Homogeneous Charge Compression Ignition Engine with Two-Stage Diesel Fuel Injection

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Abstract. A diesel-fueled homogeneous charge compression ignition (HCCI) 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 HCCI 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 HCCI engine. A two-stage injection strategy was applied to relieve these problems. Premixing of diesel fuel with air was realized mainly by the early injection strategy. 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 ml) as an ignition promoter and combustion phase controller near TDC. Effects of injection pressure and intake air temperature were studied with two-stage injection strategy. Concerning the injector geometry, a hole-type injector (5 holes) with small injection angle (100°) instead of conventional angle (150°) was applied to minimize the over-penetration that may prevent forming a premixed charge due to wall wetting. The multi-hole injector (14 holes) was also tested to maximize the atomization of spray. 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 although 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. The small injection angle was effective to minimize the over-penetration, and the multi-hole injector was effective to maximize the atomization so that the power output was increased, and exhaust gas was decreased. Exhaust results of the HCCI combustion optimized by this parametric study showed reduced NOx by more than 90% but increased HC, CO emissions compared with the conventional direct-injection (DI) diesel engine operation in the same engine. In addition, the application of the alternative gaseous fuel such as dimethyl ether (DME) was discussed implying the practical potential of HCCI operation.

1. Introduction

Homogeneous charge compression ignition (HCCI) combustion is an advanced technique for reducing the hazardous nitrogen oxide (NOx) and particulate matter (PM) in a diesel engine. NOx could be reduced by multipoint ignition through the cylinder that results low combustion temperature. PM could also be reduced by homogeneous charge that has no fuel-rich zones compared with conventional diesel combustion.
The homogeneity of the premixed charge is governed by several parameters; sufficient time for premixing, pressure and temperature of air inside the cylinder when fuel is injected, injector geometry, wall-wetting and in-cylinder flow such as swirl. These parameters are dependent on each other, and sometimes the trade-off is occurred. The preheating of intake air for vaporizing diesel fuel makes early ignition that results in low thermal efficiency. While the fuel injection timing is much more advanced than a conventional DI-diesel engine, in-cylinder air pressure and temperature is too low so that the spray impingement and wall-wetting becomes serious. There is no direct control method for HCCI combustion. Therefore, a lot of researchers have focused on a diesel-fueled HCCI combustion to solve these problems. The indirect techniques such as exhaust gas recirculation (EGR), compression ratio variation are known as the possible solutions.

Concerning the injector geometry, a swirl injector [1], a hole-type injector with small injection angle [4, 12] or impinged-spray nozzle injector [9] were applied to minimize the over-penetration of the fuel. The multi-hole injector [1, 14] was applied to promote the atomization and spatially homogeneous distribution of fuel. The fuel injection method based on a DI-diesel engine has been studied actively. The combination of early direct injection and near TDC injection was tested to achieve high power output [1, 6, 7, 12]. Previous researchers tested commonly the effects of intake air preheating to vaporize diesel fuel [1, 3-5, 7], low compression ratio [1, 4, 5, 7, 9] and EGR [1, 4-6, 9, 10] to retard ignition timing. The visualization of diesel-fueled HCCI combustion was applied with the shadowgraph technique [7, 9]. Although the achievement is limited in low load condition, advantages of NOx and PM reduction were reported [1, 3-7, 9, 10].

In this study, a hole-type injector with the small injection angle (100°) and a multi-hole injector (14 holes°) were tested in the engine, and the macroscopic spray structure was analyzed for each injector in the visualization chamber. The injection parameters – injection pressure, injection strategy and injected fuel quantity – were tested widely. The two-stage injection strategy was applied to promote ignition and to control the combustion as shown in Fig. 1. An extremely early direct-injection, named as the main injection, was applied to form a premixed charge. This injection was followed by the other direct-injection, named as the second injection, executed near TDC. The main difference between this study and previous researches [1, 6, 7, 12] is the smaller fuel quantity in the second injection (1.5 ml). The result from the pilot injection study for the DI-diesel engine [11] reported that this small amount of fuel injection showed non-luminous combustion itself. 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 the second injection timing while it did not affect the homogeneity of mixture. The dependent effect of intake air temperature was tested with this two-stage injection strategy. Also combustion images were taken to show characteristics of HCCI combustion in the single-cylinder optical engine. Most of research parameters are directed to solve the problems of diesel fuel. Alternatively, application of a gaseous fuel, DME, was tested implying the practical potential of HCCI operation. DME has the advantage of easy evaporation so that it does not need the preheating. DME results showed high power output and substantial reduction of NOx while its low PM emission was maintained.
Fig. 2 Spray chamber setup

Fig. 3 Example of macroscopic spray images and measured radial penetrations
2 Experimental apparatus

2.1 Spray chamber

Spray characteristics were investigated in a spray chamber as shown in Fig. 2. The chamber allows optical accesses through a circular window. The pressure of the chamber was controlled to meet the same condition in a HCCI engine – mostly open-valve condition (atmospheric pressure). Spray images frozen by spark light source, which had light duration of shorter than 100ns were acquired with CCD camera.

Radial spray penetrations corresponding to the liquid phase penetration were measured from macroscopic images as shown in Fig. 3. Figure shows penetrations under atmospheric chamber condition and those averaged penetrations as a fitted line. The common-rail pressure was 120MPa. The indicated time means the time after start of injection that is detected from the injection rate.

2.2 Engine

A Single-cylinder optical diesel engine (Engine Tech; RSi-090) was used to operate in HCCI combustion mode. 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 4 shows the schematic diagram of the experimental setup. Two common-rails were used for each fuel - diesel and DME. DME was pressurized by the pneumatic pump, instead of the common-rail pump, to solve the lubrication problem. 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).

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 opaque exhaust components include the liquid-phase unburned hydrocarbon in the incomplete HCCI 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.

Figure 5 depicts the optical access and an observation field to the combustion chamber. An elongated piston was applied in order to enable the mounting of a 45° mirror beneath the piston quartz window. A high speed digital video camera (Vision Research Inc.; Phantom v7.0) with an image intensifier (PROXITRONIC) was used to take images of HCCI 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 800 rpm 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 HCCI combustion fields. The maximum gain of the intensifier was 10^8W/W and the maximum light power per area was 70µW/cm². Almost 90% of the gain value was used for detecting weak HCCI combustion except highly luminous case where the gain value was decreased to 30%.

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<th>Table 1. Engine specifications</th>
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<td><strong>Engine</strong></td>
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<td>Bore x Stroke</td>
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<td>Displaced volume</td>
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<td>Compression ratio</td>
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<td>Injector</td>
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3 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 5-hole injector with injection angle 150°. A smaller injection angle 100° was applied to overcome the over-penetration problem in HCCI combustion mode. The 14 holes injector was applied to maximize atomization of diesel fuel to form a premixed charge. The injection timings were applied more widely than previous works [1, 7, 9, 12]. The fuel quantity was fixed that was adopted for 800rpm, idling condition.

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 mm³. The intake air was not preheated and the compression ratio was 18.9. The injector has 5 holes with the injection angle 150°.
Table 2. High speed digital video camera settings

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<th>Camera: Vision Research Inc. Phantom v7.0</th>
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<td>Recording rate</td>
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<td>Exposure</td>
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<td>Resolution</td>
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Table 3. Engine operating conditions

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<th>Engine speed 800 rpm/no load</th>
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<td>Injection pressure</td>
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<td>Main Injection timing</td>
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<td>Second injection timing of the two-stage injection</td>
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<td>Total quantity of fuel injection</td>
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<td>Quantity of second injection</td>
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<td>Intake air temperature</td>
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<td>Compression ratio</td>
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<td>Fuel</td>
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4 Results

4.1 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 [2]. 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 [8], the advantage of a high pressure injection such as better atomization was more dominant so that the injection pressure was fixed as 120MPa for whole experiments.

![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°](image-url)
4.2 Effect of intake air temperature

Preheating of the intake air could help the sufficient vaporization of diesel fuel, and then the vaporized diesel fuel was premixed with air. Figure 7 shows the effect of intake air temperature on the cylinder pressure and the heat release rate. The heat release rate 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 could result in low thermal efficiency and low power output compared to the DI-diesel combustion.

4.3 Effect of injection timing

The fuel injection timing should be advanced earlier than DI-diesel’s so that the time for premixing was fulfilled. If the injection starts too early, however, the initial air temperature is lower so that the vaporization rate of diesel fuel decreases. Moreover, the wall-wetting is increased by the 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. If injection starts later, the temperature is initially higher so that the vaporization rate of diesel fuel increases. The wall-wetting is decreased by the shorter spray penetration. But the time for the premixing between diesel fuel and air would be insufficient. The premixed charge is inhomogeneous resulting in higher opacity and HC in the exhaust stream.

These trade-off characteristic by variation of the injection timing on IMEP, cylinder pressure, heat release rate and exhaust emissions are described in Fig. 8. If the injection timing was more advanced than BTDC 50°C until BTDC 250°C, the opacity was decreased as the injection timing was advanced. While the NOx was dramatically reduced that was important advantage of the HCCI combustion. It explained that the premixing between the fuel and air was improving. The heat release rate explained same characteristics. At the injection timing BTDC 50°C, the ignition delay was near 10°CA as in DI-diesel operation. Its ignition was much more advanced than the other cases. For earlier timings than BTDC 50°C, the ignition of cool combustion and hot combustion were occurred in order at the same timings.

Previous works [1, 7, 9, 12] focused on the limited injection timings. One of the reasons was low power output of the advanced injection timings [7], but the results of IMEP and exhaust gases as well as the flame images explained that the more advanced than BTDC 100°C should be focused. The flame images for different injection timings and those spatial grey-level analyses are shown in Fig. 9. The grey-level data was detected from the cross line of the bowl. The line A-B of the Fig. 9(a) is an example of the sampling line. The optical investigation also proved that the injection timing should be advanced to earlier than BTDC 100°C. Injection timings of BTDC 250°C, 200°C, 150°C and 100°C showed near-uniform flame intensity. Injection timing of BTDC 200°C is
50°C showed relatively non-uniform and luminous flames, if the intensified direct-imaging technique was applied. Therefore, extremely advanced timings at the intake stroke were also preferred in this study for the successful HCCI combustion.

### 4.4 Effect of two-stage injection strategy

The aim of two-stage injection strategy was improvement of the combustion efficiency by the second injection. The second injection was expected to act as the combustion controller and the ignition promoter of the HCCI combustion. Figure 10 shows the in-cylinder pressure trace and heat release rate with the second injection timing variation compared to the single injection case. The intake air temperature was controlled 30°C for Fig. 10 (a) and 160°C for Fig. 10 (b). The second injection with small fuel quantity (1.5 ml) increased the heat release rate of hot combustion when the intake air temperature was 30°C. The auto-ignition of the early injected fuel could not be achieved because the diesel fuel was not sufficiently vaporized without intake air preheating. It was explained that the combustion of the second injection promoted vaporization of diesel fuel. Therefore, the mixing between the diesel fuel and air was also promoted. The ignition of this premixed regions resulted in high heat release rate. It was the ignition promotion by the second injection. Concerning the second injection timing, the ignition was triggered by the second injection that was the start timing of the hot combustion following cool combustion. The hot combustion 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. The highest heat release rate was measured when the second injection timing was TDC in this case. As the intake air was preheated to assist vaporization of the diesel fuel as shown in Fig. 10 (b), the second injection may not be the start of the heat.
Fig. 9 Intensified imaging of HCCI combustion under the injection timing variation; single injection, injection quantity=11.5 ml, injection pressure=120MPa, intake air temperature=160°C, compression ratio=18.9, injection angle=150°
release rate of hot combustion any more due to early ignition. In this case, the second injection timing should be advanced to the optimal timing, which was just before hot combustion, to promote the ignition. When the second injection timing was at TDC or BTDC 10°CA, there were two peaks of the heat release rate of hot combustion. 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 20°CA, which was the timing of auto-ignition of single injection case at 160° intake air temperature, the highest value of the heat release rate was detected. It showed only one high peak of the heat release rate of hot combustion. This was because the combustion efficiency was improved by optimizing the second injection timing.

From the implications of Fig. 10 and the tests, optimized second injection timings regarding the intake air temperature were selected as listed in Table 4.

### 4.5 Effect of injector geometry

The injector geometry including the injection angle and the number of holes was tested to reduce wall-wetting by the over-penetration and to improve combustion efficiency. The hole-type injector designed for DI-diesel engine (5 holes, injection angle=150°) showed long radial spray penetration in HCCI engine mode as shown in Fig. 11. It shows the radial penetrations of fuel sprays relative to the position of cylinder liner when the given injector of 150° injection angle was used. The spray of axial-direction (Pz) didn’t contact with the top of the piston if the injection timing was controlled. But the spray of radial-direction (Pr) had severe problems. The spray contacted the cylinder liner if the injection angle was 150° under the condition of low charge air density (1.23kg/m³). It hindered the complete reduction of PM. The injector with small injection angle (100°) was designed to reduce this over-penetration. 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.
Fig. 11 Effect of injection angle on radial spray penetration and wall impingement

Fig. 12 Effect of injector geometry on IMEP and exhaust emissions; 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
Figure 12 shows the combined effect of injection angle and number of holes on IMEP and exhaust gas concentration. The two-stage injection method was applied for these results. The IMEP of 5 hole injector was improved over 200 kPa as the injection angle was decreased that was the result of the reduced wall-wetting. The opacity was also reduced while the low NOx was maintained. The multi-hole injector improved more the combustion efficiency. As a result, IMEP was increased, and the opacity was improved to minimum 7%. It was because the atomization was promoted by the small nozzle diameter that affected the premixing of the homogeneous charge. The HC and CO concentrations were also reduced compared with 5 holes, 150° injection angle case.

Two-stage injection method using this optimized injector geometry showed its advantages for high power output and low exhaust gases. Now, IMEP of the extremely advanced injection timing such as BTDC 200°CA showed no big difference with more retarded injection timings that was not like single injection case, while it had the lowest opacity and NOx advantages.

4.6 DME HCCI operation

DME (CH3OCH3) is an alternative fuel of the diesel. The main advantages of DME compared with diesel are a similar order of cetane number, low weighting and extremely low PM emissions on account of the high oxygen content (34.8%). For the HCCI operation, DME is easily evaporated even in the intake stroke injection so that the preheating is not necessary.

These advantages were tested on the same engine as shown in Fig. 13. The HCCI operations using DME were compared with the single-injection operation using diesel. Now, easily evaporating DME showed no power loss due to the advantage of no wall-wetting. IMEP showed the highest value when the injection timing was most advanced that fulfilled sufficient time for achieving homogeneous charge, and its value was almost the same with DI-diesel’s. The opacity and NOx concentrations showed almost zero value that was the real advantage of the HCCI combustion. Still, the HC and CO emissions were higher than DI-diesel’s because of the low combustion temperature and hard oxidation condition. Concerning the injection pressure of DME, the higher injection pressure increased IMEP because of the improvement of the atomization.

Fig. 13 Effect of an alternative fuel DME on IMEP and exhaust emissions; single injection, total fuel quantity=11.5 ㎣, intake air temperature=30°C, compression ratio=18.9
Conclusions

HCCI combustion with two-stage injection strategy is investigated to achieve the high combustion efficiency while the exhaust advantages are maintained. 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 opacity reduction.
- The second injection timing has optimized value to promote ignition with the intake air temperature variations.
- The small injection angle and multi-hole injector could reduce the opacity while the power output is increased.
- DME could solve the hard evaporation problem of diesel so that the PM and NOx are simultaneously reduced without power loss.

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