Investigation on Effect of Offset Orifice Nozzle under Multi Pulse Ultrahigh Pressure Injection and PPC Combustion Conditions

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ABSTRACT: The effect of nozzle orifice design on combustion characteristics under multi pulse ultra high pressure injection and PPC conditions. The objective of offset orifice nozzle is to shorten spray penetration and improve mixture formation in order to increase thermal efficiency. The offset orifice nozzle was designed by shift orifice aliment from into the sac center to edge of sac follow swirl direction. The experiments were carried out on a single cylinder engine at 0.55MPa gross IMEP at 1,750 RPM and 0.7MPa gross IMEP at 2,000 RPM. The injection pulses were 3 pulses equally mass under injection pressure up to 350MPa.

KEY WORDS: Heat engine, compression ignition engine, performance/fuel economy/efficiency, offset orifice nozzle, PPC [A1]

1. Introduction

The world is facing numerous environmental problems and fossil fuel depletion. Many researchers are trying to develop new energy sources for vehicles such as electricity, fuel cells, hydrogen, etc. Unfortunately, these are difficult to implement on conventional vehicles due to cost, energy sources, vehicle performance and distances between refueling. Internal combustion engines and hybrid internal combustion engines will play an important role in powering light-duty vehicles at least until 2050 (1). CI engines provide higher thermal efficiency compared with other internal combustion engines. However, large amounts of soot and NOx are produced during the combustion process (2).

By applying partially premixed charge (PPC) or premixed charge compression ignition (PCCI) combustion, soot and NOx emissions can be simultaneously reduced (3). Conventional premixed combustion is realized by using a high Exhaust Gas Recirculation (EGR) rate combined with early fuel injection to provide sufficient time to produce a lean and homogenous mixture (4). However, the early fuel injection results in increased HC and CO emissions from fuel adherence on the piston head and cylinder wall (5). Another disadvantage of PCCI combustion is limited operating range due to misfire and engine knock (6). To control the PPC combustion, ignition timing is the critical parameter. In order to inject the fuel close to the TDC, shortening the mixing formation of fuel and air is required. One solution to improve mixture formation is increasing fuel injection pressure. This enhances mixture formation by increasing turbulence energy at the outlet of the nozzle orifice (7). Furthermore, it increases spray fuel impingement and spray penetration. Multi pulse fuel injection is a simple, effective method to reduce impingement. This results from decreased spray penetration and a fuel-rich area. On the other hand, multi pulse injection decreases thermal efficiency due to increased combustion duration (8). A previous study showed that a combination of increasing injection pressure and injection pulse expanded the operating range of PPC combustion and simultaneously increased thermal efficiency (9). However, a too high injection pressure will increase fuel impinging, thus ITE reduction. Reducing nozzle orifice diameter is a simple, effective method to reduce spray penetration. On the other hand, reducing nozzle orifice diameter decreases injection rate due to a decreased orifice cross section area. This means increasing fuel injection duration in order to deliver the same amount of fuel. Offsetting the orifice nozzle is an effective method of reducing spray penetration and also delivering the same amount of fuel without prolonging actual injection duration. The asymmetrical fuel velocities at the nozzle orifice outlet due to difference distances of each orifice side make the fuel spray bend to follow the offset direction, as shown in Figure 1.

Fig. 1 Ultra-high pressure multi pulsed injection PPC combustion concept.
Offsetting the orifice nozzle was achieved by shifting orifice alignment from the sac center to the edge of the sac, following swirl direction to create different lengths at each side of the nozzle orifice entrance. The different lengths at the nozzle orifice entrance created a different recirculation zone at each side of the nozzle orifice entrance, thus different velocities at the orifice outlet. A counterbore design was applied to offset the orifice nozzle in order to reduce orifice length, increasing from nozzle orifice position change. This also recovered flow rate from reducing flow loss inside the nozzle orifice. Counterbore distance was designed to ensure constant orifice length in both the standard nozzle and the offset orifice nozzle, as shown in Figure 2. The offset orifice nozzle provides significantly shorter spray penetration and a wider spray angle compared with the standard nozzle.

However, the offset orifice nozzle still has a disadvantage in terms of flow rate. The offset orifice nozzle has approximately a 5% lower injection rate under a fully open condition compared with the standard nozzle under all fuel injection pressures, as shown in Figure 3. This is due to higher cavitation inside the nozzle orifice.

To date, no study has investigated the effect of an offset orifice nozzle on PPC combustion. The objective of this paper is to investigate the effect of an offset orifice nozzle under multi pulse ultra-high pressure injection conditions, specifically observing combustion characteristics that have not been studied before.

This paper investigates the effects of an offset orifice nozzle under multi pulse ultra-high pressure injection conditions using heat release rate analysis, exhaust emissions measurement and heat balance analysis. Two designs of injector were tested at various injection pressures. Testing was conducted on a single cylinder engine.

2. Experimental

2.1. Experimental engine

A single cylinder DI diesel research engine was equipped with a 350 MPa common rail fuel injection system. The specifications of the research engine are given in Table 1. The fuel used in this experiment was Japanese commercial diesel fuel (JIS2). The intake air flow rate was measured by an orifice flow meter and mixed with the cooled EGR gas which was mixed with fresh air. This was boosted by an external drive supercharger before being supplied to the engine. The intake manifold was equipped with an electric heater to maintain the intake air temperature at 50°C. The engine was equipped with a piezoelectric dynamic pressure transducer (Kistler 6052C) and amplified with a charge amplifier (Kistler 5018A). Pressure data was transmitted to a personal computer (PC) at 0.125 degrees of crank angle. Resolved pressure data averaged 150 cycles. The exhaust emissions were measured with an exhaust gas analyzer (Horiba MEXA1700DEGR) and smoke meter (AVL 415SE), as shown in Figure 4.
Actual injection duration is an important parameter to determine premixed combustion. Injection rate measurement was conducted separately from the combustion experiment by using the Bosch injection rate measurement.

Table 1. Engine specifications.

| Engine type          | DI Single-cylinder Diesel engine |
|----------------------|----------------------------------|
| Bore                 | 85 mm                            |
| Stroke               | 96.9 mm                          |
| Compression ratio    | 16.3 : 1                         |
| Piston               | Low swirl type                   |

2.2. Analysis method

2.2.1. Apparent rate of heat release

Apparent rate of heat release \( (dQ/d\theta) \) was calculated from pressure rise after fuel injection \( (dP/d\theta) \) by applying the first law of thermodynamics \(^{(1)}\), using equation (1) where \( \gamma \) is the specific heat ratio, \( P \) is the initial pressure inside the combustion chamber and \( (dV/d\theta) \) is the rate of volume change in the chamber. The specific heat ratio in this study varied depending on the average gas temperature inside the combustion chamber during the combustion process \( (dV/d\theta) \). The average gas temperature was calculated from in-cylinder pressure. The apparent rate of heat release included cooling loss.

\[
dQ = \frac{1}{\gamma - 1} \left( \frac{\gamma P}{\frac{dP}{d\theta}} + \frac{\gamma P}{\frac{dV}{d\theta}} \right) - \frac{P}{\gamma - 1} \frac{dv}{d\theta}
\]

2.2.2. Heat balance

Heat balance was expressed in terms of gross work, exhaust loss, unburned loss, and cooling loss, as shown in equations (2) through (5). First, the gross work \( (W) \) was calculated from the mean effective pressure \( (\bar{P}) \) and change of volume \( (dV) \), as shown in equation (2).

\[
W = \bar{P} \cdot dV
\]

In order to calculate the exhaust and unburned loss, it is necessary to use the volumetric exhaust flow rate.

Exhaust loss \( (Q_{ex}) \) was calculated by assuming the composition and enthalpy of the intake air \( (X(i)_{mol,EX}) \) and the exhaust gas \( (X(i)_{mol,EX}) \), where \( Cp \) is constant pressure specific heat, as shown in equation (3).

\[
Q_{ex} = \int_{\theta_{in}}^{\theta_{out}} X(i)_{mol,EX} \cdot Cp(l,T) dT - \int_{\theta_{in}}^{\theta_{out}} X(i)_{mol,EX} \cdot Cp(l,T) dT
\]

The unburned loss \( (Q_{ub}) \) was calculated from heat absorption when the THC (\( m_{THC} \)) and CO (\( m_{CO} \)) formed, where \( LHV_{fuel} \) is fuel low heating value, as shown in equation (4).

\[
Q_{ub} = LHV_{fuel} \times m_{THC} + (-\Delta h'_{CO}) \times m_{CO}
\]

The cooling loss \( (Q_{c}) \) was calculated by subtracting actual work, exhaust loss and unburned loss from input energy, as shown in equation (5).

\[
Q_{c} = Q_{f} - W - Q_{ex} - Q_{ub}
\]

2.3. Experimental conditions

In this study the engine was operated under two operating conditions, 0.55 MPa gross IMEP at 1,750 RPM and 0.7 MPa gross IMEP at 2,000 RPM, in order to expand the operating length of PPC combustion. Injection quantities were adjusted in order to achieve target gross IMEP. The intake and exhaust pressure were set equally to simulate operation of a turbo charger. EGR ratio was kept constant for each operation condition. Swirl ratio was at 1.3. The injector used in this study was a DENSO G4.5S injector (solenoid type), with a nozzle hole diameter of 0.123 mm, and 8 holes on both the standard and offset orifice nozzles, as shown in Fig. 2. The injection strategy used 3 main injections. The main injection mass was equally distributed between each pulse of injection. Dwell between each pulse of injection, determined as minimum dwell time, was used to separate injection rate curves and keep the injected amount of each pulse constant. If the dwell time is too short, injection rate will be reduced as the needle does not move in the desired direction (12).

Table 2. Experimental conditions.

| Engine revolution | 1,750 RPM | 2,000 RPM |
|-------------------|-----------|-----------|
| Gross IMEP        | 0.55 MPa  | 0.7 MPa   |
| Intake. Exhaust pressure | 0.11 MPa | 0.12 MPa |
| EGR ratio         | 40 %      | 30 %      |
| Swirl ratio       | 1.3       |           |
| Injector nozzle orifice | 0.123 mm x 8 Standard, 0.123 mm x 8 Offset |
| Nozzle cone angle | 156°      |           |
| Injection pressure | 150, 200, 250, 300, 350 MPa |
| Injection pulse   | 3 pulses  |           |
| Main injection timing | -10°, -1.5°, 7° ATDC, -12°, -2°, 8° ATDC |
3. Result and Discussion

In this experiment, the effects of nozzle orifice position and fuel injection pressure were investigated in terms of apparent rate of heat release, exhaust emissions and heat balance.

3.1. Effect of nozzle orifice position and injection pressure on apparent rate of heat release

Figure 6 shows in-cylinder pressure, apparent rate of heat release and injection rate under the operating condition of 0.55MPa gross IMEP at 1,750 RPM. Apparent rate of heat release is calculated from pressure rise after fuel injection by using equation (1). The peak of in-cylinder pressure and apparent rate of heat release increased by increasing fuel injection pressure. However, the first peak of in-cylinder and apparent rate of heat release decreased with increasing injection pressure. Increasing injection pressure decreased the first pulse rate of heat release peak and increased the second pulse rate of heat release peak due to reducing accumulated fuel during injection delay due to a shorter ignition delay. In both the standard and offset orifice nozzle, the apparent rate of heat release at 1st pulse were premixed combustion followed by PPC in 2nd and diffusive combustion in 3rd pulse due to ignition delay shortening from the high in-cylinder pressure and temperature made by 1st and 2nd pulse combustion. The higher in-cylinder pressure and temperature of offset orifice nozzle promoted mixture formation thus shorter 2nd pulse ignition delay. The ignition delay of the second and third pulse caused by combustion of each pulse of injection occurred due to the differences of combustion chamber.
The offset orifice nozzle showed an advance rate of heat release compared with the standard nozzle. This is due to a higher ambient pressure, temperature and equivalence ratio inside the spray area. This effect can be clearly seen with increasing fuel injection pressure. Fuel injection pressure of 350 MPa shows a shortening of actual injection duration of each pulse of 30.0% and 26.2% respectively, compared with pressure of 150 MPa for the standard and offset orifice nozzle. This made combustion duration (CA10 to 90) become shorter.

Figure 7 shows in-cylinder pressure, apparent rate of heat release and injection under the operating condition of 0.70MPa gross IMEP at 2,000 RPM. The highest peaks of in-cylinder pressure and apparent rate of heat release were observed at the offset orifice nozzle injector. The effect of injection pressure on peak in-cylinder pressure and apparent rate of heat release also showed the same trend with 0.55MPa gross IMEP at 1,750 RPM. Under this operating condition the effects of increasing fuel injection pressure on combustion duration shortening are less compared with 0.55MPa gross IMEP at 1,750 RPM. The standard injector under fuel injection condition 350 MPa shows shorter actual injection duration (38.5%) compared with injection pressure of 150 MPa. The offset orifice nozzle also shows shorter actual injection duration (36.8%) under the same condition. The effects of increasing fuel injection pressure on actual injection are superior compared with the 0.55MPa gross IMEP at 1,750 RPM condition. This is due to a higher fuel injection amount and engine speed.
3.2. Effect of nozzle orifice position and injection pressure on exhaust emissions

Figure 8 shows the effect of nozzle orifice position and injection pressure on exhaust emissions. Exhaust emissions were measured by smoke meter and gas analyzer.

Figure 8(a) shows the exhaust emissions under the operating condition 0.55MPa gross IMEP at 1,750 RPM. The offset orifice nozzle showed a slightly higher smoke level compared with the standard nozzle at 150 and 200 MPa injection pressure. However, as the injection pressure rose, the differences were diminished. Increasing fuel injection pressure decreased smoke from enhanced mixture formation. This was due to increased nozzle orifice turbulence energy (7). Another benefit of increasing injection pressure is improved soot oxidation (15). Increasing fuel injection pressure also decreases NOx. This is due to a reduced average flame temperature (15). However, differences of NOx emissions between the standard and the offset orifice nozzle were not found under this operating condition. On the other hand, THC and CO increased with increasing fuel injection pressure. This is due to the decreased equivalence ratio inside the spray area from increasing fuel injection pressure.

Figure 8(b) shows the exhaust emissions under the operating condition of 0.70MPa gross IMEP at 2,000 RPM. Under this operating condition smoke also decreased with increasing injection pressure. However, smoke under 0.70MPa gross IMEP at 2,000 RPM operating condition was significantly higher than under 0.55MPa gross IMEP at 1,750 RPM. This is due to increased fuel impingement from an increasing fuel injection amount and less mixture formation time from increasing engine speed. The offset orifice nozzle shows a significantly higher smoke level compared with the standard nozzle. This is due to asymmetrical fuel velocity at the nozzle orifice outlet crate overlapping the flame spray. Under this operating condition NOx also decreased as injection pressure increased. There was higher NOx due to a lower EGR ratio. Increasing oxygen concentration from decreasing EGR ratio decreases flame temperature and NOx emissions (16). THC and CO emissions under this operating condition were not increased with increasing injection pressure under the 0.55MPa gross IMEP at 1,750 RPM condition. This is due to increase in combustion temperature promoting THC and CO oxidation.

Fig. 8. Effect of nozzle orifice position and injection pressure on exhaust emissions under 0.55MPa gross IMEP at 1,750 RPM (a) and 0.70MPa gross IMEP at 2,000 RPM (b).
3.3. Effect of nozzle orifice position and injection pressure on heat balance

The heat balance analysis was carried out to explore the thermal efficiency. Based on the exhaust emissions results, the mass of each exhaust emission component was calculated. The gross indicated thermal efficiency was calculated based on these results. Next, the exhaust heat loss was calculated using enthalpy of intake and exhaust temperature. Cooling loss was calculated by subtracting the heat loss from gross indicated thermal efficiency, exhaust heat loss and unburned loss from input heat energy.

Figure 9(a) shows the heat balance under the operating condition 0.55MPa gross IMEP at 1,750 RPM. The offset orifice nozzle showed a significantly higher gross ITE compared with the standard nozzle in all injection pressures. However, 350 MPa fuel injection pressure showed a decrease in gross ITE due to the lower equivalence ratio inside the spray area from increasing fuel injection pressure. This coincides with results of THC and CO. The offset orifice nozzle showed a higher exhaust loss compared with the standard nozzle due to higher exhaust gas temperature. Higher exhaust gas temperatures result in higher exhaust loss. Increasing fuel injection pressure shortens actual fuel injection duration. This caused a decrease in exhaust gas temperature, thus lower exhaust loss. The offset orifice nozzle showed a lower cooling loss compared with the standard nozzle due to shorter spray penetration. Increasing injection pressure increased cooling loss due to increasing spray penetration and fuel impingement on the piston head and cylinder. Increasing injection pressure increased unburned loss. This coincides with the results of THC and CO.

Figure 9(b) shows the heat balance under the operating condition of 0.70MPa gross IMEP at 2,000 RPM. Under this operating condition, gross ITE, exhaust loss and cooling loss show the same trend as the 0.55MPa gross IMEP at 1,750 RPM condition. However, under this operating condition gross ITE become lower compared with the 0.55MPa gross IMEP at 1,750 RPM condition due to less mixture formation time from increasing engine speed. On the other hand, unburned loss was consistent under this operating condition.

Fig. 9. Effect of nozzle orifice position and injection pressure on heat balance under 0.55MPa gross IMEP at 1,750 RPM (a), 0.70MPa gross IMEP at 2,000 RPM (b).
4. Conclusions

In this study, the experiment investigated the effect of nozzle orifice position and injection pressure on apparent rate of heat release, exhaust emissions and heat balance. Indicated thermal efficiency improvement and exhaust emissions reduction by applied offset orifice nozzle was confirmed with the base condition indicating thermal efficiency and exhaust emissions.

1. The experiment found the offset orifice nozzle can be used under multiple pulse PPC combustion conditions. This makes it possible to increase indicated thermal efficiency of PPC combustion.

2. The offset orifice nozzle provided higher in-cylinder pressure, peak of heat release rate, advanced rate of heat release and shortened combustion duration compared with the standard nozzle.

3. The combination of the offset orifice nozzle and ultra-high pressure is an effective method to increase indicated thermal efficiency and reduce NOx and soot. However, when the engine speed and load were increased, smoke emissions significantly increased.

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