with a post-injection dwell of more than 30 crank angle degrees (CAD) in experiments by Ojeda, while the exhaust soot level with close-coupled posts was the same or higher compared to single injections. The effect of post-injected fuel quantity on exhaust-soot emissions was studied in a heavy-duty diesel engine by Payri et al. [4]. They concluded that the optimum fuel mass in the post-injection pulse is around 7-10% of the total fuel mass for full-load conditions and 15-20% for partial-load modes. Very late post injections also can be useful in regeneration of the catalyst in after treatment systems [9].

Various and sometimes contradicting hypotheses have been offered in the literature to explain improvements in emission performance with post injections, and a comprehensive understanding of mechanisms responsible for this improvement has not been established. Chen et al. [10] suggested that particulate oxidation is improved later in the combustion cycle due to increased in-cylinder bulk temperature and enhanced air-fuel mixing. Hotta et al. [11], proposed that the jet flame from a small post injection enhances mixing of the soot from the main injection by carrying it into the squish area where it is more readily oxidized. By contrast, Bruneaux [12] observed that when individual injections that did not produce soot were closely coupled, the second injection formed soot due to earlier ignition caused by enhanced mixing with hot ambient gases and radical pools left over from the first injection. These studies show that interactions between multiple injections can have opposite effects on soot formation depending on operating conditions. On the other hand, Han et al. [13] and Desantes et al. [14] have argued that under some operating conditions, there is no significant interaction between the main and the post injections. Their tests were done at a constant load, where the main injection duration was reduced as the post injection was added. They concluded that the post injection was neutral for soot emissions, and that the reduction in exhaust soot was due to less fuel burning in the soot-producing main injection. Desantes termed this behavior as “split flame” behavior.

Part of the confusion regarding the interactions between multiple injections results from practical aspects of engine operation. Typically, the effects of multiple injections at a given engine load are of interest. However, to maintain constant load, the amount of fuel in the main injection must be reduced as the post injection is added. The result is that both injections are changed, so that any change in soot emissions could be attributed to one or both of the injections, or to their interactions. Furthermore, measurements of exhaust emissions alone do not provide direct evidence of the in-cylinder mechanisms responsible for soot emission reductions. Likewise, measurements of in-cylinder soot during combustion are not necessarily representative of exhaust soot.

To provide a comprehensive picture of both (i) the effects of post injections on exhaust soot and (ii) observations of the in-cylinder mechanisms responsible, we report here quantitative in-cylinder soot and exhaust smoke measurements for a parametric study of post injections in an optical heavy-duty engine. The in-cylinder soot mass and temperature are measured using a 2-color thermometry technique in either the piston bowl or in the squish volume. Also, the evolution of soot in the piston bowl and squish regions is visualized using high-speed imaging of natural soot-luminosity. Smoke emissions in the engine exhaust are measured using the filter-paper blackening technique. To study the role of the separation between injections on exhaust soot and on possible interactions between injections, the timing of both the main and post injections are varied parametrically. Throughout the experiments, the fuel mass in each injection, rather than engine load, is held constant, so that the fundamental fluid-mechanical effects of post injections may be studied. Also by keeping the fuel mass in each injection constant, any change in in-cylinder and exhaust soot performance with injection timing can be mainly attributed to changes in the injection strategy, rather than to variations in engine global equivalence ratio.

EXPERIMENTAL SETUP

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. [15,16].

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*
Intake valve opening	17° BTDC Exhaust [‡]
Intake valve closing	195° ATDC Exhaust [‡]
Exhaust valve opening	235° BTDC Exhaust [‡]
Exhaust valve closing	27° ATDC Exhaust [‡]
Combustion chamber	Quiescent, direct injection
Swirl ratio	0.5 (approx.)
Bore	139.7 mm [5.5 in]
Stroke	152.4 mm [6.0 in]
Bowl width	97.8 mm [3.85 in]
Displacement	2.34 liters [142 in ³]
Connecting rod length	304.8 mm [12.0 in]
Piston pin offset	None
Geometric compression ratio	11.2:1
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°
Orifice treatment	None (square-edged)
Nominal orifice diameter	0.15 mm

**In this optically accessible diesel engine, one of the two exhaust valves of the production cylinder head was replaced by a window and periscope (see Fig. 1). ‡All valve timings correspond to the crank angle when the valve first starts to move from fully closed.*

ENGINE OPERATING CONDITIONS

The engine operating conditions employed here to investigate changes in soot formation with post injections are listed in Table 2. The top dead center (TDC) conditions (27.7 kg/m³ and 914 K) correspond to medium-boost operation in production diesel engines. The nominal TDC conditions listed in the table were calculated using a classical polytropic compression from the intake manifold conditions as described in [17]:

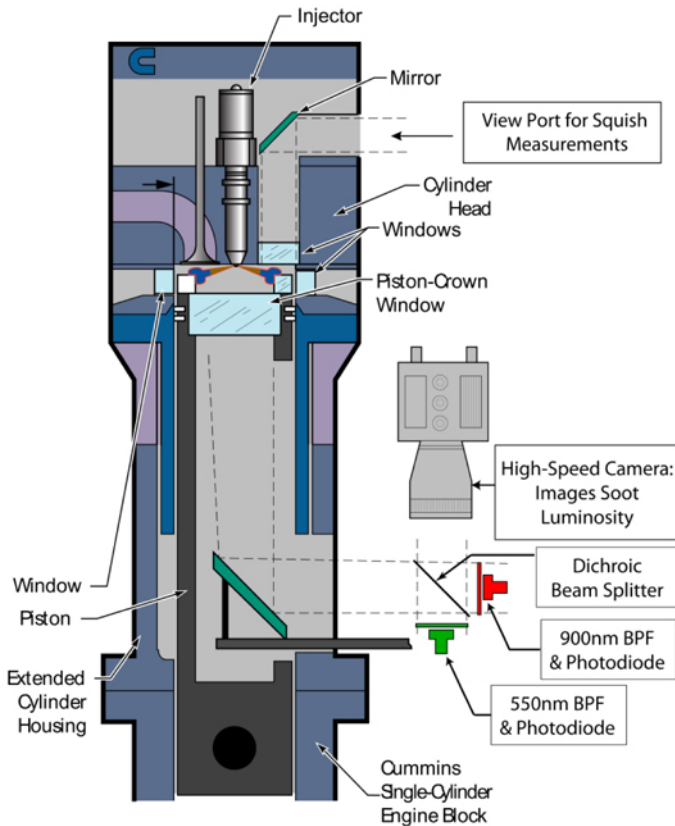


Figure 1. Schematic of the heavy-duty DI diesel engine and layout of the two-color soot thermometry and high-speed imaging diagnostics

The Cummins N-series engine is typical of heavy-duty size-class diesel engines. 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-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. Several windows also are located around the top of the cylinder-wall for introduction of laser light for other diagnostics (not used in the current study). The engine is equipped with a Cummins XPI 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.

Table 2. Operating Conditions

Engine Speed	1200 RPM
Engine Load Range	9-10 bar $\bar{g}IMEP^{\dagger}$
Intake O_2	12.7% (Sim. 55% EGR)
Fuel Pressure	1600 bar
TDC Motored Density	27.7 kg/m ³
TDC Motored Temperature	914 K
TDC Motored Pressure	74.3 bar
Intake Pressure	276 kPa
Intake Temperature	116.5°C
(16:1 Compression Ratio Intake Pressure)	170 kPa (abs)
(16:1 Compression Ratio Intake Temperature)	70°C
Command Start of Main Injection (SOI1)	-18 to 2 ATDC
Command Duration of Main Injection (DOI1)	1500 μ s (~11 CAD)
Post-Injection Dwell	~5-60 CAD
Fraction of Fuel in Post Injection	17% (~25 mg/stroke)

$\dagger \bar{g}IMEP$ = Gross Indicated Mean Effective Pressure, calculated using indicated work done during compression and expansion strokes only.

Note that the intake temperatures and pressures in [Table 2](#) 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 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 [Table 2](#) 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 than in the production engine at crank positions where the piston is away from TDC, 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. 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 [18]).

The in-cylinder soot and exhaust smoke from the engine are measured for a matrix of main- and post-injection timings. The command start of the main injection (i.e., the rising edge of the logical command signal) for these tests occurs at -18, -13, -8 or 2° after top dead center (ATDC), while the command start of the post-injection in each case is swept from about 5 CAD after the end of the main injection command to 60 CAD into the expansion stroke. For each injection, the delay between the command start of injection and the actual start of fuel injection (i.e., when fuel first

emerges from the injector) is approximately 460 microseconds, or 3.3 CAD at 1200 crankshaft rotations per minute (RPM). The 1500-microsecond (11 CAD at 1200 RPM) command-duration of the main injection (DOI1) delivers 115 mg of fuel. The nominal 400 microsecond post-injection command-duration is adjusted to account for rail dynamics so that the injected mass remains approximately constant at 25 mg of fuel, corresponding to 17% of total fuel mass in the post. When using multiple injections, pressure oscillations triggered by the nozzle closure after each injection can induce disturbances in the fuel injection profiles during subsequent injections (thereby altering the fuel mass in the post injection), as the dwell between injections is varied [19]. Hence to be able to study the effect of post injections on engine performance at constant fuel mass as desired, the amount of fuel injected during the post injection is monitored using a “rail-pressure correlation,” described in [Appendix A](#). The post-injected fuel mass is held constant by making small changes to its command duration according to the rail-pressure analysis. [Figure 2](#) shows the extent of the required variation in the command duration of the post injection (DOI2) as the post injection is retarded following a main injection timed at -8°ATDC. The figure also highlights the effect of these DOI2 changes on the global equivalence ratio of the engine cylinder as the post injection is delayed. The maximum deviation in the post-injected fuel mass is within 3 mg (out of nominally 25 mg in the post injection) across the cases, so that the global equivalence ratio is virtually constant for all the cases presented in this work.

The soot and visualization measurements also are performed for two single-injection cases as illustrated in [Fig. 3](#). First is a case with only the main injection of 1500-microsecond command duration and hence with a lower injected fuel mass compared to a case with a post injection. The second is a long main injection with a command duration of 2070 microseconds matching the total fuel mass injected in the main and post injections together. The single-injection cases form a reference dataset that is helpful for interpreting the results with post injections. The terminology shown in the figure is used for these cases through the rest of the paper.

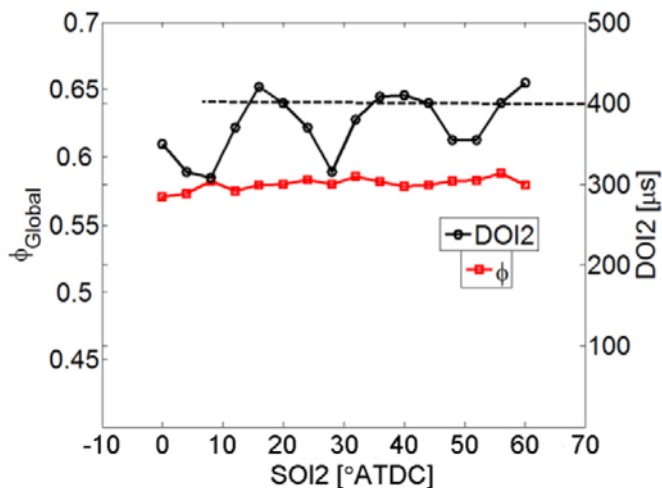


Figure 2. Post-injection command-durations employed at various post-injection timings to maintain a relatively constant injected fuel mass for main start of injection at -8°ATDC . The red curve shows the variation of the overall engine equivalence ratio (ϕ) as a result of using the shown DOI2 values.

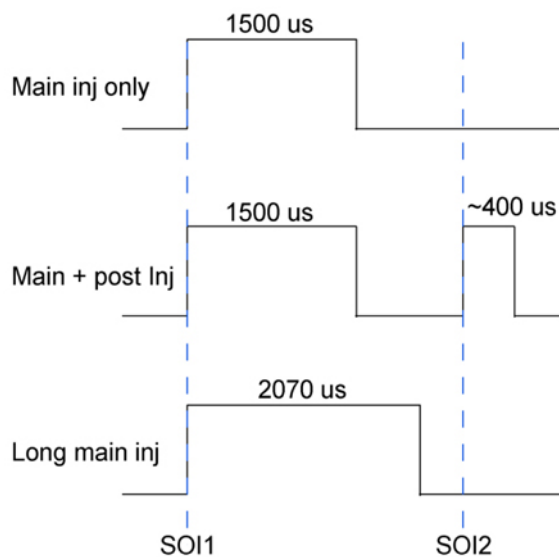


Figure 3. Illustration of various injection schedules and terminology used in the paper.

All the tests are performed at a high simulated EGR rate of 55% for a 16:1 compression ratio engine (intake oxygen concentration of 12.7%). Based on earlier studies (for example, Ojeda et al. [8], Helmantel [20], and Horibe et al. [21]), the exhaust NO_x emission at these high EGR rates (although not measured directly), is expected to be sufficiently below the US 2010 heavy-duty truck limit of 0.2 g/bhp-hr. 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 90 seconds before the start of fuel injection to minimize variability of results due to transients. A petroleum-based diesel fuel has been used in this study. The fuel properties provided by the manufacturer are listed in Table 3.

Table 3. 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	6.53
Net Heat of Combustion	43 MJ/kg
Viscosity @40°C	2.35 cS

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 typically primarily composed of soot incandescence, with some chemiluminescence, as discussed in detail in Ref. [22]. The temperature and optical density of the soot may be discerned from the spectral measurements of the combustion luminosity, using the two-color soot thermometry technique. The mathematics of 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. [23]. The technique works well as long as soot incandescence dominates the total 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 [22]. This assumption is true for many diesel conditions, but not necessarily for late cycle soot temperature measurements when using a very late post injection. The

chemiluminescence due to combustion of the post-injected fuel can be comparable to very weak soot luminosity in these cases, thereby introducing some uncertainty in the temperature measurement.

For soot-pyrometry measurements in the piston bowl, light emitted by glowing soot is reflected off the piston mirror, captured by two photodiodes, and imaged by an IDT-XS-4 high-speed digital CMOS camera (manufactured by Integrated Design Tools, Inc.) as shown in Fig. 1. The two photodiodes equipped with band-pass filters (BPF) record light intensity in 10-nm-wide bands located about their respective center wavelengths of 550 nm and 900 nm. A dichroic beam-splitter directs a part of the collected soot luminosity to each of the photodiodes as shown in Fig. 1. Most of the soot luminosity reflects back from both band-pass filters, passes through or reflects off the beam-splitter again, and thus can be captured and imaged using the high speed camera, as shown in Fig. 1. This arrangement records luminosity from soot only in the piston bowl and the region directly above it.

For measurements in the squish volume, the 2-color setup is moved to the upper periscope window, which provides a view of the squish region, along with a small part of the piston bowl. To provide measurements of the squish region only, the small portion of the piston bowl region visible through the periscope port is blocked using a mask at the cylinder head window. High-speed soot luminosity imaging data are recorded both with and without the mask to study possible dynamics of soot at the edge of the piston bowl and in the squish region.

Note that the optical diagnostic measures soot within the bowl and the volume directly above it for piston bowl measurements (see Fig. 1). Measurement from the periscope view (with mask added over the visible piston bowl region) covers one of the diesel jets (out of 8) and hence is multiplied by 8 to represent the total soot volume in the squish region when these measurements are presented later in the paper.

EXHAUST SMOKE MEASUREMENTS

Exhaust smoke from the engine is measured using an AVL 415S variable sampling smoke meter. The smoke meter draws a known volume of the exhaust gas through a filter paper and the resulting paper blackening is used to determine the filter smoke number (FSN). All the exhaust smoke measurements reported in the paper are averaged over 10 fired engine cycles and have been corrected to account for skip-fired operation, so that the reported FSN values correspond to continuously fired operation.

RESULTS AND DISCUSSION

APPARENT HEAT RELEASE RATE ANALYSIS

Shown in Fig. 4a-c are apparent heat release rates (AHRR) for three main start-of-injection timings (SOI1) of -18° , -8° and 2 ATDC respectively, and for a range of post-injection start of injection timings (SOI2) in each case. The AHRR for a run with “main-injection only” also is provided for each SOI1 as a reference for comparison. AHRR data presented in the figure are ensemble-averaged over 20 fired engine cycles for all cases, and the various data series in the figure are color-coded to indicate the same SOI2 for all three SOI1 timings.

Figure 4 shows that the start of combustion is delayed as the main injection is retarded. The early main-injection timing (SOI1= -18° ATDC) has a larger peak AHRR during the premixed burn, indicating enhanced premixing of main-injection fuel for the early main-injections during the longer ignition delay. With the addition of a post, a second distinct bump in AHRR is evident, representing the additional combustion of fuel in the post injection for each case. The AHRR analysis shows that for the late main-injection (SOI1= 2° ATDC), post injections as late as SOI2 = 48° ATDC have significant heat release, while for an early main injection (SOI1= -18° ATDC), post injections injected after 44° ATDC do not have significant heat release.

For each of the main-injection timings, the magnitude of the post-injection AHRR peak (over that of “no-post” case) increases as the post is delayed, reaching a peak near SOI2 = 32° ATDC and thereafter decreasing as the delay is further increased. Such behavior is consistent with greater premixing of the post-injected fuel before combustion. Furthermore, for a given post-injection timing (e.g., SOI2= 32° ATDC), the AHRR peaks grow narrower and taller as the main-injection timing is retarded, suggesting that post-injected fuel with late main-injections is more premixed than with early main-injections.

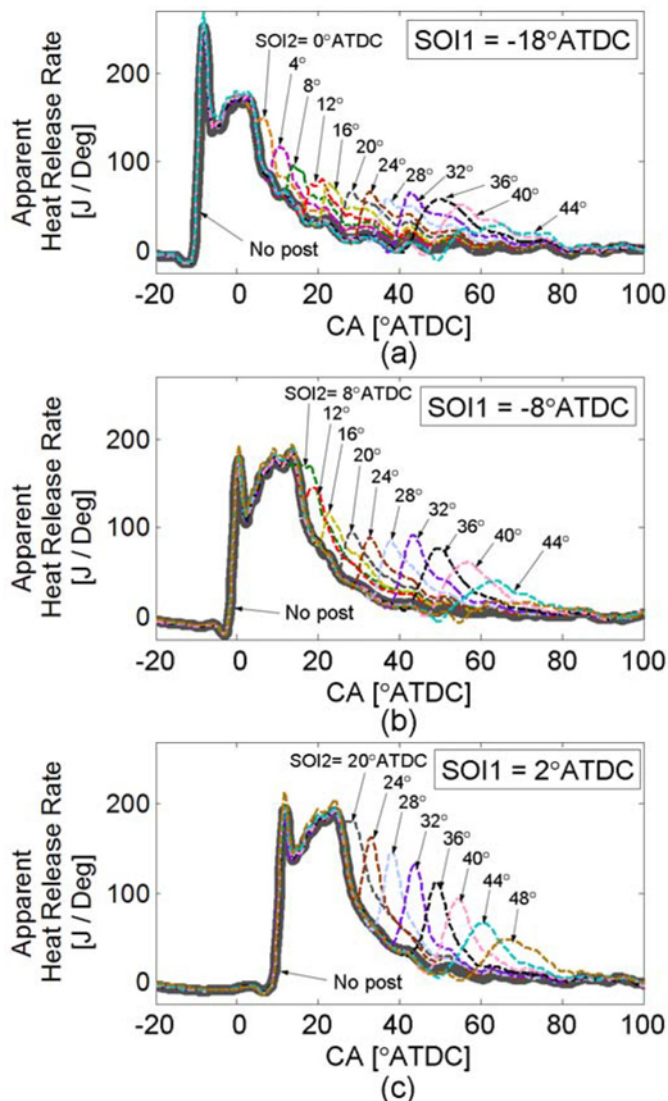


Figure 4. Apparent heat release rates for post-injection timing sweeps (dashed lines) with start of main injection at (a) -18° , (b) -8° , and (c) 2° ATDC. Traces for “main-injection only” operation (thick gray curves) also are presented in each case.

EFFECT OF MAIN AND POST INJECTION TIMING ON EXHAUST SMOKE

Figure 5 shows the variation of engine-out smoke emissions and gIMEP with timing of the post injection for three different cases: 1) Main injection + post, 2) main-injection-only and 3) a long main injection with same injected fuel mass as with post (see Fig. 3). These emission trends are explored for four different SOIs of main injection ranging from -18 to 2° ATDC. The smoke emissions are quite high because the engine is operated near the smoke limit, with a global equivalence ratio near 0.57 (see Fig. 2). This high-

sooting condition was intentionally chosen to examine the behavior of post-injections under difficult sooting conditions.

For a small delay between the main and post injections ($\text{SOI}_2 < 20^\circ$ ATDC), the post injection increases the engine-out smoke, even above that of the long main injection. Therefore, for these high-EGR operating conditions, a close-coupled post-injection has a detrimental effect on engine-out soot. For longer delays, where $\text{SOI}_2 > 20^\circ$ ATDC, the exhaust soot drops below the long main-injection levels, and in some cases, even drops below the soot produced by the standard main injection alone.

For an early main injection case ($\text{SOI}_1 = -18^\circ$ ATDC), the smoke emissions for later post injections asymptotically reach the same value as obtained for the main-injection only case. So for an early main-injection, the best a post injection can do is to reduce the exhaust smoke down to the value from main-injection only. This observation is consistent with previous studies [13,14] that concluded that the injections can be treated as independent. Of course, some interaction may still occur, but if so, its net effect on exhaust smoke is negligible for early injections.