Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering
http://pid.sagepub.com/

Premixed compression ignition combustion with various injector configurations in a heavy duty diesel engine

Yongjin Jung, Choongsik Bae, Seibum B Choi and Hyun Dong Shin

Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering published online 4 October 2012
DOI: 10.1177/0954407012457628

Original Article

Institution of MECHANICAL ENGINEERS

Journal of AUTOMOBILE ENGINEERING

Proc IMechE Part D:
J Automobile Engineering
0(0) 1–11

DOI: 10.1177/0954407012457628
pid.sagepub.com
SAGE

Premixed compression ignition combustion with various injector configurations in a heavy duty diesel engine

Yongjin Jung, Choongsik Bae, Seibum B. Choi and Hyun Dong Shin

Abstract

Premixed compression ignition combustion was implemented using early injection timing and exhaust gas recirculation in a direct injection single cylinder diesel engine and was evaluated with respect to the injector configurations. A baseline injector with an injection angle of 146° and eight nozzle holes showed premixed compression ignition combustion at the injection timing of 40° crank angle before top dead centre among three distinct combustion regimes. The burn duration in premixed compression ignition combustion was shortest among the regimes. Premixed compression ignition combustion at an injection timing of 40° crank angle before top dead centre was achieved at an exhaust gas recirculation rate ranging from 0% to approximately 40%. Two different injector configurations were applied to investigate the effect of injection angle and the number of nozzle holes on premixed compression ignition combustion: one had an injection angle of 70° and eight nozzle holes and the other had an injection angle of 70° and 14 nozzle holes. These two injectors could implement premixed compression ignition combustion as well as the baseline injector under the injection timing and exhaust gas recirculation conditions. In case of both injectors with an injection angle of 70°, the indicated mean effective pressures for 8 and 14 nozzle holes increased by 26% and 11%, respectively, because of the increased fuel participating in combustion and the reduced negative work in premixed compression ignition combustion. On the other hand, the injector with an injection angle of 70° and 14 nozzle holes showed the lowest levels of hydrocarbon, carbon monoxide and smoke emissions, which decreased by 82%, 92% and 81%, respectively, in premixed compression ignition combustion. However, the nitrogen oxides emission for the injectors with eight and 14 holes increased by 82% and 68%, respectively, in premixed compression ignition combustion. Natural luminosity from an in-cylinder visualization reveals pool fire of fuel films on the base of the piston bowl when both injectors had a narrow injection angle. For the injector with an injection angle of 70° and eight nozzle holes, a more vigorous pool fire at an exhaust gas recirculation rate of 0% is attributed to a larger amount of fuel film. At an exhaust gas recirculation rate of 40%, however, the portion of unburned fuel films increased the hydrocarbon and carbon monoxide emissions, and the rest of the diffusive pool fire can increase smoke emission surviving under a lack of oxygen. On the other hand, for the injector with an injection angle of 70° and 14 nozzle holes, the hydrocarbon and carbon monoxide emissions were maintained at lower levels due to less formation of fuel film and better air utilization.

Keywords

Exhaust gas recirculation, premixed compression ignition, narrow injection angle, pool fire, in-cylinder visualization

Date received: 28 March 2012; accepted: 18 July 2012

Introduction

Stringent legislation concerning diesel engines has led to efforts to reduce emissions that are harmful to people and the environment.[1] Harmful emissions from diesel engines include nitrogen oxides (NOx) and smoke. Advanced combustion technologies have been applied to diesel engines to reduce these emissions: one of these

Korea Advanced Institute of Science and Technology, Daejeon, Republic of Korea

Corresponding author:
Choongsik Bae, Department of Mechanical Engineering, Korea Advanced Institute of Science and Technology (KAIST), 291 Daehak-ro, Yuseong-gu, Daejeon 305-701, Republic of Korea.
Email: csbae@kaist.ac.kr

technologies is homogeneous charge compression ignition (HCCI) combustion.

HCCI combustion is achieved by a creating a homogeneous premixed charge before the combustion begins so that combustion occurs as a result of spontaneous ignition at multiple points. The combustion of the lean charge in an HCCI engine results in low combustion temperature, and the NOx emission is thus reduced dramatically. In addition, low smoke emission can be achieved because of the well-premixed charge. In spite of the reduction of NOx and smoke, other hazardous emissions are still present, such as unburned hydrocarbon (HC) and carbon monoxide (CO) resulting from incomplete combustion.[2]

Injecting the fuel into the intake such as in a port fuel injection system has been used to obtain a premixed homogeneous charge.[3,4] However, direct injection systems are currently being used in most conventional diesel engines, and as a result mixture preparation including premixed charge through direct injection has emerged. Attempts to prepare a homogeneous charge and to implement HCCI combustion in direct injection diesel engines have led to early injection strategies.[5–8] The early injection strategies allow more time for mixing during the inherently longer ignition delay period. However, longer spray penetration due to low air density can cause fuel impingement on the cylinder wall, leading to deterioration of combustion efficiency. Furthermore, it is difficult to control combustion phasing due to spontaneous ignition. The load is also limited to a low level by the high pressure rise rate in the form of knocking. Another strategy of late injection using a high swirl ratio and a large amount of exhaust gas recirculation (EGR) was introduced to prepare a homogeneous charge.[9] Late injection strategies rely on the EGR to increase ignition delay. These injection strategies employ a certain period to prepare the mixture of fuel and air using ignition delay from injection to autoignition as an early injection strategy, and prolongation of ignition delay due to the EGR as a late injection strategy, respectively.

Premixed compression ignition (PCI) or partially premixed compression ignition (PPCI) strategies are based on premixed charge preparation in a compression ignition engine. Lee and Reitz[10] reported that premixed charge compression ignition (PCCI) can be identified with short burn durations and an early start of combustion timing. Multiple injection strategies, such as split injection, have attracted attention in relation to PCI combustion because the strategy can extend the operating range.[11,12] A two-stage injection strategy with an early main injection and a short second injection at near top dead centre (TDC) was used in a direct injection diesel engine to control the combustion phase by using the second injection as an ignition promoter.[13] Injection strategy for partially compression ignition combustion was investigated in a commercial multi-cylinder engine.[14] In addition, the injection system configurations, including the injector type, injector position and injector tip details, have been varied to mitigate challenges related to early injection strategies.[5,7,8,10,12,15] Comprehensive study with respect to the load and compression ratio has been performed in partially premixed combustion (PPC),[16] and the application of injection angles of 148°, 120° and 100° in PCCI combustion showed that the operation in conventional diesel combustion is limited by high smoke emission.[17]

The optical engine technique using an injector with a narrow injection angle showed that fuel impingement on the bowl wall with early injection timing can result in fuel film combustion, such as pool fire.[18,19] The pool fire is strongly connected with the emission results.[20,21] The source of NOx emission could be the near-stoichiometric region of fuel film combustion, which could increase the NOx emission.[20] However, it could lead the decrease in the NOx emission due to the rich mixture of the fuel film layer.[22]

In the present study, an early injection strategy and EGR were applied to implement PCI combustion in a direct injection single cylinder diesel engine. Combustion characteristics and exhaust emissions were investigated while varying the injection timing and EGR rate. PCI combustion was verified on the basis of a burn duration analysis. Various injector configurations were used to study the effects of the injection angle and the number of nozzle holes on the PCI combustion and exhaust emissions. A pressure rise rate analysis was performed to inspect the noise characteristics in PCI combustion.

Experimental set-up

A four-stroke, single-cylinder, direct injection diesel engine was used to investigate PCI combustion. A schematic diagram of the experimental setup is shown in Figure 1, and the engine specifications are listed in Table 1. The engine speed was controlled by an 82 kW alternating current (AC) dynamometer at the rated speed.

Diesel fuel was supplied into a common-rail system through a filter. The pressure in the common-rail was adjusted by a pressure controller (ZB-1200, Zenobalti Co., Seoul, ROK). The fuel injection duration was controlled by a programmable injector driver (IDU 5000B, Zenobalti Co.). The ambient air was introduced through an intake surge tank to settle fluctuation. The temperature of the intake air, which was naturally-aspirated, and the coolant was maintained at 25° C and 80° C, respectively.

For the EGR, an exhaust gas recirculation line was installed downstream from the exhaust surge tank to reduce fluctuation. The intake temperature of the mixture of exhaust gas and fresh air was cool enough to be 25–50° C depending on the rate of EGR, because naturally convective heat transfer occurred in the exhaust surge tank. The amount of EGR was controlled by a three-way valve.

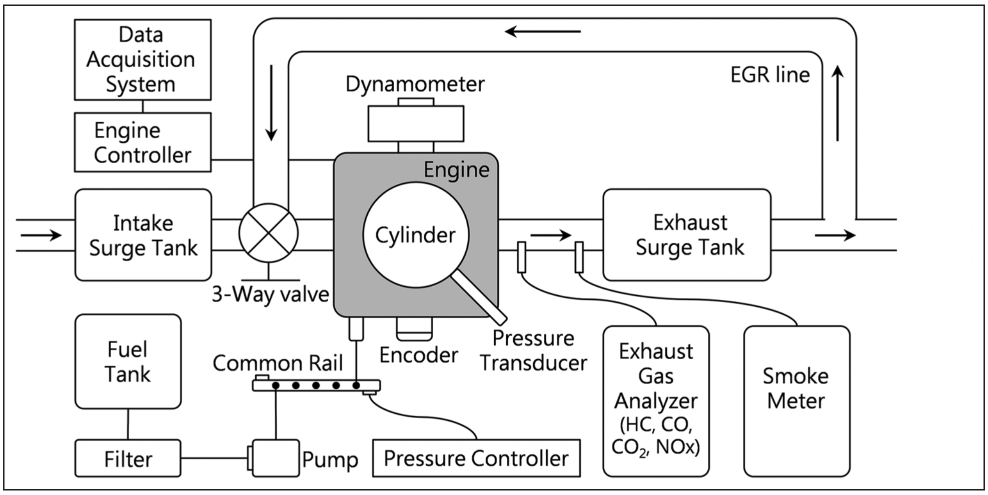

Figure 1. Schematic diagram of the experimental set-up.

Table 1. Engine specifications.

Cylinder	1
Valves per cylinder	4 (2 intake and 2 exhaust)
Cycle	4-stroke
Bore (mm)	100
Stroke (mm)	125
Displacement (cm^3)	981
Compression ratio	17.4

In-cylinder pressure was measured using a piezoelectric pressure transducer (6052C, Kistler, Anyang City, ROK). The flush-type mounted pressure transducer was mounted in cylinder head located 46 mm away from the centre of the bore. The bush was intentionally designed to integrate the adaptor of the pressure transducer, because the commercial engine used in this experiment does not have a glow plug, the glow-plug-type adaptor for the pressure sensor was not used in this experiment. The relative value from the piezoelectric pressure transducer (4045A, Kistler) was scaled by the absolute value from the piezo-resistive pressure transducer installed in the intake manifold. The cylinder pressure at intake bottom dead centre was set to be equal to the intake manifold pressure. The injection signal was synchronized by the cyclic encoder for 1800 pulses (E50S8, Autonics, Mundelein, IL, USA). The in-cylinder pressure data were averaged for 100 cycles to calculate the heat release rate and the data were acquired with a crank angle (CA) resolution of 0.2°.

An analysis of the gaseous emissions was carried out using an exhaust gas analyzer (MEXA 1500D, Horiba, Kyoto, Japan) consisting of a non-dispersive infrared absorption (NDIR) analyzer for carbon monoxide (CO) and carbon dioxide (CO_2) measurements, a chemi-luminescence detector (CLD) analyzer for NOx measurements, and a flame ionization detector (FID) analyzer for hydrocarbon (HC) measurements.

A smoke meter (415S, AVL, Seoul, ROK) was used to measure smoke emissions. Part of the exhaust gas flow was sampled with a probe in the exhaust line and drawn through filter paper. The resultant blackening of the filter paper was measured by a reflectometer and represented the smoke content in the exhaust gas. One filter smoke number (FSN) corresponded to a concentration of 17.84 mg/m^3 at the sampling point.

The engine operating conditions are given in Table 2. The engine speed and injection pressure were set at 1200 r/min and 160 MPa, respectively. The fuel injection quantity was fixed at 30 mg/stroke by adjusting the injection duration. This injection quantity corresponded to about 0.6 MPa of IMEP for the convention diesel combustion in this research engine. The injection timing was varied from -5° to 100° CA BTDC to study the combustion characteristics, and then an injection timing of 40° CA BTDC was selected to achieve PCI combustion. The EGR rate, which was calculated using equation (1), was varied from 0% to the maximum limit to study the effect of the EGR rate on combustion and emission characteristics. The EGR rate was set to around 40% to achieve PCI combustion

$$EGR\ rate\ (\%) = \frac{[CO_2]_{in}}{[CO_2]_{ex}} \times 100 \tag{1}$$

Various injector tip configurations were applied to investigate the effects of the injection angle and the number of nozzle holes on PCI combustion. The baseline injector, which is currently used in commercial multi-cylinder engines, has an injection angle of 146°

Table 2. Operating conditions.

Engine speed (r/min)	1200
Injection pressure (MPa)	160
Injection quantity (mg/stroke)	30
Injection timing (°CA BTDC)	−5–100
Air/fuel ratio	32
Intake pressure	Naturally aspirated
EGR rate (%)	0–max

Table 3. Specifications of the injector configurations.

Injector	8H146	8H70	14H70
No. of holes	8	8	14
Injection angle (°)	146	70	70
Diameter (mm)	0.146	0.146	0.107
Hydraulic flow rate (cm^3/30s @ 100 bar (10 MPa))	460	460	460

and eight nozzle holes (8H146). The injection angle in the present study is defined as the spray included angle. An injector with an injection angle of 70° and eight nozzle holes (8H70) was used to evaluate the effects of the injection angle. Moreover, an injector with an injection angle of 70° and 14 nozzle holes (14H70) was used to assess the effects of the nozzle diameter. The hydraulic flow rate (HFR) of all the injectors within 30 s was the same at 460 cm^3 when the injection pressure was 10 MPa, which implies that an increase in the number of nozzle holes brings about a decrease in the nozzle diameter to maintain a constant flow rate. All injector tips used a micro-sac design. Specifications of the injector configurations are listed in Table 3.

The estimations of burn duration and start of combustion are important because the PCI combustion has short burn duration and early start of combustion, as discussed later. The burn duration and start of combustion including CA10 and CA90 in this experiment are defined. The start of combustion is defined as CA10, which indicates the crank angle when the accumulated heat release reached 10% of the total heat release. The burn duration is defined as crank angle between CA10 and CA 90, which indicates the crank angle when the accumulated heat release reached 90% of the total heat release.

Macroscopic spray visualization was used to quantitatively analyze the spray tip penetration and the spray cone angle. The spray cone angle was measured by images, which were taken by direct photography of Mie-scattering. This experimental method is described in detail in Kang et al.[23] and Lee et al.[24]

The in-cylinder flame image through the piston quartz window was reflected by a 45° mirror mounted in the elongated piston. The window allowed visualization within a circle of 65 mm diameter in a 100 mm bore. A high speed camera (Phantom V.7.1, Vision Research Inc., Wayne, NJ, USA) was used to take images of the natural luminosity produced by combustion. This high speed imaging system can record at 26,000 frames/s, with a resolution of 256 × 256 pixels, thus images could be taken at every crank angle of 0.28° at an engine speed of 1200 r/min. For image sharpness, the exposure duration was set to 9 μs to prevent the flame from streaking. The in-cylinder flame image was taken at the fourth cycle during five consecutive cycle firing conditions. The lens attached to the camera was a Nikon 105 mm lens with f/16 to allow a sufficient brightness and depth of field.

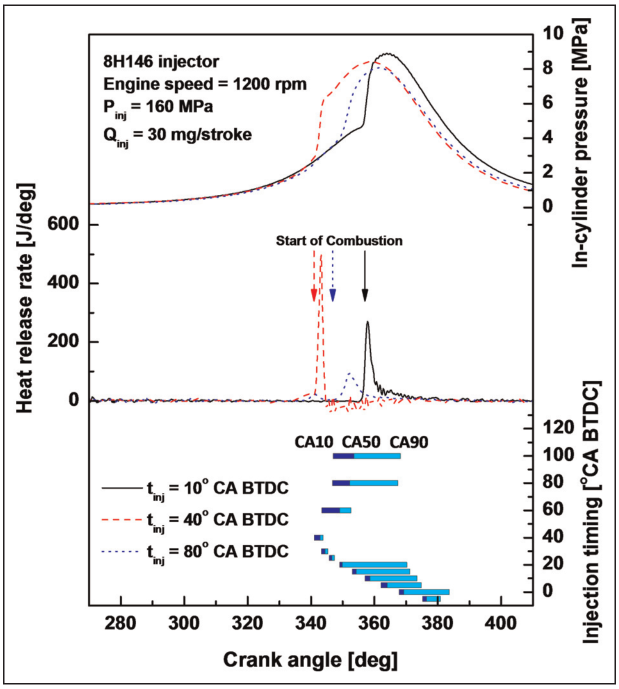

Figure 2. In-cylinder pressure, heat release rate and burn duration according to the injection timing including the injection timing of 10°, 40° and 80°CA BTDC for baseline injector (8H146). Engine speed, injection pressure and injection quantity was 1200 r/min, 160 MPa and 30 mg/stroke, respectively.

Results and discussion

Premixed compression ignition combustion

A baseline injector with eight nozzle holes and an injection angle of 146° (8H146) was used initially among three injector configurations to investigate the combustion characteristics and emissions. The injection timing was varied from −5° to 100° CA BTDC to study the combustion characteristics without an EGR strategy. Figure 2 shows the in-cylinder pressure, heat release rate and burn duration according to the injection timing. The other parameters such as engine speed, injection pressure and injection quantity were fixed at 1200 r/min, 160 MPa, and 30 mg/stroke, respectively. Three combustion regimes can be found as a result of the experiment from injection timing. First, conventional diesel combustion with injection in close vicinity to TDC was observed when the injection timing was 10° CA BTDC. Premixed burn of conventional diesel combustion was detected, followed by diffusion burn. Second, the peak of the heat release rate was higher than the others and the injected fuel burned rapidly when the injection timing was 40° CA BTDC. In addition, the combustion phase was the most advanced among those with other injection timings. The combustion regime with this injection timing can be regarded as PCI combustion. Lastly, HCCI combustion with low and high temperature oxidation can be observed when the injection timing was 80° CA BTDC. In this study, focus was placed on combustion regime with an

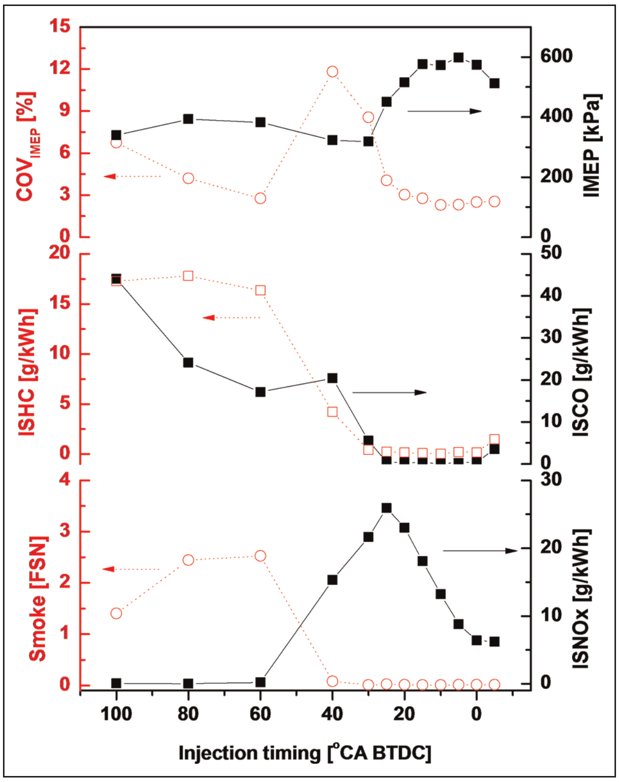

Figure 3. IMEP, COV of IMEP and emissions according to the injection timing for baseline injector (8H146). Engine speed, injection pressure and injection quantity were 1200 r/min, 160 MPa and 30 mg/stroke, respectively.

injection timing of 40° CA BTDC. The bottom graph in Figure 2 shows the burn duration to distinguish the combustion regime according to the injection timing. The burn duration, which is expressed by the degree of crank angle, can be defined as the period from 10% to 90% of the accumulated heat release. The burn duration results indicate that the duration between injection timings of 25° and 40° CA BTDC were shorter than those of the other injection timings. The burn durations for injection timings of 25°, 30° and 40° CA BTDC were 1.6°, 2.2° and 2.8° CA, respectively. Moreover, the start of combustion, which can be represented by CA10, was earlier than those of other injection timings. CA10 for injection timings of 25°, 30° and 40° CA BTDC were 345.8°, 343.2° and 341.0° CA, respectively. PCI combustion in the present study can be verified by a short burn duration and early start of combustion timing.[10]

Figure 3 shows the indicated mean effective pressure (IMEP) and coefficient of variation (COV) of IMEP with respect to the injection timing. The IMEP of conventional diesel combustion with injection in close vicinity to TDC remains above 500 kPa. However, the IMEP decreased sharply after the conventional diesel combustion regime and deteriorated during PCI combustion because negative work during the compression stroke increased due to the early start of combustion, as shown in Fig.ure 2. In addition, the COV of IMEP, a combustion instability indicator, was high. Low IMEP and unstable combustion are disadvantages that should be overcome in PCI combustion. The IMEP increased slightly again as the injection timing was advanced to 40° CA BTDC due to the occurrence of HCCI combustion. A retarded combustion phase for HCCI combustion can reduce the negative work, and consequently the IMEP for HCCI combustion can be improved slightly. On the other hand, PCI combustion had advantages of lower HC and CO emissions compared with HCCI combustion, as shown in Figure 3. However, the levels of HC and CO emissions in PCI combustion were still high relative to conventional diesel combustion. High levels of HC and CO emissions for PCI combustion necessitate the application of various injector configurations, especially with narrow injection angle, which will be discussed later. The level of NOx emission for PCI combustion with an injection timing of 40° CA BTDC was slightly higher than that for conventional diesel combustion with an injection timing of 10° CA BTDC. Smoke emission for PCI combustion remained at a low level, similar to the conventional diesel combustion. These trends of emissions related to the spray interaction with walls are interpreted based on diesel models.[25] The mass of the spray that impinges on the walls is not favourable to the formation of the anchored diffusion flame envelope surrounding the spray, as a result, it would be expected to reduce NOx emission. In addition, the mass that impinges on the walls cannot participate in the premixed combustion, which is supported by the highest NOx level of vigorous premixed combustion for the injection timing of 25° CA BTDC with little wall impingement due to the higher air density. Soot would reduce due to the fact that the early injection allows for better mixing, and that the wall-impinged fuel unavailable for combustion burns, albeit partially, at low bulk temperatures much later in the expansion stroke. Therefore, this might increase the HC and CO emissions but not the soot. The comprehensive trends of the IMEP, COV of IMEP and emissions for the other injectors with narrow injection angle according to the injection timing are similar to those for the baseline injector except for NOx emission. The peaks of the NOx emission were much higher than those of the baseline injector, and the phase of the NOx emission was shifted toward early injection timing.[26]

EGR was applied to investigate the effects of EGR on PCI combustion with a fixed injection timing of 40° CA BTDC, as shown in Figure 4. The heat release rate in Figure 4 shows that the retarded combustion phase and decreased peak of heat release rate as the EGR rate increased due to the dilution effect of the EGR.[27–29] Oxygen in the intake was replaced with inert gases such as nitrogen and carbon dioxide in the exhaust, which contributed to a lower combustion temperature and relatively inactive reaction. In addition, this causes the decrease in the negative work that allows the IMEP for PCI combustion to improve, as shown in Figure 5. On the other hand, the

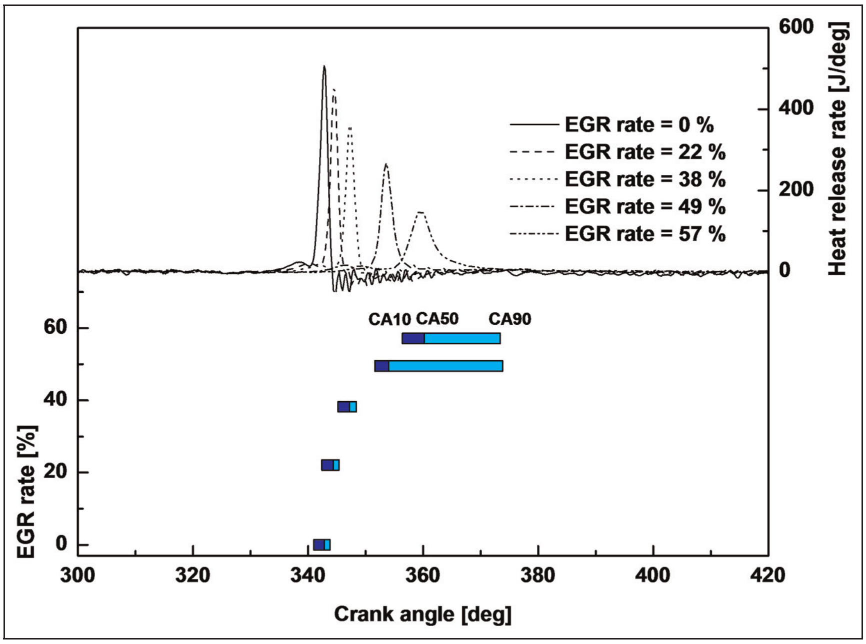

Figure 4. Heat release rate and burn duration with the EGR rate at injection timing of 40°CA BTDC for baseline injector (8H146). Engine speed, injection pressure and injection quantity were 1200 r/min, 160 MPa and 30 mg/stroke, respectively.

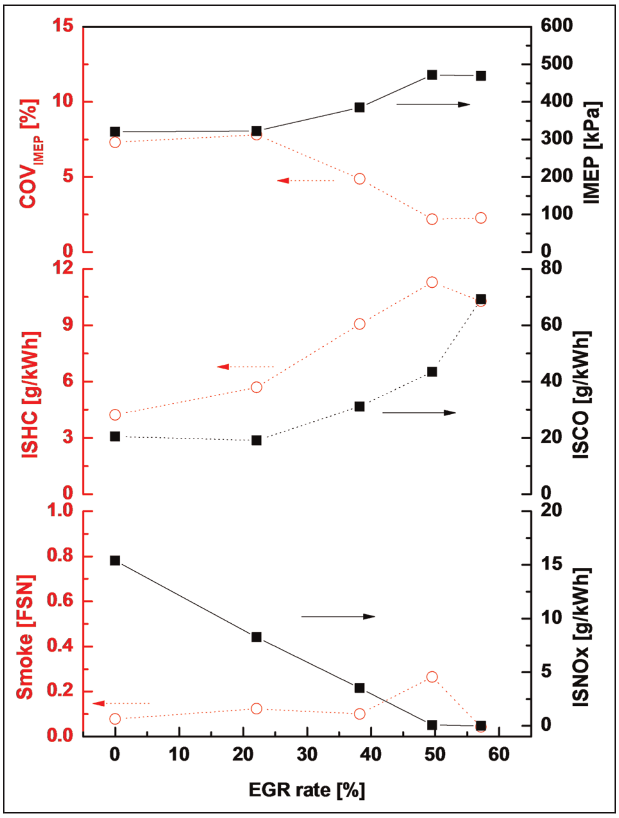

Figure 5. IMEP, COV of IMEP and emissions at injection timing of 40°CA BTDC with the EGR rate for the baseline injector (8H146). Engine speed, injection pressure and injection quantity were 1200 r/min, 160 MPa and 30 mg/stroke, respectively.

characteristics for PCI combustion such as short burn duration and early start of combustion timing occurred up to around a 40% EGR rate. Burn durations with a 0%, 22% and 38% EGR rate were 2.8°, 3.0° and 3.2°CA, respectively. The CA10 for 0%, 22% and 38% EGR rates were 341.0°, 342.4° and 345.2°CA, respectively. Thus, PCI combustion could be realized when the EGR rate reached around 40%.

The increase in the IMEP resulting from the reduced negative work is regarded as an advantage of the EGR application, as shown in Figure 5. The IMEP at a 38% EGR rate increased by around 20% compared with the IMEP at a 0% EGR rate. Furthermore, the COV of IMEP was lower than 5% for an EGR rate of 38%. This implies that stable combustion could be achieved with EGR due to the slower reaction rate. NOx emission decreased as the EGR rate increased due to the lower combustion temperature, and the level of NOx emission at a 38% EGR rate was consequently similar to that of HCCI combustion. However, high levels of HC and CO emissions, which are regarded as a drawback that should be overcome for PCI combustion, still increased continuously as the EGR rate increased, due to the lack of oxygen.

Effect of injector configuration

In the previous section PCI combustion was characterized by short burn duration and early start of combustion from the results of the experiment on injection timing and the application of EGR. PCI combustion was optimized in this experiment at an injection timing of 40° CA BTDC and a 38% EGR rate. EGR could compensate for the decreased IMEP due to the reduced negative work. NOx emission was maintained at a low level due to a low combustion temperature. Various injector configurations of the injection angle and the

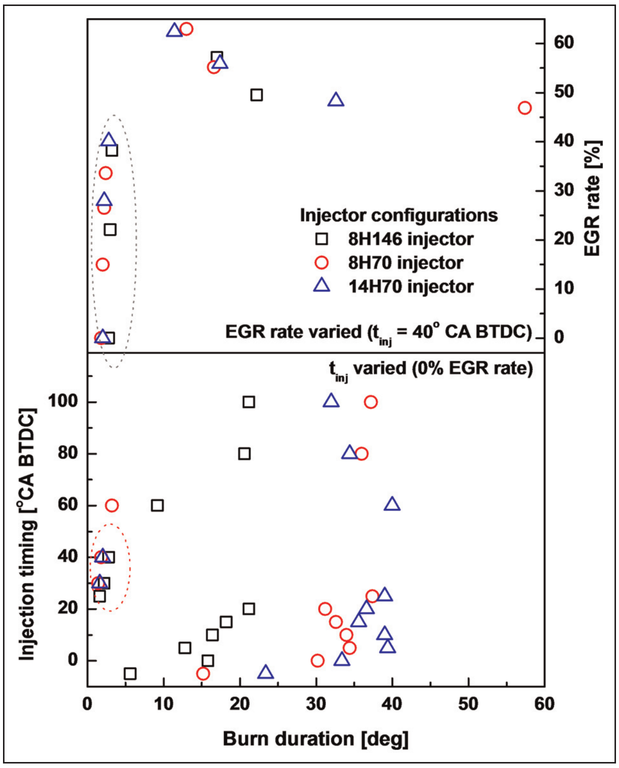

Figure 6. Burn duration for the various injector configurations with respect to EGR rate at a fixed injection timing of 40°CA BTDC (top) and to the injection timing at EGR rate of 0% (bottom). Engine speed, injection pressure and injection quantity were 1200 r/min, 160 MPa and 30 mg/stroke, respectively.

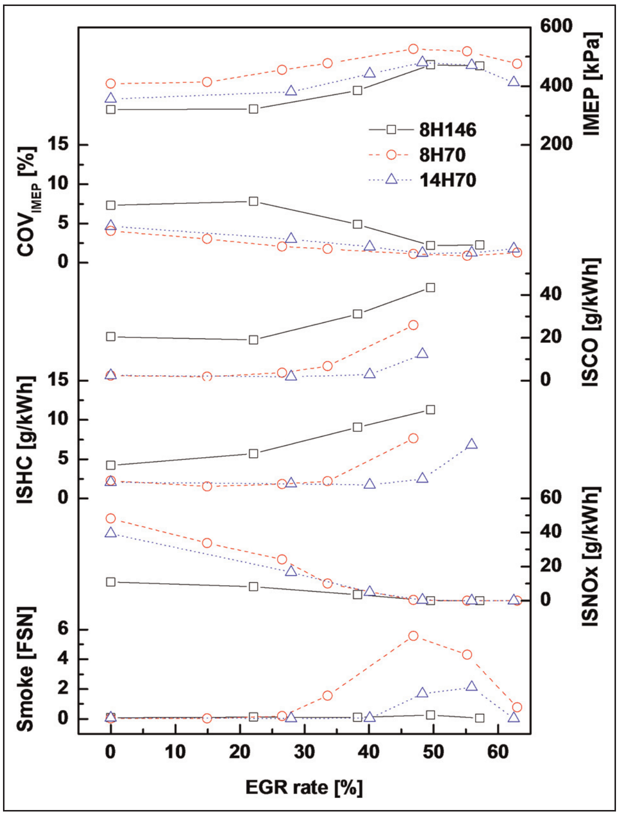

Figure 7. IMEP, COV of IMEP and emissions at injection timing of 40°CA BTDC according to the EGR rate for various injectors. Engine speed, injection pressure and injection quantity were 1200 r/min, 160 MPa and 30 mg/stroke, respectively.

number of nozzle holes were employed to address the problem of large HC and CO emissions. The detailed injector specifications used in the experiment are described in the experimental set-up.

Figure 6 shows the achievement of PCI combustion in terms of injection timing and EGR rate from the analysis of the burn duration. All injectors with different injection angles and numbers of nozzle holes had short burn duration in the range of injection timing from 30° to 40° CA BTDC. At an injection timing of 25°CA BTDC, in contrast with the baseline injector, burn durations for both injectors with a narrow injection angle were considerably longer, regardless of the number of nozzle holes. Likewise, short burn durations occurred in an EGR range from 0% to around 40%. Injection timing and the EGR rate were determined at 40° CA BTDC and around 40%, respectively, to achieve PCI combustion for all injector configurations.

Figure 7 shows a cross-sectional schematic of the injected fuel trajectory in the cylinder when the piston was located at 40° CA BTDC. It is seen that the fuel from the injector with a wide injection angle (146°) can easily move toward the boundary region between the cylinder (liner) wall and piston crown beyond the piston bowl lip. On the other hand, the fuel from the injector with a narrow injection (70°) can easily travel toward the piston bowl and impinge on the bottom surface of the combustion chamber.

Figure 7 shows IMEP, COV of IMEP, and emissions with respect to the EGR rate at a fixed injection timing of 40° CA BTDC for all three injector configurations. The IMEP for two injectors (8H70, 14H70) with a narrow injection angle was higher than that for the baseline injector (8H146), because a larger amount of fuel can be contained within the piston bowl and participate in combustion. Likewise, for the baseline injector, the fuel is directed to the cylinder wall beyond the piston bowl and can impinge on the wall, and consequently the amount of fuel that participates in combustion can be relatively reduced. The in-cylinder pressure for the baseline injector was lower than that of the two injectors with narrow injection angle, as shown in Figure 8. On the other hand, the IMEP for the injector with eight nozzle holes was slightly higher than that for the injector with 14 nozzle holes in the same narrow injection angle. The peak of the heat release rate for the 14H70 injector was slightly higher than that for the 8H70 injector. That is, premixed burn for the 14H70 injector was more vigorous than that for the 8H70 injector. This implies that the negative work for the 14H70 injector was larger than that for 8H70 injector. Accordingly, the negative work exerted from the premixed burn in the compression stroke can cause a lower IMEP for the 14H70 injector.

The COV of IMEP for injectors with a narrow injection angle was lower than that for the baseline injector,

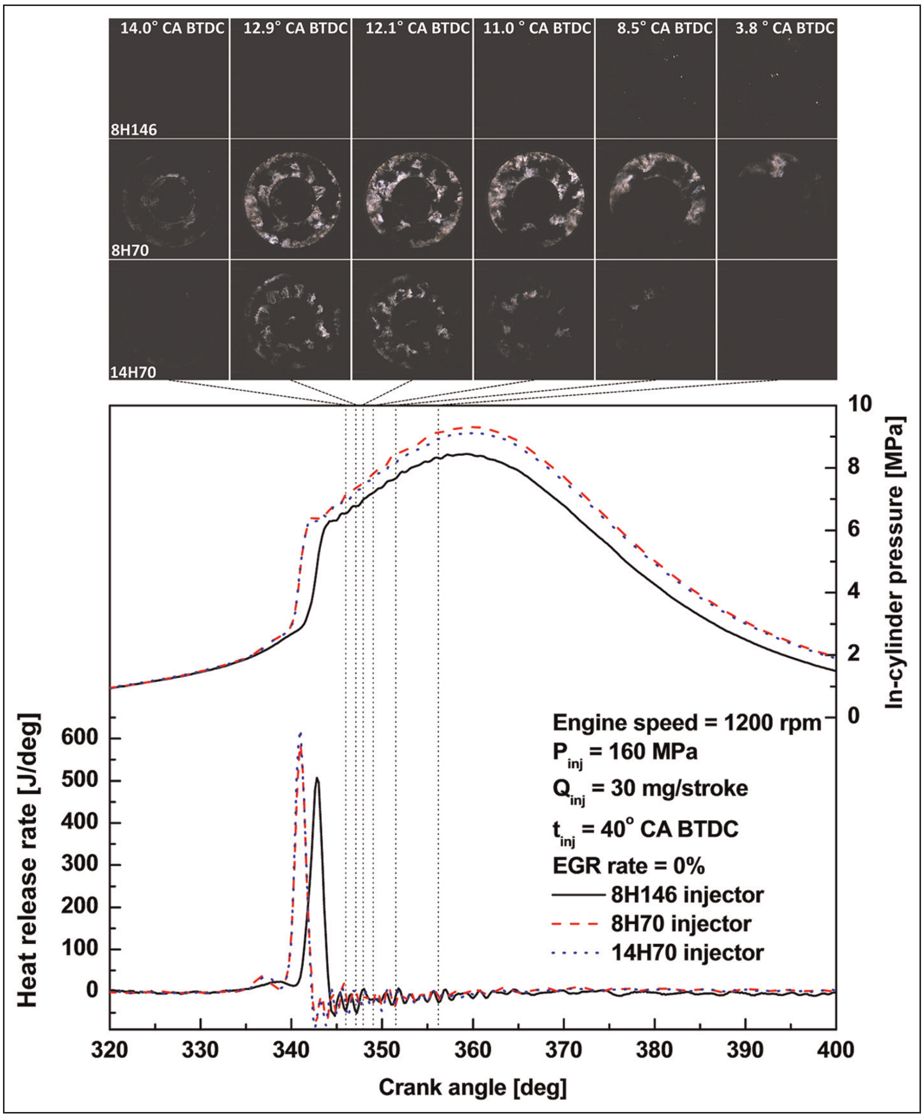

Figure 8. In-cylinder visualization, pressure and heat release rate in terms of the injector configurations at injection timing of 40°CA BTDC and without EGR. Engine speed, injection pressure and injection quantity were 1200 r/min, 160 MPa and 30 mg/stroke, respectively.

which implies that PCI combustion can be more stable with the use of injectors with a narrow injection angle. For the injector with an injection angle of 146°, some fuel can impinge on the cylinder wall and the rest of the fuel can participate in combustion, because the fuel trajectory is located at the boundary region between the cylinder wall and the piston crown beyond the piston bowl lip. As the piston moves to TDC, the intensity of squish flow can be enhanced, thus making it difficult to estimate the amount of fuel that participates in combustion – and this varyies cycle by cycle.

The in-cylinder visualization in Figure 8 depicts the natural luminosity of the flame for each injector, together with the corresponding crank angle under the in-cylinder pressure and heat release rate. After the end of injection of 31° CA BTDC, most of the premixed fuel was burned between a crank angle of 20° and 15° CA BTDC, corresponding to rapid heat release. The flame luminosity in the case of rapid heat release for premixed burn was too weak to be detected, and thus the first image with appropriate luminosity distinguishable from black could be detected at a crank angle of 14.0° and 12.9° CA BTDC for the 8H70 and 14H70 injectors, respectively. The vigorous flame luminosity after rapid heat release occurred and lasted before the piston approached the TDC. The locations of flame luminosity correlated well with the fuel spray, which implies that the fuel that impinged on the base of the piston bowl burned vigorously. Some of the fuel that impinged on the piston bowl also burned along the periphery of the optical quartz window. These flames were reported to be pool fire of liquid films and strongly affect the amount of emissions.[18–20] In contrast with the injectors with a narrow injection angle, vigorous flame luminosity was not shown for the baseline injector. From brightness- and contrast-improved images

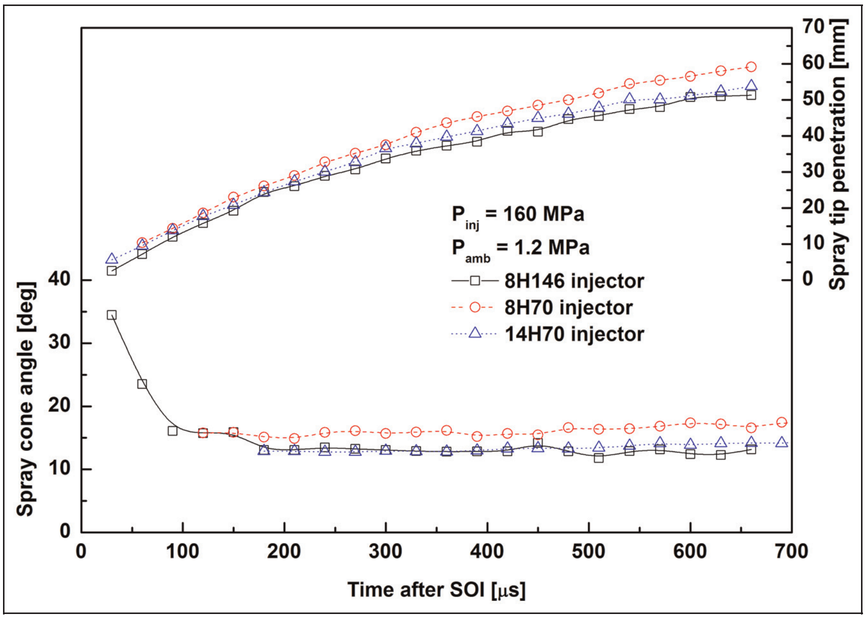

Figure 9. Spray tip penetration and spray cone angle versus time after the start of injection with respect to the injector configurations.

after rapid heat release, however, several small flame fragments with luminosity floated in a clockwise direction from 11° CA BTDC to 12° CA ATDC. The luminous fragments may have resulted from the fuel-rich dense mixture.[30]

HC and CO emissions of the baseline injector in Figure 7 mostly resulted from adsorbed fuel on the cylinder wall relative to the injector with a narrow injection angle. On the other hand, the HC and CO emissions for both injectors with a narrow injection angle were lower than for the baseline injector at all EGR rates, including around 40%, even though the emissions increased as the EGR rate increased. This is because pool fire activity can reduce the HC and CO emissions originating from the fuel film,[20] At around a 40% EGR rate, corresponding to approximately a 15% oxygen concentration at the intake, the HC and CO emissions for the 8H70 injector increased sharply relative to those for the 14H70 injector. Because the pool fire luminosity decreased as the EGR rate increased, the vapour or droplet cloud detached from the fuel films, which do not ignite, can increase the HC and CO emissions due to the lack of oxygen and the pressure drop during the expansion stroke.[18,20] For the 14H70 injector, however, reduced nozzle diameter causes a decrease of the spray momentum, which resulted in a shorter spray tip penetration length, as shown in Figure 9. This contributes to reduced formation of a fuel film on the base of the piston bowl. In addition, reduced nozzle diameter can enhance the spray atomization. Air utilization per unit spray for the 14H70 injector can meanwhile be improved, because the injected fuel quantity per nozzle hole is smaller than that for the 8H70 injector, even though the spray cone angle for the 14H70 injector showed a somewhat low value, as presented in Figure 9. Accordingly, HC and CO emissions for the 14H70 injector can be maintained at lower levels than in the other injector configurations, with the achievement of PCI combustion at a 40% EGR rate.

Higher pool fire luminosity for the 8H70 injector, as seen in Figure 8, results in higher NOx emission, as presented in Figure 7, relative to the 14H70 injector. Martin et al.[20] reported that there is a strong correlation of the NOx emission with the pool fire luminosity, because near-stoichiometric conditions in diffusion combustion of a pool fire can cause high NOx emission. On the other hand, the NOx emission for the baseline injector is lower than that of the other injectors due to the absence of pool fire. Comprehensively, the NOx emissions at around 40% EGR rate for both 8H70 and 14H70 injectors are two-thirds those obtained with conventional diesel combustion using the baseline injector.

Smoke emission for the 14H70 injector at a 40% EGR rate was much lower than that for the 8H70 injector, as shown in Figure 7. The difference is attributed to the amount of fuel film and its diffusion flame. For the 8H70 injector, a larger fuel film is formed due to larger spray tip penetration, and some of the fuel film burns as a diffusion flame, leading to smoke. The smoke cannot oxidize easily because of the lower oxygen concentration and decreased combustion temperature at a 40% EGR rate. Thus, the remaining smoke for the 8H70 injector is greater than that for the 14H70 injector due to the larger amount of fuel film for the 8H70 injector. Smoke emission for the baseline injector was kept at a low level compared with the injectors with a narrow injection angle due to the absence of pool fire. However, it is noted that smoke results should be carefully interpreted because of oil dilution.

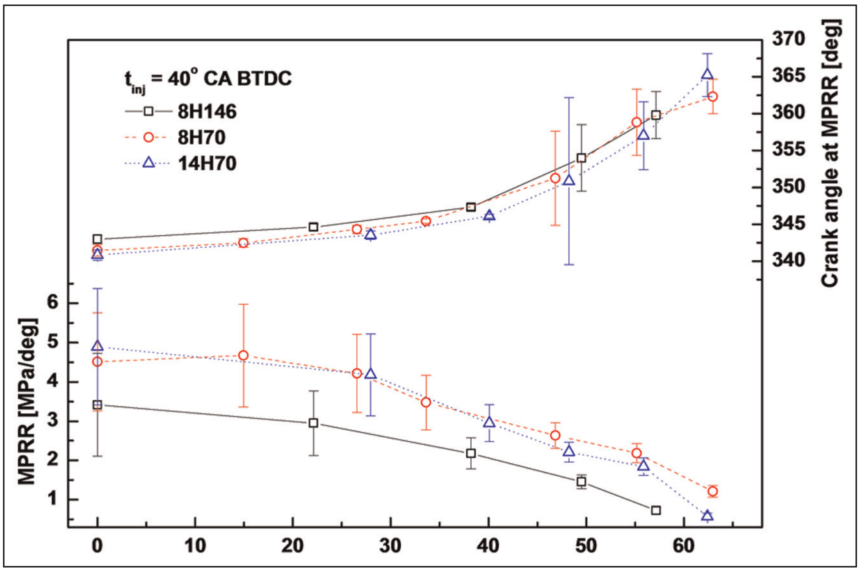

Figure 10. The maximum pressure rise rate (MPRR) and crank angle corresponding to the MPRR according to the EGR rate for three injector configurations at injection timing of 40°CA BTDC. Engine speed, injection pressure and injection quantity were 1200 r/min, 160 MPa and 30 mg/stroke, respectively.

Figure 10 shows the maximum pressure rise rate (MPRR) and the crank angle corresponding to the MPRR with respect to the EGR rate for three injector configurations. On the whole, MPRR decreased and the crank angle corresponding to the MPRR was retarded as the EGR rate increased. The MPRR was 3.01 MPa/° for PCI combustion at a 40% EGR rate. The MPRR for conventional diesel combustion at an injection timing of 10° CA BTDC without EGR was 1.96 MPa/°. The MPRR for PCI combustion was still higher than that for a conventional diesel engine. The MPRR for the injectors with a narrow injection angle was higher than that for the baseline injector. For the injectors with a narrow injection angle, there was no difference in the MPRR of PCI combustion, regardless of the number of nozzle holes.

Conclusion

Premixed compression ignition (PCI) combustion with an early injection strategy and exhaust gas recirculation was investigated in a direct injection, single-cylinder diesel engine. The effect of the injection angle and the number of nozzle holes on premixed compression ignition combustion and exhaust emissions were studied, together with in-cylinder visualization. The major findings from our study are summarized as follows:

1. Three combustion regimes according to the injection timing were verified by a heat release rate analysis: conventional diesel combustion, PCI combustion and HCCI combustion. The premixed compression ignition combustion was characterized by a short burn duration and was achieved when EGR was applied at a rate of around 40%.
2. When PCI combustion was implemented with an injection timing of 40° CA BTDC and an EGR rate of around 40%, the IMEP for the injector with an injection angle of 70° and eight nozzle holes was the highest, because a larger portion of the injected fuel can participate in combustion. On the other hand, HC, CO and smoke emissions for the injector with an injection angle of 70° and 14 nozzle holes showed the lowest levels as a result of the better utilization due to the reduced nozzle diameter.
3. In-cylinder visualization shows pool fire of fuel films on the base of the piston bowl for both injectors with a narrow injection angle (70°). At EGR rate of 0%, more vigorous pool fire for the injector with an injection angle of 70° and eight holes was observed and is attributed to a larger amount of fuel film due to the longer spray tip penetration length. Near-stoichiometric conditions in diffusion combustion of a pool fire can cause highest NOx emission for this injector. NOx emission decreased with increased EGR; however, NOx–HC/CO and NOx–smoke trade-off occurred for both injectors with a narrow injection angle when the EGR was increased. At EGR rate of about 40% with lower oxygen concentration, unburned fuel film due to deteriorated pool fire activity can increase the HC and CO emissions. Smoke for an injector with an injection angle of 70° and eight holes is attributed to the relatively larger fuel film. A portion of the fuel film causes a diffusion flame, which causes the smoke, which cannot be oxidized due to the lack of oxygen and decreased combustion temperature, and is emitted. For an injector with an injection angle of 70° and 14 holes at EGR rate of 40%, the shorter penetration length due to the reduced nozzle diameter caused less fuel film formation;

moreover, better air utilization leads to a lower level of HC and CO emissions, including smoke emission.

Funding

The authors would like to express their appreciation for the financial support by the Ministry of Knowledge Economy for project no.10033392.

Acknowledgement

The authors thank Dr Jungseo Park and the Zenobalti Company for their cordial advice.

References

1. Johnson TV. Diesel emission control in review. SAE paper 2009-01-0121, 2009.
2. Zhao F, Asmus TN, Assanis DN, et al. *Homogeneous charge compression ignition (HCCI) engines: key research and development issues*. Warrendale, PA: SAE International, 2003, PT-94.
3. Ryan TW and Callahan TJ. Homogeneous charge compression ignition of diesel fuel. SAE paper 961160, 1996.
4. Gray AW and Ryan TW. Homogeneous charge compression ignition (HCCI) of diesel fuel. SAE paper 971676, 1997.
5. Takeda Y, Keiichi N and Keiichi N. Emission characteristics of premixed Lean diesel combustion with extremely early staged fuel injection. SAE paper 961163, 1996.
6. Yokota H, Kudo Y, Nakajima H, et al. A new concept for low emission diesel combustion. SAE paper 970891, 1997.
7. Iwabuchi Y, Kawai K, Shoji T, et al. Trial of new concept diesel combustion system – premixed compression-ignition combustion. SAE paper 1999-01-0185, 1999.
8. Walter B and Gatellier B. Development of the high-power NADI™ concept using dual-mode diesel combustion to chieve zero NOx and particulate emissions. SAE paper 2002-01-1744, 2002.
9. Kimura S, Aoki O, Kitahara Y, et al. Ultra-clean combustion technology combining a low-temperature and premixed combustion concept for meeting future emission standards. SAE paper 2001-01-0200, 2001.
10. Lee S and Reitz RD. Spray targeting to minimize soot and CO formation in premixed charge compression ignition (PCCI) combustion with a HSDI diesel engine. SAE paper 2006-01-0918, 2006.
11. Okude K, Mori K, Shiino S, et al. Premixed compression ignition (PCI) combustion for simultaneous reduction of NOx and soot in diesel engine. SAE paper 2004-01-1907, 2004.
12. Lechner GA, Jacobs TJ, Chryssakis CA, et al. Evaluation of a narrow spray cone angle, advanced injection timing strategy to achieve partially premixed compression ignition combustion in a diesel engine. SAE paper 2005-01-0167, 2005.
13. Kook S and Bae C. Combustion control using two-stage diesel fuel injection in a single-cylinder PCCI engine. *SAE Int J Engines* 2004; 1(1): 563–578.
14. Ojeda W, Zoldak P, Espinosa R, et al. Development of a fuel injection strategy for partially premixed compression ignition combustion. SAE paper 2009-01-1527, 2009.
15. Reveille B, Kleemann A, Knop V, et al. Potential of narrow angle direct injection diesel engines for clean combustion: 3D CFD analysis. SAE paper 2006-01-1365, 2006.
16. Noehre C, Andersson M, Johansson B, et al. Characterization of partially premixed combustion. SAE paper 2006-01-3412, 2006.
17. Vanegas A, Won H and Peters N. Influence of the nozzle spray angle on pollutant formation and combustion efficiency for a PCCI diesel engine. SAE paper 2009-01-1445, 2009.
18. Kashdan JT, Mendez S and Bruneaux G. On the origin of unburned hydrocarbon emissions in a wall guided, low NOx diesel combustion system. SAE paper 2007-01-1836, 2007.
19. Fang T, Coverdill RE, Lee CF, et al. Smokeless combustion within a small-bore HSDI diesel engine using a narrow angle injector. SAE paper 2007-01-0203, 2007.
20. Martin GC, Mueller CJ, Milam DM, et al. Early direct-injection, low-temperature combustion of diesel fuel in an optical engine utilizing a 15-hole, dual-row, narrow-included-angle nozzle. *SAE Int J Engines* 2008; 1(1): 1057–1082.
21. Fang T-G, Coverdill RE, Lee C-F, et al. Low-sooting combustion in a small-bore high-speed direct-injection diesel engine using narrow-angle injectors. *Proc IMechE, Part D: J Automobile Engineering* 2008; 222(10): 1927–1937.
22. Fang T, Coverdill RE, Lee CF, et al. Effects of injection angles on combustion processes using multiple injection strategies in an HSDI diesel engine. *Fuel* 2008; 87(15–16): 3232–3239.
23 Kang J, Bae C and Lee KO. Initial development of non-evaporating diesel sprays in common-rail injection systems. *Int J Engine Res* 2003; 4(4): 283–298.
24. Lee J, Park J, Jeon J, et al. Assessment of JP-8 and diesel in direct injection diesel engine. In: *THIESEL 2008 conference on thermo- and fluid dynamic processes in diesel engines*, 9–12 September 2008, Valencia, Spain, pp.355–365. Spain: Universidad Politécnica de Valencia.
25. Dec JE. A conceptual model of DI diesel combustion based on laser-sheet imaging. *SAE Int J Engines* 1997; 1(1): 1319–1348.
26. Jung Y, Bae C, Jang J, et al. Improvement of premixed compression ignition combustion using various injector configurations. SAE paper 2011-01-1357, 2011.
27. Ladommatos N, Abdelhalim S, Zhao H, et al. The dilution, chemical and thermal effects of exhaust gas recirculation on diesel engine emissions – Part 1: effect of reducing inlet charge oxygen. SAE paper 961165, 1996.
28. Ladommatos N, Abdelhalim SM, Zhao H, et al. The effects of carbon dioxide in exhaust gas recirculation on diesel engine emissions. *Proc IMechE, Part D: J Automobile Engineering* 1998; 212(1): 25–42.
29. Zhao H, Hu J and Ladommatos N. In-cylinder studies of the effect of CO_2 in exhaust gas recirculation on diesel combustion and emissions. *Proc IMechE Part D: J Automobile Engineering* 2000; 214(4): 405–419.
30. Kook S, Bae C and Kim J. Diesel-fuelled homogeneous charge compression ignition engine with optimized premixing strategies. *Int J Engine Res* 2007; 8(1): 127–137.