Combustion Control Using Two-Stage Diesel Fuel Injection in a Single-Cylinder PCCI Engine

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ABSTRACT

A diesel-fueled premixed charged compression ignition (PCCI) combustion technique using a two-stage injection strategy has been investigated in a single cylinder optical engine equipped with a common-rail fuel system. Although PCCI combustion has the advantages of reducing NOx and PM emissions, difficulties in vaporization of a diesel fuel and control of the combustion phase hinder the development of the PCCI engine. A two-stage injection strategy was applied to relieve these problems. The first injection, named as main injection, was an early direct injection of diesel fuel into the cylinder to achieve premixing with air. The second injection was a diesel injection of a small quantity (1.5 \text{ms}) as an ignition promoter and combustion phase controller near TDC. Effects of injection pressure, injected fuel quantity and compression ratio were studied with variation of an intake air temperature. The experimental results showed that the two-stage injection could be used as a combustion phase controller only in the case of a low intake air temperature but maintained the effect of ignition promotion while the ignition timing was advanced by the high intake air temperature. The main injection timing should be advanced earlier than BTDC 100°CA for the homogeneous and non-luminous combustion. Results of the PCCI combustion showed reduced NOx by more than 90% but increased fuel consumption and HC, CO emissions compared to direct injection (DI) diesel engine. The base injection angle 150° was modified to 100° for the reduction of smoke and unburned fuel made from the over-penetration of fuel spray, which implies the way of optimized injector selection to achieve the successful early injection strategy.

INTRODUCTION

Premixed charged compression ignition (PCCI) combustion is an advanced combustion process characterized by the near-homogeneous mixing between diesel fuel and air in a cylinder before compression ignition. PCCI 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 PCCI combustion are the difficulties in vaporization of a diesel fuel and the lack of a combustion phase control method. The fuel injection timing of PCCI 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 PCCI 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 PCCI 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 PCCI combustion.

Concerning the fuel injection method to form a premixed charge, the early direct-injection of diesel fuel has been studied actively\(^{15}\). A port injection is also tried\(^{6}\) 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\(^{33}\). 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. 1. 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 PCCI combustion and DI-diesel combustion\(^1,2,5\). The main difference between this study and previous works is the smaller fuel quantity in the second injection (1.5 \(\text{mL}\)). 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 PCCI combustion and a combustion controller by the variation of injection timing.

![Fig. 1 Two-stage injection strategy; injection signals detected by a high current amperemeter](image1)

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

**EXPERIMENTAL APPARATUS**

**RESEARCH ENGINE**

A Single-cylinder optical diesel engine (Engine Tech; RSi-090) was used to operate in PCCI combustion mode. The engine specifications are shown in Table 1. Fuel injection parameters including injection pressure, injected quantity and injection timings were controlled in a common-rail fuel system operated by a programmable injector driver (TEMS; TDA3000H) with a pressure controller. Figure 2 shows the schematic diagram of the experimental setup. The intake air was preheated with an electrical heater to test the effect of intake air temperature. Cylinder pressure was recorded with a piezoelectric pressure transducer (KISTLER; 6052A). A rotary encoder (Koyo; 3600 pulses/revolution) attached to a camshaft was used to control the operating timings of multiple fuel injections and a camera. Its resolution was 0.2 crank angle (CA).

**VISUALIZATION**

Figure 3 depicts the optical access to the combustion chamber. An elongated piston was applied in order to enable the mounting of a 45° mirror beneath the piston quartz window. An observation field on the piston window is shown in Fig. 4. A high speed digital video camera (Vision Research Inc.; Phantom v7.0) with an image intensifier (PROXITRONIC) was used to take images of PCCI combustion. It affords 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. Detailed camera settings are shown in Table 2. The exposure times of the camera were optimized to obtain clear images. The images were highly amplified through the intensifier to take non-luminous PCCI combustion fields. The maximum gain of the intensifier was 10\(^8\)W/W and the maximum light power per area was 70\(\mu\)W/cm\(^2\). Almost 90% of the gain value was used for detecting weak PCCI combustion except highly luminous case where the gain value was decreased to 30%.

![Fig. 2 Experimental setup](image2)

<table>
<thead>
<tr>
<th>Table 1 Engine specifications</th>
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<tbody>
<tr>
<td>Engine</td>
</tr>
<tr>
<td>Bore x Stroke</td>
</tr>
<tr>
<td>Displaced volume</td>
</tr>
<tr>
<td>Compression ratio</td>
</tr>
<tr>
<td>Injector</td>
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</table>

Fig. 1 Two-stage injection strategy; injection signals detected by a high current amperemeter
EXHAUST GAS MEASUREMENT

An opacimeter (EplusT; OP-120) satisfying SAE J1667 was used to measure the exhaust gas opacity. If the opacimeter is used for the DI-diesel engine, the opacity of the exhaust gas represents the smoke intensity. The opacity value can be translated to the Bosch smoke number by SAE J1667. The principle of opacimeter is such as an absorption photometry; while a light emitting diode emits 563nm light and a crystal diode receives it, the opacity of the exhaust gas changes the light intensity which is translated to a voltage signal. Because of this principle, the opacity value in this study represents not only the smoke intensity but also the heterogeneity of the exhaust gas. The heterogeneous exhaust components include the liquid-phase unburned hydrocarbon in the incomplete PCCI combustion case. An emission measurement system (HORIBA MEXA1500D) sampling the exhaust gas after the opacimeter was also used to measure the concentrations of NOx, HC and CO.

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 PCCI combustion mode. The injection timings were applied more widely than previous works1,3,5,8). The fuel quantity of the second injection was fixed at 1.5 m³. The total quantity of fuel was changed to investigate the effect of the air/fuel ratio in PCCI combustion. 11.5 m³ was the actual fuel quantity at the idling condition of the corresponding diesel engine. The higher compression ratios were tried to promote auto-ignition at this idling condition. It was achieved by the modification of the piston block as shown in Fig. 5.

Table 3 Engine operating conditions

<table>
<thead>
<tr>
<th>Engine speed 800 rpm/no load</th>
<th>Injection pressure</th>
<th>30, 120MPa</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injection timing of the single injection</td>
<td>BTDC 250, 200, 150, 100, 50° crank angle (CA)</td>
<td></td>
</tr>
<tr>
<td>Main injection timing of the two-stage injection</td>
<td>BTDC 250, 200, 150, 100, 50° CA</td>
<td></td>
</tr>
<tr>
<td>Second injection timing of the two-stage injection</td>
<td>BTDC 20, 15, 10° CA and TDC</td>
<td></td>
</tr>
<tr>
<td>Total quantity of fuel injection</td>
<td>11.5, 16, 20.9 m³</td>
<td></td>
</tr>
<tr>
<td>Quantity of second injection</td>
<td>1.5 m³</td>
<td></td>
</tr>
<tr>
<td>Intake air temperature</td>
<td>30, 80, 120, 160°C</td>
<td></td>
</tr>
<tr>
<td>Compression ratio</td>
<td>18.9, 23, 27.7</td>
<td></td>
</tr>
<tr>
<td>Injection angle</td>
<td>150°, 100°</td>
<td></td>
</tr>
</tbody>
</table>
The cylinder pressure was recorded at every 0.16˚CA at 800 rpm. The cylinder pressures of 150 engine cycles were recorded and an average of those was used to calculate IMEP and heat release rate under these operating conditions.

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 mℓ. 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˚.

RESULTS

EFFECT OF INJECTION PRESSURE

Figure 6 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, BTDC 200˚CA. The intake air was preheated to 160˚C for assisting vaporization of diesel fuel. 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 9). 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.

EFFECT OF INTAKE AIR TEMPERATURE

Figure 7 shows non-luminous images of PCCI combustion with the variations of intake air temperature. This is for the single injection case where the fuel was injected early at BTDC 200˚CA. The flame intensity seems nearly uniform over the whole combustion chamber that represents multi-point ignition. Flame intensity was found to be higher when the intake air temperature is higher. Preheating of the intake air could achieve the sufficient vaporization of diesel fuel so that the vaporized diesel fuel was premixed with air. The mixture between diesel fuel and air was auto-ignited at multiple points. It results in the high flame intensity as shown in Fig. 7. When the intake air temperature was lower than 120˚C, the flame intensity was too low to detect clearly even with highest amplification of the image intensifier. Figure 8 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. 6 Effect of injection pressure on cylinder pressure and heat release rate; single injection, injection timing=BTDC 200˚CA, injection quantity=11.5 mℓ, intake air temperature=160˚C, compression ratio=18.9, injection angle=150˚

Fig. 7 Intensified imaging of PCCI combustion with the intake air temperature variation; single injection, injection timing=BTDC 200˚CA, injection quantity=11.5 mℓ, injection pressure=120MPa, compression ratio=18.9, injection angle=150˚
Figure 9 shows the IMEP and exhaust emissions. As the intake air temperature was increased from 30°C to 160°C, IMEP was increased more than 100%. The opacity and HC was decreased which represented the improvement of combustion efficiency. NOx maintained low value independent of intake air temperature which is the biggest advantage of the PCCI combustion. It should be noted that the efficiency and power output is still lower than the normal DI-diesel combustion supposedly due to the less utilization of the fuel attributed to cylinder wall wetting mentioned above.

TWO-STAGE INJECTION STRATEGY

Direct images of the DI-diesel combustion are shown in Fig. 10. The diesel fuel was injected at BTDC 15°CA and the injection pressure was 120MPa. The images explained the principle of DI-diesel combustion. Evidence of the flame occurred at BTDC 8°CA. The combustion started from local regions and was accompanied with a luminous flame that was attributed to the thermal radiation from soot. The mixing between fuel and air was not sufficient in certain regions. It is known that these regions are surrounded by stoichiometric mixtures, where the flame temperature is very high, resulting in the formation of NOx. This implied that the mixing between fuel and air was not sufficient in certain regions.

The direct imaging was also executed for the PCCI combustion with the two-stage injection strategy as shown in Fig. 11. The results showed luminous flame only in heterogeneous combustion regions of the second injection. The intake air (30°C) was not preheated in this case. If the PCCI combustion with the single injection was conducted without the intake air preheating, the diesel fuel could not
be vaporized so that the subsequent combustion is weak with very low IMEP value.

But if the two-stage injection was applied without the intake air preheating, the IMEP value could be improved. It was explained that the combustion of the second injection promoted vaporization of diesel fuel. So the mixing between the diesel fuel and air was also promoted. The ignition of this premixed regions resulted in high IMEP value. It was the ignition promotion by the second injection.

The second injection also controlled the combustion phase. The auto-ignition of the early injected fuel could not be achieved because the diesel fuel was not sufficiently vaporized without intake air preheating. The ignition was triggered by the second injection. Figure 12 shows the in-cylinder pressure trace and heat release rate for two-stage injection PCCI case with two different second injection timings. The main heat release was started near BTDC 10˚CA if the second injection timing was BTDC 10˚CA. It was started near TDC if the second injection timing was TDC.

EFFECT OF INJECTION TIMING

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.

These effects of the main injection timing on IMEP and exhaust emissions are described in Fig. 13. In the single injection case, IMEP was higher as the injection timing was retarded. As the main injection was advanced, the opacity and HC were reduced while NOx maintained its low value. Previous works\(^1,^3,^5,^8\) limit the main injection timing near BTDC 100 °CA. One of the reasons is low power output of the advanced main injection timing\(^5\). But in this study, low IMEP could be improved by the second injection. The two-stage injection result was also compared in Fig. 13. The extremely advanced timings like BTDC 200 °CA and BTDC 250 °CA showed a high value of IMEP and low value of opacity, HC and NOx. Figure 14(a) shows heat release rate with main injection timing variation in case of the single injection. The peak rate of heat release was highest when the main injection timing was BTDC 50 °CA. Although the ignition was severely advanced as shown, it indicated the highest IMEP value. Figure 14(b) shows extended combustion duration by the second injection. The BTDC 50 °CA injection case still showed highest peak rate of heat release but the BTDC 200 °CA case showed highly increased peak rate of heat release and longest combustion duration resulting in highest IMEP. The main injection timing variation did not affect the start of cool combustion and main heat release. The cool combustion kept the same value and had no relation with main injection timing.

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 timings of BTDC 250 °CA, 200 °CA and 150 °CA (Fig. 15(a), (b) and (c)) showed near-uniform flame intensity. This is the case for the more advanced injection timings than the previous works\(^3,^5,^8\). Previous researchers, Yoshinori Iwabuchi et al.\(^3\) and Ryo Hasegawa et al.\(^5\), used shadowgraph techniques and showed that luminous flame was not detectable. Injection timings of BTDC 100 °CA and 50 °CA (Fig. 15 (d) and (e)) 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. The line A-B of the Fig. 15(a) is an example of the sampling line. 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 PCCI combustion.

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**Fig. 13** Effect of fuel injection timing on IMEP and exhaust emissions; single injection and two-stage injection (second injection timing=BTDC 20 °CA), total fuel quantity=11.5 \(\text{ml}\), main injection quantity=10 \(\text{ml}\), second injection quantity=1.5 \(\text{ml}\), injection pressure=120 MPa, intake air temperature=160 °C, compression ratio=18.9, injection angle=150 °
Fig. 14 Effect of main injection timing on cylinder pressure and heat release rate; single injection and two-stage injection (second injection timing=BTDC 20°CA), total fuel quantity=11.5 m³, main injection quantity=10 m³, second injection quantity=1.5 m³, injection pressure=120MPa, intake air temperature=160°C, compression ratio=18.9, injection angle=150°

(a) Single injection

(b) Two-stage injection

Fig. 15 Intensified imaging of PCCI combustion under the main injection timing variation; single injection, injection quantity=11.5 m³, injection pressure=120MPa, intake air temperature=160°C, compression ratio=18.9, injection angle=150°
Effect of second injection timing

The aim of applying the two-stage injection strategy was the improvement of IMEP by the second injection. The second injection was expected to act as the ignition controller and the ignition promoter of the PCCI combustion. The second injection timing was the start timing of the main heat release in a low intake air temperature. As the intake air was preheated to assist vaporization of the diesel fuel, the second injection may not be the start of main heat release any more due to early ignition. 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.

These optimized second injection timings were tested as shown in Fig. 17(a). The intake air was preheated to 120°C. When the second injection timing was at TDC, there were two peaks of the main heat release. The first peak was from the auto-ignition of a premixed charge. The second peak was from the ignition of remaining charge by the second injection. As the second injection timing was advanced until the BTDC 15°CA, which was the timing of auto-ignition at 120°C intake air temperature, the highest value of the main heat release was detected. It showed only one high peak of the main heat release. This was because the combustion efficiency was improved by optimizing second injection timing.

The optimized second injection timing was further advanced as the intake air was preheated as shown in Fig. 17(b). The intake air temperature was 160°C. The highest value of the main heat release was detected when the second injection timing was BTDC 20°CA. It was also the timing just before the auto-ignition.

The combustion duration became longer than the case of the single injection. In this case, IMEP was highly improved while exhaust emissions maintained low values as shown in Fig. 13. From the implications of Fig. 12 and the tests, optimized second injection timings were selected as listed in Table 4.
EFFECT OF COMPRESSION RATIO

When the intake air temperature was 30°C, a higher compression ratio increased IMEP as shown in Fig. 18. This was because the thermal efficiency increased with the higher compression ratio. But the opacity, HC and CO still showed high value in this low intake air temperature. This was because of the lower combustion efficiency than DI-diesel combustion. NOx showed no correlation with the compression ratio variation and kept its low value. When the intake air temperature was 160°C, unstable combustion of the high compression ratio operation increased the opacity and HC. The IMEP showed limited increase although the compression ratio increased. This was explained by the heat release rate analysis as shown in Fig. 19. It shows severely advanced ignition timing with a high intake air temperature and high compression ratio. The ignition timing of the compression ratio 27 was advanced more than 5° crank angle compared to the ignition timing of the 18.9 compression ratio. This problem hindered the improvement of the IMEP though the main heat release increased.

### Table 4 Optimal second injection timings for ignition promotion under intake air temperature variation

<table>
<thead>
<tr>
<th>Intake Air Temperature</th>
<th>Optimal Second Injection Timing</th>
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<tr>
<td>30°C</td>
<td>TDC</td>
</tr>
<tr>
<td>80°C</td>
<td>BTDC 10°CA</td>
</tr>
<tr>
<td>120°C</td>
<td>BTDC 15°CA</td>
</tr>
<tr>
<td>160°C</td>
<td>BTDC 20°CA</td>
</tr>
</tbody>
</table>

Fig. 18 Effect of compression ratio on IMEP and exhaust emissions both 30°C and 160°C intake air temperature; two-stage injection, main injection timing=BTDC 200°CA, second injection timing=BTDC 10°CA, main injection quantity=10 ml, second injection quantity=1.5 ml, injection pressure=120MPa, injection angle=150°

Fig. 19 Effect of compression ratio on heat release rate; single injection, injection timing=BTDC 200°CA, total fuel quantity=11.5 ml, injection pressure=120MPa, intake air temperature=160°C, injection angle=150°
The effect of air fuel ratio variation with the changes of injected fuel quantity was studied at fixed idling condition. The corresponding DI-Diesel engine uses 11.5 \text{mL} diesel fuel at the idling condition. Table 5 represents the fuel quantities and the relative air/fuel ratios. Maximum fuel quantity was almost double of idling quantity.

Figure 20 shows that the flame intensity was increased as the more fuel was injected when the single injection in 160°F intake air temperature was applied. As the fuel quantity increased more than 11.5 \text{mL}, getting relatively non-uniform flame intensity was detected for 16 and 20.9 \text{mL} cases as shown in Fig. 21. This was because the time for the premixing was insufficient as the fuel quantity increased.

If more fuel was injected, the higher IMEP was achieved as shown in Fig. 22. In this case, the intake air temperature was 30°F. If the compression ratio was 23, IMEP showed the same value with DI-Diesel’s when the doubled fuel quantity was injected with second injection at TDC. This result showed bad fuel economy to achieve the high IMEP. This bad fuel economy could be improved by intake air preheating as shown in Fig. 23. But the 23 compression ratio operation with this high intake air temperature hindered the increase of IMEP though the injected fuel quantity was increased. This was because the ignition was significantly advanced as shown in Fig. 24.

**Table 5 Relative air/fuel ratio by fuel quantity variations**

<table>
<thead>
<tr>
<th>Relative air/fuel ratio (air excess ratio), $\lambda$</th>
<th>4</th>
<th>3</th>
<th>2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel quantity (\text{mL})</td>
<td>11.5</td>
<td>16.0</td>
<td>20.9</td>
</tr>
</tbody>
</table>

**EFFECT OF INJECTED FUEL QUANTITY**

Fig. 20 Intensified imaging of PCCI combustion with the fuel quantity variation; single injection, injection timing=BTDC 200°CA, injection pressure=120MPa, intake air temperature=160°F, compression ratio=18.9injection angle=150°

Fig. 21 Flame intensity of the PCCI combustion under the fuel quantity variation, detected from the Fig. 20
Fig. 22 Effect of fuel quantity on IMEP and exhaust emissions both 18.9:1 and 23:1 compression ratio; two-stage injection, main injection timing=BTDC 200’CA, second injection timing=TDC, second injection quantity=1.5 mℓ, injection pressure=120MPa, intake air temperature=30°C, injection angle=150°

Fig. 23 Effect of fuel quantity on IMEP and exhaust emissions both 18.9:1 and 23:1 compression ratio; two-stage injection, main injection timing=BTDC 200’CA, second injection quantity=1.5 mℓ, injection pressure=120MPa, intake air temperature=160°C, injection angle=150°
OVER-PENETRATION PROBLEM

Premixed combustion enabled the low combustion temperature resulting in NOx reduction by more than 90% compared to DI-Diesel's. HC and CO are increased by low exhaust gas temperature. The opacity that represented PM concentration was higher than DI-Diesel's. The high pressure injection (120MPa) using common-rail fuel system enabled the minimized opacity that was almost 0% at idling condition of DI-Diesel engine. But the opacity of this study only reached a minimum of 14% though the intake air was preheated to 160°C and the optimized second injection was applied. This was the result of over-penetration of in-cylinder fuel injection. The hole-type injector designed for DI-diesel engine showed long spray penetration length in PCCI engine mode as shown in Fig. 25 (a)

Previous researchers tried to solve this problem with new injectors like the small nozzle-diameter injector, swirl injector, impinged-spray injector or the small injection-angle injector. Considering the movement of piston and the position of cylinder liner, injection angle 70° is required to avoid wall wetting. The virtual penetration with small injection-angle injector, with 70° injection angle, was compared with the solid confinement of this engine (bore 83mm) as shown in Fig. 26.
Fig. 26 Optimized Injection Angle for PCCI engine

(a) Axial spray penetration in Pz-direction

(b) Radial penetration in Pr-direction

Fig. 27 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 ml, main injection quantity=10 ml, second injection quantity=1.5 ml, injection pressure=120MPa, intake air temperature=160°C, compression ratio=18.9
EFFECT OF SMALLER INJECTION ANGLE

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. As shown in Fig. 26, the spray of axial-direction (Pz) didn’t contact with the top of the piston unless the injection timing was advanced earlier than BTDC 290˚CA case. For a spray of radial-direction (Pr), a period for spray impingement to a cylinder liner was shortened from 0.4˚CA to 0.1˚CA.

Figure 27 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.

CONCLUSIONS

PCCI combustion with two-stage injection strategy is 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 two-stage injection strategy could act as an ignition promoter so that the combustion efficiency is improved with higher IMEP.
- The two-stage injection strategy could act as a combustion controller in a low intake air temperature, while an advanced auto-ignition prevents this effect in high intake air temperatures.
- 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.
- The high intake air temperature and low compression ratio with a little increase of fuel quantity is the optimized operating condition for the high IMEP and the low exhaust emissions in this study.
- 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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REFERENCES


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