Diesel Combustion Strategy and Its Extension

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ABSTRACT

DI-Diesel engine has expanded its market owing to the benefit of fuel economy. The development of high pressure injection systems such as common-rail system enables reduction of emissions and improvement of fuel consumption. The high pressure injection could reduce particulate matters (PM) by the improved fuel atomization followed by better mixing with air. Electronically controlled injectors are incorporated in common-rail fuel injection equipments (FIE) which realize the high response injection control. The faster injection operation can enhance the flexibility of fuel spray pattern and combustion phase control. Application of multiple injection strategy has been tested in an optical single-cylinder engine. Combustion characteristics and exhaust emissions were studied as a function of injection scheme.

The operational excellence of diesel combustion could be potentially extended to achieve better performance through homogeneous charge compression ignition (HCCI) combustion. It has the advantages of reduced NOx emission resulting from the low combustion temperature under homogeneous charge lean burn conditions, and reduced PM by the premixed combustion. The application of early diesel injection for near-homogeneously premixed charge was tested in the same optical engine, where the combination of cool-flame and non-luminous premixed combustion. HCCI combustion has been investigated with the variety of fuel injection parameters.

I. MULTIPLE INJECTIONS IN A HSDI ENGINE

I.1 INTRODUCTION

DI-Diesel engine has advantages of high thermal efficiency and low fuel consumption as a modern light duty vehicle, as well as a power plant for heavy duty vehicle. The high flexibility of common rail injection system enables reducing emissions and improving fuel consumption. High injection pressure over 100MPa could reduce particulate matters (PM) by its high momentum resulting in optimized atomization, evaporation and mixing. Injection timing and fuel quantity could be controlled electronically.[1] The control of these injection parameters had no concern with engine speed or load. Nevertheless, the acceptance of diesel engine is still dependent on hazardous emissions. It is well known that simultaneous reduction of PM and nitric oxides (NOx) is difficult.[2] One of the possible solutions is the application of multiple injection strategy. Multiple injection strategy has been known as a useful tool to achieve noise and emission reduction, while maintaining low fuel consumption.[3] The pilot-injection provides environment with high temperature and pressure for the main injection resulting in smoother pressure rise and faster ignition of main injection.[4] The after-injection oxidizes unburned fuel that reduces hydrocarbons (HC) and carbon monoxide (CO) as well as PM.[5]

In this study, an optically-accessible single cylinder diesel engine equipped with a common-rail injection system was built to investigate the effect of multiple injection and concerned combustion characteristics. Optical access was possible with the elongated piston and bottom-view quartz. High speed digital video camera was used to visualize combustion process inside the cylinder. Experiments were carried out with a wide range of injection parameters including injection pressures and injection timings. Effect of pilot- and after-injection was analyzed separately and optimized result was applied to operate at the triple injection mode. Direct-imaging results for all operating parameters showed details of spray and combustion characteristics inside the cylinder. Emissions including PM, NOx, HC were also measured for various operating conditions.

I.2 EXPERIMENTAL SETUP

OPTICAL RESEARCH ENGINE

The engine used in the tests was a single-cylinder, direct injection, 4-stroke optical research diesel engine equipped with a common rail injection system. The specification of the research engine is listed in Table 1. A schematic of the engine structure is shown in Fig. 1. The piston has been modified with respect to the original design in order to permit an optical access to the combustion chamber. To this end the elongated piston design presents two distinct parts: crown and body. This structure permits experiments with different combustion chamber geometry by replacing piston crown without modifying the piston body. A window in the cylinder head provides a view of the piston bowl through a piston-crown window. The quartz inserted in the piston crown (used as bowl bottom) permits a full bowl view of the combustion chamber. To collect spray and combustion images, the elongated piston accommodates an elliptical mirror. The combustion system was refined to match with the newly installed common rail fuel injection system. For the experiments presented here, the injector was equipped with a 5-hole tip. The nominal hole diameter was 0.168mm. The injection angle was 150°.

Table 1 Engine specifications

Specifications	Resources
Bore x Stroke (mm)	83x92
Displacement (cc)	489
Compression Ratio	18.9
FIE	Common-rail (max. 150MPa)
Injector	sac-type, 5 holes, injection angle 150˚

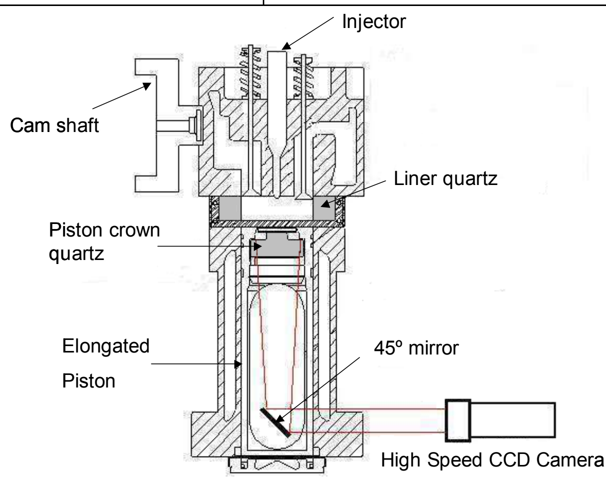


Fig. 1 Schematic of the single-cylinder optical diesel engine.

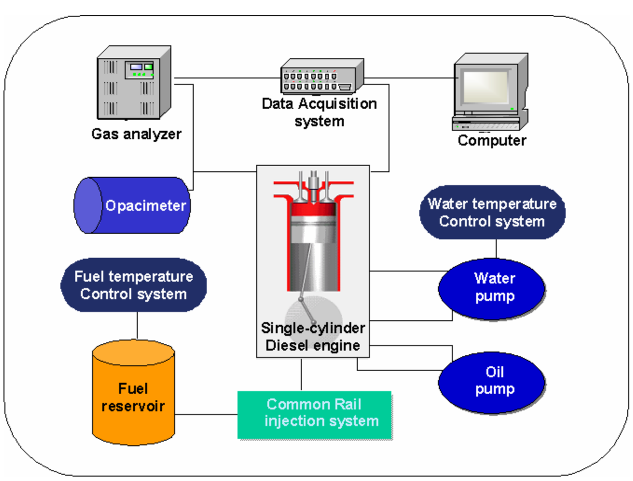


Fig. 2 Schematic diagram of the engine test bench.

ENGINE INSTRUMENTATION

A schematic drawing of the engine test bench is shown in Fig. 2. According to its functional behavior, it includes the following modules: diesel engine, fuel injection line, data acquisition and control units as well as the emission measurement system. The electrical motor allowed operation both in motoring and firing conditions. The pressure regulator valve and the solenoid injector were driven by an electronic injector operating system (TDA 3000H, TEMS Ltd.). A shaft encoder was used to transmit the crank shaft position to the injection system for the electronic control. The information was in digital pulses, the encoder had two outputs, the first (top dead center (TDC) index signal) had a resolution of 1 pulse/revolution, the second 1pulse/0.2degree. The encoder gives as output two TDC signal per engine cycle so as to have the right crank shaft position, one pulse was suppressed via hardware. A sampling probe was installed in the exhaust pipe: this was connected to the HORIBA MEXA1500D exhaust gas analyzer. And the in-line type opacimeter (OP100 EplusT Ltd.) was employed to measure the smoke emissions. If the opacimeter was used for the DI-diesel engine, the opacity of the exhaust gas represented the smoke intensity. The opacity value could be translated to bosch smoke number by SAE J1667. The principle of opacimeter was such as an absorption photometry; while a light emitting diode emits 563nm light and a crystal diode received it, the opacity of the exhaust gas changed the light intensity which was translated to a voltage signal. During the tests, the injection parameters were controlled by a personal computer (PC) that was directly connected to the fuel injection system, while a Kistler 6052A piezoelectric pressure transducer was used for the measurement of the in-cylinder pressure. The cylinder pressure data were digitized and recorded at 0.16 crank-angle-degree increments and ensemble-averaged over 130 engine cycles. The apparent heat release rate was calculated from the ensemble-averaged cylinder pressure data.

Table 2 Engine operating conditions

Engine Speed	800 rpm
Engine Load	Idling
Injection pressure	30, 60, 90, 120MPa
Main Injection Quantity	10mm^3/stroke
Pilot/After-injection Quantity	1.5mm^3/stroke
Main Injection Timing	BTDC18°CA ~ ATDC5°CA
Pilot/After-injection Timing	BSOI/ASOI 10°CA~60°CA

BTDC : before top dead center, ATDC : before top dead center, BSOI : before start of injection, ASOI : after start of injection

The in-cylinder images were acquired using a high speed CCD camera (Vision Research Inc.; Phantom v7.0). It afforded the high speed imaging up to 10,000 frames per second so that the images could be taken every 0.48°CA at 800RPM engine speed. The exposure times of the camera were optimized to obtain clear images. The camera images were digitized by a frame grabber in a PC to a resolution of 512 by 384 pixels. Synchronization between the engine and camera was controlled by another PC and a digital delay generator, with the master signal coming from the engine shaft encoder. The camera allowed the video frame acquisition to be synchronized with the engine. This synchronization system could be adjusted to obtain images at any desired crank angle within the 0.48 degrees resolution.

OPERATING CONDITIONS

Reference tests were run before starting the optical measurements. All the data presented in this article were taken at an engine speed of 800rpm. Before conducting the experiments, the engine was heated to 80°C by means of electrical heaters on the cooling water and the lubricating oil circulation systems. The operating conditions are summarized in Table 2. The amount of pilot and/or after fuel injected was fixed as 1.5mm^3/stroke to reduce the dwell time between consecutive injections with benefits on the engine performance and emissions. The total quantity of fuel injected was maintained at 11.5mm^3/stroke.

I.3 THE EFFECT OF PILOT-INJECTION

Figure 3 shows the effects of pilot-injection on combustion characteristic curve of the engine running at 30MPa injection pressure, at main injection timing of BTDC 6.6°CA and pilot-injection timing of BTDC 26°CA. It is well known that the pilot-injection is effective in shortening the ignition delay of the main injection. Therefore, by using the pilot-injection, the main injection timing can be retarded without affecting the ignition delay so that NOx emissions and combustion noise can be suppressed. For this purpose, a small amount of fuel, as little as possible, is to be injected just before the main injection to prevent the deterioration of diffusion combustion resulting in the deterioration of both smoke emissions and fuel consumption.

The result of IMEP and emissions with pilot-injection is shown in Fig. 4. With pilot-injection, the changes in CO and HC emissions with the main injection timing are similar to those of only main injection case. IMEP and smoke emissions were increased due to the extension of the diffusion-controlled combustion duration. This means that smoke generation is increased by high temperatures in the fuel-rich zone during diffusion combustion. The smoke emissions can be reduced by shortening the diffusion combustion phase, because of less time for soot formation and more time for soot oxidation.

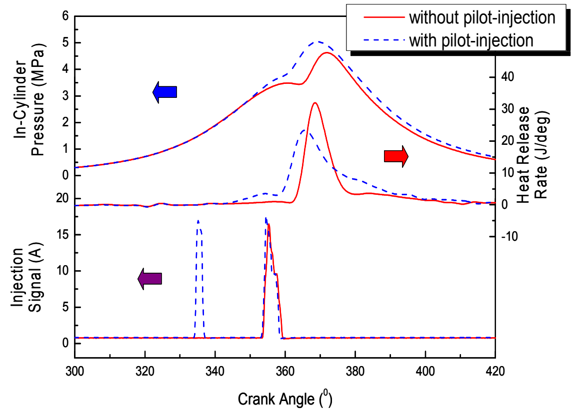


Fig. 3 Effect of pilot-injection on combustion characteristic curve; 30MPa, 6.6 degree BTDC main injection timing, 26 degree BTDC pilot-injection timing.

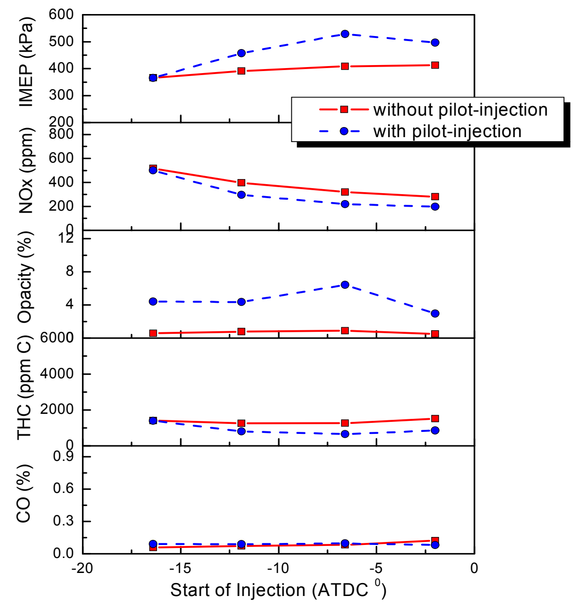


Fig. 4 Effect of pilot-injection on performance and emissions; 30MPa, 20 degree BSOI pilot-injection timing.

Figure 5 illustrates a temporal sequence of flame images obtained with pilot-injection. The pilot fuel injected at BTDC 25°CA starts to burn at BTDC 12°CA, showing a thin flame. At BTDC 2°CA after the start of the main injection, the liquid-phase fuel is surrounded with a number of flames into which the main fuel is injected. During the main injection, burned gas is entrained in the atomized fuel spray. With the pilot-injection used, the atomized fuel spray is lacking oxygen because the main injection entrains burned gas of pilot-injection. As a result, combustion progress gradually, causing a relatively low peak of heat release rate. In addition, slow combustion rate during diffusion combustion causes smoke emission to increase.

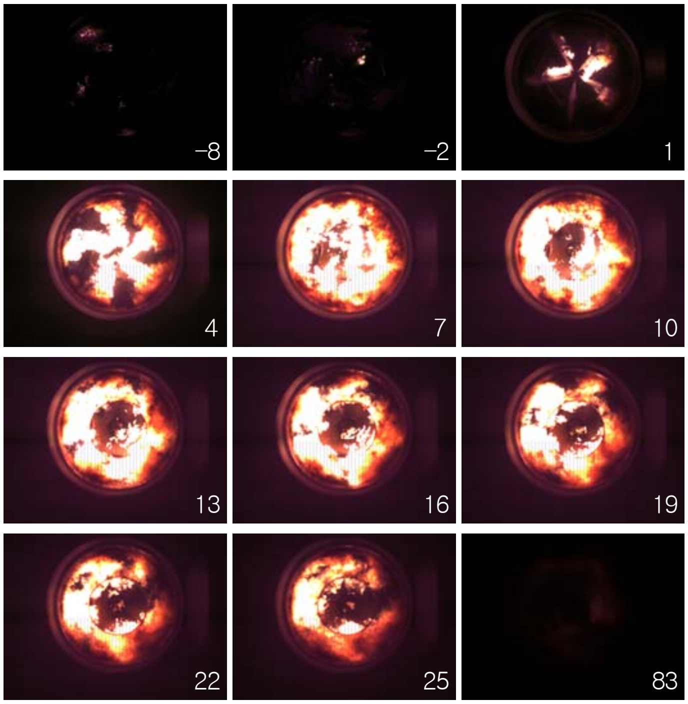


Fig. 5 High-speed images of combustion (With crank angle reference to TDC) Test condition : 30MPa injection pressure, 5 degree BTDC main injection timing, 25 degree BTDC pilot-injection timing

I.4 THE EFFECT OF AFTER-INJECTION

After-injection is typically used as a means to generate hydrocarbons for an active lean NOx catalyst[6]. However, in this study, a series of tests were carried out to determine the benefits of using after-injection to reduce PM. Oxidation process would be improved due to increased cylinder gas temperature and enhanced fuel-air mixing resulted from a second combustion event late in the cycle. The results shown in Fig. 6 indicate the effects of after-injection on combustion characteristic curve of the engine running at 30MPa injection pressure, at main injection timing of BTDC 10°CA and after-injection timing of ATDC 10°CA. Because the initial mass of main fuel is too small to maintain fuel consumption, the maxima of apparent heat release and cylinder pressure curves indicate the lower value than the single injection case. At and after TDC, the in-cylinder temperature will be decreased by the drop of the peak of heat release rate and work at this point is presumed to be reduced. Furthermore, approximately constant ignition delay time remains even when the main fuel decreases. Figure 7 shows the effects of after-injection at low pressure injection condition. With the start of main injection on the horizontal axis, IMEP and emissions were showed. The results demonstrate that injecting a small amount of fuel at after the start of main injection can reduce particulate emissions. The figure also indicates that improvement in particulate emissions does not come at the expense of decreased IMEP. In addition, the hydrocarbon emissions do not appear to increase significantly with after-injection at this injection timing.

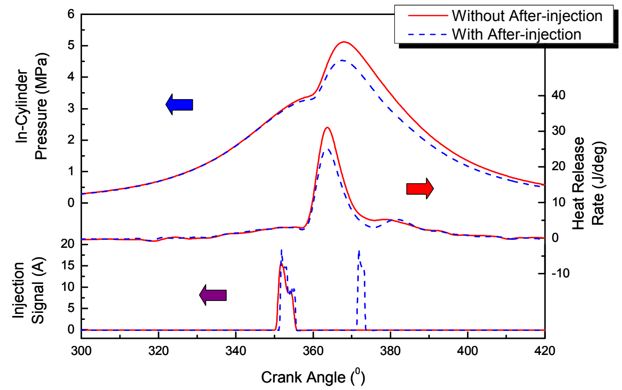


Fig. 6 Effect of after-injection on combustion characteristic curve, 30MPa injection pressure, 10 degree BTDC main injection timing, 10 degree ATDC after-injection timing

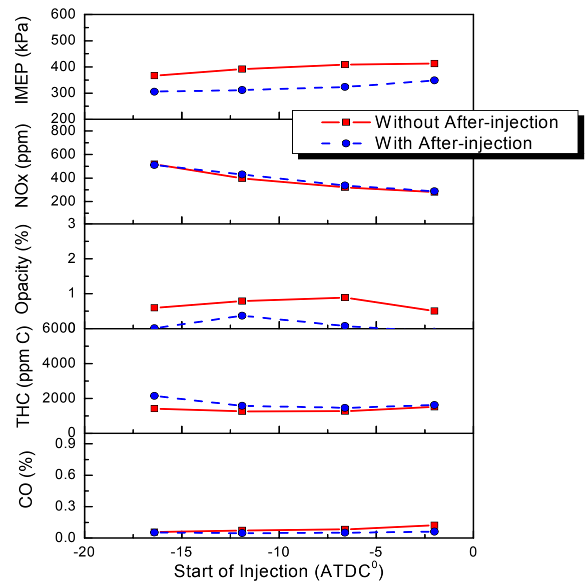


Fig. 7 Effect of after-injection on performance and emissions; 30MPa injection pressure, 10 degree ASOI after-injection timing.

Visualization of the after-injection are displayed in the Fig. 8. The after-injection introduces an interaction between the after-injection's spray plumes and the spot flames of the main combustion. The frames that the spray plume can be seen are relative to after-injection event.

I.5 THE EFFECT OF TRIPLE INJECTION

To study the combined effect of pilot and after-injections additional tests were conducted where both injections were done simultaneously with flame visualization as shown in Fig. 9. Also figure 10 and 11 showed cylinder pressure, apparent heat release rate, emissions and performance for the main and triple injection.

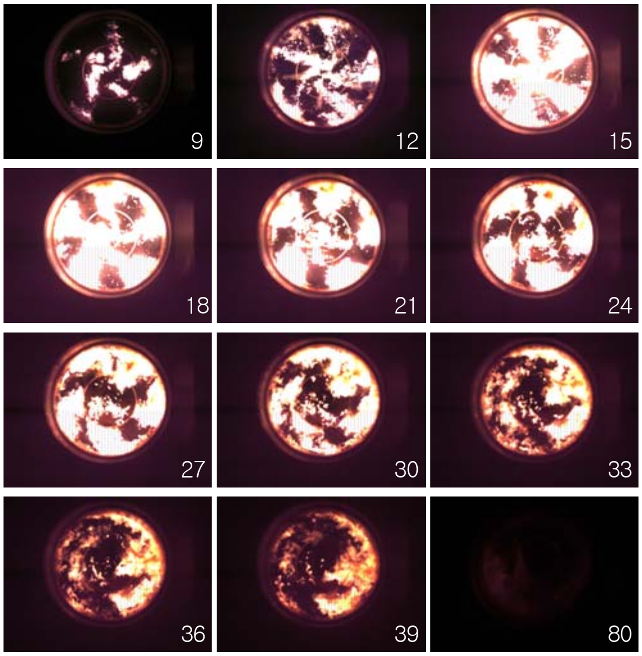


Fig. 8 High-speed images of combustion (With crank angle reference to TDC) Test condition : 30MPa injection pressure, 5 degree BTDC main injection timing, 5 degree ATDC after-injection timing

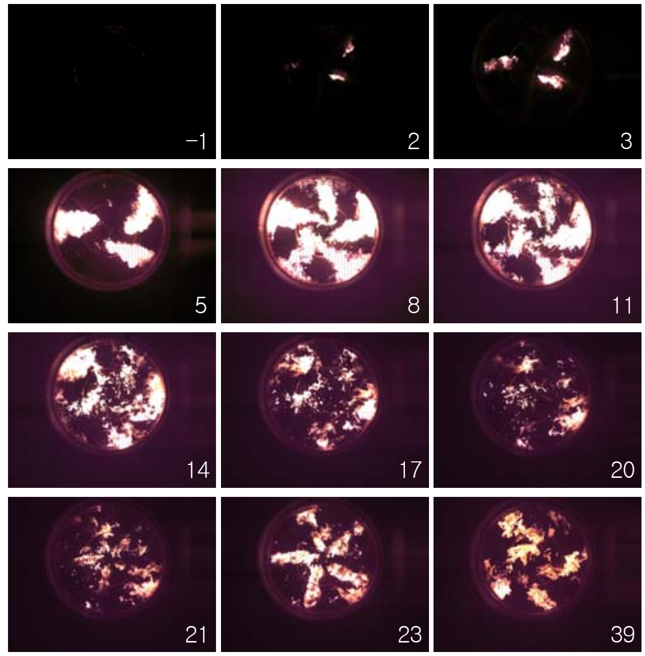


Fig. 9 High-speed images of combustion (With crank angle reference to TDC) Test condition : 30MPa injection pressure, 5 degree BTDC main injection timing, 35 degree BTDC pilot-injection timing, 15 degree ATDC after-injection timing

As shown in Fig. 9, when multiple injections are employed, the flame evolution relative to the main combustion is very similar to the main- and after-injection test case. The after-injection introduces an interaction between the after-injection's spray plumes and the spot flames of the main combustion. A very short ignition delay seems to characterize the after-injection. When the after injection is 20 degree crank angle retarded, soot flames mainly form the visible plumes of after-injection. During the first phase of after combustion, the flames are located near the wall of cylinder. This is an important factor that contributes to improve fuel-air mixing with multiple injections. Figure 10 shows that there was a difference in cylinder pressure and apparent heat release rate for single and triple pulsed injections. As expected, when the interval between main and pilot-/after-injection is more than 20 degrees, the triple injection is less effective at high pressure injection condition. However, with triple injection at low injection pressure, ignition delay decreases by half and the peak of heat release rate falls due to the effect of pilot-injection. And, an additional small peak of heat release rate occurred late in the cycle resulting form the last injection pulse for the triple injection case. Figure 11 shows effect of injection type on performance and emissions. At high pressure injection, the rate of NOx formation is nearly equivalent to the single injection case and smoke emissions are somewhat increased, under the idling conditions of the current experiments. One important conclusion drawn these results is that multiple injections are used in combination with low pressure injection, a significant reduction in NOx and smoke emissions can be achieved without decrease in IMEP. With low injection pressure, NOx reduction of 30% and smoke reduction of 40% were observed for triple injection. The penalty for IMEP in this case is between only 3 to 4%.

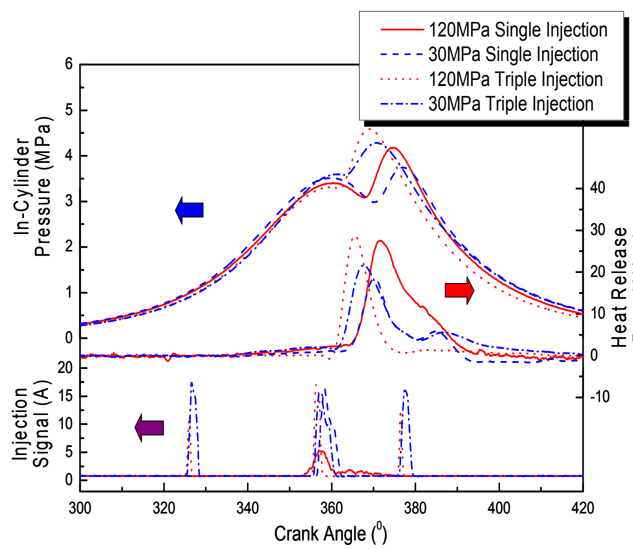


Fig. 10 Effect of triple injection on combustion characteristic curve; 5 degree BTDC main injection timing, 35 degree BTDC pilot-injection timing, 15 degree ATDC after-injection timing.

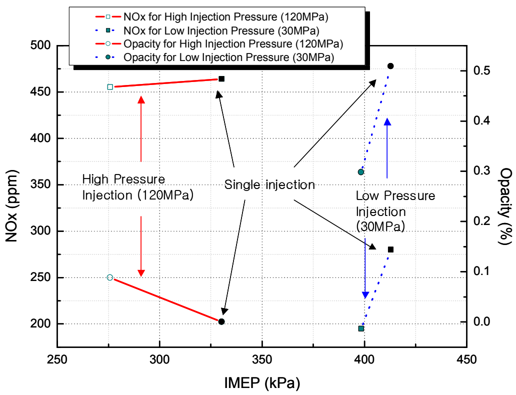


Fig. 11 Effect of triple injection on performance and emissions; 5 degree BTDC main injection timing, 35 degree BTDC pilot-injection timing, 15 degree ATDC after-injection timing.

II. DIESEL HCCI OPERATION

II.1 INTRODUCTION

Homogeneous charge compression ignition (HCCI) combustion is an advanced combustion process characterized by the near-homogeneous mixing between diesel fuel and air in a cylinder before compression ignition. HCCI combustion has the advantages of reducing NOx emission by spontaneous ignition at multiple points with lean premixed mixture resulting in low combustion temperatures. PM emissions can also be reduced by the premixed combustion without fuel-rich zones that characterize the heterogeneous combustion process in the conventional DI-diesel engine.

The main problems that hinder the realization of HCCI combustion are the difficulties in vaporization of a diesel fuel and the lack of a combustion phase control method. The fuel injection timing of HCCI combustion is much more advanced than DI-diesel combustion. When the diesel fuel is injected, the cylinder pressure and temperature is close to the atmospheric conditions. The viscous diesel fuel is not vaporized under these conditions. Therefore, the preheating of the intake air is required for the vaporization of diesel fuel. There is no direct control method of the HCCI combustion with the fuel injection timing in a DI-diesel engine. The indirect techniques such as exhaust gas recirculation (EGR), compression ratio variation are known as the possible solutions. Diesel may not be an appropriate fuel for HCCI combustion because of these problems but it still has high fuel efficiency and commercial benefits. Therefore, a lot of researches have focused on a diesel-fueled HCCI combustion.

Concerning the fuel injection method to form a premixed charge, the early direct-injection of diesel fuel has been studied actively[7-10]. A port injection is also tried though direct-injection still has the advantages such as the controllability of wide-range fuel injection timing, injection quantity and injection pressure with a common-rail fuel system. A common-rail fuel system is well known as a very versatile component for reducing gaseous and noise emissions as well as fuel consumption of DI-diesel engines[11]. It is due to the high pressure direct injection of the diesel fuel that is independent from the engine speed or load conditions.

In this study, the two-stage injection strategy based on a direct injection technique was executed as shown in Fig. 12. An extremely early direct-injection, named as the main injection, using common-rail fuel system was applied to form a premixed charge. This injection was followed by the other direct-injection, named as the second injection, executed near TDC. This kind of injection strategy has already been studied as a combination of HCCI combustion and DI-diesel combustion[7,10]. The main difference between this study and previous works is the smaller fuel quantity in the second injection (1.5 ㎣). Previous works used almost a half of total diesel fuel to form a premixed charge. The rest half of diesel fuel was injected in the similar injection timing with DI-diesel's resulting in a partially premixed and heterogeneous combustion. These injection strategies report limited reduction of PM and NOx emissions. The small fuel quantity of second injection was chosen from the result of pilot injection study in a DI-Diesel engine[7]. The results of a small quantity of diesel fuel injection show non-luminous combustion near TDC. This combustion is characterized by virtually zero soot. Referred to these combustion characteristics, the second injection was expected to act as an ignition promoter of the HCCI combustion and a combustion controller by the variation of injection timing.

A number of combustion images were taken to show characteristics of HCCI combustion. Effects of injection pressure, intake air temperature, injection timings were studied with the two-stage injection strategy. The indicated mean effective pressure (IMEP), heat release rate and emissions were analyzed as a function of input parameters.

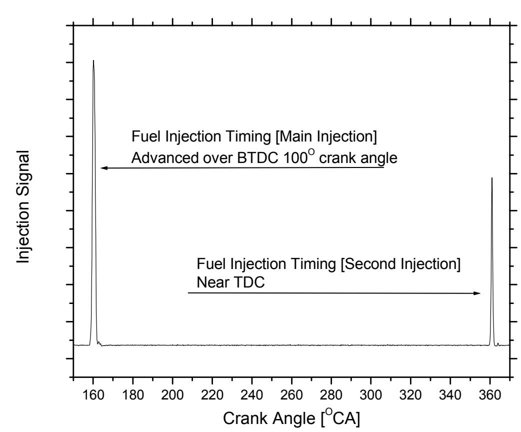


Fig. 12 Two-stage injection strategy; injection signals detected by a high current amperemeter

Table 3 Engine operating conditions

Engine speed 800 rpm/no load	
Injection pressure	30, 120MPa
Injection timing of the single injection	BTDC 250, 200, 150, 100, 50˚crank angle (CA)
Main injection timing of the two-stage injection	BTDC 250, 200, 150, 100, 50˚CA
Second injection timing of the two-stage injection	BTDC 20, 15, 10˚CA and TDC
Total quantity of fuel injection	11.5 ㎣
Quantity of second injection	1.5 ㎣
Intake air temperature	30, 80, 120, 160˚C
Injection angle	150˚, 100˚

II.2 ENGINE OPERATING CONDITIONS

The engine was operated at 800 rpm under both motored and fired conditions. The coolant temperature was set to 80˚C and diesel fuel temperature was kept at 40˚C. This represents an idling condition of a diesel engine. Various engine operating conditions are described in Table 3. The base engine was equipped with a sac-type 5 hole nozzle injector with injection angle 150˚. A smaller injection angle 100˚ was applied to overcome the over-penetration problem in HCCI combustion mode. The injection timings were applied more widely than previous works[7,8,10]. The fuel quantity of the second injection was fixed at 1.5 ㎣. The total quantity of fuel was 11.5 ㎣ that was the actual fuel quantity at the idling condition of the corresponding diesel engine. All the results were compared to the DI-diesel combustion at the idling condition with the conditions as follows; the injection timing was BTDC 15˚CA. The injection pressure was 120MPa and total fuel quantity was 11.5 ㎣. The intake air was not preheated and the compression ratio was 18.9. The sac-type injector has 5 holes with the injection angle 150˚.

II.3 EFFECT OF INJECTION PRESSURE

Figure 13 shows the effect of the injection pressure on the cylinder pressure and the heat release rate for the single injection case. The fuel was supplied by the single injection at the early timing. It shows that the main heat release rate of the 120MPa injection pressure case was much higher than the 30MPa injection pressure case even for the heat release rate of the cool combustion. By increasing the injection pressure, the spray impulse was increased providing smaller droplets. It enhanced vaporization of diesel fuel. Therefore, the mixing between diesel fuel and air was promoted. This well-premixed charge was auto-ignited resulting in high heat release rate. Although the hole-type injector used in this study had the over-penetration problem with cylinder wall wetting which could be more serious when the injection pressure is high, the advantage of a high pressure injection such as better atomization was more dominant.

II.4 EFFECT OF INTAKE AIR TEMPERATURE

Figure 14 shows the effects of intake air temperature on cylinder pressure and heat release rate. The heat release was increased as intake air temperature is higher. But early ignition problem was raised. The start of main heat release was advanced more than 20˚CA with the increase in the intake air temperature from 30˚C to 160˚C. This was because the cool combustion, which led early main combustion, was advanced as the charge air temperature was increased. It results in low thermal efficiency and low power output compared to the DI-diesel combustion.

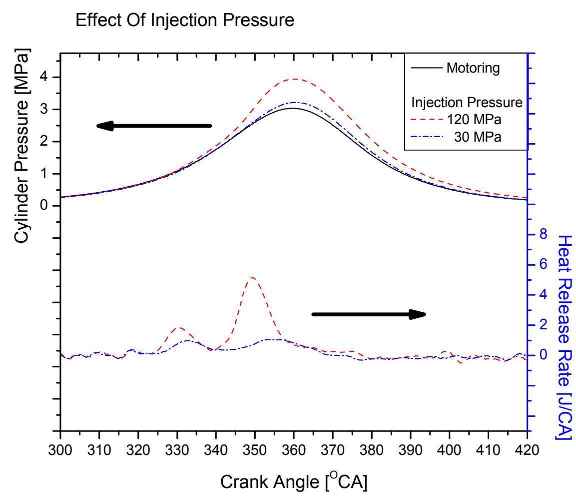


Fig. 13 Effect of injection pressure on cylinder pressure and heat release rate; single injection, injection timing=BTDC 200˚CA, injection quantity=11.5 ㎣, intake air temperature=160˚C, compression ratio=18.9, injection angle=150˚

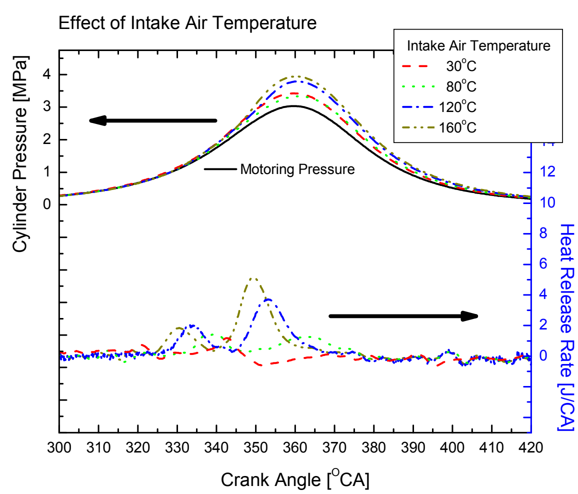


Fig. 14 Effect of intake air temperature on cylinder pressure and heat release rate; single injection, injection timing=BTDC 200˚CA, injection quantity=11.5 ㎣, injection pressure=120MPa, compression ratio=18.9, injection angle=150˚

II.5 TWO-STAGE INJECTION STRATEGY

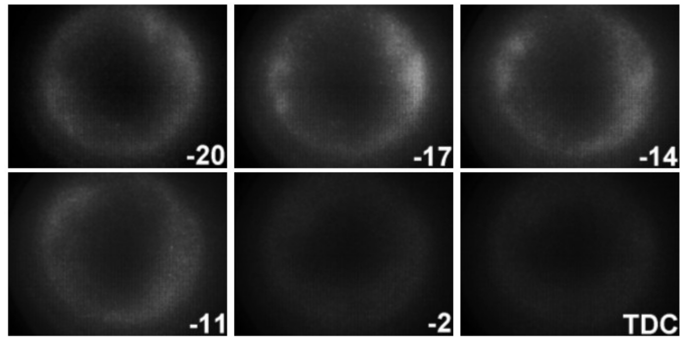
EFFECT OF MAIN INJECTION TIMING

Sufficient time is needed for a homogeneous premixing between the diesel fuel and air before ignition which could be achieved by advancing the main injection timing. However if the injection starts too early (over BTDC 100˚CA), the initial air temperature is lower so that the vaporization rate of diesel fuel could decrease. The wall wetting is increased by long spray penetration and low cylinder liner temperature. Some portion of the fuel is exhausted as the unburned hydrocarbon. It results in low combustion efficiency and low IMEP showing higher opacity in the exhaust stream. If injection starts later (near BTDC 50˚CA) the temperature is initially higher so that the vaporization rate of diesel fuel could increase. The wall wetting is decreased by the shorter spray penetration. But the time for the premixing between diesel fuel and air will be insufficient. The premixed charge is inhomogeneous resulting in high exhaust emissions.

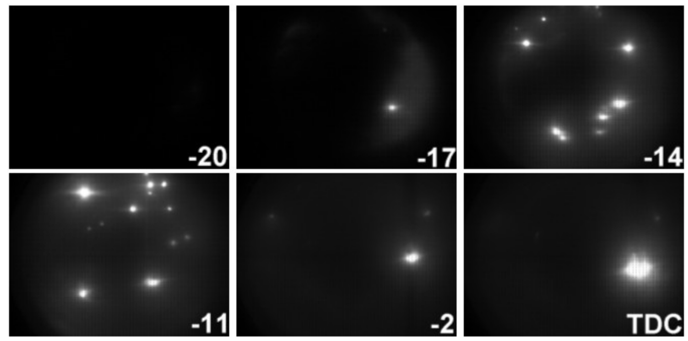
The flame images for different main injection timings are shown in Fig. 15. The optical investigation also proved that the main injection timing should be advanced to earlier than BTDC 100˚CA. Injection timing of BTDC 100˚CA (Fig. 15(a)) showed near-uniform flame intensity. This is the case for the more advanced injection timings than the previous works[7,8,10]. Previous researchers, Yoshinori Iwabuchi et al.[8] and Ryo Hasegawa et al.[10], used shadowgraph techniques and showed that luminous flame was not detectable. Injection timing of BTDC 50˚CA (Fig. 15(b)) showed relatively non-uniform and luminous flames if the intensified direct-imaging technique was applied. Figure 16 shows the flame intensity by the grey level. Each data was detected from the cross line of the bowl. Injection timings of BTDC 250˚CA, 200˚CA and 150˚CA showed more uniform distributions of the flame intensity than BTDC 100˚CA and 50˚CA cases through this sampling line. Therefore, extremely advanced timings at the intake stroke were preferred in this study for the successful HCCI combustion.

EFFECT OF SECOND INJECTION TIMING

As the intake air was preheated to assist vaporization of the diesel fuel, the auto ignition of injected fuel became earlier. In this case, the second injection timing should be advanced to the optimal timing that was just before the main heat release to promote the ignition. Figure 17 shows the heat release rate with second injection timing variations. When the second injection timing was at TDC, there were two peaks of the main heat release. The first peak was from the cool combustion and the second peak from the auto-ignition of a premixed charge and the third peak was from the ignition of remaining charge by the second injection. As the second injection timing was advanced until the BTDC 20˚CA, which was the timing of auto-ignition at 160˚C intake air temperature, the highest value of the main heat release was detected. The combustion duration became longer than the case of the single injection.



(a) Main injection BTDC 100˚CA



(b) Main injection BTDC 50˚CA

Fig. 15 Intensified imaging of HCCI combustion under the main injection timing variation; single injection, injection quantity=11.5 ㎣, injection pressure=120MPa, intake air temperature=160˚C, compression ratio=18.9, injection angle=150˚

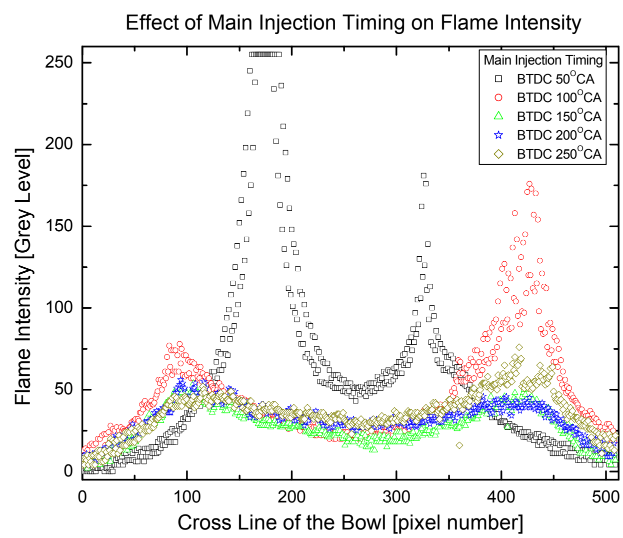


Fig. 16 Flame intensity variation along a center line of the HCCI combustion as a function of main injection timing, detected from the Fig. 15

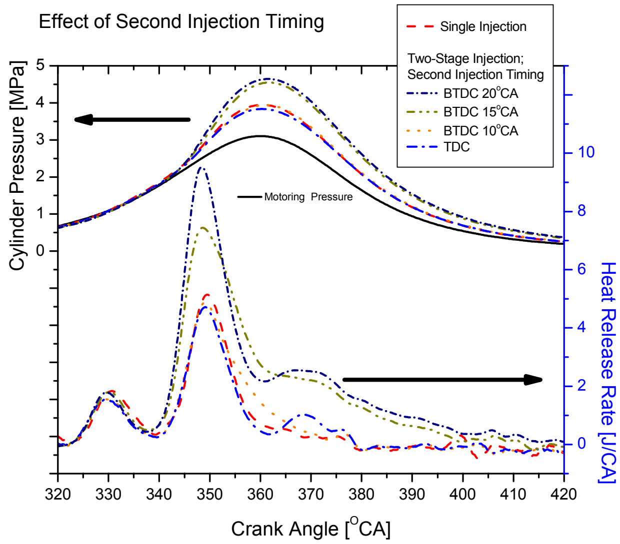
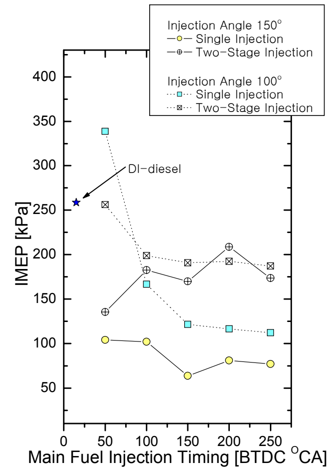


Fig. 17 Effect of second injection timing on cylinder pressure and heat release rate; single injection and two-stage injection, main injection timing=BTDC 200˚CA, total fuel quantity=11.5 ㎣, main injection quantity=10 ㎣, second injection quantity=1.5 ㎣, injection pressure=120MPa, compression ratio=18.9, injection angle=150˚, intake air temperature=160˚C

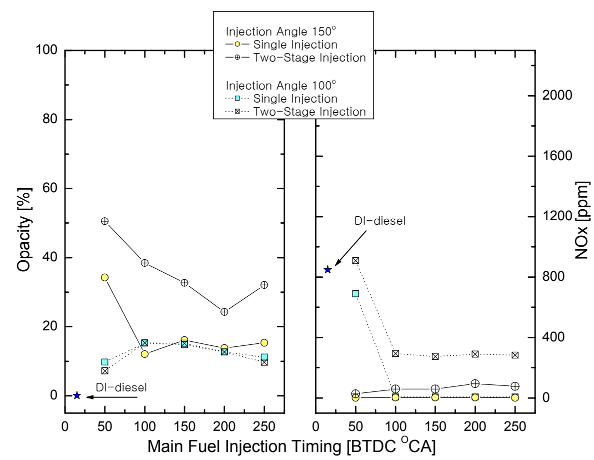
II.6 EFFECT OF INJECTOR GEOMETRY

The injector with small injection angle was tested to reduce wall wetting by the over-penetration. The number of holes and hole diameter was maintained to 5 holes and 0.168mm. The spray penetration of injection angle 150˚ injector contacted with cylinder liner because the spray penetration became longer because the cylinder pressure and temperature is low when the fuel is early injected. It could be a severe source of opacity and HC emissions. So the small injection angle 100˚ was applied to reduce the radial spray penetration so that the spray didn't contact with cylinder liner.

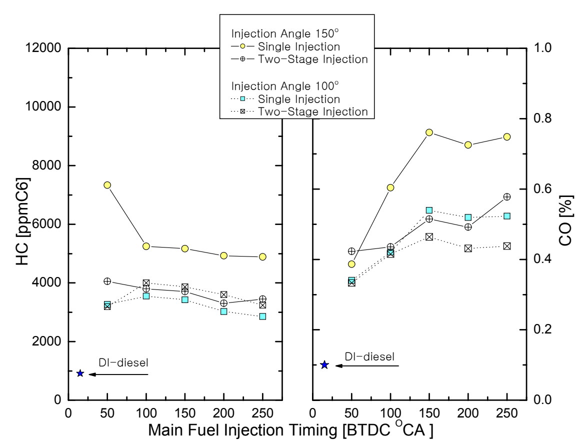
Figure 18 shows the effect of injection angle for both single injection and two-stage injection method. IMEP and exhaust gas concentration were presented with injection timing variation. The opacity was improved to minimum 7% because the spray impingement was reduced. HC and CO concentration was also reduced compared to 150˚ injection angle case while the power output was increased. Two-stage injection method still shows its advantages for power output and opacity reduction. Concerning the main injection timing, the high opacity of a 150˚ injection angle injector in BTDC 50˚CA injection timing was reduced but NOx was highly increased. The high NOx concentration was a result of incomplete and non-uniform premixed charge because a time for premixing between the fuel and air was insufficient. So the main injection timing for the injector with small injection angle should also be advanced earlier than BTDC 100˚CA.



(a) IMEP



(b) opacity and NOx



(c) HC and CO

Fig. 18 Effect of injection angle on IMEP and exhaust emissions with main injection timing variation; single injection and two-stage injection (second injection timing=BTDC 20˚CA), total fuel quantity=11.5 ㎣, main injection quantity=10 ㎣, second injection quantity=1.5 ㎣, injection pressure=120MPa, intake air temperature=160˚C, compression ratio=18.9

III. MAJOR FINDINGS

Experiments were carried out to analyze the effect of different multiple injection strategies on the reduction of the emissions and the IMEP improvement of a small DI diesel engine. A high speed CCD camera allowed for visualization of sprays and their combustion evolution. Visualization was performed at fixed engine speed and idling condition with various injection strategies. From these investigations, major findings could be summarized as follows;

- Pilot-injection provides a significant reduction in the peak of heat release rate. This is due to the fact that pilot-injection is effective in shortening the ignition delay, which reduces the portion of pre-mixed combustion, and in slowing down the combustion rate entraining the pilot-injection burned gas by the atomized main spray.
- The in-cylinder temperature is increased by the adoption of pilot-injection and work is presumed to contribute to the improvement of IMEP.
- The heat release rate curve rises to higher value in the latter half of diffusion combustion period than the values seen on the curve obtained without pilot-injection. This slow-down in combustion process causes some deterioration in smoke.
- The after-injection was found to be very effective in completing the oxidation processes and reducing the particulate emissions.
- The results indicate when multiple injections are used in combination with low pressure injection, a significant improvement in the reduction of NOx and smoke emissions can be achieved.

HCCI combustion with two-stage injection strategy is also investigated to achieve the high combustion efficiency while the exhaust advantages are maintained. Intensified direct-imaging technique is applied to visualize non-luminous combustion. The exhaust emission (opacity, NOx, HC, CO) concentrations are measured under various operating conditions with various parameters; injection pressures, injection timing, intake air temperature, the compression ratio, fuel quantity and injection angle. From these investigations, major findings could be summarized as follows;

- The high pressure injection has an advantage of achieving a premixed state between the diesel fuel and air.
- The main injection timing should be advanced earlier than BTDC 100˚CA to achieve premixed charge resulting in exhaust emission reduction.
- The second injection timing has optimized value to promote ignition with the intake air temperature variations.
- Although HC and CO emission is increased compared to DI-Diesel's, NOx is reduced by more than 90%.
- The opacity as a result of the over-penetration problem could be reduced when the injection angle 150˚ was modified to 100˚.

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