Spray Targeting to Minimize Soot and CO Formation in Premixed Charge Compression Ignition (PCCI) Combustion with a HSDI Diesel Engine

Sangsuk Lee and Rolf D. Reitz
University of Wisconsin - Madison

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ABSTRACT

The effect of spray targeting on exhaust emissions, especially soot and carbon monoxide (CO) formation, were investigated in a single-cylinder, high-speed, direct-injection (HSDI) diesel engine. The spray targeting was examined by sweeping the start-of-injection (SOI) timing with several nozzles which had different spray angles ranging from 50° to 154°. The tests were organized to monitor the emissions in Premixed Charge Compression Ignition (PCCI) combustion by introducing high levels of EGR (55%) with a relatively low compression ratio (16.0) and an open-crater type piston bowl.

The study showed that there were optimum targeting spots on the piston bowl with respect to soot and CO formation, while nitric oxide (NOx) formation was not affected by the targeting. The soot and CO production were minimized when the spray was targeted at the edge of the piston bowl near the squish zone, regardless of the spray angle. Targeting this spot is believed to enhance the pre-ignition mixing of air and the spray effectively with the help of the squish flow. The results from the narrow angle nozzles (50° and 85°) indicated that soot could be optimized when the spray was targeted at the bottom of the piston bowl which provided the longest spray travel distance. However, CO emission increased but was significantly reduced when the spray was targeted at the inner surface of the bowl with a corresponding increase in soot emission. In the standard diesel combustion regime, the soot and CO increased as SOI was retarded, and the minimum soot was achieved with SOI of about -20 degree ATDC. This SOI timing provides a rough boundary between conventional diesel and PCCI combustion as seen from the heat release rate data.

INTRODUCTION

Several recent studies [1-5] have demonstrated that the PCCI combustion strategy can be a practical solution in diesel engines to satisfy stringent future emission regulations since the strategy achieves a good soot and NOx trade-off with an acceptable impact on fuel economy and carbon monoxide emissions. However, the key parameters that determine the soot and NOx trade-off have not been completely identified.

The objectives of this study were to achieve low emission PCCI operation and to identify key parameters that enable the simultaneous reduction of soot, NOx, and carbon monoxide emissions.

Six nozzles with different spray included angles ranging from 50 to 154 degrees were examined to identify the effects of the spray targeting on emissions. The results from the six different spray included angles were evaluated for both PCCI and standard diesel combustion. PCCI combustion was differentiated from standard diesel combustion by noting its combustion characteristics.

Late injection or MK combustion, which is a well-established low emissions strategy, is not considered in this study since it has similar combustion characteristics to PCCI due to its long ignition delay. Thus, MK combustion may be considered as a special case of PCCI combustion. It was hard to evaluate the characteristics of MK combustion with different spray angles since the stability of MK combustion was strongly affected by only slight changes of the injection timing and EGR.

EXPERIMENTAL SETUP

The present test engine consists of a Hydra single-cylinder engine from Ricardo Research and a single-cylinder version of a production Fiat cylinder head which was designed for a 2.4L five-cylinder engine. The cylinder head is equipped with double over-head camshafts and two exhaust ports and two intake ports, which consist of a helical port and a directed port to control the swirl ratio. The original Fiat piston was...
Table 1. The Engine Specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Type</td>
<td>4 valve DI diesel</td>
</tr>
<tr>
<td>Bore x Stroke</td>
<td>82.0 x 90.4 mm</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>16.0:1</td>
</tr>
<tr>
<td>Displacement</td>
<td>477 cm³</td>
</tr>
<tr>
<td>Piston Geometry</td>
<td>Open-Crater-bowl</td>
</tr>
<tr>
<td>Intake Ports</td>
<td>1 swirl, 1 tumble</td>
</tr>
<tr>
<td>Swirl ratio (at IVC)</td>
<td>1.83</td>
</tr>
<tr>
<td>Bowl diameter</td>
<td>48mm</td>
</tr>
<tr>
<td>IVO</td>
<td>10° BTDC</td>
</tr>
<tr>
<td>IVC</td>
<td>38° ABDC</td>
</tr>
<tr>
<td>EVO</td>
<td>38° BBDC</td>
</tr>
<tr>
<td>EVC</td>
<td>8.5° ATDC</td>
</tr>
</tbody>
</table>

Table 2. Common rail system specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injector type</td>
<td>Electro-hydraulically</td>
</tr>
<tr>
<td></td>
<td>controlled injector</td>
</tr>
<tr>
<td>Nozzle type</td>
<td>Dual guided VCO</td>
</tr>
<tr>
<td>Flow Number</td>
<td>400cm³/30s @100bar</td>
</tr>
<tr>
<td>Number of nozzle holes</td>
<td>6</td>
</tr>
<tr>
<td>Hole diameter</td>
<td>0.154mm, 0.133mm</td>
</tr>
<tr>
<td>Spray included angle</td>
<td>50°, 85°, 120°, 130°, 140°</td>
</tr>
</tbody>
</table>

replaced with an open-crater-type bowl piston which was designed using CFD combined with a genetic algorithm (GA) optimization [6] to realize the NADI concept [7]. Specifications of the engine are indicated in Table 1.

The injection system used for this study was a Bosch common-rail injection system. The specifications of the common rail injection system are summarized in Table 2. The nozzles had eight holes except for the nozzles with narrow spray included angle (50 and 85 degree) which had 6 holes due to machining restrictions. The difference in the number of nozzle holes affects the level of emissions [8-10]. However, the focus of the present study was on trends to evaluate the effect of spray targeting in qualitative manner.

Intake air was supplied from pressurized building air to simulate a turbocharger. The flow rate was measured with a critical orifice system. EGR flow was driven with the pressure difference between the exhaust and the intake surge tanks and was controlled by regulating the openings of the valves. The emission data recorded during the experiments included gaseous and particulate emissions. The gaseous emissions, including NO, NOx, CO, Exhaust CO2, were measured with a Nicolet Rega model 7000, FTIR emissions analyzer. Intake CO2 was assessed with a Horiba PIIR-9000 infrared gas analyzer. Exhaust smoke levels were sampled with a Bosch RTT100 instrument. The instrument measures visual opacity and converts it into mass concentration in mg/m³ through the use of internal tables. The mass concentration was converted into specific soot emission in g/kW-hr. Measurement of the concentration ranges from 10 to 1966 mg/m³ with a 1 mg/m³ resolution. The lower bound of the measuring range, 10 mg/m³ mass concentration, corresponds to around 0.05 g/kW-hr at the operating conditions in this study. Special care is required when the levels of soot achieved in PCCI combustion are compared to the lower bound of the unit. However, this does not impact the results of this study since the main objective of the study is to investigate optimum ways to achieve low emissions PCCI combustion in qualitative manner. The quantitative values achieved in the study are of less importance since they depend strongly on the operating conditions, including equivalence ratio, injection pressure, EGR, and piston geometry.

Cylinder pressure was measured with a Kistler 6125A piezo-electric pressure transducer and the measured signal was amplified and converted into a voltage with a Kistler 5010 charge amplifier. The output signal from the charge amplifier was sampled with a National Instruments AT-MIO-16E-1 data acquisition board every quarter degree crank angle. The pressure traces were averaged over 300 cycles to compensate for cycle-by-cycle variations. The IMEP includes pumping loss during the gas exchange process by considering the whole engine cycle. Heat loss was not considered and the specific heat was assumed to be constant, 1.33, when the heat release rate was evaluated.

OPERATING CONDITIONS

The operating conditions for this study are summarized in Table 3. The conditions correspond to representative operation for small bore HSDI engines in the New European Drive Cycle (NEDS) [5].
Table 3 Operating Condition Summary

<table>
<thead>
<tr>
<th>Operating Parameters</th>
<th>Test Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Speed</td>
<td>2000 rev/min</td>
</tr>
<tr>
<td>Test Mode</td>
<td>Constant IMEP (IMEP = 4.8 bar)</td>
</tr>
<tr>
<td>Intake Pressure</td>
<td>151 kPa</td>
</tr>
<tr>
<td>Exhaust Pressure</td>
<td>169 kPa</td>
</tr>
<tr>
<td>EGR</td>
<td>55%</td>
</tr>
<tr>
<td>Intake Air Temperature</td>
<td>90 °C</td>
</tr>
<tr>
<td>Injection Pressure</td>
<td>150 MPa</td>
</tr>
<tr>
<td>Start of Injection (SOI)</td>
<td>-43° ~ 0° ATDC</td>
</tr>
<tr>
<td>Swirl Ratio</td>
<td>1.85</td>
</tr>
<tr>
<td>Oil and Coolant Temperature</td>
<td>80 °C</td>
</tr>
<tr>
<td>Fuel Temperature</td>
<td>55 °C</td>
</tr>
</tbody>
</table>

A high level (55%) of EGR was chosen to achieve PCCI combustion which required a long ignition delay due to the early start-of-injection timing used to promote mixing. The high level of EGR is also required to reduce NOx emissions significantly, as mandated in future emission regulations. The baseline equivalence ratio in this study was 0.55 and this condition allowed the effects of PCCI combustion and spray angles to be evaluated effectively.

The start-of-injection timing was advanced to the timing at which the spray was targeted to stay within the piston bowl. This was done to minimize liner wetting and oil dilution problems. Advancing the injection timing with the use of narrow spray angle nozzles was limited by the peak pressure rise rate which was recommended not to exceed 2 MPa/degree [11]. In addition, the start-of-injection timing could not be retarded beyond TDC since the engine could not keep its power as the timing approached TDC. The injection pressure was set at the maximum recommended value of 150 MPa which also provided better soot and NOx trade-off results [5].

DIFFERENTIATING PCCI COMBUSTION FROM STANDARD DIESEL COMBUSTION

PCCI combustion was assumed when the start-of-injection timing was advanced moderately relative to standard diesel combustion. However, the definition relies on subjective choice since there is no precise demarcation boundary between PCCI and standard diesel combustion. In this study, PCCI and standard diesel combustion were distinguished by considering their combustion characteristics. PCCI combustion was identified with short burn durations and an early start-of-combustion timing. For example, the burn duration in this study was about 4 crank angle degrees and the combustion starts around -11 degree ATDC in PCCI combustion, as will be discussed later.

Standard diesel combustion consists of a premixed combustion phase and a mixing-controlled combustion phase [12]. The fuel and air mixture in the premixed phase reaches a flammable limit during the ignition delay period, and ignition and combustion are controlled by the thermodynamic properties and chemical reactions in the fuel-air mixture. As a result, burning rates in the premixed combustion phase are relatively faster than those in the mixing-controlled phase, and the duration is determined by the concentration of reactants, and the amount of fuel mass that is premixed. On the other hand, reactions in the mixing-controlled phase are determined by the mixing process of fuel and air which provides a proper mixture to be burnt. Consequently, the duration of the mixing controlled phase is controlled by the mixing process which is dominated by the fuel spray development, mixing of the fuel and air, and chemical reactions.

BURN DURATION ANALYSIS

Burn durations for the six spray angles considered in the experiments (see Table 2) are shown in Figure 2 with respect to the start-of-injection timing. The burn durations, which are defined by the duration between the 10% to 90% points of the accumulated heat release rate, are about 4 crank angle degrees when the injection timing is advanced earlier than -25 degrees ATDC. This is a convenient border definition between PCCI and standard diesel combustion. The durations increase as the injection is retarded beyond -25 degrees ATDC, and engine operation can be considered to belong to standard diesel combustion. The combustion duration under standard diesel combustion conditions becomes longer with retarded injection since the proportion of the mixing-controlled phase becomes bigger due to the shorter ignition delay.

According to results by Siebers [13], the higher air temperature and pressure accompanying retarded injection timings decreases the flame lift-off length and
results in less air within the spray and longer mixture residence times in the mixing-controlled phase. Different slopes in the standard diesel combustion regime with the different spray angles imply that the spray angle affects the mixture preparation and mixing significantly. For example, the slope for the 50 degree spray angle case in Figure 2 is 1.45 degree of burn duration per degree of injection timing while the slopes for other cases are from 0.36 to 0.55. The steeply increasing slope for the 50 degree spray angle case indicates that the spray plume for that nozzle does not penetrate as far as the other sprays and the poorer mixing slows the combustion process.

Spray travel distances of the 50, 85, 120, and 140 degree spray angle cases are compared with respect to the start-of-injection timing in Figure 3. The spray travel distance in this plot is defined by the distance between the injector nozzle exit and the location where the center of the spray plume is aligned so as to hit the piston surface at the injection timing. Though the definition contains uncertainty, the estimation is satisfactory for the purpose of nozzle-by-nozzle comparison since the spray reaches the piston surface within about a crank angle degree, and the injection lasts only a few crank angle degrees. Geometric considerations indicate that the 50 degree spray angle case travels less than half of the distance than the other nozzles do before the fuel collides with the piston wall in standard diesel combustion, as shown in Figure 3.

10 to 50% burn durations, 50 to 75% burn durations, and 75 to 90% burn durations for the 120 degree spray angle case are compared with the 10 to 90% burn durations in Figure 4 to evaluate the relation between the premixed burn durations and the mixing-controlled phase. The durations of the 10% to 50% burn in Figure 4 become slightly shorter as the injection is retarded. Higher cylinder temperature and pressure, accompanied with the retarded injection may accelerate the combustion which is assumed to take place in the premixed regime. However, the durations of the 75% to 90% burn duration become longer when the injection timing is retarded beyond -25 degree ATDC. The mixing controlled phase, which extends the combustion duration due to its slower burn rate, starts affecting the combustion at SOI -25 degrees ATDC and transitions from PCCI combustion. Interestingly, the 50 to 75% burn durations becomes longer starting at -20 degrees ATDC, which is a little later than that for the 75 to 90% burn durations. This implies that 75% of the fuel was burnt at least in the premixed regime at SOI -20 degrees ATDC. In conclusion, as also observed by Miles et al. [14], the burn duration analysis provides a good indication of when the mixing controlled phase starts to become influential and how PCCI combustion can be differentiated from standard diesel combustion.

**START OF BURN ANALYSIS**

The timing of the start-of-combustion may be another indicator of the existence of premixed combustion, in addition to the combustion duration. The ignition in the premixed regime is controlled by thermodynamic properties like temperature and pressure within the cylinder. The ignition in this study is expected to start at the same crank angle when cases are within the PCCI combustion regime, since the engine was operated at the same intake and exhaust pressures, intake temperature, and EGR rate. Figure 5 reports the start of combustion timings, which are defined by the crank angle at which the accumulated heat release reaches 10%. The combustion starts at around -11 degrees ATDC in PCCI combustion (except for the 85 degree spray angle case). On the other hand, the start-of-combustion is retarded linearly with retarded injection timing in the standard diesel combustion regime. The temperature and pressure in the standard diesel combustion regime exceed the thresholds to trigger high temperature reactions before the spray has
formed a flammable mixture. Thus, the start of combustion in standard diesel combustion is controlled mostly by the injection timing. The spray angles do not have a significant effect on the start-of-combustion timing, even though they change spray trajectories and spray travel distances (i.e., vaporization times), which the spray travels before impinging on the piston surface. This implies that the trajectory and travel distance dependence on the spray angle do not change the formation process of the rich premixed mixture, as conceptualized by Dec [15], but does affect the amount of air entrained within the spray. The deviations of the 10 to 90% burn durations in Figures 2 and 5 reflect the amount of air entrained and its dependence on the spray angles.

The start-of-combustion timing for the 85 degree spray angle case keeps advancing with injection timing advance, as shown in Figure 5, while the other cases stay around -11 degrees ATDC in the PCCI combustion regime. It is hard to explain the reason at this moment since the average temperature and pressure in the cylinder evidently do not reach the thresholds to trigger the high temperature reactions.

One possible reason could be due to the piston surface temperature distribution. The temperature distribution on the piston bowl surface may affect the start of ignition by transferring heat to the mixture near hot spots. As shown in Figure 6, the spray of the 85 degree spray angle case is directed toward the deep bottom of the piston bowl where most of the combustion takes place. In this case, the piston surface is likely to have a relatively hot spot in this location, while the other spray angles target spots which deviate from this hot spot. Ladommatos et al. [16] showed that the piston temperature significantly affects engine emissions by controlling the mixture formation in vicinity of the piston surface in the case of a small bore diesel engine. However, their results do not give a direct explanation for an advance of the start-of-combustion timing.

**EFFECT OF INJECTION TIMING ON EMISSIONS**

Injection timing is a traditional and widely accepted parameter to control engine-out emissions. The present six different spray angle nozzles were examined to evaluate the effects of injection timing on engine-out emissions in each case. The injection timings were swept from a timing as early as could be achieved without damaging the engine to around TDC. The results shown in this section belong to the same data set discussed previously.

**SOOT AND CARBON MONOXIDE EMISSIONS**

Soot and carbon monoxide emissions for the six different spray angle nozzles are indicated in Figure 7 and Figure 8 with respect to the injection timing. Both soot and carbon monoxide emissions in the PCCI combustion regime, (i.e., when the SOI is earlier than -25 degree ATDC), are seen to depend strongly on the spray angle, in addition to the injection timing. It is hard to describe a trend regarding the influence of spray angle since each spray angle shows a different behavior. For example, the 120 degree spray angle case shows peak soot emission at SOI -30 degrees ATDC, while the soot emissions for the 140 degree spray angle start to increase when the injection is advanced beyond that SOI. The 85 degree spray angle case does not show significant variation in soot emissions. However, the carbon monoxide emission for the 85 degree spray angle increases tremendously as the injection is advanced.
The results imply that a new approach is needed to analyze the emissions in the PCCI regime. The combustion details and its products in the PCCI regime should be determined by the equivalence ratio distribution within the premixed mixture, which may not be directly linked to the injection timing. The equivalence ratio distribution results from an interaction between the spray and in-cylinder flow before the combustion starts. Thus, the results will be interpreted with respect to the spray targeting, which links the spray interaction with the injection timing and spray angles.

On the other hand, the soot and carbon monoxide emissions in the standard diesel combustion regime increase with different slopes depending on the spray angles as the injection timing is retarded. The carbon monoxide emissions in the standard diesel combustion regime are less sensitive to the injection timing than the soot emission (except for the 50 degree nozzle). The trend within the standard diesel combustion regime is consistent with the results of Siebers [13] and Pickett [17] who investigated emissions in a constant volume chamber. According to their results, the higher temperatures and pressures that accompany retarded injection result in shorter flame lift-off lengths and increased soot emissions due to the resulting increased equivalence ratios within the spray.

The results for the 50 degree spray angle nozzle indicate much higher soot and carbon monoxide emissions than the others since the sprays for the 50 degree nozzle travel less distance before they collide with the piston bowl which is closer to the nozzle (see Figure 3). Spray impingement near the nozzle exit prohibits spray plumes from entraining air by shortening the spray travel distance and this results in rich mixtures. The results are also consistent with the longer burn duration of the 50 degree spray angle shown in Figure 2, which indicates that less fuel is burnt in the premixed combustion regime due to lack of entrained air.

NOx EMISSIONS

The NOx emissions for the six spray angle nozzles are shown in Figure 9 with respect to the injection timing. The NOx emissions in the standard diesel combustion regime decrease as the injection timing approaches TDC, as expected. However, the emissions are not controlled by the injection timing alone in the PCCI combustion regime. The emissions are also plotted in Figure 9 with respect to the overall equivalence ratio to help identify the controlling parameter. As shown in Figure 10, the NOx emissions strongly depend on the equivalence ratio. The effect of the injection timing is minor when compared to the effect of the equivalence ratio.
Figure 10. Indicated specific NOx emissions with respect to equivalence ratio

The strong dependency on the equivalence ratio of the NOx emissions explains the complex behavior of the NOx emissions when the injection timing is advanced. Advanced injection increases cylinder pressure and temperature during combustion while it deteriorates fuel economy when the injection timing is earlier than -15 degrees BTDC. The overall equivalence ratio becomes richer due to the poor fuel economy with the earlier injection timing since the tests were done by keeping IMEP constant (i.e., more fuel was injected). Consequently, the richer equivalence ratio contributes to the decreasing NOx emission even though the higher pressure and temperature would lead to increasing NOx emission. The complex behavior of the NOx emissions with the earlier injection results from the trade-off between both factors.

SPRAY TARGETING TO MINIMIZE EMISSIONS

As discussed in the previous section, the injection timing is not the only factor that explains the emissions in the PCCI combustion regime. When the fuel is injected into the cylinder, it reacts with the flow field and responds to the properties of the surrounding air. As far as the flow field is concerned, the injection timing is not a controlling parameter. However, spray targeting is a key linkage between the injection timing and the controlling parameters like the flow field and air properties. Nevertheless, analysis of spray targeting is complicated since it requires the injection timing to be combined with the spray angle and the geometries of the piston and engine.

As discussed previously, emissions in the standard diesel combustion regime depend strongly on the injection timing and only the 50 degree spray angle nozzle showed significant deviations from the others due to its extraordinarily short spray travel length. However, the emissions in the PCCI combustion regime could not be explained by the injection timing alone since the sprays in PCCI combustion are believed to interact with the flow details, such as the swirl and squish flow due to the long ignition delays. Figure 11 and Figure 12 show how the emissions in the PCCI combustion regime are influenced by the location where the corresponding sprays are targeted.

Soot and carbon monoxide emissions for three selected three spray angles (50, 85, and 120 degrees) are plotted normal to the spray targeting point on the piston bowl surface in Figure 11 and Figure 12 to help reveal the effects of spray targeting on emissions. The three spray angle nozzles shown are three representative cases for which the sprays target the inner, bottom, and outer surface of the piston bowl. Sprays for the 50 degree nozzle hit the inner surface, while the sprays for the 85 and 120 degree cases impinge on the bottom and outer surfaces, respectively. The relative magnitudes of the emissions in Figure 11 and Figure 12 are indicated by the distance perpendicular to the piston bowl surface, and the corresponding injection timings are shown at the locations where the centers of the spray plumes are aligned so as to hit at these timings.

The spray targeting results reveal two “sweet spots” in Figure 11 when the soot emissions are considered. The soot emissions record much lower values when
the sprays are targeted toward the bottom of the bowl and at the edge near the squish volume. However, targeting the bottom surface increases carbon monoxide emissions significantly, as seen in Figure 12, while targeting the bowl edge shows minimum levels of both soot and carbon monoxide emissions. The inner surface, which the 50 degree spray angle nozzle targets, shows better carbon monoxide emissions but poorer soot emissions. The side wall of the bowl shows poor soot and carbon monoxide emissions, except for when the spray targets near the edge. The emission trends in the standard diesel combustion regime, indicated by the dotted lines, are not explained by the spray targeting since in that case the emissions are determined by air entrainment in the spray and mixing processes after the ignition, instead of by spray targeting and pre-ignition mixing.

Though the differences regarding the targeting are very interesting, it is hard to explain how the targeting actually influences emissions. According to Miles et al. [14], the combustion details and its products in cases of early injection like in the PCCI regime are affected strongly by the swirl and squish flows. It is a great challenge to predict the details of the spray, swirl flow, squish flow and their interactions for each specific case.

Spray targeting at the piston bowl edge (e.g., the optimum targeting for the 120 degree spray angle nozzle) is of special interest since it provides better carbon monoxide emissions, in addition to an excellent soot and NOx trade-off, and the location is very close to the squish flow. The squish flow heads toward the spray plumes and transports fuel-air mixture to the center of the piston bowl where the mixture rarely locates itself without the help of the squish flow. Consequently, the squish flow enhances the fuel-air mixing and prevents local rich spots from being formed by spreading the mixture out through the piston bowl.

Once the spray is targeted above the top of the piston by advancing the fuel injection timing, the engine torque drops suddenly since the fuel hits the top of the piston and moves to the cylinder liner, leading to wasted fuel and diluted engine oil. This limits the use of advanced injection timing, especially for the cases of the 120 degree and higher spray included angles.

### SPRAY TARGETING AND SQUISH FLOW INTERACTION

Soot and carbon monoxide emissions for three spray angle nozzles (130, 140, 154 degrees) are compared with the 120 degree spray angle in Figure 13 to verify the advantage of targeting at the bowl edge. Both emissions show their optimum spots around the edge of the piston bowl, as in the case of the 120 degree spray angle. However, the emissions for the other spray angles are much less sensitive to the spray targeting than the 120 degree spray angle case, and the optimum spots move down deeper into the piston bowl as the spray angle becomes wider. The optimum spots for the 140 and 154 degree spray angles are placed lower than the spots for the 120 and 130 spray angles. Two factors might cause the optimum spots to move down.

First, the upper part of the spray plume periphery tends to be directed at the squish area on the top of the piston when the targeting approaches the bowl edge. The targeting shown in the plots indicates the location at which the centerline of the spray hits the piston bowl. A nozzle with a wider spray angle transfers more momentum into the radial direction and forces more fuel to flow into the squish region, where the fuel does not burn properly. Therefore, the radial momentum determines the upper boundary of the optimum spot. It is clear that the soot and carbon monoxide emissions with a wider spray angle start to increase at locations closer to the edge in Figure 13.

Next, the axial component of momentum of the spray also affects the location of the optimum spot. More
axial momentum transports more fuel to the bottom of the piston bowl which is far from the squish flow and gives less chance for the spray to interact with the squish flow. Therefore, the lower boundary of the optimum spot can be determined by the axial momentum and the optimum spots for a nozzle with narrower spray angle might be closer to the bowl edge.

However, both theories are hard to prove with the results of this study alone, and are not sufficient to explain the big jumps of both soot and carbon monoxide emissions seen with the 120 degree spray angle as the targeting moves down the bowl edge. There are complex interactions between the swirl flow, the squish flow, and the spray, as discussed in the previous section and analysis of these flows is beyond the scope of this study.

The spray targeting and squish flow interaction for nozzles with narrower spray included angle than 120 degrees may suggest interesting results since they provide more room for advancing the injection timing, and consequently more mixing time which is favorable to PCCI combustion. However, the results may not be easily predicted since they are strongly affected by the spray targeting in PCCI combustion as discussed previously. The details will be explored in future studies.

**CONCLUSIONS**

In the present study, PCCI combustion was differentiated from standard diesel combustion by evaluating the burn duration and the start-of-combustion timing from heat release rate analysis. The use of this approach led to the following conclusions:

1. The burn duration is a good indicator to differentiate PCCI combustion from standard diesel combustion.
2. The burn duration in the standard diesel combustion regime reflects the amount of air entrainment by the spray.
3. The start-of-combustion timing is a supplementary indicator of PCCI combustion.

The emissions for six different spray included angles were evaluated with respect to spray targeting to link the interaction between the fuel spray and the in-cylinder flow that controls the fuel-air mixing process in the PCCI combustion regime. The following conclusions were reached:

1. The start-of-Injection timing alone could not explain the emission trends in PCCI combustion.
2. NOx emissions strongly depend on the equivalence ratio. The effect of the injection timing is minor when compared to the effect of the equivalence ratio.
3. An excellent soot and NOx trade-off in addition to better carbon monoxide emissions could be achieved in the PCCI combustion regime when the spray was directed at the piston bowl edge near the squish region.
4. Spray targeting to the bottom of the piston bowl, which provided the longest spray travel distance, gave better soot emissions. However, CO emissions increased.
5. The 50 degree spray included angle case showed poor soot and carbon monoxide emissions in the standard diesel combustion regime due to its extraordinary short spray travel distance. However, its CO emission was significantly reduced in the PCCI combustion regime by targeting the spray at the inner surface of the piston bowl, consistent with the NADI concept [7].

The optimum spray targeting near the piston bowl edge in PCCI combustion is likely to be observed even under different engine operating conditions such as different engine speeds and loads since injection at this location help the squish flow to promote mixture preparation in general. However, the details are not easy to predict since the mixing is controlled by the ignition delay and the spray momenta which depend on thermodynamic properties and the flow field details, respectively. Future study is needed to investigate the benefits of spray targeting for other operating conditions by scaling the parameters in terms of the ignition delay and spray momenta.

**CONTACT**

Sangsuk Lee sslee@cae.wisc.edu

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