Diesel-fuelled homogeneous charge compression ignition engine with optimized premixing strategies

Sanghoon Kook, Choongsik Bae and Jangheon Kim

International Journal of Engine Research 2007 8: 127
DOI: 10.1243/14680874JER02506

Diesel-fuelled homogeneous charge compression ignition engine with optimized premixing strategies

Sanghoon Kook[1], **Choongsik Bae**[1]*, and **Jangheon Kim**[2]

[1] Department of Mechanical Engineering, Korea Advanced Institute of Science and Technology, Daejeon, Republic of Korea

[2] Powertrain Laboratory, Hyundai Motor Company, Kyunggi-Do, Republic of Korea

The manuscript was accepted after revision for publication on 16 October 2006.

DOI: 10.1243/14680874JER02506

Abstract: The operation of a diesel-fuelled homogeneous charge compression ignition (HCCI) engine was studied in a single-cylinder, direct-injection diesel engine with regard to three key parameters: spray penetration, time for premixing, and dilution of the premixed charge. The relationships between these parameters were clarified through spray measurements, flame imaging, and combustion analysis. The spray penetration was optimized by a small included angle to avoid wall impingement at low pressure and temperature in the cylinder. However, the hole diameter did not affect spray penetration. Sufficient time for premixing was realized by advanced injections earlier than 100 crank angle degrees (CAD) before top dead centre (BTDC) at 800 r/min. Dilution of the premixed charge to control ignition timing was investigated by adopting exhaust gas recirculation (EGR). The optimized premixing strategies of two-stage injection, with a small amount of the ignition-promoting fuel (1.5 mm^3) – which was injected near TDC to assist the combustion of a premixed charge (10 mm^3) – resulted in an indicated mean effective pressure of up to 250 kPa within 3 per cent fluctuation, along with a significant reduction in particle matter and nitrogen oxides emissions, while 46 per cent EGR rate was applied to the premixed charge with preheated intake air at 433 K.

Keywords: homogeneous charge compression ignition (HCCI), direct-injection, spray wall impingement, small included angle, multi-hole injector, injection timing, exhaust gas recirculation

1 INTRODUCTION

Diesel-fuelled homogeneous charge compression ignition (HCCI) combustion is an advanced technique for reducing hazardous nitrogen oxides (NO_x) and particulate matter (PM) emissions at low load operation, while providing high thermal efficiency in a diesel engine. NO_x emissions can be reduced by achieving a lean homogeneous mixture through the entire cylinder volume, resulting in low combustion temperature. A homogeneous charge having no fuel-rich zones, unlike conventional diesel combustion, also reduces PM emissions. There are two main approaches to supplying a homogeneous charge before ignition: port-injection where diesel fuel is supplied to the intake port [**1**, **2**], and in-cylinder injection where fuel is injected directly into the combustion chamber. Although port-injection theoretically guarantees sufficient time for premixing and enables HCCI combustion, wall wetting on account of the low temperature condition in the intake port is a major problem in practical applications. Moreover, the mode transition between HCCI and conventional diesel combustion is impossible with one fueling system [**3**]. For better premixed charge preparation, in-cylinder direct injection has been widely tested with the most advanced common-rail fuel injection system [**4–12**].

The accomplishment of homogeneous charge with the in-cylinder direct-injection method is particularly influenced by three parameters.

1. Spray penetration: the fuel spray penetration should be long enough to achieve spatially

** Corresponding author: Department of Mechanical Engineering, Korea Advanced Institute of Science and Technology, 373-1 Guseong-Dong, Yuseong-Gu, Daejeon 305-701, Republic of Korea. email: csbae@kaist.ac.kr*

uniform fuel distribution but, simultaneously, short enough to impede spray impingement to the cylinder liner or the piston. This parameter is controlled by the injector geometry. Multi-hole injectors have been applied to promote the atomization and spatial distribution of the mixture [**4**, **5**]. A swirl injector with a pintle-type nozzle and two impinging side injectors has been tested to minimize the spray wall impingement [**6**]. In addition, an impinged-spray nozzle injector [**7**], or a hole injector with a small included angle [**8**, **9**] were investigated to resolve the same problem.

2. The time for premixing: generally in HCCI engine studies, the fuel injection timing is extremely advanced, and is earlier than conventional diesel combustion so as to guarantee sufficient time for premixing [**4–10**]. However, at this early injection timing, in-cylinder air pressure and temperature are so low that spray impingement and wall wetting becomes serious. Some studies employed multiple injection strategies using only one injector [**4**, **9**, **10**] or several injectors [**5**, **6**]. The earlier injection was performed to form a premixed charge without the spray-impingement problem, and was followed by the injection near the top dead centre (TDC). Other studies, meanwhile, have focused on injections near TDC, while premixing is promoted by high swirl ratio and long ignition delay employed by a high rate of exhaust gas recirculation (EGR) [**11**, **12**].
3. The dilution of the premixed charge: generally, the ignition timing of HCCI combustion is earlier than that of conventional diesel combustion because of the hot combustion driven by the cool combustion [**1**, **3**, **13**]. This early ignition problem becomes more severe if the intake air is preheated to assist vaporization of the diesel [**1**, **2**, **6**]. Previous researchers have used the charge dilution by EGR to retard the combustion phase [**2**, **6**, **7**, **8**, **10**]. Diluting the intake charge extends the ignition delay and lowers the combustion temperature due to the reactant being replaced by the inert gases with increased heat capacity.

The objective of this study is to suggest premixing strategies that optimize the three parameters noted above. Injection rate measurements and macroscopic spray visualization were conducted for sac nozzle injectors with various included angles ($70^\circ \sim 150^\circ$) and hole numbers ($5 \sim 14$ holes) in a spray chamber. The fuel injection timing was widely controlled from the compression stroke to the intake stroke with a common-rail fuel injection system incorporated in a single-cylinder diesel engine at 800 r/min. In addition, the flame images were obtained over a wide range of injection timing. EGR was employed up to 46 per cent, so as to control combustion phase. The fluctuation of power outputs was carefully monitored with EGR rate change. In addition, a small amount of fuel (1.5 mm^3) was injected near TDC to promote ignition of in-cylinder charge (total fuel amount = 11.5 mm^3) with preheated intake air of 433 K. Its effect on power output and combustion stability was investigated at a fixed EGR rate of 46 per cent.

2 EXPERIMENTS

2.1 Spray measurements in a chamber

The spray penetrations of sac-nozzle injectors were measured in a spray chamber with a volume of 13 804 cm^3 under atmospheric pressure and temperature. Figure 1 shows the experimental set-up. The chamber allows optical access through circular windows 89 mm in diameter. Spray images frozen by a spark light source (MVS-2601-CE96; EG&G Optoelectronics, Fremont, California), which had a light duration of shorter than 100 ns, were acquired with a CCD camera (SensiCam; PCO CCD Imaging, Kelheim, Germany). The macroscopic spray structures were manifested by this imaging technique [**14**]. The air density or pressure was controlled to meet the conditions at early injection timing, generally an open-valve condition, which is close to atmospheric pressure. At first, the 5-hole injector (Bosch, Stuttgart, Germany) with an included angle of 150° was tested, and then smaller included angles of 100° and 70° were applied to overcome spray wall impingement on the cylinder liner. Injectors with 8 and 14 holes were tested to enhance the atomization as well as to shorten the spray penetration. The main injector geometries are listed in Table 1. The

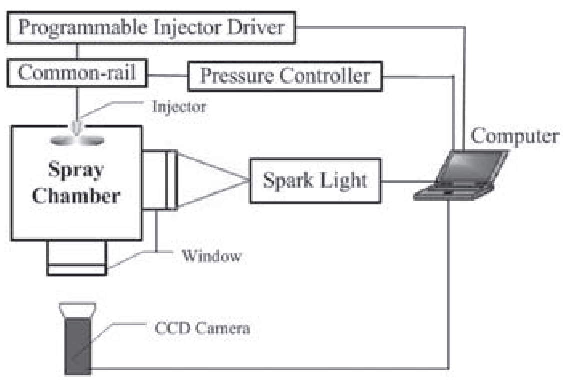

Fig. 1 Schematic diagram of spray chamber set-up

Table 1 Injector geometries (sac-nozzle injectors; Bosch)

Case	Hole number	Hole diameter (mm)	Included angle (deg)
I	5	0.168	150
II	5	0.168	100
III	8	0.133	100
IV	14	0.100	150
V	14	0.100	100
VI	14	0.100	70

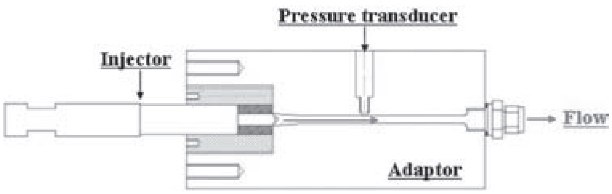

Fig. 2 Cross-sectional view of the injection rate meter (Bosch method)

injection rate was measured with an injection rate meter based on the Bosch method, as shown in Fig. 2 [**15**]. The principle of this method is to measure the pressure difference (dp) with a pressure transducer, while the injected diesel fuel passes through the tube. This pressure difference was converted to the velocity difference (du) and subsequently the volume flowrate of the diesel fuel following mass and momentum conservation laws.

$$\mathrm{d}p = c\rho_1 \mathrm{d}u \qquad (1)$$

where c is the velocity of sound and ρ_1 is liquid diesel density.

2.2 Research engine

The single-cylinder optical HCCI engine (RSi-090; Engine Tech, Hwasung, Korea) used for this study is based on a typical direct-injection diesel engine [**9**]. The bore and stroke were selected to be comparable with the specifications of a typical passenger car engine. The engine's specifications are summarized in Table 2. A schematic diagram of the engine set-up is shown in Fig. 3. The temperature of the intake air was controlled by an electrical heater and k-type thermocouple with a temperature control unit. Fuel injection parameters – including injection pressure, quantity, and timings – were controlled by a programmable injector driver (TDA3000H; TEMS, Daejeon, Korea) and a pressure regulator. A rotary encoder (Koyo; 3600 pulses/rev, Hoffman Estates, Illinois) attached to a camshaft was used to control the operating timings of the multiple fuel injections and the camera. Cylinder pressure was recorded with a piezoelectric pressure transducer (6052A; Kistler, Ostfildern, Germany) at every 0.16 crank angle degrees (CAD). An opacimeter (OP-120; EplusT, Seoul, Korea) satisfying SAE J1667 regulations was used to measure the exhaust opacity that was representing PM emissions. An emission measurement system (MEXA1500D; Horiba, Koyto, Japan), sampling the exhaust gas after the opacimeter, was employed to measure the concentrations of NO_x, HC, and CO. An EGR line was installed from the exhaust pipe to the intake pipe, with a surge tank to minimize the pressure fluctuations. The cylinder pressures and exhaust gas concentrations of 138 engine cycles were recorded and averaged to calculate indicated mean effective pressure (IMEP) and apparent heat release rate.

Table 2 Specifications of the research engine

Engine type	Single-cylinder, direct injection, four-valves, optical diesel engine
Bore	83 mm
Stroke	92 mm
Displacement	498 mm^3
Compression ratio	18.9
Fuel injection equipment	Bosch common rail

Figure 3 also illustrates the optical access to the combustion chamber and the observation field. An elongated piston was installed in order to enable the mounting of a 45° mirror beneath the piston quartz window, which allowed a 50 mm diameter visual field in 83 mm bore. A high-speed digital video camera (Phantom v7.0; Vision Research, Wayne, New Jersey) with an image intensifier (Proxitronic, Bensheim, Germany) was used to acquire the combustion images inside the optical engine. This set-up was corresponding to the high-speed imaging up to 10 000 frames per second with resolution of 512×384 pixels, thus the images could be taken each 0.48 CAD. The exposure time of the camera was optimized to 35 μs to obtain flame images without blurring. The images were highly amplified through an intensifier to show the non-luminous flame. The maximum gain of the intensifier was 10^8 W/W and the maximum light power per area was 70 μW/cm^2. An intensity gain of 50 per cent was used for detecting weak HCCI combustion, except in the highly luminous case where the gain value was decreased to 30 per cent.

2.3 Engine operating conditions

The engine operating conditions are listed in Table 3. The engine was operated at 800 r/min under both motored and fired conditions. In order to maintain consistency of performance data and flame images, compression pressure was kept at 3.2 MPa for every

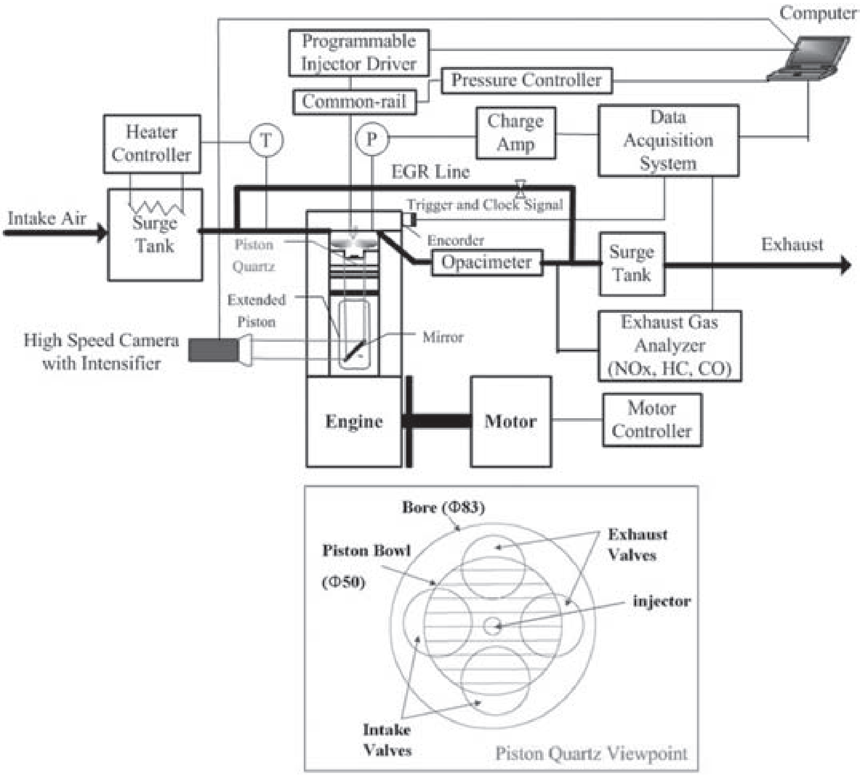

Fig. 3 Schematic diagram of the engine set-up and visual field

Table 3 Engine operating conditions

Coolant temperature	353 K
Diesel fuel temperature	313 K
Common-rail pressure	120 MPa
Intake air temperature	433 K
EGR rate	0 ~ 46%
Main injection timing	250 ~ 50 CAD BTDC
Second injection timing of the two-stage injection	20, 10 CAD BTDC and TDC
Total quantity of fuel injection	11.5 mm^3
Quantity of second injection	1.5 mm^3
Fuel	Conventional diesel (cetane no. = 50)

Engine speed = 800 r/min.

experiment. This was achieved by the motoring procedure before fuel injection. During the motoring procedure, the peak pressure kept rising and then reached a stable condition in 120 s. The coolant temperature was controlled at 353 K and diesel fuel temperature at 313 K. The common-rail pressure that was close to the injection pressure was selected as 120 MPa. This high-pressure injection could aggravate the wall-impingement problem. However, our earlier study showed that the total heat release was increased with increasing injection pressure due to the improved atomization and the better mixing with air [**9**]. Therefore, the injection pressure was maintained at a relatively high value in order to achieve as high a power output as possible, which was comparable with that in the conventional diesel engine. Intake air temperature was controlled and held constant at 433 K. This temperature was high enough to assist the vaporization of diesel fuel, such that the power output was high even with the early ignition [**9**]. In this study, the early ignition problem was resolved by introducing EGR. The EGR rate is defined as the ratio between the CO_2 concentration of the intake air, $X_{CO_2}^{in}$, and the CO_2 concentration of the exhaust gas, $X_{CO_2}^{ex}$

$$\text{EGR rate} = \frac{X_{CO_2}^{in}}{X_{CO_2}^{ex}} \times 100\ (\%) \tag{2}$$

EGR rate was varied from 0 to 46 per cent. The injection timings were widely changed from 50 CAD BTDC to 250 CAD BTDC. For near 50 CAD BTDC, in particular, more precise control was conducted, as this timing showed transition characteristics between conventional diesel combustion and homogeneous combustion. The fuel quantity of 11.5 mm^3 was fixed for each injection, this quantity was selected from conventional engine data corresponding to the idling condition at 800 r/min.

All data for performance and flame images were compared with direct-injection diesel combustion under the following conditions: the injection timing was 15 CAD BTDC; the injection pressure was 120 MPa and total fuel quantity was 11.5 mm^3 for a

5-hole injector with an included angle of 150°; the intake air was not preheated so that the temperature was kept at 303 K.

3 RESULTS AND DISCUSSION

3.1 Spray penetration in the spray chamber

The corresponding spray penetrations were measured from macroscopic liquid spray images in the spray chamber. Figure 4 shows an example; because the line of sight was longitudinal to the injector axis, it depicts radial penetrations for the 14-hole injector. The position of the cylinder wall is superimposed. The ten images were averaged to estimate the penetration. This procedure was repeated for each hole. The injectors used are listed in Table 1; the injection rate was also measured. Figure 5 shows an injection delay of 320 μs which represents the time lag between the injection start signal and the actual start of injection; this arises mainly from the time required to build enough magnetic field to open the valve and the mechanical response for the needle to open. The injection signal was detected by a high-current probe, which measured the current flowing to the solenoid valve of the injector. The injection delay was estimated in order to correct the corresponding time after start of injection for the spray images. The rectangular-shaped injection signal induced a linear increase and a sudden subsequent decrease in the injection rate. The total amount of injected fuel was carefully maintained at 11.5 mm^3 for all injectors. The common-rail pressure was monitored during the injection. The pressure decreased during the injection and recovered to 120 MPa. This indicated that the injection pressure was not constant but fluctuating during the injection event.

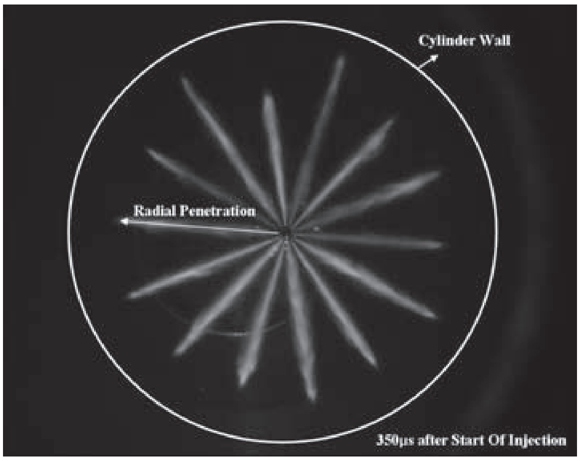

Fig. 4 The macroscopic spray visualization in the spray chamber; 14-hole injector, included angle = 100° (case V)

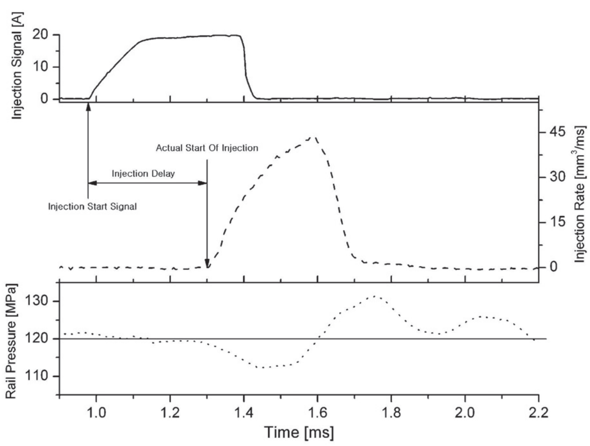

Fig. 5 Injection signal, injection rate and common-rail pressure; 14-hole injector, included angle = 100° (case V)

As investigated previously [**14**], spray penetration was proportional to the square root of the hole diameter and time, when sufficient time had elapsed from the start of injection at room temperature. It was also reported that a 16-hole centre injector (small-hole diameter of 0.080 mm) could be used for the purpose of short penetration with the help of two additional side injectors [**5**]. Shorter penetration by smaller hole diameter, however, could not be achieved in HCCI combustion due to the lower in-cylinder air density and pressure. Figure 6 shows the spray penetrations as a function of time for various numbers of holes, i.e. hole diameters. The ambient conditions were close to the atmospheric pressure and density (1.189 kg/m^3 at 298 K). There was no significant difference in spray penetration regardless of the hole

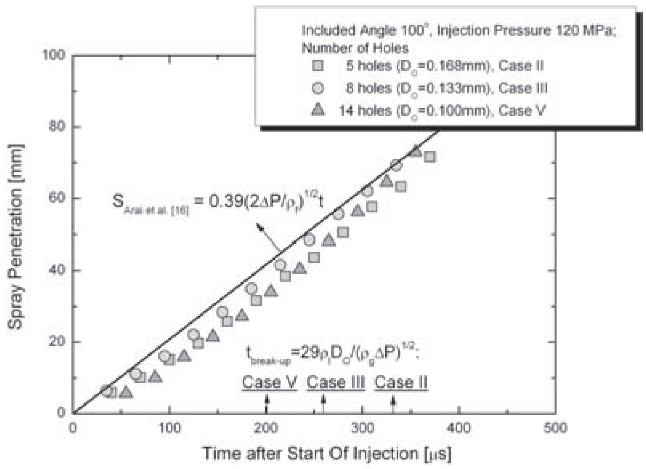

Fig. 6 Effect of hole diameter on initial spray penetrations at low density ambient conditions; injection pressure = 120 MPa

diameter. The spray penetrations were almost linearly proportional to the time after the start of injection until the end of injection.

The results were comparable with the previously reviewed correlations of initial spray penetration [**14**]. Arai *et al.* [**16**] reported that penetration (S) increased linearly with time within the jet break-up time ($t_{\text{break-up}}$), as shown in Fig. 6. Naber and Siebers [**17**] reported the same type of correlation for the initial injection time as follows:

$$S = 0.39(2\,\Delta P/\rho_l)^{1/2}t \tag{3}$$

$$t_{\text{break-up}} = 2\rho_l D_o/(\rho_g \Delta P)^{1/2} \tag{4}$$

For the estimations in each case, the pressure difference between the inside of the hole and the chamber (ΔP) was fixed to 120 MPa. The diesel density (ρ_l) and the charge air density (ρ_g) were maintained at 839 and 1.189 kg/m^3 respectively. From Fig. 6, measured spray penetrations follow equation (3). The slight over-prediction could be attributed to the variation in the injection pressure as captured in Fig. 5. Note that equation (3) was derived from the constant-injection-pressure experiment.

Generally, the initial spray penetration is independent of the hole diameter (D_o) [**14**, **16**, **17**]. In Fig. 6, $t_{\text{break-up}}$ for corresponding conditions given by equation (4) was 200.5$\sim$336.8 μs, and hence much of the injection occurred within the initial spray-penetration range. Consequently, the wall impingement at HCCI operation could not be reduced because the spray penetration was not shortened by the small hole diameter.

A small included angle, extending the axial spray penetration and shortening the radial spray penetration, was examined to resolve the problem of spray wall impingement. Figure 7 shows the axial and radial spray penetrations relative to the position of the piston and cylinder liner at various included angles both for 5-hole and 14-hole injectors. For the 5-hole injector (Fig. 7(b)), the penetration of a conventional diesel injection was shown under a high charge air density of 33.8 kg/m^3. The spray wall-impingement problem occurred under a low charge air density of 1.189 kg/m^3. The axial development of spray (S_z) did not impinge on the top of the piston if the injection timing was controlled near BDC, but the radial development of spray (S_r) predicted severe wall impingement. The spray impinged the cylinder liner if the included angle was 150°. The increased wetting could cause poor mixing and air utilization resulting in low combustion efficiency and high PM emission. The spray with the included angle of 100° impinged less than the included angle of 150°, as shown in Fig. 7(b). The conflict was eased with the

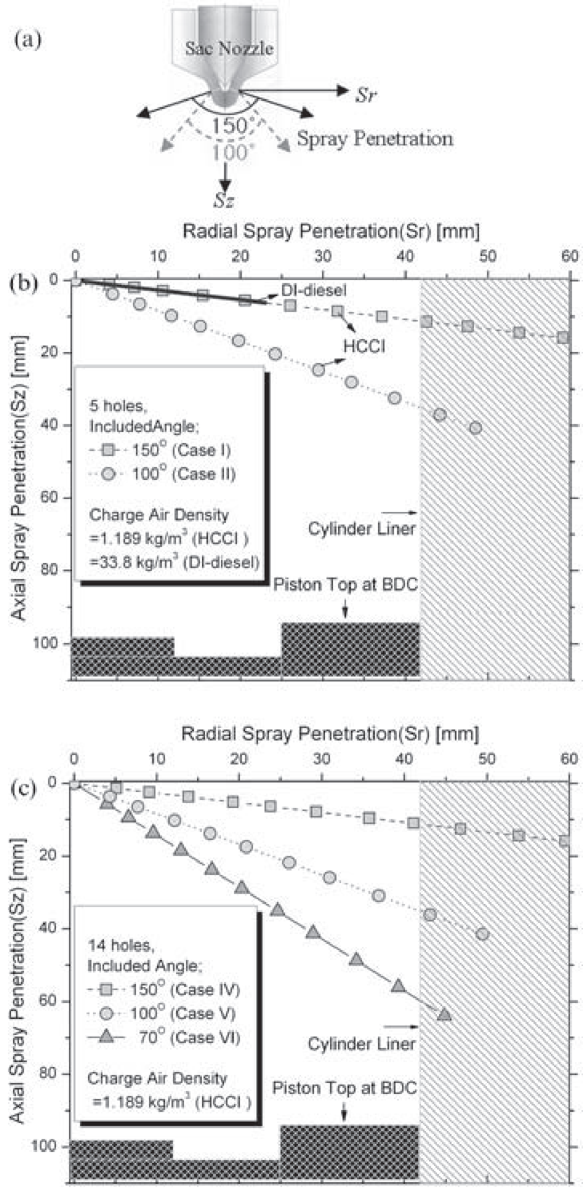

Fig. 7 Effect of included angle on spray wall impingement, 5- and 14-hole injectors. (a) Definition of included angle; (b) penetrations with 5-hole injector; (c) penetrations with 14-hole injector

included angle of 70°, as shown in Fig. 7(c) for the 14-hole injector.

It has been reported that application of a multi-hole injector (e.g. 30 holes, 0.100 mm hole diameter) for HCCI combustion promotes atomization of the spray [**4**]. It was also reported that supplying higher injection pressure through a smaller hole facilitates better fuel atomization and evaporation [**18**], and the smaller hole diameter tended to increase air entrainment into the spray [**19**]. On the other hand, some researchers found that small hole diameter reduced spray penetration, resulting in poor spatial distribution of fuel [**20**], and created a fuel-rich region

[21]. This would not be a problem for HCCI combustion on account of the independence of the hole diameter as noted above. However, as the mode transition with only one fueling system is the most important advantage of the early direct-injection method, poor air utilization for the conventional diesel combustion mode, corresponding to warm-up, cold-start, or high load conditions, would be a serious source for high PM emission. From this point of view, an overly small included angle would also be a problem for near-TDC injection. Consequently, the 5-hole injector with an included angle of 100° (case II) was selected for further investigation through the following experiments.

3.2 The time for premixing

The fuel injection timing is a control parameter for the time of fuel/air premixing. It should be early enough to achieve a thoroughly homogeneous charge before ignition. However, the wall wetting increases and fuel vaporization becomes more difficult as the in-cylinder temperature is lowered. Previous researchers limited the injection timing to near 50 CAD BTDC [**10**] or near TDC [**11**, **12**] due to low power output possibly associated with the wall wetting. In this study, the wall-wetting problem was resolved by a modified injector with a small included angle along with preheated intake air of 433 K. Injection timings were then varied from 50–250 CAD BTDC to determine the minimum premixing time for HCCI combustion.

Figure 8 shows the effect of injection timings on IMEP, exhaust emissions, in-cylinder pressure trace, and heat release rate. The performance of conventional diesel combustion is also shown. It is notable that IMEP was higher than the reference value at

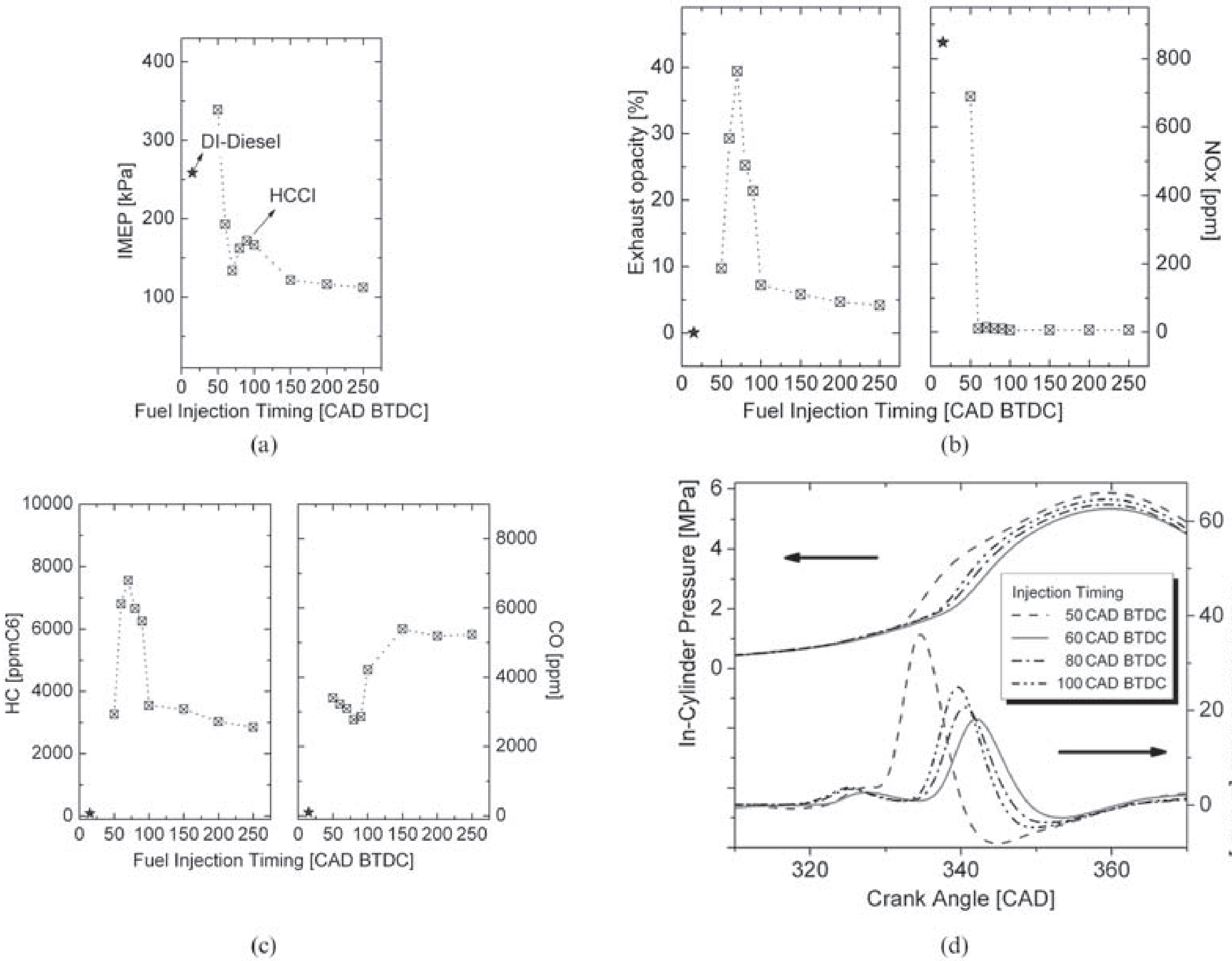

Fig. 8 Effect of time for premixing (fuel injection timing) on IMEP, emissions, in-cylinder pressure trace, and heat release rate; single injection, 5-hole injector, included angle = 100° (case II), intake air temperature = 433 K, injection pressure = 120 MPa. (a) IMEP; (b) exhaust opacity and NO_x emissions; (c) HC and CO emissions; (d) in-cylinder pressure trace and heat release rate

the injection timing of 50 CAD BTDC. However, the exhaust opacity increase and little reduction in NO_x indicated that the combustion characteristics are still similar to those of conventional diesel combustion. When the injection was more advanced than 50 CAD BTDC, up to 100 CAD BTDC, NO_x emission was decreased. IMEP was decreased and then slightly increased while exhaust opacity and HC emissions were subsequently increased and then decreased with advancement of the injection timing. This could be interpreted as evidence of a mode transition from conventional diesel combustion to HCCI combustion. It was anticipated that the in-cylinder pressure and temperature at fuel injection were too low to complete the conventional diesel combustion while the time for premixing was not sufficient to form homogeneous charge. As the injection timing was advanced more than 100 CAD BTDC, IMEP was decreased, and the exhaust opacity was decreased because fuel/air premixing was improved so that a lean homogeneous mixture was achievable. NO_x emission maintained negligible values. Note that CO and HC emissions show trade-off. The CO emission increased as HC emission decreased. The combustion temperature should be slightly increased so that HC could be oxidized to CO.

The in-cylinder pressure trace and calculated heat release rate showed the same characteristics. For injection timing of 50 CAD BTDC, the ignition delay was about 10 CAD as in conventional diesel combustion, and hence its combustion phase was more advanced than in the other cases. As the injection timing was more advanced than 50 CAD BTDC, the combustion phase was retarded. It is notable that the injection timings of 80 and 100 CAD BTDC yielded long ignition delay, thus providing adequate time for premixing. These timings resulted in the same start timing of cool combustion. Moreover, the following hot combustion took place at the same timing for different injection timings. This implies that fuel injection timings do not affect the ignition timings.

Visualization of combustion clarified whether the time for premixing was adequate. Figure 9 shows the flame images, corresponding to the results shown in Fig. 8. For the conventional diesel combustion, the flames were highly luminous because of the thermal radiation from soot. As such, direct imaging without the image intensifier was possible, as shown in Fig. 9(a). The combustion started from local regions where the flame temperature was relatively high resulting in the formation of NO_x. The scattered luminous regions show and imply that the premixing between the diesel fuel and air was not sufficient in certain regions. As the injection timing advanced to

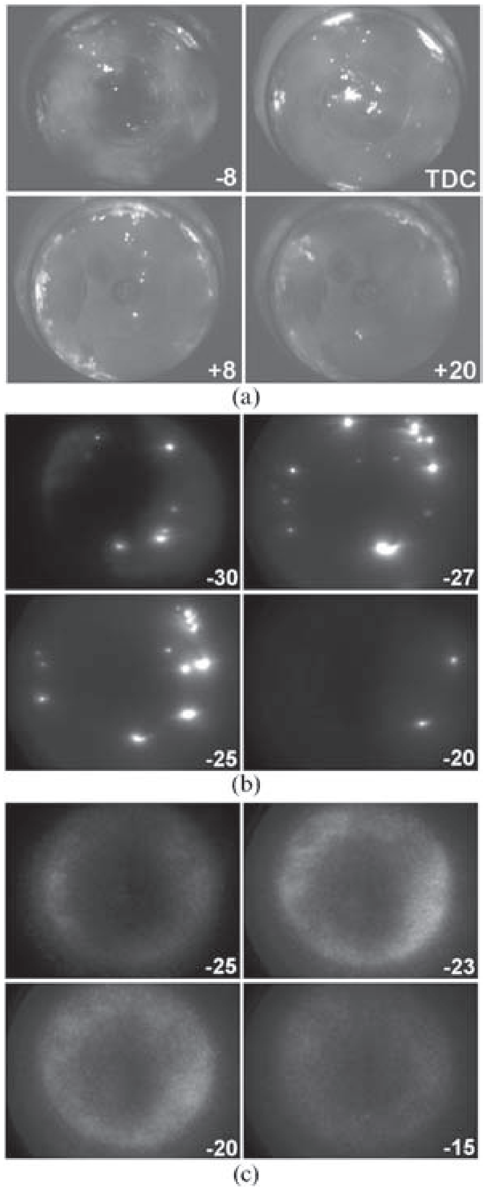

Fig. 9 Flame images under injection timing variation; single injection, 5-hole injector, injection pressure = 120 MPa. (a) Injection timing = 15 CAD BTDC; without intensifier, 5-hole injector, included angle = 150° (Case I), intake air temperature = 303 K. (b) Injection timing = 50 CAD BTDC; intensifier gain of 30 per cent, 5-hole injector, included angle = 100° (case II), intake air temperature = 433 K. (c) Injection timing = 100 CAD BTDC; intensifier gain of 50 per cent, 5-hole injector, included angle = 100° (Case II), intake air temperature = 433K. Crank angle (ATDC) is at lower right

50 CAD BTDC with intake air preheating (Fig. 9(b)), the flames lost much of the luminosity. For this reason, an image intensifier with a level of 30 per cent was installed in front of the high-speed camera. Although the time for premixing was extended,

luminous fragments can still be detected in Fig. 9(b). The remaining luminosity provided proof of the rich mixture. The flame intensity became spatially near-uniform without a luminous flame at the injection timing of 100 CAD BTDC, as shown in Fig 9(c). Cool combustion was not detected through the direct imaging, while an image intensifier with a level of 50 per cent was used to detect the weak flame intensity. From the results of engine performance and combustion visualization, it became clear that sufficient time for premixing induced homogenous combustion resulting in low NO_x and PM emissions.

3.3 The dilution of the premixed charge with two-stage injection

Diesel injection without spray wall impingement, which was conducted at sufficiently early timing, supplied a well-premixed charge into the cylinder, but the early ignition induced low thermal efficiency as the combustion phase was not controlled effectively. Previous researchers focused on the effect of EGR and reported retarded ignition timing [**6–8**].

Figure 10 shows that ignition timing is retarded as the EGR rate increases from 15 to 46 per cent. The peak of the heat release rate was decreased, but the combustion period was extended resulting in high thermal efficiency. It was explicit that the fresh air in the intake pipe was diluted by the exhaust gas and, due to the reactant being replaced by the inert gases with increased heat capacity, the chemical reaction of the premixed charge was slowed down.

Figure 11(a) shows the increase of power output with increasing EGR rate for the single-injection case.

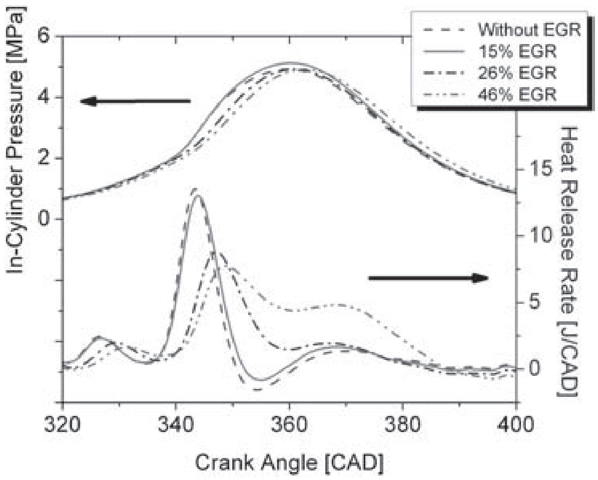

Fig. 10 Effect of EGR on in-cylinder pressure trace and heat release rate; single injection, 5-hole injector, included angle = 100° (case II), intake air temperature = 433 K, injection pressure = 120 MPa

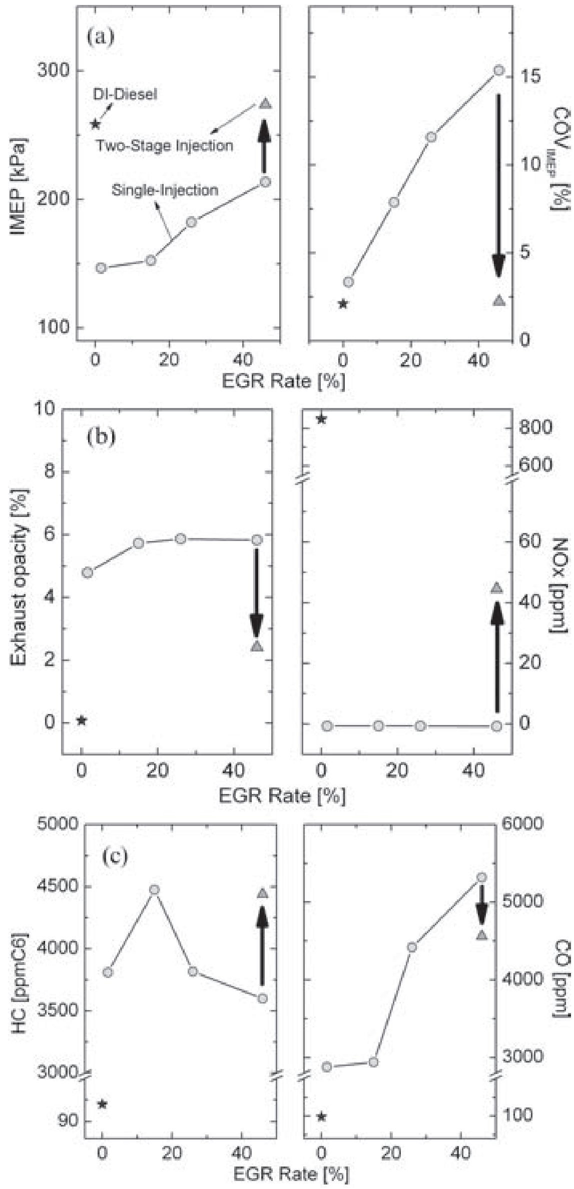

Fig. 11 Effect of EGR on IMEP and emission; single (injection timing = 250 CAD BTDC) and two-stage injection (main injection timing = 250 CAD BTDC and second injection timing = TDC), 5-hole injector, included angle = 100° (case II), intake air temperature = 433 K, injection pressure = 120 MPa. (a) IMEP and COV of IMEP; (b) exhaust opacity and NO_x emissions; (c) HC and CO emissions

The problem in applying EGR to HCCI combustion is the transient response and temperature instability of recycled gas [**7**]. As shown in the figure, the coefficient of variation (COV) of IMEP was detrimentally increased with high EGR. A two-stage injection could be a potential solution. Previously, the two-stage injection strategy was introduced as a concept of ignition enhancing [**9**]. It was achieved by injecting a small amount of fuel (1.5 mm^3) (termed the second injection) near TDC, while most of the fuel (10 mm^3) (termed the main injection) was injected earlier than 100 CAD BTDC to make a lean homogeneous charge. Figure 11(a) shows that the two-stage injection increases the IMEP compared with the single injection for the same total amount of fuel (11.5 mm^3). Moreover, the COV of IMEP is improved to 3 per cent or less, which is comparable with the conventional diesel combustion.

With increasing EGR for the single injection, the exhaust opacity slightly increased while the NO_x emission was negligible, as shown in Fig. 11(b). The two-stage injection reduced the exhaust opacity by improving the late-cycle oxidation. The increase in combustion temperature resulted in an increase in NO_x emission. The increased NO_x emission, however, was less than 10 per cent of that of conventional diesel combustion. Figure 11(c) shows the HC and CO emissions. The one drawback of the two-stage injection was increased HC due to fuel injection into the burning gas, which resulted in a locally rich flame, i.e. an under-mixed mixture [**18**]. The CO emission showed the reverse trend to HC emission, as it did in Fig. 8(c). The CO emission decreased as less HC could be oxidized to CO. It was anticipated that the optimized combustion phasing by the two-stage injection along with high EGR would result in high power output although the combustion efficiency decreased.

4 CONCLUSIONS

The three key parameters – spray penetration, time for premixing, and dilution of premixed charge with the two-stage injection – for the operation of a diesel-fuelled HCCI engine were tested in a single-cylinder direct-injection diesel engine. The major findings from these investigations are summarized below.

1. At the ambient condition under low air pressure and density, the initial spray penetrations were found to be independent of the hole diameter. Therefore, a small hole diameter (0.100 mm for the 14-hole injector) could not be a solution to resolve the spray wall impingement. However, an injector with a small included angle of 70° reduced the spray wall impingement by extending the axial spray penetration and shortening the radial spray penetration.
2. Sufficient time for premixing was guaranteed by extremely early injection timing. Heat release rate and flame images verified that homogeneous combustion could be achieved with injection timings earlier than 100 CAD BTDC at 800 r/min.
3. The EGR of the maximum 46 per cent EGR rate with the preheated intake air of 433 K retarded the ignition timing and extended the combustion period, and consequently the thermal efficiency was significantly improved.
4. Optimized premixing strategies with a small amount of the ignition-promoting fuel (1.5 mm^3), which was injected near TDC to assist the combustion of a premixed charge (10 mm^3), resulted in IMEP of up to 250 kPa within 3 per cent fluctuation, along with significant reduction in PM and NO_x emissions.

ACKNOWLEDGEMENTS

The authors would like to express their appreciation for support by the National Research Laboratory scheme, Korea and in part by the Hyundai Motor Company.

REFERENCES

1 **Christensen, M., Hultqvist, A.,** and **Johansson, B.** Demonstrating the multi fuel capability of a homogeneous charge compression ignition engine with variable compression ratio. SAE technical paper 1999-01-3679, 1999.

2 **Gray, A. W.** and **Ryan III, T. W.** Homogeneous charge compression ignition (HCCI) of diesel fuel. SAE technical paper 971676, 1997.

3 **Stanglmaier R. H.** and **Roberts, C. E.** Homogeneous charge compression ignition (HCCI): benefits, compromises, and future engine applications. SAE technical paper 1999-01-3682, 1999.

4 **Yokota, H., Kudo, Y., Nakajima, H., Lalegawa, T.,** and **Suzuki, T.** A new concept for low emission diesel combustion. SAE technical paper 970891, 1997.

5 **Takeda, Y., Keiichi, N.,** and **Keiichi, N.** Emission characteristics of premixed lean diesel combustion with extremely early staged fuel injection. SAE technical paper 961163, 1996.

6 **Akagawa, H., Miyamoto, T., Harada, A., Sasaki, S., Shimazaki, N., Hashizume, T.,** and **Tsujimura, K.** Approaches to solve problems of the premixed lean diesel combustion. SAE technical paper 1999-01-0183, 1999.

7 **Iwabuchi, Y., Kawai, K., Shoji, T.,** and **Takeda, Y.** Trial of new concept diesel combustion system – premixed compression-ignited combustion. SAE technical paper 1999-01-0185, 1999.

8 **Walter, B.** and **Gaterllier, B.** Development of the high power NADITM concept using dual mode

diesel combustion to achieve zero NO_x and particulate emissions. SAE technical paper 2002-01-1744, 2002.
9 **Kook, S.** and **Bae, C.** Combustion control using two-stage diesel fuel injection in a single-cylinder PCCI engine. SAE technical paper 2004-01-0938, 2004.
10 **Hasegawa, R.** and **Yanagihara, H.** HCCI combustion in DI diesel engine. SAE technical paper 2003-01-0745, 2003.
11 **Shimazaki, N., Tsurushima T.,** and **Nishimura, T.** Dual mode combustion concept with premixed diesel combustion by direct injection near top dead center. SAE technical paper 2003-01-0742, 2003.
12 **Kimura, S., Aoki, O., Kitahara Y.,** and **Aiyoshizawa, E.** Ultra-clean combustion technology combining a low-temperature and premixed combustion concept for meeting future emission standards. SAE technical paper 2001- 01-0200, 2001.
13 **Yanagihara, H.** A study on combustion structure of premixed compression ignition diesel engines. In Proceedings of THIESEL 2002 Conference on *Thermo- and fluid-dynamic processes in diesel engines, homogeneous charge C. I.*, 2002, pp. 111–129.
14 **Kang, J., Bae., C.,** and **Lee, K. O.** Initial development of non-evaporating diesel sprays in common-rail injection systems. *Int. J. Engine Res.*, 2003, **4**(4), 283–298.
15 **Bosch, W.** The fuel rate indicator: a new measuring instrument for display of the characteristics of individual injection. SAE technical paper 660749, 1966.
16 **Arai, M., Tabata, M., Hiroyasu, H.,** and **Shimizu, M.** Disintegrating process and spray characterization of fuel jet injected by a diesel nozzle. SAE technical paper 840273, 1984.
17 **Naber, J. D.** and **Siebers, D. L.** Effects of gas density and vaporization on penetration and dispersion of diesel spray. SAE technical paper 960034, 1996.
18 **Heywood, J. B.** *Internal combustion engine fundamentals*, 1988 (McGraw-Hill, New York).
19 **Siebers, D. L.** and **Higgins, B. S.** Flame lift-off on direct-injection diesel sprays under quiescent conditions. SAE technical paper 2001-01-0530, 2001.
20 **Hiroyasu, H.** and **Arai, M.** Structures of fuel sprays in diesel engines. SAE technical paper 9000475, 1990.
21 **Rente, T., Gjirja, S.,** and **Denbratt, I.** Experimental study on the influence of nozzle orifice number on combustion and emissions formation of a heavy duty DI diesel engine. SAE_NA technical paper 2003-01-67, 2003.

APPENDIX

Notation

ATDC	after top dead centre
BTDC	before top dead centre
c	velocity of sound
CAD	crank angle degree
COV	coefficient of variation
dp	pressure difference in the pipe of the injection rate meter
du	velocity difference in the pipe of the injection rate meter
D_o	nozzle exit diameter
EGR	exhaust gas recirculation
HCCI	homogeneous charge compression ignition
IMEP	indicated mean effective pressure
S	spray penetration
S_r	radial spray penetration
S_z	axial spray penetration
t	time after start of injection
$t_{break\text{-}up}$	break-up time
TDC	top dead centre
ΔP	pressure difference between the exit nozzle and the ambient chamber
ρ_g	ambient gas density
ρ_l	liquid fuel density