EFFECTS OF ETHANOL ADDED FUEL ON EXHAUST EMISSIONS AND COMBUSTION IN A PREMIXED CHARGE COMPRESSION IGNITION DIESEL ENGINE

by

Yungjin KIM [a], ***Eiji TOMITA*** [b], ***and Kihyung LEE*** [a*]

[a]Department of Mechanical Engineering, Hanyang University, Sangnok-gu, Ansan-si, Kyeonggi-do, Republic of Korea
[b]Department of Mechanical Engineering, Okayama University, Tsushima-Naka, Kita-ku, Okayama, Japan

Original scientific paper
DOI: 10.2298/TSCI140831120K

The use of Diesel engines for vehicle has been increasing recently due to its higher thermal efficiency and lower CO_2 emission level. However, in the case of Diesel engine, NO_x increases in a high temperature combustion region and particulate matter is generated in a fuel rich region. Therefore, the technique of premixed charge compression ignition is often studied to get the peak combustion temperature down and to make a better air-fuel mixing. However it also has got a limited operating range and lower engine power produced by the wall wetting and the difficulty of the ignition timing control. In this research, the effect of injection strategies on the injected fuel behavior, combustion, and emission characteristics in a premixed charge compression ignition engine were investigated to find out the optimal conditions for fuel injection, and then ethanol blended diesel fuel was used to control the ignition timing. As a result, the combustion pressures and rate of heat release of the blended fuel became lower, however, indicated mean effective pressure showed fewer differences. Especially in the case of triple injection, smoke could be reduced a little and NO_x emission decreased a lot by using the ethanol blended fuel simultaneously without much decreasing of indicated mean effective pressure compared to the result of 100% diesel fuel.

Key words: *premixed charge compression ignition, diesel emission, ethanol, early injection, indicated mean effective pressure*

Introduction

As the air pollution and the petroleum resource exhaustion mainly caused by an increase in vehicle uses have become worldwide problems recently, the demands of harmful emission reduction and high thermal efficiency of vehicle engines have been increasing. In light of this, Diesel engines which have a low CO_2 emission level and high fuel efficiency have been attracting attention. However NO_x and diesel particulate matter (PM) generated from the process of diesel combustion are the big problems of Diesel engines. Therefore, to solve the problems, reducing the peak combustion temperature in order to reduce NO_x emissions and making better

* Corresponding author; e-mail: hylee@hanyang.ac.kr

air-fuel mixture to reduce PM generation are very important. The premixed charge compression ignition (PCCI) is a technique that can satisfy such conditions [1] and many researches on homogeneous charge compression ignition combustion including PCCI combustion are still in progress to achieve a higher engine efficiency and to meet emission standards [2, 3]. However, the fuel supply system of conventional Diesel engines is difficult to reduce harmful emissions as much as expected and it is not properly acceptable for PCCI engines because of the spray collision with the cylinder wall occurred by a low pressure condition of the combustion chamber at the early injection timing. Also the engine efficiency is lower than that of the conventional Diesel engine due to the control difficulties of ignition timing which is usually earlier than that of the conventional Diesel engine.

In some studies narrow injection angle injectors are investigated in conjunction with early multiple injection to reduce wall wetting and to create a more homogeneous mixture [4, 5]. Many kinds of methods [6] and fuels for the ignition timing control are still under study to improve the engine efficiency of PCCI engines besides [7, 8]. In this research, injected fuel behavior, combustion and emission characteristics in accordance with injection strategies are clarified using a common rail-type high-pressure injector with a narrow injection angle for a PCCI engine. Diesel fuel blended with ethanol was used to delay and control the ignition timing as a method to expand the operating range of the PCCI engine. From the result, we verified that fuel injection strategies largely affected the air-fuel mixing and combustion and also the use of ethanol affected the ignition timing, engine efficiency, and emissions.

Experimental apparatus and method

In this study, the effects of injection strategies and the use of diesel fuel blended with ethanol on engine combustion and emission were investigated using a common rail direct injection (CRDI) type PCCI engine. First, injection quantity and spray test using diesel fuel were carried out to obtain baseline data. Then engine experiments were executed to investigate the effects of varying injection strategies and to determine the optimal fuel injection conditions. Finally the effects of diesel fuel blended with ethanol on the engine efficiency and emissions were evaluated using the optimized injection conditions.

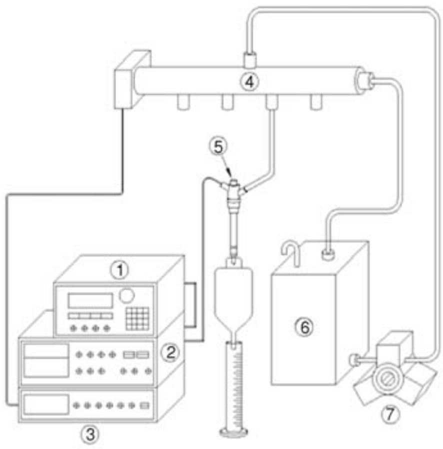


Figure 1. High pressure common rail type fuel injection system
1 – pulse generator, 2 – injector driver, 3 – presure contol valve (PCV) controller, 4 – common rail, 5 – CRDI injector, 6 – fuel tank, 7 – high pressure pump

Injection quantity measurement

To obtain fundamental data for injection, injection quantity tests were carried out for several injection conditions and ethanol mixing rates. The single injection quantity varying with injection duration was first measured from 200 timed injections then single and triple injection quantities were compared to determine appropriate injection durations for both conditions which made the total fuel amount same over one cycle. The injection pressure was fixed at 100 MPa throughout. Injection durations changed from 0.3 ms to 1.2 ms for single injection, and from 0.3 ms to 0.6 ms repeated three times for triple injection. In the triple injection case, the interval between injections was fixed at 10 degrees of crank angle at the engine speed of 1,400 rpm. The high pressure common rail type injection system shown in fig. 1 was used to obtain injection quantity data, the injector specifications and experimental conditions are summarized in tab. 1.

Table 1. Injector specifications and experimental conditions of the injection test

Injector specifications		Experimental conditions	
Injector type	Narrow angle CRDI	Injection duration	0.3 ms-1.2 ms
		Used fuel (Ethanol mixing rate)	Diesel 100%, (Ethanol 5%, 10%)
Hole diameter	0.168 mm	Injection strategy	Single, triple
Injection angle	100°	Injection pressure	50 MPa-130 MPa

Spray visualization and image processing

As shown in fig. 2, a spray visualization system was used to compare the injected fuel spray behavior and evaporation characteristics of single and triple injection. The chamber used in this research simulated the in-cylinder conditions of the experimental engine, including high temperature and high pressure. The ambient pressures were varied from 0.1 MPa to 2.0 MPa using N_2 gas to simulate the cylinder pressure at the timing of fuel injection, other test conditions are shown in tab. 2. In order to synchronize the image capture signal with the injection trigger, a pulse generator was used. The spray images were taken from the bottom of the chamber using a high speed camera with a xenon lamp for a light source. The spray penetration length which is an important parameter affecting spray impingement on the cylinder wall was measured from the spray images and injected fuel evaporation characteristics were compared using image processing.

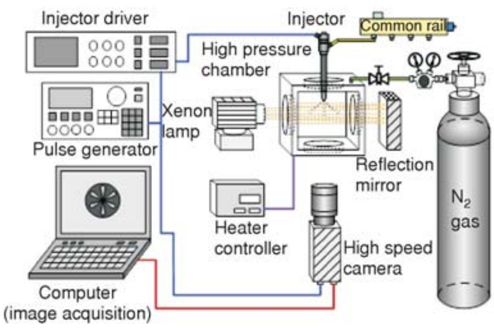


Figure 2. Injected fuel spray visualization system

Table 2. Experimental conditions for the injected fuel spray test

Injection quantity	18 mm^3
Fuel	Diesel
Injection strategy	Single, triple
Injection pressure	100 MPa
Ambient temperature	520 K
Ambient pressure	0.1 MPa-2.0 MPa

Image processing is a useful technique for comparing spray characteristics [9]. In this study, root-mean-square (RMS) image processing, which provides a framework for quantitative investigation of spray characteristics, was used over the entire injection period, MATLAB (Mathworks Co.) code was employed for the implementation. The RMS image processing assists in understanding the periodicity of the spray and produces high-contrast images because periodicity makes it more difficult to produce high contrast images using the mean averaging method. The arithmetic mean image (S_{mean}) was obtained using the arithmetic mean of pixels of each image from eq. (1), while the RMS average image (S_{RMS}) was obtained using eq. (2) in terms of the value of each frame pixel.

$$S_{\text{mean}}(x, y) = \frac{\sum_{k=1}^{n} S_{\text{k}}(x, y)}{n} \tag{1}$$

$$S_{\text{RMS}}(x, y) = \sqrt{\frac{\sum_{k=1}^{n} [S_k(x, y) S_k(x, y)]}{n}} \tag{2}$$

where, S_{mean}(x, y) is the mean averaged image, S_{RMS}(x, y) – the RMS averaged image, S_{k}(x, y) – the image (foreground) after correcting and de-noising, and n – the number of images.

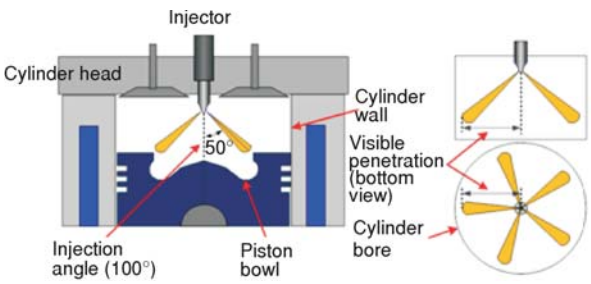


Figure 3. Definition of injection angle and spray penetration

Figure 3 shows the definition of injection angle and penetration for the CRDI injector. The injection angle is defined as twice the angle between the vertical line extended from the center of the injector tip and the center line of one of the spray cones. The penetration length, in this study, refers to visible penetration which primarily affects spray collision with the cylinder wall.

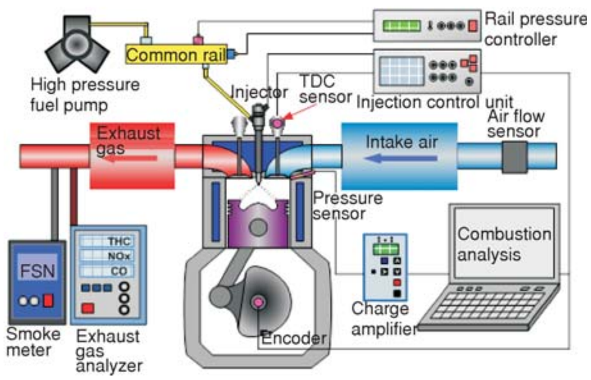


Figure 4. Schematic diagram of the common rail type PCCI engine system

Table 3. Engine specifications

Engine system	Single cylinder (DOHC 4-valve)
Bore × stroke	91 × 96 mm
Compression ratio	17.7
Injection type	Bosch CRDI
Combustion chamber	Re-entrant type
Intake condition	n. a.

Engine performance and exhaust emission test

Figure 4 contains a schematic diagram of a common rail type PCCI engine system; the experimental engine specifications are shown in tab. 3. As shown in fig. 4, a common rail injector with injection angle of 100° was fitted vertically in the center of the cylinder head. An encoder and a top dead center (TDC) sensor were installed to detect the crank position. A PCV controller was used to control injection pressure and injection timing and duration were controlled by an injection control unit. A cooling water temperature control system was used to control the engine temperature. Combustion pressure was measured by a pressure sensor installed at the combustion chamber. Finally, an exhaust gas analyzer (Horiba, Ltd.) and a smoke meter (AVL Co., Ltd.) were used to measure engine emission levels. The experimental conditions of the engine test are shown in tab. 4.

Table 4 Experimental conditions for the engine test

Engine speed	1400 rpm
Injection pressure	100 MPa
Injection timing	65° bTDC (Single) 75°, 65°, 55° bTDC (Triple)
Injection angle	100 °
Injection strategy	Early single and early triple
Fuel	Diesel and ethanol blended
Coolant temperature	90 °C

Results and discussion

Volumetric injection quantity for varying injection conditions

The effects of injection duration and pressure on volumetric injection quantity were investigated using a common rail injection system. The results of the injection quantity tests for diesel fuel are shown in figs. 5 and 6; the error was less than 2%. It is obvious that the single injection quantity is increased with a simultaneous increase in injection duration and pressure, as shown in fig. 5. In addition, the injection quantity for continuous triple injection is not equal to triple value of single injection, as can clearly be seen in fig. 6. Therefore the injection duration for triple injection was newly determined to make the same injection quantity. Ethanol blended fuel was then used to compare the volumetric injection quantities. From the results shown in fig. 7, it can be seen that ethanol mixing rate does not substantially affect the injection quantity according to injection duration.

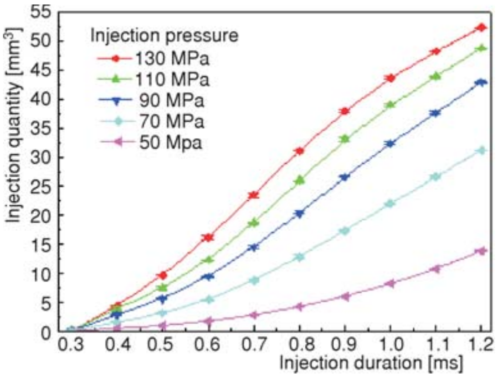


Figure 5. Injection quantities for varying injection pressures

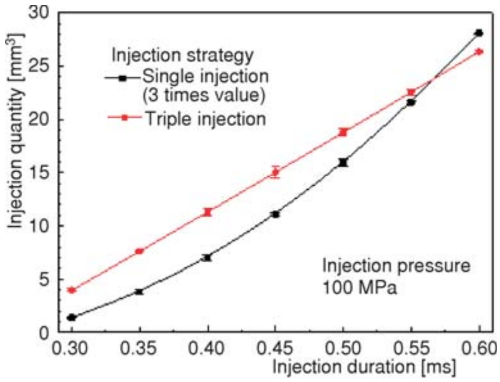


Figure 6. Injection quantities for varying injection strategies

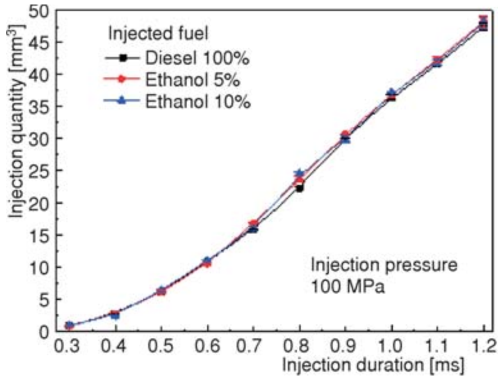
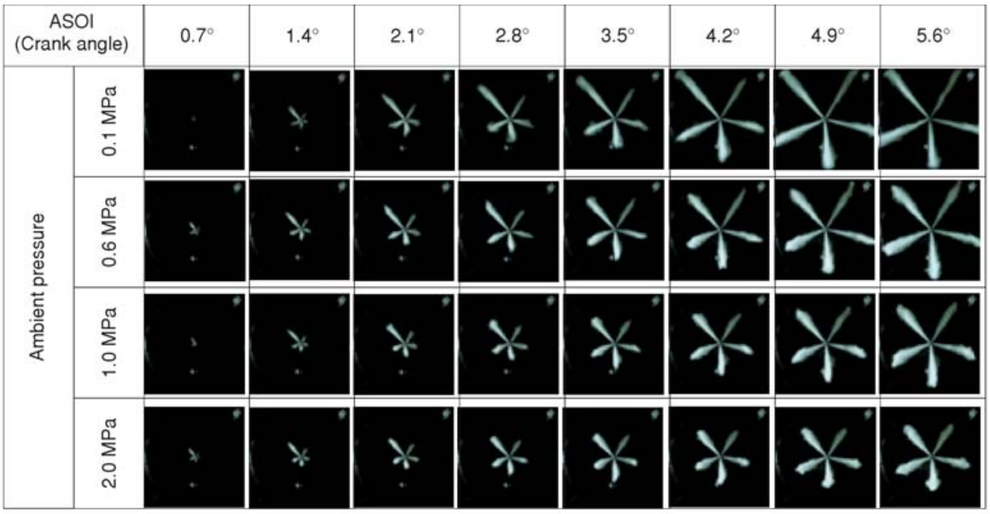


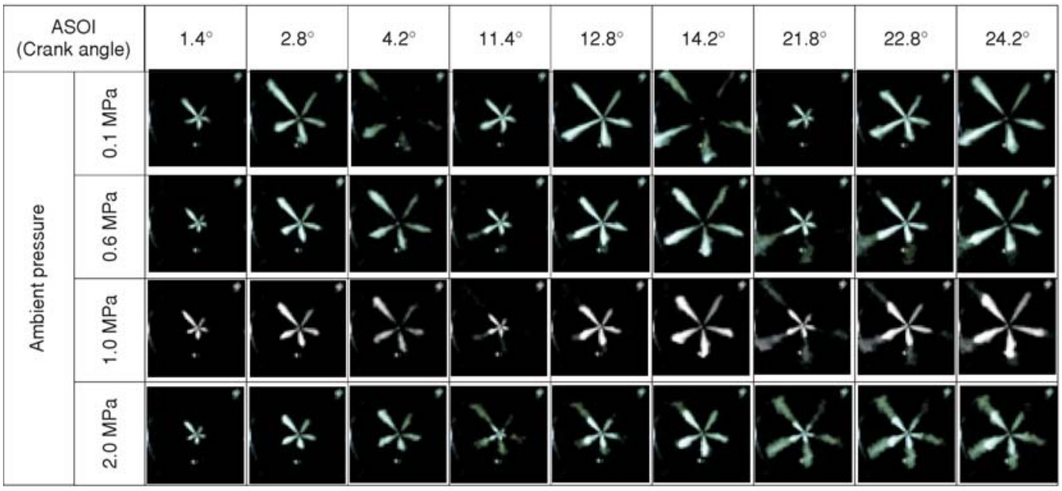
Figure 7. Injection quantities for varying ethanol mixing rates

Injected fuel spray characteristics

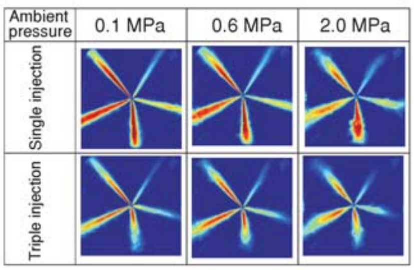
Figure 8 shows the spray images of test injector under single and triple injection conditions. The total injected fuel quantity and injection pressure were fixed at 18 mm^3 and 100 MPa, respectively, and the ambient temperature was designated as 520 K for the test. In particular the



(a) Spray images of single injection



(b) Spray images of triple injection



(c) The RMS images of single and triple injection spray

Figure 8. Injected fuel spray and RMS images

ambient pressure of 0.6 MPa and temperature of 520 K are very similar to the in-cylinder conditions at the timing of injection. The spray behavior and evaporating characteristics of single and triple injection strategies were compared. The spray images show that the spray penetration length can be reduced with triple injection. In addition, based on RMS image processing comparisons of injected fuel evaporation characteristics, triple injection appears to show improved spray atomization, leading to better air-fuel mixing.

Spray penetration length, which is an important parameter affecting spray impingement on the cylinder wall, was measured quantitatively from the spray images. The effects of injection strategy and ambient pressure on spray penetration are clearly seen from the result shown in fig. 9. Figure 9(c) shows the average spray penetration of triple injection for comparison with single injection. The data for single and triple injection show that, as the ambient pressure increases, the spray penetration length decreases. However, in the case of triple injection, the penetration length is shorter overall than that of single injection for the same injection quantity, and therefore less wall wetting could be expected.

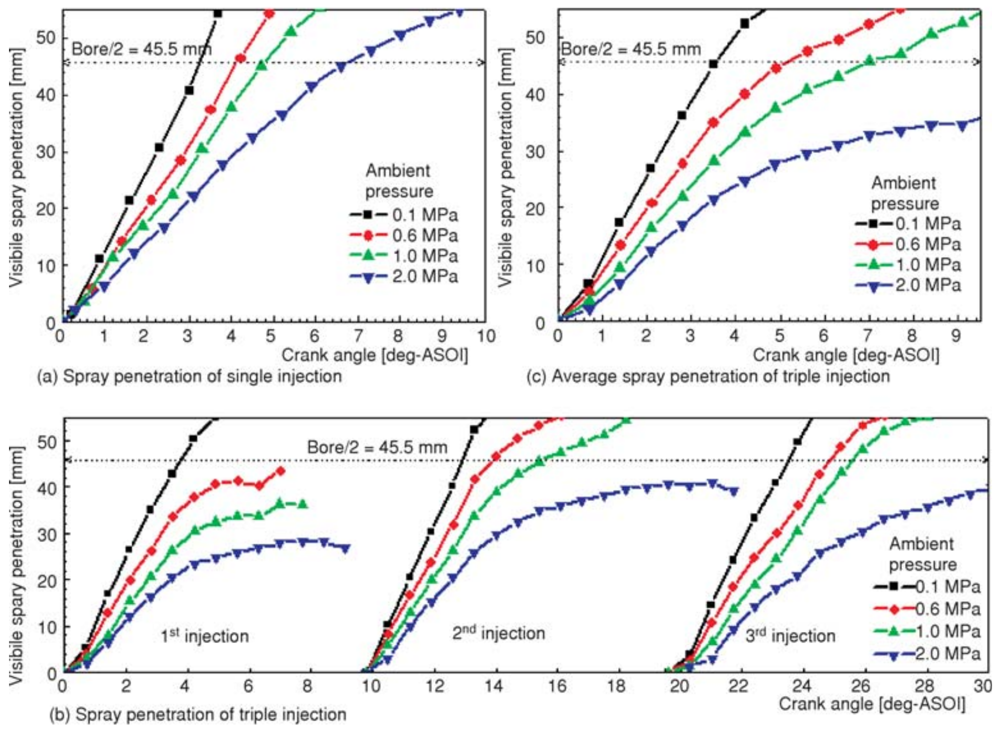


Figure 9. Visible spray penetration of injected fuel

IMEP and smoke characteristics of the main injection timings

To identity optimal conditions for main injection timing, smoke and indicated mean effective pressure (IMEP) characteristics were first investigated using diesel fuel. In general, when IMEP increases, the amount of smoke tends to increase, similar to the tendency shown in fig. 10. However the smoke level rapidly increased at injection timing of 50° bTDC compared to 60° bTDC. Thus 55° bTDC was chosen for the last injection timing, 75° bTDC for the first and 65° bTDC became the main injection timing for the optimal triple injection timings.

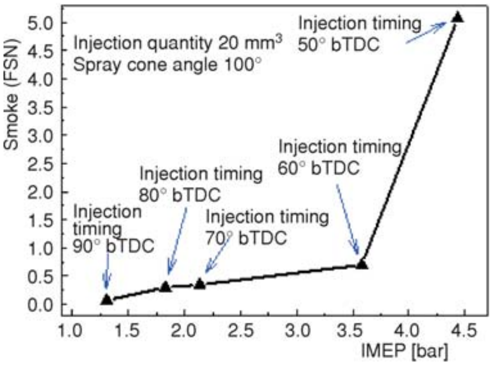


Figure 10. Smoke and IMEP characteristics for varying injection timings

Combustion and emission characteristics

In this section, the effects of ethanol mixing rates on the characteristics of ignition, combustion and emissions are evaluated, using the optimized injection timings of 65° bTDC for single injection and 75°, 65°, 55° bTDC for triple injection. The combustion pressure and rate of heat release (ROHR) characteristics of the blended fuel are indicated in fig. 11. The pressure and ROHR results were averaged and calculated from 200 engine operating cycles after the engine stabilized. In the case of singe injection, the fuel composition has only slight effect on the combustion pressure and ROHR patterns.

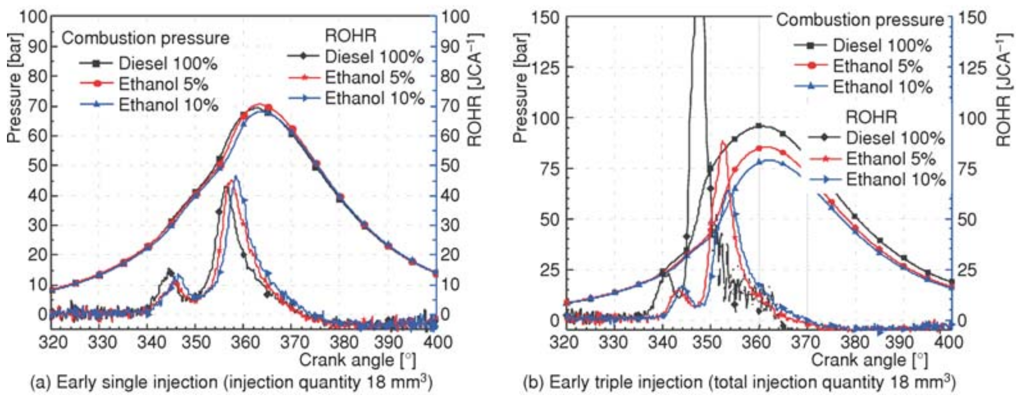


Figure 11. In-cylinder pressure and ROHR characteristics

However, differences are clearly seen with triple injection. In each case, combustion starts earlier than with single injection, and it is expected that the first injection starts earlier, and the triple injection yields better fuel evaporation and a larger region of stoichiometric air-fuel mixture than dose single injection. The ROHR pattern is delayed with increases in the ethanol mixing rate and moving closer to TDC. However, the combustion pressure and overall ROHR decrease, presumably owing to the lower heating value (approximately 64% of diesel fuel) and the lower cetane number of ethanol.

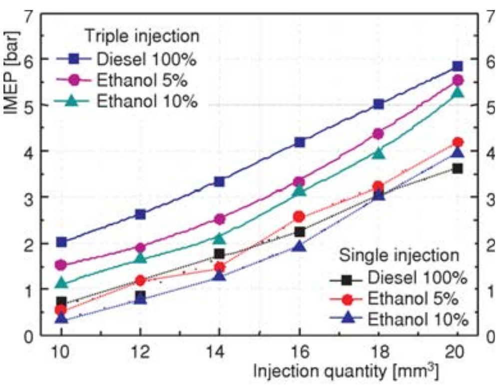


Figure 12. The IMEP characteristics for varying injected fuel quantity

The IMEP characteristics of the blended fuel according to the total injected fuel quantity are shown in fig. 12. In general, the single injection IMEP is lower than that of triple injection at the same injection quantity. Moreover, the singe injection IMEP characteristics are not as well defined as those of ROHR and combustion pressure. In the case of triple injection, even though the combustion pressure and ROHR of the blended fuel are lower as shown in fig. 11, the IMEP values show smaller differences. As the injection quantity increases, the gap between IMEP values narrows. This indicates that even if the ROHR values are lower, the main combustion shifts closer to TDC.

Figures 13 and 14 show representative exhaust emission characteristics found in PCCI combustion. It can be seen from fig. 13 that, in most cases except for triple injection with 100% diesel fuel, NO_x emission levels are extremely low. Add more, in triple injection cases, smoke level decreases with an increase in ethanol mixing rate and those of blended fuels are generally much lower than that of 100% diesel fuel. It is thought that this result is attributable to the lower combustion peak temperature caused by the lower heating value of ethanol, and carbon oxidation promoted by the O_2 contained within the ethanol molecule. As shown in fig. 14, when blended fuel is used with triple injection, smoke levels are slightly decreased relative to 100% diesel fuel with triple injection. However, NO_x emissions are decreased substantially without a significant lessening of IMEP.

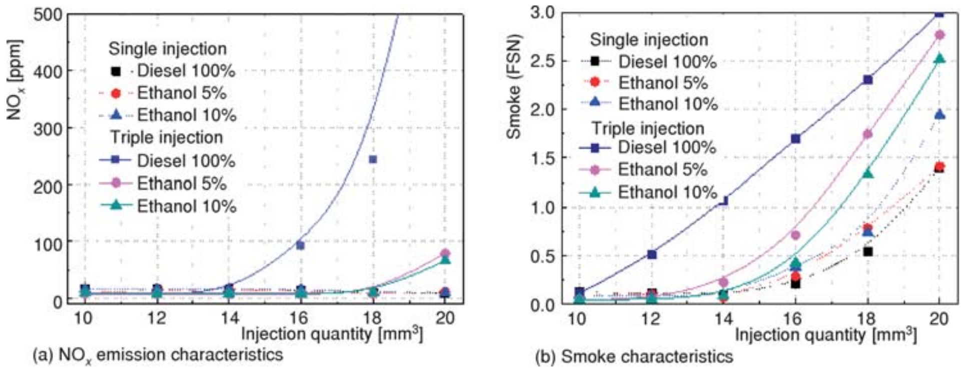


Figure 13. Exhaust emission characteristics for varying injected fuel quantities

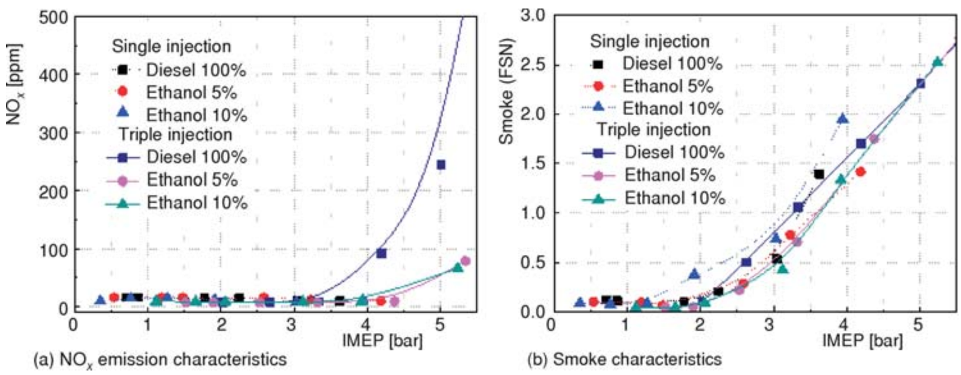


Figure 14. Exhaust emission characteristics according to IMEP values

Conclusions

In this work, the characteristics of injected fuel spray and quantity according to various injection conditions were investigated using a PCCI Diesel engine. Under identified optimal conditions, the effects of diesel blended ethanol fuel on combustion and exhaust emissions were examined, leading to the following conclusions.

- The ethanol blended fuel was used for comparison of the volumetric injection quantity. However, the ethanol mixing rate did not significantly affect the injection quantity according to injection duration.
- Base on RMS image processing results and quantitative spray characteristics, triple injection seems to demonstrate the benefits of spray atomization. Moreover, the penetration length was shorter overall than that of single injection for the same injection quantity, indicating that better air-fuel mixing and less wall wetting could be expected .
- An engine test to determine optimal injection timings showed that injection timing of 60° bTDC produced less smoke and higher IMEP, with smoke levels rapidly increasing for timing of 50° bTDC, thus the first, main and last injection timings were set at 75°, 65° and 55° bTDC, respectively.
- From the engine test for the ethanol blended fuel, it was determined that single injection did not lead to a substantial difference in combustion pressure and ROHR, but that triple injection induced larger differences. Moreover, combustion was delayed with increased ethanol mixing rate, shifting closer to TDC.
- Although combustion pressures and ROHR were lower for the blended fuel, IMEP showed a smaller difference. In particular, in the case of triple injection, smoke levels were slightly decreased relative to 100% diesel fuel when blended fuel is used. However, NO_x emissions were decreased substantially without a significant lessening of IMEP.

Acknowledgments

This work was supported by Mid-Career Researcher Program through National Research Foundation of Korea (NRF) grant funded by the Korea government (MSIP). (No. NRF-2014R1A2A2A01005055)

References

[1] Therkelsen, P., Dunn-Rankin, D., Small-Scale HCCI Engine Operation, *Combustion Science and Technology, 183* (2011), 9, pp. 928-946
[2] Brijesh, P., Sreedhara, S., Exhaust Emissions and its Control Methods in Compression Ignition Engines, *International Journal of Automotive Technology, 14* (2013), 2, pp. 195-206
[3] Jung, G., *et al.*, Major Sources of Hydrocarbon Emissions in a Premixed Charge Compression Ignition Engine, *International Journal of Automotive Technology, 13* (2012), 3, pp. 347-353
[4] Nandha, K., Abraham, J., Dependence of Fuel-Air Mixing Characteristics on Injection Timing in an Early-Injection Diesel Engine, SAE paper No. 2002-01-0944, 2002
[5] Walter, B., Gatellier, B., Development of the High Power NADITM Concept using Dual Mode Diesel Combustion to Achieve Zero NO_x and Particulate Emissions, SAE paper No. 2002-01-1744, 2002
[6] Magand, S., *et al.*, Use of Ethanol/Diesel Blend and Advanced Calibration Methods to Satisfy Euro 5 Emission Standards without DPF, *Oil & Gas Science and Technology – Rev. IFP Energies Nouvelles, 66* (2011), 5, pp. 855-875
[7] Giakoumis, E. G., *et al.*, Exhaust Emissions with Ethanol or N-Butanol Diesel Fuel Blends during Transient Operation: A Review, *Renewable and Sustainable Energy Reviews, 17* (2013), Jan., pp. 170-190
[8] Pidol, L., *et al.*, Ethanol as a Diesel Base Fuel – Potential in HCCI Mode, SAE paper No. 2008-01-2506, 2008
[9] Kim, Y., *et al.*, Experimental Study on Liquid Phase LPG Spray and Icing Phenomenon for a Middle Class Diesel Engine Application, *Atomization and Sprays, 21* (2011), 7, pp. 611-624

Paper submitted:August 31, 2014
Paper revised: June 25, 2015
Paper accepted: August 28, 2015