A Parametric Study of Low-Temperature, Late-Injection Combustion in a HSDI Diesel Engine*

Dae CHOI**, Paul C. MILES**, Hanho YUN*** and Rolf D. REITZ****

A parametric study of automotive diesel combustion in a low-temperature, late-injection combustion regime is described. Injection pressure was varied from 600 – 1 200 bar, swirl ratio from 1.44 – 7.12, and intake temperature from 30 – 110°C. In-cylinder pressure records, heat release analysis, spatially-integrated soot luminosity, and images of the spatial distribution of combustion luminosity are employed to study the influence of these parameters on the combustion and soot formation/oxidation processes. Load points of 3 and 6 bar gross IMEP at 1 500 RPM and an O₂ concentration of 0.15 are considered. Increased injection pressure is found to enhance the early mixture formation process, resulting in increased peak apparent heat release, generally decreased soot luminosity, and modestly increased light-load soot oxidation rates. At lower injection pressures, more soot luminosity is observed from the squish volume. In contrast, variation of flow swirl impacts the latter half of the combustion process, and affects the initial combustion only slightly. An optimum Ricardo swirl ratio of roughly 3 is found for best moderate-load efficiency and soot oxidation. A marked reduction in early heat release rates and peak soot luminosity is observed with decreased intake temperature. Nevertheless, significant in-cylinder soot luminosity is observed even at the lowest intake temperatures, indicating that complete suppression of in-cylinder soot formation is difficult with the fuel injection and combustion system characteristics employed.

Key Words: Low-Temperature Combustion, Late-Injection, Chemical Kinetics, Turbulent Mixing, Experiment

1. Introduction

Low-temperature diesel combustion regimes (e.g. Refs. (1) – (6)) have received considerable attention due to their potential for reducing engine-out emissions to levels commensurate with future emission standards. One of these regimes, popularized by Nissan as “Modulated Kinetics (MK) Combustion”(2),(3), relies on high rates of cooled EGR, modest compression ratios, and late-

injection timing to achieve low in-cylinder temperatures and hence an extended ignition delay. High injection pressures are also employed to decrease the injection duration, thus maximizing the time available for fuel-air mixing prior to ignition. Additionally, unusually high swirl levels are adopted and are found to increase fuel economy — an important consideration due to the efficiency loss associated with the delayed combustion phasing.

Several attractive benefits are associated with adoption of this combustion regime, namely: the hardware employed (i.e. fuel injection equipment and bowl geometry) is that of a typical diesel combustion system, combustion timing is controlled by the fuel injection event, and low NOₓ and PM emissions can be obtained simultaneously.

The combustion process in this late-injection, low-temperature regime is often considered to be predominantly premixed. Examination of the rate-limiting processes in the first portion of the combustion event confirms this expectation. However, a variety of factors —
including observations of significant in-cylinder soot formation and scaling of the late-cycle heat release rate with engine speed\(^{(7)}\) — suggest that the early mixing is far from complete and that during the latter part of combustion (beyond approximately 50% burned) the rate of heat release is controlled by turbulent mixing processes. Accordingly, this type of combustion process might best be viewed as a high-EGR, retarded-injection calibration of conventional diesel combustion, rather than a predominantly pre-mixed, HCCI-like combustion process.

Nevertheless, we anticipate that intake temperature, injection pressure, and flow swirl will strongly influence the degree of initial premixing — and that the latter two will also influence turbulent mixing processes later in the combustion event. The primary objective of this study, then, is to study the influence of these parameters on the progress of the combustion event and on emissions formation, in particular, soot formation.

2. Engine and Operating Conditions

Measurements were made in an optically-accessible diesel engine with geometry typical of modern light-duty HSDI diesel engines intended for automotive application, i.e.: four-valves; a central, vertical injector; a concentric, re-entrant bowl; and a displacement of 422 cm\(^3\)/cylinder. The bore, stroke and compression ratio were selected to be typical of passenger car applications. The engine is equipped with a Bosch common-rail fuel injection system, capable of a maximum injection pressure of 1350 bar. Table 1 summarizes the main geometric characteristics of the engine and fuel injection system, including nozzle-specific details.

A schematic view of the engine and the engine operating conditions are shown in Fig. 1 and Table 2, respectively. Note that a realistic piston bowl geometry has been retained, which is an important requirement to ensure that flow structures influencing the mixture formation and oxidation processes are not significantly altered from those existing in a production engine. Although care was taken to ensure that the engine operation conditions are also representative of an operating production engine, some unavoidable differences exist: i.e. lower combustion chamber wall temperatures and a larger top ring land crevice volume in the skip-fired optical engine. To compensate for these differences, the engine is pre-heated to 361 K and is motored for 90 s prior to the start of data acquisition. Motoring permits stabilization of intake plenum pressures and preheats the combustion chamber walls.

The engine has two helical intake ports which allow the amount of swirl to be varied from roughly 1.5 – 3.5 by throttling the high-swirl port. To extend the range of achievable intake swirl, a shrouded valve was fitted in the high-swirl intake port, allowing a swirl ratio of 7.12 to be achieved.

EGR is simulated in the optical engine by adding N\(_2\) and CO\(_2\) to the intake air stream to achieve the desired [O\(_2\)] of 15%.

Data presented here were obtained at an engine speed of 1500 rpm and at loads of 3 and 6 bar gross IMEP, thus

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<td>Bore: 7.95 [cm]</td>
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<td>Stroke: 8.50 [cm]</td>
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<td>Comp. Ratio: 18.7</td>
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<td>Bowl Vol.: 17.3 [cm(^3)]</td>
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<td>Lip Dia.: 3.625 [cm]</td>
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<td>Squish Height: 0.067 [cm]</td>
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<th>Valve Events (0.15 mm lift)</th>
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<tr>
<td>EVO: 145 °CA ATC</td>
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<th>Fuel Injection Equipment</th>
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<td>Included Angle: 145°</td>
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<tr>
<td>Number of Holes: 6</td>
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<td>Hydro-erosion: 10%</td>
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<tr>
<td>Nozzle Style: Cylindrical Minisac (Bosch DLLA)</td>
</tr>
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Table 1 Engine geometry and features

| Speed: 1500 RPM             |
| Load: 3 and 6 bar IMEP\(_g\) |
| λ (In Cyl): 2.66 (3 bar load); 1.66 (6 bar load) |
| MAP: 1.2 bar                |
| T\(_\text{Cylinder}\): 88°C |
| [O\(_2\)]: 0.15              |
| SOI: Varied to match combustion phasing |
| Fuel: US D2 (CN47)          |
| P\(_\text{inlet}\): 600, 800, 1000, 1200 bar |
| Swirl Ratio (R\(_w\)): 1.44, 2.59, 3.77, 4.94, 7.12 |
| T\(_\text{inlet}\): 30, 50, 70, 90, 110°C |

Table 2 Operating conditions

Fig. 1 Schematic of optically-accessible test engine. Note the re-entrant combustion bowl geometry typical of automotive diesel engines
covering the typical and upper load range of this combustion regime. The parameters varied are detailed in the lower portion of Table 2. At the baseline operating condition of $P_{\text{int}} = 800$ bar, $R_s = 3.77$, and $T_{\text{in}} = 90^\circ\text{C}$, the start of injection (SOI) was $-0.25^\circ\text{CA}$ and injection ended at approximately $4.5^\circ\text{CA}$. As each parameter was varied, the injection timing was varied to fix the crank angle at which 10% of the cumulative heat release was obtained. Emissions of PM and NO$_x$ measured in a companion, non-optical engine$^7$ were both acceptably low, corresponding to a BSU of 0.6 and 0.3 g/kW-hr NO$_x$$^{11}$, at the 3 bar load and the operating conditions. Fuel economy measured at this injection timing was not perceptibly degraded from that seen at more typical diesel combustion injection timings.

3. Data Acquisition and Analysis

Cylinder pressure data are acquired with a water-cooled piezoelectric pressure transducer, and are pegged to the measured intake manifold pressure at $-180$ CAD. The injector is fitted with a sensor to monitor movement of the control rod, which is employed to determine the start and end of the injection event. A spatially-integrated measure of the total combustion luminosity is also obtained with a photodiode viewing the combustion chamber through a port in the cylinder liner (see Fig. 1). Because the luminosity striking the photodiode is attenuated with an OD = 3.0 neutral density filter, there is little response to natural combustion chemiluminescence and this signal is dominated by luminous soot. The cylinder pressure, control rod movement, and combustion luminosity are sampled with a resolution of 0.25 °CA.

Images of the spatial distribution of natural flame luminosity are also obtained to provide qualitative information on the progression of the combustion process. These images are acquired through the extended piston, employing an ICCD camera (Fig. 1). The camera has sufficient sensitivity to capture the natural chemiluminescence associated with the ignition process, and images obtained during the cool-flame portion of the apparent heat release are dominated by this natural chemiluminescence. Otherwise, the images are dominated by emissions from hot, radiant soot detected over a wavelength range of about 300–900 nm.

Prior to obtaining cylinder pressure and needle-lift measurements in the optical engine, the engine is motored for a 90 s warm-up period to pre-heat the combustion chamber walls and allow the intake plenum pressure to stabilize. Subsequently, the engine is skip-fired for 15 s, with fuel injection occurring on only one of every four engine cycles. After this 15 second firing period, pressure, control rod movement, and combustion luminosity are sampled for 50 fired cycles. Further analysis and presentation of these data is based on the average of these 50 cycles. Images of combustion luminosity are obtained shortly after commencement of firing operation, before significant window fouling by soot can occur.

The in-cylinder pressure data are analyzed to obtain the apparent heat release rate $dQ_{\text{app}}/d\theta$, following the procedure given by Heywood$^{8}$ and employing a constant specific heat ratio of 1.33. The effects of heat transfer and crevice flows are partially accounted for by subtracting the apparent heat release rate calculated from a corresponding motored pressure trace. Because the optical engine is skip-fired, this procedure is expected to provide a very accurate correction prior to the release of a significant amount of heat energy. Note that the heat release rate calculated is a net apparent heat release, and thus includes energy losses due to heat transfer as well as sensible energy increases associated with chemical heat release.

The apparent heat release rate is integrated from SOI until just prior to exhaust valve opening (EVO) to obtain the total, cumulative apparent heat release — which is normalized by the maximum value. Burn times and burn angles reported here are based on this normalized cumulative apparent heat release. For example, the 10–50% burn time is the duration in °CA between a normalized heat release of 0.1 and 0.5, while the specific crank angle at which the normalized heat release equals 0.1 is referred to as the 10% burned angle.

Following previous workers$^{2,3}$, we employ a work conversion efficiency $\eta_{\text{work}}$ to indicate the proportion of the apparent heat release $Q_{\text{app}}$ converted to gross indicated work:

$$\eta_{\text{work}} \equiv \left( D'\text{IMEP}_g \right) \int_{\text{SOI}}^{\text{EVO}} \left( dQ_{\text{app}}/d\theta \right) d\theta$$

(1)

$D$ represents the engine displacement. The gross indicated work can further be written as

$$D'\text{IMEP}_g = \int_{\text{SOI}}^{180} \left( P - P_{\text{no comb}} \right) (dV/d\theta) d\theta + D'\text{IMEP}_g, \text{no comb}$$

(2)

The first term on the right-hand-side (RHS) of Eq. (2), normalized by the cumulative apparent heat release, represents an ideal $\eta_{\text{work}}$ that would be found if no heat or mass losses occurred over the compression and expansion strokes of a non-combusting cycle.

Finally, some discussion of the interpretation of the spatially-integrated soot luminosity detected by the photodiode is merited. The detected soot luminosity is in general dependent on the spatial distribution of the soot in the cylinder, the soot volume fraction, the optical density of the medium through which the luminosity passes, and the
soot temperature. Clearly, the luminosity cannot be interpreted as a quantitative marker of the in-cylinder soot mass. Nevertheless, for optically thin media, and similar peak soot temperatures (similar adiabatic flame temperatures), we anticipate that the behavior of the soot luminosity can serve as a qualitative indicator of the formation and oxidation of in-cylinder soot. Strong correlation recently observed between spatially-integrated soot luminosity at EVO and engine-out soot in a heavy-duty engine\textsuperscript{9} provide support for this perspective.

4. Results and Discussion

4.1 Injection pressure effects

The effect of injection pressure on the combustion development was investigated at the base swirl ratio $R_s = 3.77$ for injection pressures of 600, 800, 1000, and 1200 bar. In-cylinder pressure traces measured at both the 3 and 6 bar load conditions are shown in Fig. 2. As noted previously, these data were obtained with SOI adjusted to fix the 10% burned location at each load. SOI was the same for the 3 and 6 bar loads at $P_{inj} = 800$ bar. Due to the higher combustion chamber surface temperatures, the near-TDC cylinder pressure is slightly higher for the 6 bar load. Figure 2 clearly demonstrates that the higher injection pressures promote more rapid heat release, evidenced by the higher peak cylinder pressures observed at the higher injection pressures, as well as the larger $\frac{\partial P}{\partial \theta}$ seen early in the combustion process.

The trends observed in Fig. 2 are reinforced by the corresponding $\frac{dQ_{app}}{d\theta}$ and cumulative heat release presented in Fig. 3. For both engine loads, the peak heat release rate increases with injection pressure, more markedly so for the 6 bar load. As discussed in the introduction, we find the rate of early heat release ($\approx 10$ – 50\% burned) is dominated by the rate of chemical reactions (“modulated by kinetics”). The observed injection pressure dependency of the early heat release rate is not inconsistent with this view. At higher injection pressures a greater amount of fuel/air premixing occurs during the ignition delay period. A greater quantity of fuel prepared for combustion, as well as mixture equivalence ratios closer to unity — promoting more rapid reaction — could explain the increased peak heat release at higher injection pressures. Nevertheless, we cannot discount the possibility that at high injection pressures large instantaneous turbulent mixing rates may also be promoting more rapid heat release.

Beyond 11 – 12°CA ($\approx 50$ – 60\% burned), little effect of injection pressure can be seen in the instantaneous $dQ_{app}/d\theta$ of Fig. 3. Accordingly, the cumulative heat release curves are nearly parallel at this time. Increased injection pressure thus predominantly influences the early heat release, while having little impact on late-cycle combustion. Nevertheless, the overall heat release is more advanced at higher injection pressures, and some increase in the work conversion efficiency is to be expected. The effectiveness of increased injection pressure on advancing the overall heat release is diminishing, however, as $P_{inj}$ is increased. By evaluating $\eta_{work}$, we find that injection pressures higher than 1000 bar are unlikely to provide significant additional benefit to the cycle efficiency, and may be detrimental at lower loads.

Figure 4 depicts the variation in combustion luminosity as injection pressure is varied. At the 3 bar load, a smooth variation in the spatially-integrated natural luminosity is observed, and exhibits a monotonic decrease with increased injection pressure. The elevated cylinder pressures observed in Fig. 2 for the higher injection pressures are indicative of higher bulk cylinder temperatures, and...
suggests that fuel in the head of the jet that is transported with increasing injection pressure, dropping from about 80 to 20% of its peak is found to decrease modestly rate. The time required for the soot luminosity to decay be employed as a qualitative measure of the soot oxidation rates inferred from the decay in natural luminosity. The 80–20% decay times tend to increase slightly with injection pressure, from near 25 °CA at 600 or 800 bar to 30 °CA at the higher pressures. Thus, increased injection pressure does not appear beneficial for late-cycle soot oxidation at the higher load. Note that at the higher load, the 80–20% oxidation period is significantly more retarded (about 35–60 CAD) than at the lower load (about 20–30 CAD). In the high-load case, significantly more time has elapsed since EOI, and injection-generated turbulence has likely decayed to a large extent. In this light, the difference in the luminosity decay between the 3 and 6 bar loads is not unexpected.

4.2 Swirl ratio effects

The effect of swirl ratio on the combustion development was investigated in a manner similar to that employed to study the effect of injection pressure: SOI was identical for the 3 and 6 bar loads at $R_s = 3.77$, and at other swirl ratios was adjusted to maintain the same combustion phasing observed at $R_s = 3.77$. The injection pressure was held fixed at 800 bar. In-cylinder pressure traces measured at both load conditions are shown in Fig. 6. Unlike the variation in cylinder pressure as $P_{inj}$ was varied, relatively little variation is observed with $R_s$. Note, however, that the near-TDC cylinder pressure is lower at the higher swirl ratios due to increased heat loss during the compression stroke.

Changes in the combustion process with swirl level are more clearly observed by examining the characteristics of the heat release rates presented in Fig. 7. At the 3 bar load, differences in the peak $dQ_{op}/d\theta$ are relatively minor, and the peak rates achieved are non-monotonic in $R_s$. This cannot be interpreted as evidence that an optimum swirl ratio exists for achieving the best mixture preparation, as such behavior could be due to a competition between two different processes. For example, high swirl may provide better mixture preparation, but the benefits into the squish volume by the reverse squish flow has been mixed to sufficiently lean equivalence ratios ($\phi < 2$) at the higher injection pressure to avoid soot formation.

At the 6 bar load, the variation in luminosity seen in Fig. 4 is more complex. For injection pressures of 800 bar and above, the evolution of the luminosity is similar to that seen at the 3 bar load, although significantly higher luminosity is observed. With $P_{inj} = 600$ bar, however, the soot luminosity initially increases much more slowly, and reaches a peak later than that seen at the higher injection pressures. This behavior could be associated with a difference in the overall soot formation rate, or in the details of the spatial distribution of the soot luminosity. Images of the soot luminosity were not obtained at this load condition.

Unlike the lighter load, at the 6 bar load increased injection pressure does not increase the late-cycle soot oxidation rates inferred from the decay in natural luminosity. The 80–20% decay times tend to increase slightly with injection pressure, from near 25 °CA at 600 or 800 bar to 30 °CA at the higher pressures. Thus, increased injection pressure does not appear beneficial for late-cycle soot oxidation at the higher load. Note that at the higher load, the 80–20% oxidation period is significantly more retarded (about 35–60 CAD) than at the lower load (about 20–30 CAD). In the high-load case, significantly more time has elapsed since EOI, and injection-generated turbulence has likely decayed to a large extent. In this light, the difference in the luminosity decay between the 3 and 6 bar loads is not unexpected.
Fig. 6  The effect of swirl ratio on cylinder pressure

Fig. 7  The effect of swirl ratio on the apparent heat release rate and the cumulative heat release

May not be seen due to lower TDC temperatures associated with the higher heat losses, which slow the chemical reaction rates. Through approximately 11°CA (≈ 50% burned) there is little difference in the cumulative heat release as the swirl level is varied. Beyond this time, however, the cumulative heat release curves diverge, with higher swirl ratios generally providing the more rapid heat release. Much of this divergence occurs between roughly 11 and 18 CAD (≈ 50–75% burned), at which time the rate of heat release is expected to be strongly influenced by turbulent mixing processes.

At the 6 bar load the initial behavior of the heat release rates are similar to the behavior seen at the 3 bar load. As seen in Fig. 7, the high-load heat release rates also exhibit relatively small variations in the peak heat release that are non-monotonic in swirl ratio. Likewise, the cumulative heat release varies little with \( R_s \) through 11°CA. In contrast to the 3 bar load results, the divergence in the cumulative heat release beyond this point is no longer monotonic with swirl ratio and varies in a complex manner. For example, at 20°CA, the cumulative heat release for \( R_s = 1.44 \) and \( R_s = 4.94 \) are nearly equal and are considerably lower than that seen for the highest heat release rates at \( R_s = 2.59 \) and \( R_s = 7.12 \). By 40°CA, however, the cumulative heat release at \( R_s = 3.77 \) and \( R_s = 4.94 \) is largest, having surpassed the heat released at \( R_s = 7.12 \), while the heat released at \( R_s = 2.59 \) has lagged behind.

This behavior can be examined more conveniently by considering the variation with \( R_s \) of the burn times required for various portions of the cumulative energy release, as shown in Fig. 8. Prior to 50% burned, little difference in the burn times is observed, as is evident from the cumulative heat release curves of Fig. 7. The period required to release 50–70% of the energy is less for \( R_s = 2.59 \) and \( R_s = 7.12 \), consistent with the levels of the cumulative heat release near 20°CA discussed above. The major differences are seen in the late-cycle period, however, where the 70–90% burn angles are significantly shorter for the intermediate swirl ratios \( R_s = 3.77 \) and \( R_s = 4.94 \).

Accelerated late-cycle burning improves the work conversion efficiency \( \eta_{\text{work}} \) as shown in Fig. 9. Note that the highest \( \eta_{\text{work}} \) is seen at \( R_s = 2.59 \) and \( R_s = 3.77 \), rather than \( R_s = 4.94 \), as might be expected based on the burn durations shown in Fig. 8. The explanation for this difference lies in the additional heat losses associated with the higher swirl levels, as can be inferred by the lower near-TDC pressures of Fig. 6. Indeed, the ideal \( \eta_{\text{work}} \) (Eq. (2)) exhibits behavior that is consistent with the burn durations. However, at the higher swirl ratios any benefits achieved in late-cycle mixing are overwhelmed by increased heat losses. Likewise, \( \eta_{\text{work}} \) determined at the 3 bar load decreases monotonically with swirl ratio. Although at this load the late-cycle heat release advances monotonically with swirl ratio, the improvement in combustion phasing is insufficient to compensate for the increased heat losses.
Fig. 9 The effect of swirl on the work conversion efficiency $\eta_w$. The scattered points indicate the results from multiple tests.

The variation with $R_s$ of the spatially-integrated soot luminosity is depicted in Fig. 10 for the 3 and 6 bar loads. Like the effect of increased $P_{inj}$ seen in Fig. 4, increased swirl monotonically reduces the peak soot luminosity observed at low load. The 80–20% decay times follow suit, decreasing from about 14 °CA at $R_s = 1.44$ to 8 °CA at $R_s = 7.12$. The bulk of this decrease, 4 °CA, occurs between $R_s = 1.44$ and $R_s = 2.59$. At the higher, 6 bar load a much more complex variation is seen. The highest levels of soot luminosity are seen at the intermediate swirl ratios $R_s = 3.77$ and $R_s = 4.94$. The decay of soot luminosity is very rapid at these swirl ratios, however, such that beyond roughly 60 °CA, the soot luminosity is less than that observed at $R_s = 7.12$, despite significantly lower peak luminosity at the higher swirl ratio. Note that the rapid decay of soot luminosity at the intermediate swirl ratios — and the relatively slow decay of the low and high swirl ratio luminosity — correlates well with the burn durations shown in Fig. 8. This correlation suggests that the rapid mixing responsible for the increased heat release is also the mechanism responsible for soot oxidation.

The influence of increased swirl on the development of the combustion process is examined via images of the spatial distribution of combustion luminosity at the 3 bar load condition shown in Fig. 11. There is little influence of swirl on the ignition process, as evidenced by the images obtained during the period of low-temperature heat release near 5 ºCA. At later crank angles, lower swirl results in a greater volume of luminous soot and higher peak intensities, in much the same manner as seen for the lower injection pressure of Fig. 5.

4.3 Intake temperature effects

Intake temperature $T_{in}$ (and, hence, TDC temperature) is expected to influence the combustion and emissions formation processes in two distinct ways. First, at lower temperatures the ignition delay is extended and enhanced fuel-air premixing will occur. Second, at lower initial reactant temperatures, the adiabatic flame temperature of a fuel parcel which has mixed to a given equivalence ratio is reduced. Thus, the trajectory of a fuel element on the $\phi$–$T$ plane (e.g. Ref. (10)) is generally displaced away from the zones of soot formation.

To investigate the effect of $T_{in}$ on the combustion development, temperatures of 30, 50, 70, 90, and 110°C were employed at the base $R_s$ and $P_{inj}$ of 3.77 and 800 bar, respectively. Multi-dimensional numerical simulation indicated that peak TDC temperatures varied from 805 K to 938 K for this range of $T_{in}$. Like the tests involving $P_{inj}$ or $R_s$ variations, SOI was adjusted to maintain a constant combustion phasing by fixing the 10% burn angle. In conducting these tests, the mass flow into the engine was maintained fixed, such that the in-cylinder gas density did not vary.

Figure 12 depicts the variation in $dQ_{app}/d\theta$ and cumulative heat release with $T_{in}$. The peak $dQ_{app}/d\theta$ increases monotonically with increasing $T_{in}$, despite the smaller mixing time available for mixing. The smaller mixing time is associated with both a shortened ignition delay as well as a decreased time period from ignition until the an-
angle of peak heat release. The decrease in the latter period is evident in the advanced phasing of the main heat release at the higher temperatures, and is evidence of the kinetics-controlled nature of the combustion process at this time. Beyond a cumulative heat release of approximately 60–70%, the heat release becomes more advanced for the lower intake temperatures. We believe this is predominantly due to lower heat losses associated with the lower in-cylinder temperatures.

Figure 13 presents the variation in the spatially integrated soot luminosity observed for the various intake temperatures. A striking dependency of the soot luminosity on $T_{in}$ is clearly apparent, likely due to both lower soot temperatures as well as reduced soot formation. It is noteworthy, however, that soot luminosity is clearly observed even at $T_{in} = 30\, ^\circ C$, where the TDC temperature is only 805 K. That is, with typical fuel injection system parameters, avoiding in-cylinder soot formation will be difficult — even with very low near-TDC temperatures. Also interesting is the near constant 80–20% luminosity decay times observed for the various $T_{in}$ (9.7±0.5 °CA). The near constancy of the decay times is consistent with the postulate that soot oxidation is not limited by chemical kinetic processes, but rather by turbulent mixing — which is not expected to vary significantly with the changes made in $T_{in}$ or SOI.

The spatial development of the combustion process is illustrated in Fig. 14, which presents images of the natural combustion luminosity which have been normalized to the same luminous camera gain. The ignition and soot formation processes clearly differ little with $T_{in}$. Like the images presented earlier for various swirl ratios and injection pressures, the high luminous soot operating conditions (high $T_{in}$) generally exhibit a broader distribution of soot luminosity, but evidence of dramatic changes in the development of the luminous soot distributions is not present.

5. Summary and Conclusions

A parametric study of HSDI diesel combustion in a low-temperature, late-injection combustion regime is performed, in which $P_{inj}$ was varied from 600–1 200 bar, $R_s$ from 1.44–7.12, and $T_{in}$ from 30–110 °C. Two load points characterized by a gross IMEP of 3 and 6 bar at 1 500 RPM are considered. A re-entrant bowl geometry and a common-rail injection system were employed.

Increased injection pressure is found to enhance the early mixture formation process, resulting in increased peak apparent heat release and generally decreased soot luminosity. Soot oxidation rates, inferred from the relative decay rates of the integrated luminosity, are modestly influenced by injection pressure only at the 3 bar load. The spatial distribution of soot luminosity is characterized by increased luminosity observed from the squish volume at the lower injection pressures.

In contrast, variation of flow swirl impacts the lat-
ter half of the combustion process, and affects the initial combustion only slightly. At the 3 bar load, increased swirl monotonically advances the late-cycle apparent heat release rate, reduces peak soot luminosity, and increases soot oxidation rates. Due to increased heat losses the work conversion efficiency decreases, however. At the 6 bar load, both the late-cycle heat release and the soot luminosity suggest that an optimum swirl ratio exists, at which the work conversion efficiency and soot oxidation rates are maximized.

Tests conducted at variable intake temperature indicate a marked reduction in early heat release rates and peak soot luminosity with decreased intake temperature. Soot luminous decay rates are unchanged. Significant soot luminosity is observed in the cylinder even at the lowest intake temperatures, indicating that complete suppression of in-cylinder soot formation is difficult with the injection system characteristics employed.

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References