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Study on Characteristics of Combined PCCI and Conventional Diesel Combustion

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ABSTRACT: The main objective of this study is to evaluate the characteristics of combustion that combine premixed charge compression ignition (PCCI)–based combustion with conventional mixing controlled combustion. In this type of combustion, it is supposed that the combustion duration is shortened due to the synchronization of the timing of two types of combustions. In addition, the cooling loss caused by spray impingement is expected to decrease by the reduction of the proportion of mixing controlled combustion. In this study, the effect of injection pressure, injection timing, and split injection on thermal efficiency and emissions were investigated in order to determine the appropriate injection parameters for PCCI-based combustion to realize the proposed combustion concept.

KEY WORDS: PCCI, Cooling loss, Multi-injection, Spray combustion, Combustion characteristics [A1]

1. Introduction

To date, much effort has been directed towards the simultaneous improvement of thermal efficiency and the reduction of the emissions for internal combustion engine. In terms of the reduction of emissions, several combustion concepts such as premixed charge compression ignition (PCCI) [1-5], homogeneous charge compression ignition (HCCI) [6, 7], and low temperature combustion (LTC) technique [8, 9] have been proposed to address the trade-off between NOx and particulate matter (PM) emissions in diesel engine. In the PCCI engine, a lean and homogenous air-fuel mixture is ignited by compression. NOx can be reduced by decreasing the combustion temperature and soot formation can be avoided if the equivalence ratio is below approximately 2 when combustion occurs. However, the PCCI combustion technique is restricted to low load conditions. As such, dual mode combustion concept combined PCCI combustion at low load with conventional diesel combustion at high load has been proposed [10].

Traditionally, early direct injection has been applied to realize a homogeneous mixture but this technique may cause high unburnt emissions. When too early injection is applied with the conventional spray included angle, the spray targets out of the piston cavity. Hence, the mixture distributes around the squish area and crevice. As a result, the fuel oxidization will be slower due to the low temperature around the walls and cause high unburnt emissions [11, 12]. In order to avoid this problem, the piston bowl shape and nozzle spray angle were optimized to improve the PCCI technique [12,13]. Besides, the narrow angle direct injection (NADI) combustion concept has been developed [14]. In this technique, the combustion chamber was deliberately designed to optimize the spray formation to facilitate both early injection and conventional near top dead center (TDC) injection. The narrow injection angle (lower than 100 degrees) allows the fuel to be injected early in the compression stroke when the density and temperature are relatively lower, without causing cylinder liner wetting. In addition, it can operate at full load in conventional combustion for a higher load. This dual mode operation achieved low NOx and soot emissions at the maximum brake specific power of more than 60 kW/l with a high efficiency and acceptable unburnt emissions [15].

Another solution for producing a homogeneous mixture while avoiding wall wetting is called premixed lean diesel combustion (PREDIC) system. This system consists of two injectors located at two sides of the combustion chamber. Thus, the spray impingement helps to form an air-fuel mixture at the center of the combustion chamber, resulting in a decrease in cylinder wall wetting with a long ignition delay period [1]. As a result, low NOx and smoke emissions were simultaneously achieved. However, this technique can only be applied to partial load condition. Thus, multiple stage diesel combustion (MULDIC) was developed [16]. The MULDIC system adopted a multiple stage injection strategy, which introduced fuel spray from three injectors: two located at the side of the chamber and one center injector. NOx reduction was achieved even under high load conditions although the thermal efficiency was lower than that of conventional operation. Furthermore, by utilizing low cetane fuel (cetane number 19) for the 1st stage injection, MULDIC improved the thermal efficiency to some extent. Nevertheless, the thermal efficiency was less than that of conventional operation because the center injection timing was considerably later than that of conventional operation.

Our study aims to establish a combustion strategy to improve the thermal efficiency of diesel engines by both cooling loss reduction and an increase in the degree of constant volume (DCV) under high load conditions. PCCI-based combustion is achieved by early direct injection (sub-injection) and features a short combustion duration that leads to high DCV. As such, the cooling loss is possibly lower than that of conventional spray combustion because of the lean mixture at the combustion phase. Furthermore, the ignition timing of the PCCI-based combustion by early direct injection was intended to be synchronized with the ignition timing of the injection for conventional mixing controlled combustion (main injection). For this purpose, a series of experiments were conducted using a single cylinder diesel engine with a dual injector system, which enabled independent control of the fuel injections for PCCI-based and mixing-controlled combustions. In order to examine the basic characteristics of this combustion strategy, fuel injection pressure, number of injections, injection quantity, injection timing, and fuel property of the PCCI-based combustion were investigated. The results showed that when diesel fuel was used for sub-injection, the sub-injection with early timing provided lean mixtures leading to PCCI combustion; however, the ignition timing of sub-injection spray was too early and the smoke emissions were too high since part of the fuel adhered to the piston bowl. To address this problem, the influence of the fuel property, including the self-ignitabiliy and volatility, was examined.

2. Combustion concept

The schematic of the combustion concept in which PCCI combustion and conventional diesel combustion are combined is shown in Figure 1. The injection at an early stage is called sub-injection, and the conventional injection near TDC is called the main injection. The features of this combustion concept are as follows.

・By combining the PCCI-based combustion and conventional diesel combustion, the cooling loss caused by main spray impingement could be reduced.

・The overall combustion duration can be shortened by synchronizing two types of combustion.

・The fuel used for PCCI combustion is distributed avoiding the center of combustion chamber in order not to prevent the oxygen entrainment into the main injection.

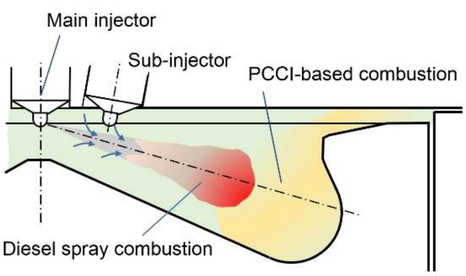

Fig. 1 Schematic of combustion concept.

From the perspective of cooling loss, when the spray flame reaches the piston wall, the heat transfer coefficient increases [17, 18]. Therefore, we introduced early injection PCCI combustion to combine this process with conventional diesel combustion to reduce the amount of fuel impingement while maintaining the engine output. In addition, since the ignition delay of the sub-injection mixture is significantly long and the mixture becomes sufficiently lean, its low momentum and low combustion temperature could further reduce the amount of cooling loss compared to conventional diesel combustion.

Normally, the combustion duration of a premixed mixture is shorter compared to that of the diesel spray combustion process [6, 7]. In addition, the conventional diesel combustion duration is reduced by decreasing the quantity of the main injection fuel. If we could adjust two combustion phases to allow for the overlapping, as shown in Figure 2, the overall combustion duration could be shortened. The shortening of the combustion phase is beneficial to the improvement of the DCV. Therefore, the thermal efficiency may increase. The key to synchronizing the combustion phase is to ensure that PCCI-based combustion occurs close to TDC. However, the unburnt emissions can reach a high level for PCCI combustion. Therefore, in this study, the main idea is to form a mixture by sub-injection thereby avoiding the center of the combustion chamber in order to preclude the entrainment of the sub-injection mixture into the main injection. The schematic of the fuel distribution of this concept is shown in Figure 1.

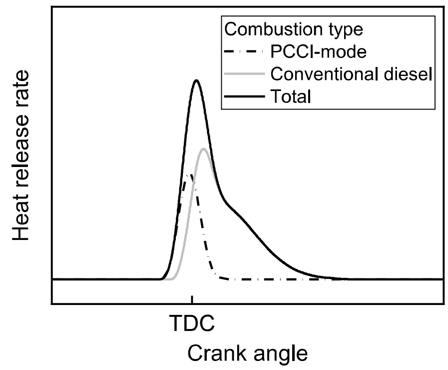

Fig. 2 The ideal heat release rate shape for the combination of PCCI-based combustion and conventional combustion.

Although there are several advantages associated with the combination of PCCI combustion and diesel combustion, the early injection spray will certainly impinge on the cylinder wall, which causes oil dilution, using an injector with a normal injection angle. Therefore, modification of the injection system is required to realize the combustion concept. We developed a dual injection system with two sets of injection system, which injection parameters could be adjusted separately. One injector with a narrow included angle (sub-injector) is used for early direct injection to realize PCCI-based combustion. The other injector with a conventional included angle (main injector) is used for the main injection near TDC. Since the symmetric disposition of the injectors caused high soot emission in the previous study [19], the main injector is set at the center of the combustion chamber and the sub-injector is set next to it. The detailed parameters for the injectors will be introduced in the experimental setup section.

3. Experimental setup

A single cylinder diesel engine with a dual-injection system; namely, a high-pressure main injection system and an early narrow-angle injection system was designed to realize the proposed combustion concept. The main injection and sub-injection parameters can be changed separately. The schematic of experimental system is shown in Figure 3. The base internal combustion engine was a four-stroke diesel engine with a water cooling system. Its bore and stroke were 85 mm and 96.9 mm respectively, and the engine displacement was 0.55 L. In order to ensure sufficient ignition delay of PCCI-based combustion, the compression ratio was set to a rather low level of 15.5. The piston shape was a step lip type.

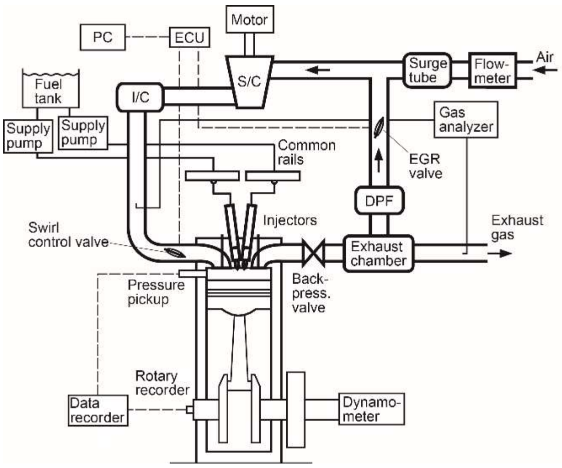

Fig. 3 Schematic of experimental setup.

A piezo injector with a common rail system was utilized and the main injector was located at the center of the combustion chamber while the sub-injector has 15 mm offset from the center. The two injectors had the same nozzle diameter of 0.104 mm but different injection angle and the number of nozzle hole. The sub-injection angle was set to 50 ° based on the results of preceding study using computational fluid dynamics (CFD) simulation. This adequate injection angle allowed the premixed fuel to distribute at the periphery of the combustion chamber while avoiding oil dilution with early injection (approximately −60 °ATDC). The main injection angle was set to 156 ° and the nozzle number was 10. However, the nozzle number of sub-injector was set to 5 due to the manufacturing limitation. The injection direction and its relation with combustion chamber are illustrated on Figure 4. The fuel pump was driven by external driving electric motor.

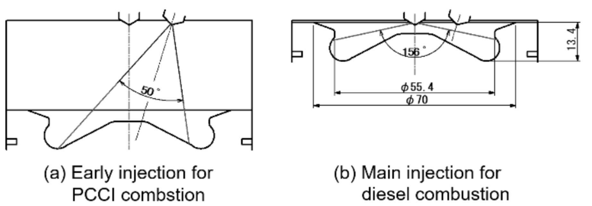

Fig. 4 The injection direction of sub-injection and the main injection and their relation to the combustion chamber.

The maximum efficiency point of the original engine was 2250 rpm with gross indicated mean effective pressure (IMEP) of 1440 kPa (fuel rate 48 mm^3/cycle). Therefore, in this study, we examined the possibility of improving thermal efficiency by implementing the combustion concept at this point. Intake and exhaust pressure were set to 180 kPa and the intake temperature was 50 ℃. The coolant temperature and lubricating oil temperatures were kept at 80 ℃. It was possible to change the swirl ratio of the engine in the range of 1.3–3.2 by adjusting a swirl control valve. Given that a gentle swirl ratio is beneficial to the improvement of the thermal efficiency, the ratio was set at 1.3. In addition, the intake oxygen concentration was modified by changing the exhaust gas recirculation (EGR) rate during the experiments. For most of the experiments, the intake oxygen concentration was set to 18% because a specific amount of EGR is required to extend the ignition delay of PCCI combustion.

Table 1 The characteristics of fuels used during the experiments: diesel, low cetane fuel, and n-heptane.

	JIS #2	Low cetane fuel	n-heptane
Cetane number	55 (cetane index)	40	53 [20]
Lower heating value [MJ/kg]	42.9 ~ 43.3	43.3	44.6
Density at 15 °C [kg/m³]	837.4 ~ 844.2	821.0	688.3
C [mass%]	85.8 ~ 86.7	85.7	84.0
H [mass%]	13.6 ~ 13.8	14.0	16.0
N [mass%]	< 0.1	< 0.1	< 0.1
10% distilling temperature [K]	484.15	486.65	-
50% distilling temperature [K]	557.65	516.15	-
90% distilling temperature [K]	616.65	578.15	-
Boiling point [K]	-	-	371.55

The fuel characteristics of diesel and low cetane fuel are listed in Table 1. The JIS (Japanese Industrial Standards) #2 diesel fuel with cetane index 55 (CN55) was used for the main injection, while several types of fuel were used for sub-injection. As described in the next chapter, the proposed combustion concept was not achieved since the ignition delay was too short and serious fuel adhesion occurred when diesel fuel was used for sub injection. Therefore, the diesel fuel with cetane number 40 (CN40) and the fuel with low boiling temperature (n-heptane) were also tested as sub fuel. In addition, 200 vol.ppm fatty acid ethyl ester fuel type lubricity improver (Infineum R655) was added to the fuel to protect the fuel pressure pump when using n-heptane. Given that the mixing ratio of the lubricity improver was infinitesimal, its influence on combustion was neglected.

An engine exhaust gas analyzer (Horiba MEXA 1600DEGR) was used to measure the NOx, THC, CO, CO_2, and O_2 concentrations. In addition, the smoke emissions were acquired using a filter-type smoke meter (AVL 415S). In this study, we set the upper limit of the smoke and CO emissions to be 2.0 filter smoke number (FSN) and 5000 ppm, respectively. The heat release rates were calculated from the in-cylinder pressure history (50-cycle average) recorded by a pressure sensor (Kistler 6052C).

4. Results and discussion

4.1. Characteristics of combustion of sub-injection

4.1.1. Effect of sub-injection pressure

Initially, in order to evaluate the characteristics of PCCI combustion for early injection, the effect of sub-injection on engine performance and emissions was investigated without the main injection. The injection mass was set to 15 mm^3/cycle, which is approximately 1/3 of the expected total injection mass. The injection timing was –45 °ATDC, and the injection pressure (p_j) values were 90, 120, 150 MPa. The intake oxygen concentration (rO_2) was modified from 16, 17, to 18%. Normal JIS#2 diesel fuel (CN55) was used for this series of experiment and the experimental parameters are summarized in Table 2.

Table 2 The parameters of the experiments to investigate the effect of sub-injection pressure.

	Sub-injection	Main injection
Injection quantity [mm^3/cycle]	15	-
Injection pressure [MPa]	90, 120, 150	-
Injection timing [°ATDC]	–45	-
Fuel type	CN55	
O_2 concentration [%]	16, 17, 18	

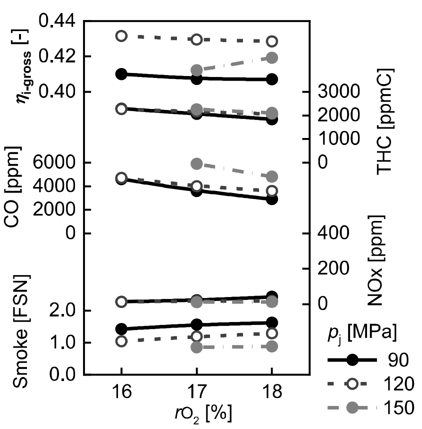

Fig. 5 The effect of oxygen concentration and sub-injection pressure on engine performance and emissions

The performance and emission characteristics are shown in Figure 5. The NOx emission was lower than 50 ppm for all operating conditions. However, the CO and THC emissions levels were high, especially for the conditions of low intake oxygen concentration. A higher injection pressure results in a leaner mixture, therefore, the CO and THC emissions showed an increase with the increase of the injection pressure. The injection pressure 150 MPa, intake oxygen concentration 16% experiment data was missing since its CO emission was expected to exceeded 5000 ppm according to the trend. The smoke emission was 0.8 FSN and 1.5 FSN for injection pressures of 150 MPa and 90 MPa, respectively. For PCCI combustion, these results are common but the high smoke emission is not typical.

In order to evaluate the combustion characteristics of the sub-injection fuel, the effect of injection pressure and intake oxygen concentration on the in-cylinder pressure, heat release rate, and average in-cylinder temperature are shown in Figure 6. For all the injection conditions, ignition occurred after the end of the injection process. After a slight amount of heat release due to a low-temperature oxidization reaction, a rapid heat release due to a high-temperature oxidization reaction followed. Although this means that the combustion was similar to PCCI combustion which ignites after the end of injection, the low heat release of the diffusion type combustion lasted until 20 °ATDC indicating that a fuel-rich region remains in the combustion chamber. This combustion mode and the wall wetting due to the injection at the early stage is considered to be the cause of high smoke emission. Furthermore, high-pressure injection and low oxygen concentration can extend the ignition delay of high temperature reaction and retard the timing of the heat release rate peak. However, overall, the ignition before TDC could not be avoided. On the other hand, the injection pressure and intake oxygen concentration seemed to have little effect on heat release of low temperature reaction.

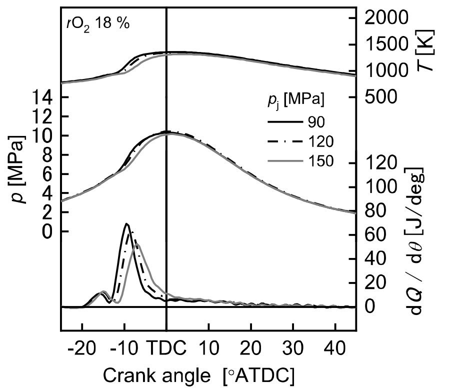

Fig. 6a Heat release rate, in-cylinder pressure, and average in-cylinder temperature as function of crank angle when sub-injection pressure was modified.

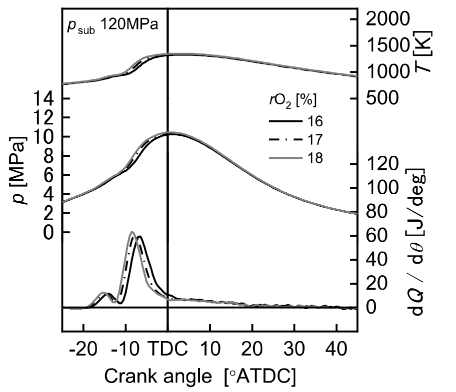

Fig. 6b Heat release rate, in-cylinder pressure, and average in-cylinder temperature as function of crank angle when intake oxygen concentration was modified.

In conclusion, smoke emission and premature ignition are two vital problems to be solved in order to improve thermal efficiency. Increasing the EGR rate is usually adopted to address the premature ignition problem. However, in this case, the CO and THC emissions were too high such that further reducing oxygen concentration was impossible. Given that the compression ratio should not be decreased from the perspective of thermal efficiency, the fuel type and sub-injection strategy were modified in the next series of experiments to address these problems.

4.1.2. Effect of sub-injection splitting

When injection at an early stage was adopted, the low in-cylinder density may cause the spray impingement on piston bowl. The fuel adhered onto the piston bowl may remain unevaporated and assist in the formation of a fuel-rich mixture during the combustion process. Picket et al. indicated that splitting injection can suppress the liquid phase length of diesel spray [21]. Therefore, the effect of splitting the sub-injection was investigated with injection pressures ranging from 90 to 150 MPa. Moreover, single injection, two-stage injection (7.5 mm^3/cycle x 2) and three-stage injection (5 mm^3/cycle x 3) strategies were adopted for sub-injection. The injection parameters are shown in Table 3. The injection timing of the first-stage injection was set at –45 °ATDC while the injection interval was reduced to a minimum to ensure a sufficient duration for fuel-air mixing. As a result, the interval of start of injection (SOI) between the two-stage injection was 9 °CA while that of the three-stage injection was 8 °CA. According to the previous section, the intake oxygen concentration should be further reduced to achieve the new combustion concept. However, considering the EGR rate limitation for actual multi-cylinder engine operation at the proposed condition and CO emission, the intake oxygen concentration was set to 18%.

Table 3 The parameters of the experiments for the investigation of the effect of sub-injection splitting.

	Sub-injection	Main injection
Injection quantity [mm^3/cycle]	15; 7.5 + 7.5; 5 + 5 + 5	-
Injection pressure [MPa]	90, 120, 150	-
Injection timing [°ATDC]	–45	-
Fuel type	CN55	
O_2 concentration [%]	18	

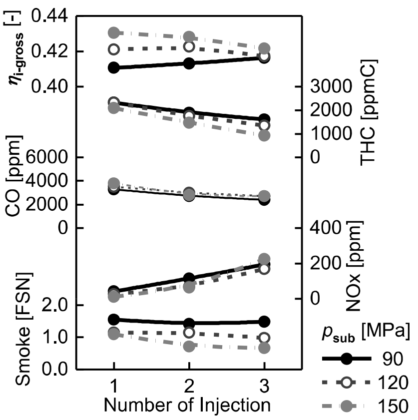

Fig. 7 The effect of split sub-injection and sub-injection pressure on eingine performance and emissions

The emission results are presented in Figure 7. According to these results, it could be deduced that a reduction in the fuel quantity that reached the piston bowl surface and cylinder wall was achieved by splitting sub-injection, because the smoke emission was lower compared to single sub-injection. Although unburnt emissions were largely produced by sub-injection, their re-oxidization could be expected due to the introduction of high-pressure main injection. In comparison, the NOx emission showed an increase by splitting sub-injection. As shown in Figure 8, the splitting injection advanced the ignition timing and extended the total injection duration. A longer injection duration means shorter fuel-air mixing duration. Thus, the fuel concentration uniformity was deteriorated in combustion chamber due to the splitting sub-injection. This could explain the advanced ignition timing for split sub-injection give that a greater portion of the mixture was still near stoichiometric ratio state and therefore, the NOx emission showed an increase.

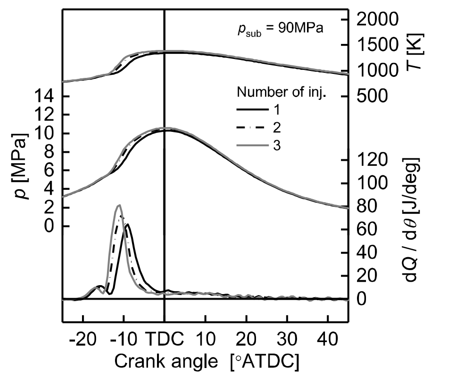

Fig. 8a Heat release rate, in-cylinder pressure, and average in-cylinder temperature as a function of crank angle when split sub-injection was applied for an injection pressure of 90 MPa.

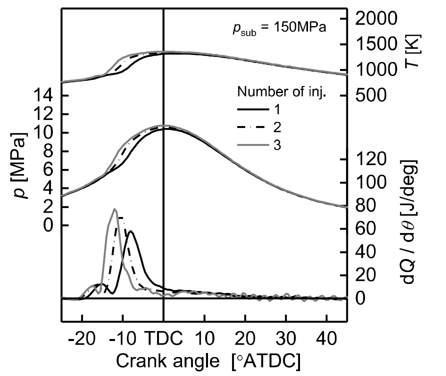

Fig. 8b Heat release rate, in-cylinder pressure, and average in-cylinder temperature as a function of crank angle when split sub-injection was applied for an injection pressure of 150 MPa.

4.2. Characteristics of combination of PCCI and diesel combustion

In this section, high-pressure main injection was introduced to investigate the combustion characteristics when sub-injection was combined with the main injection.

4.2.1. Effect of sub-injection quantity and timing

Table 4 The parameters of the experiments for the investigation of the effect of sub-injection quantity and timing.

	Sub-injection	Main injection
Injection quantity [mm^3/cycle]	7.5 + 7.5; 5 + 5	25
Injection pressure [MPa]	150	200
Injection timing [°ATDC]	–80, –70, –60, –50	–1.8
Fuel type	CN55	
O_2 concentration [%]	18	

Initially, the effect of sub-injection quantity and timing on engine performance and emissions were investigated for the introduction of main injection based on previous experimental setup. According to the soot and NOx emission results acquired from prior experiments, the two-stage sub-injection strategy was selected to combine it with main injection. The interval of SOI between the two sub-injection was fixed to 9 °CA. The main injection timing was set to −1.8 °ATDC and the injection quantity was 25 mm^3/cycle. For sub-injection, the total injection quantity was 15 mm^3/cycle, which was evenly divided. The sub-injection pressure was 150 MPa while the main-injection pressure was increased to 200 MPa. In this series of experiments, the sub-injection timing (θ_{1st}) was changed from −50 to −80 °ATDC. However, due to high smoke emission, further advancing of the sub-injection timing beyond −70 °ATDC was interrupted. To reduce smoke emission, we also investigated the injection setup with reduced sub-injection quantity (from 15 mm^3/cycle to 10 mm^3/cycle) while maintaining the main-injection quantity unchanged (25 mm^3/cycle). The experimental parameters are summarized in Table 4.

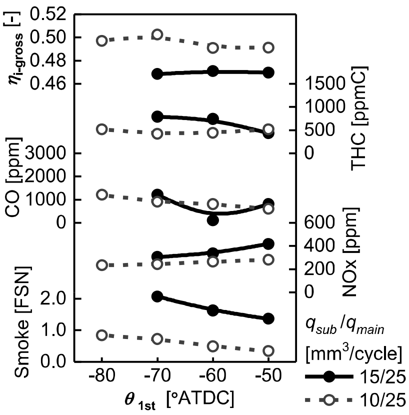

Fig. 9 The effect of the quantity and injection timing of sub-injection on engine performance and emissions

The engine performance and emission results are shown in Figure 9. By introducing the main injection, CO and THC emissions decreased dramatically, which could be attributed to the oxidization of unburnt species formed by sub-injection due to the in-cylinder temperature rise associated with the combustion of main injection spray. However, when the sub-injection quantity was set to 15 mm^3/cycle, the smoke emission result was over 1.5 FSN. This is a critical problem when combining sub-injection with main injection. When the injection timing of sub-injection is advanced, the spray liquid length is expected to increase because the in-cylinder temperature and density at the injection timing are lower. Therefore, the quantity of the fuel that reaches the piston bowl surface will increase and as a result, the smoke emission showed an increase with the advance of sub-injection timing.

As previously indicated, the total sub-injection quantity was reduced to 10 mm^3/cycle (5 mm^3 each for two-stage injection) to suppress the cylinder impingement quantity. The experimental results are summarized in Figure 9. As a result, the smoke emission was suppressed as intended because the fuel impingement quantity was considered to be reduced. In addition, the thermal efficiency was higher for sub-injection with 10 mm^3/cycle compared to 15 mm^3/cycle. However, the heat release rate shown in Figure 10 indicates that the ignition timing of sub-injection was the same for different sub-injection quantities. The ignition at −15 °ATDC was still too early to realize the goal of shortening the combustion duration. In order to increase the thermal efficiency, the ignition timing of the sub-injection should be retarded until approximately TDC.

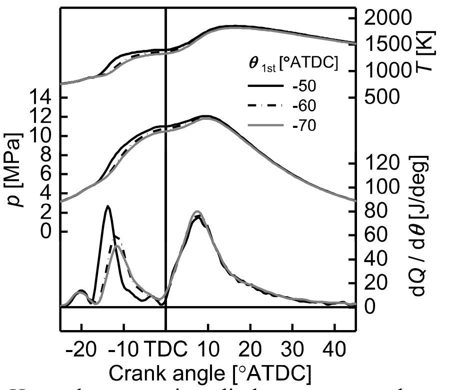

Fig. 10a Heat release rate, in-cylinder pressure, and average in-cylinder temperature as a function of crank angle when sub-injection timing was modified. (sub-injection quantity 15 mm^3/cycle)

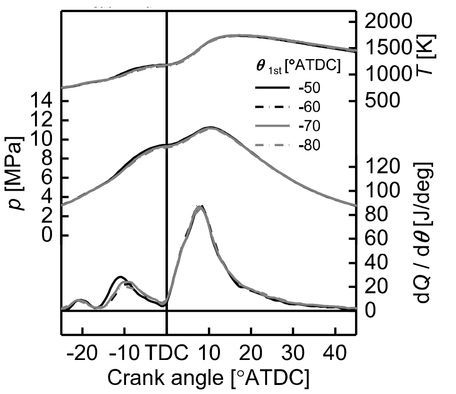

Fig. 10b Heat release rate, in-cylinder pressure, and average in-cylinder temperature as a function of crank angle when sub-injection timing and quantity were modified. (sub-injection quantity 10 mm^3/cycle)

Furthermore, no matter the injection quantity of sub-injection, the sub-injection timing had little impact on thermal efficiency. Although, the heat release peak was retarded when the sub-injection timing was advanced from −50 °ATDC to −60 °ATDC, the heat release shape showed little change when the injection timing was further advanced. This phenomenon indicates that although the local differences in the state of the mixture may exist due to the difference in the quantity of fuel impingement, overall, a homogenous and lean mixture was successfully formed in the combustion chamber.

4.2.2. Effect of self-ignitability of sub-injection fuel

As discussed in the last section, the ignition timing of the sub-injection must be retarded until approximately TDC in order to improve the thermal efficiency. Ickes et al. investigated a wide range of fuel with cetane numbers of 42, 47, 50 to 53 for PCCI combustion experiments under low load [22]. Their study suggests that the fuel cetane number has a great impact on ignition delay. The result shows that by reducing the cetane number from 53 to 42, although the low temperature oxidization timing is only retarded for 2 ℃A, the ignition delay retarded for 7 ℃A. In addition, Ickes et al. proposed that the cetane number does not have a direct effect on emissions, unlike the combustion phase and EGR rate.

Therefore, the fuel with reduced cetane number, CN40 (cetane number 40), was selected as the sub-injection fuel while the main injection fuel (CN55) was unchanged. The total sub-injection quantity was determined to be 10 mm^3/cycle and the two-stage injection (5 mm^3/cycle + 5 mm^3/cycle) was selected for sub-injection to control the emission level based on the previous section. The injection interval of SOI between two-stage sub-injection was still 9 ℃A. The injection pressure was 150 MPa and 200 MPa for sub-injection and main injection respectively. The injection timing of the sub-injection was advanced from −50 °ATDC to −80 °ATDC in steps of 10 ℃A to investigate the effect of cetane number on extension of ignition delay as well as the improvement of the homogeneity of the fuel-air mixture. The experimental parameters are summarized in Table 5.

Table 5 The parameters of the experiments for the investigation of the effect of self-ignitability of sub-injection fuel.

	Sub-injection	Main injection
Injection quantity [mm^3/cycle]	5 + 5	25
Injection pressure [MPa]	150	200
Injection timing [°ATDC]	−80, −70, −60, −50	−1.8
Fuel type	CN40	CN55
O_2 concentration [%]	18	

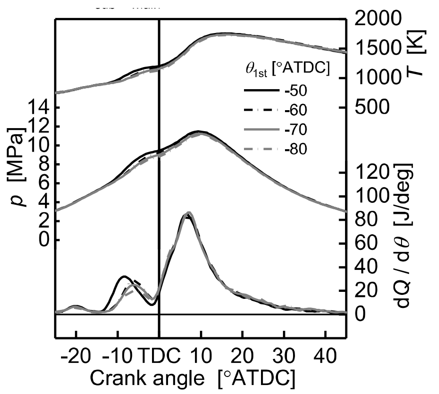

Fig. 11 Heat release rate, in-cylinder pressure, average in-cylinder temperature, and pressure rise rate as a function of crank angle when low cetane fuel was adopted for sub-injection.

The experimental results shown in Figure 11 indicate that the low temperature oxidization reaction occurred from approximately −23 °ATDC and the ignition timing was approximately −15 °ATDC for sub-injection at −50 °ATDC and approximately −12 °ATDC for the other conditions. When compared with the results of sub-injection with CN55 (Figure 10b), this series of studies proved that the reduction of the cetane number of the sub-injection fuel could retard its ignition delay duration, thus, the heat release peak timing of the PCCI combustion retarded by approximately 3 ℃A. However, the ignition timing of the sub-injection was still premature. The sub-injection fuel property also affected the ignition timing of the main injection.

4.2.3. Effect of sub-injection quantity with CN40

Figure 9 shows that when the sub-injection quantity is increased, the smoke and unburnt emissions increases and the thermal efficiency decreases. To determine the reasons for these trends, the effect of sub-injection quantity on engine performance and emissions were investigated. Single sub-injection with an injection pressure of 150 MPa was selected and the injection timing was set to −64 °ATDC. The sub-injection quantity was reduced from 10 mm^3/cycle to 4 mm^3/cycle in steps of 2 mm^3/cycle while the total injection quantity was maintained. The main injection pressure was 270 MPa and this parameter was modified slightly to adjust the total gross IMEP to 1440 kPa. The aforementioned parameters mentioned above are summarized in Table 6. The heat release rate, in-cylinder pressure, and average in-cylinder temperature as a function of crank angle are illustrated in Figure 12 and the characteristics of the engine performance and emissions are presented in Figure 13.

Table 6 The parameters of the experiments for the investigation of the effect of self-ignitability of the sub-injection fuel.

	Sub-injection	Main injection
Injection quantity [mm^3/cycle]	10	38
	8	40
	6	42
	4	44
Injection pressure [MPa]	150	270
Injection timing [°ATDC]	−64	−1.8
Fuel type	CN40	CN55
O_2 concentration [%]	18	

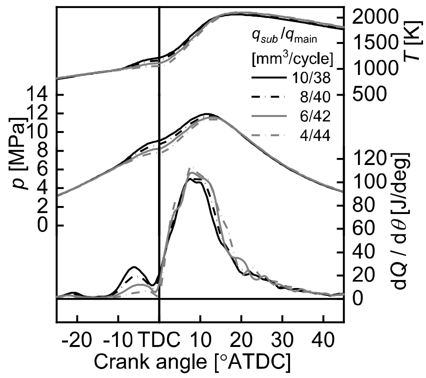

Fig. 12 Heat release rate, in-cylinder pressure, and average in-cylinder temperature as a function of crank angle when low cetane fuel was adopted for sub-injection and its quantity was reduced from 10 to 4 mm^3/cycle while the total gross IMEP was maintained at 1440 kPa.

The heat release rate graph indicates that the main part of PCCI-based combustion of sub-injection fuel started at approximately −12 °ATDC and the main injection fuel ignited at approximately TDC. With the decrease in the sub-injection quantity, heat release of sub-injection was reduced and the in-cylinder temperature became lower. That is thought to be the cause of the longer ignition delay of the main injection, as shown in Figure 12. In addition, the larger main injection quantity induced a sharper increase in the heat release rate as well as a higher peak and longer duration of the heat release.

The results for engine emissions show that the smoke emission and other unburnt emissions were reduced by decreasing the proportion of sub-injection quantity. This may be due to the reduced spray impingement quantity and liquid surface combustion when the sub-injection quantity was small. Previously, the reduction of the main injection was suggested to minimize the cooling loss since the procedure could suppress spray flame impingement. However, the increase of the main injection quantity due to the reduction of sub-injection quantity did not reduce the thermal efficiency expected according to Figure 13.

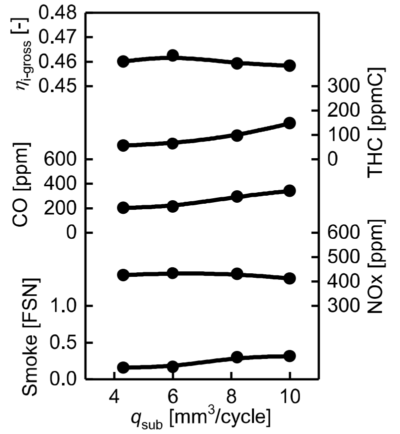

Fig. 13 The effect of the sub-injection quantity on gross thermal efficiency, CO, THC, NOx, and smoke emissions when low cetane fuel was adopted for sub-injection and the total gross IMEP was maintained at 1440 kPa.

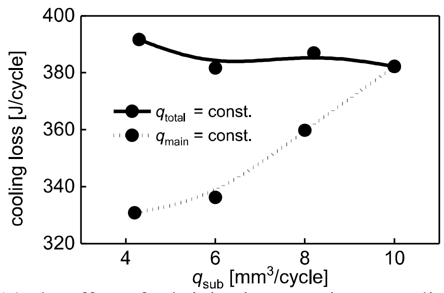

Fig. 14 The effect of sub-injection quantity on cooling loss amount for two experimental conditions: maintaining the main injection quantity and maintaining total gross IMEP.

The cooling loss as a function of the sub-injection quantity is given in Figure 14. This parameter is calculated by subtracting friction loss, unburnt loss, indicated work, and exhaust loss from the input energy. The exhaust loss was calculated using the exhaust temperature and the exhaust flow rate. In the case where the total injection quantity was fixed and the sub-injection quantity was increased by reducing the main injection quantity, the cooling loss was substantially constant. As such, the goal of this study: reduction of the cooling loss by reducing the injection quantity of diesel combustion and allocating it to PCCI-base combustion, had not been achieved.

The reasons for the failure of cooling loss reduction were investigated by reducing the sub-injection quantity while maintaining the main injection quantity at 38 mm^3/cycle. The results are also shown in Figure 14. It is evident that, the cooling loss gradually decreased with a reduction of the sub-injection quantity. Therefore, the reasons why the cooling loss could not be reduced is presumed to be attributed to the adherence of the early injected fuel on the surface of the piston bowl, forming a partial fuel-rich region in the piston bowl and the presence of high-temperature combustion gas near the piston bowl. This is probably because the in-cylinder gas density and temperature are low and therefore, the injected fuel impinges on the piston bowl while it is still in the liquid phase when the sub-injection occurs from −80 to −50 °ATDC.

4.2.4 Effect of low boiling point fuel for sub-injection

Table 7 The parameters of the experiments for the investigation of the effect of low boiling point fuel for sub-injection.

	Sub-injection	Main injection
Injection quantity [mm^3/cycle]	12	29
Injection pressure [MPa]	150	200
Injection timing [°ATDC]	−80, −70, −60, −50	−1.8
Fuel type	n-heptane	CN55
O_2 concentration [%]	19	

The n-heptane was used as sub-injection fuel as a means of suppressing fuel adhesion. The injection parameters are summarized in Table 7. Given that the possibility of using fuel with low boiling point to reduce unburnt emissions from sub-injection was the primary focus, total input energy was set to be the same compared to 39 mm^3/cycle diesel fuel. Therefore, the sub-injection quantity was set to 12 mm^3/cycle which corresponded to 10 mm^3/cycle of diesel fuel. The injection pressure was maintained at 150 MPa and 200 MPa for sub-injection and main injection respectively, as the section "Effect of sub-injection quantity and timing". Single sub-injection was utilized and the main injection timing was fixed at −1.8 °ATDC while the sub-injection timing was adjusted from −50 to −80 °ATDC. The results of emissions and thermal efficiency are shown in Figure 15 and the results of combustion analysis are represented in Figure 16.

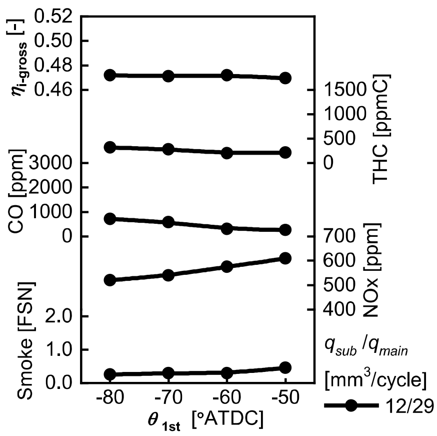

Fig. 15 The effect of the sub-injection timing on gross thermal efficiency, cooling loss ratio, CO, THC, NOx, and smoke emissions when n-heptane was adopted for sub-injection.

The Figure 15 shows that smoke emission level dropped dramatically compared to previous sections (Figure 9 qsub/qmain:

10/25) although the total injection quantity was larger. The THC and CO emissions are also relatively lower than previous section. In addition, according to Figure 16, the maximum heat release timing of PCCI combustion retarded slightly with the advancing of sub-injection timing. This indicates the fuel-air mixture formation was affected by injection timing since the fuel evaporated in the combustion chamber. However, when compared to Figure 10b, the high temperature reaction timing of sub-injection was advanced slightly. Therefore, further efforts, such as adapting low ignitability fuel for sub-injection, should be invested to ensure a longer ignition delay for sub-injection so that its combustion phase could overlap with the main injection. As a result, thermal efficiency may be improved.

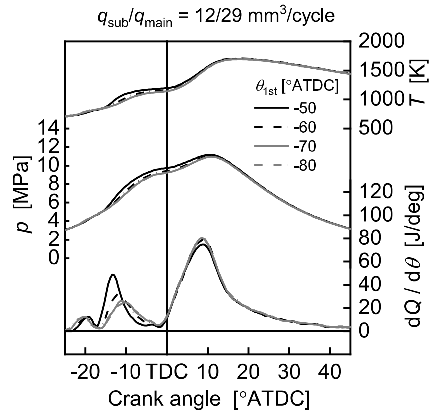

Fig. 16 Heat release rate, in-cylinder pressure, and average in-cylinder temperature as a function of crank angle when n-heptane was adopted for sub-injection and its injection timing was modified.

5. Conclusion

In this report, a combustion concept was proposed to improve thermal efficiency while maintaining low emissions. A dual injection system was introduced to a single cylinder engine to combine PCCI-based combustion with a conventional diesel combustion strategy. A homogeneous mixture was formed by early timing injection (sub-injection) thus the quantity of main injection near TDC could be reduced in order to suppress the spray-flame impingement on the piston wall. The effect of sub-injection pressure, timing, split injection, and main injection quantity were examined with JIS#2 diesel fuel. Then, the effect of the property of sub-injection fuel was also investigated. The main conclusions are as shown below.

1. Without main injection near TDC, the ignition timing could be retarded by reducing intake O_2 concentration but the unburnt emissions may increase as a result. Splitting sub-injection could help to reduce unburnt emissions but will shorten the ignition delay.
2. The early ignition of the sub-injection mixture is a critical problem in the realization of the combination of PCCI-based combustion and diesel spray combustion. By reducing the cetane number of the sub-injection fuel from 55 (cetane index) to 40, the heat release peak retarded for 3 °CA.
3. When combined with conventional direct injection near TDC, the ignition timing could be retarded by advancing sub-injection beyond −60 °ATDC. However, this could deteriorate the unburnt emissions.
4. If the total injection amount was kept constant, modifying the proportion of sub-injection and main injection quantity did not lead to any changes in cooling loss amount.
5. To control the cooling loss and improve the thermal efficiency, the impingement and adherence of the sub-injection fuel should be avoided.
6. By reducing wall wetting, for example, using n-heptane as sub-injection fuel, the smoke emission could be reduced dramatically and the thermal efficiency could be increased due to the reduction of cooling loss ratio.

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References

(1) Takeda, Y., Keiichi, N., and Keiichi, N., "Emission Characteristics of Premixed Lean Diesel Combustion with Extremely Early Staged Fuel Injection," SAE Technical Paper 961163, 1996, doi: 10.4271/961163.
(2) Nakagome, K., Shimazaki, N., Niimura, K., and Kobayashi, S., "Combustion and Emission Characteristics of Premixed Lean Diesel Combustion Engine," SAE Technical Paper 970898, 1997, doi: 10.4271/970898.
(3) Suzuki, H., Koike, N., and Odaka, M., "Combustion Control Method of Homogeneous Charge Diesel Engines," SAE Technical Paper 980509, 1998, doi: 10.4271/980509.
(4) Yanagihara, H. "A simultaneous reduction of NOx and soot in diesel engines under a new combustion system (Uniform Bulky Combustion System-UNIBUS)." 17th, Int. Viena Motor Symposium, 1996.
(5) Okude, K., Mori, K., Shiino, S., and Moriya, T., "Premixed Compression Ignition (PCI) Combustion for Simultaneous Reduction of NOx and Soot in Diesel Engine," SAE Technical Paper 2004-01-1907, 2004, doi: 10.4271/2004-01-1907.
(6) Dec, J., "A Computational Study of the Effects of Low Fuel Loading and EGR on Heat Release Rates and Combustion Limits in HCCI Engines," SAE Technical Paper 2002-01-1309, 2002, doi: 10.4271/2002-01-1309.
(7) Eng, J., "Characterization of Pressure Waves in HCCI Combustion," SAE Technical Paper 2002-01-2859, 2002, doi: 10.4271/2002-01-2859.
(8) Akihama, K., Takatori, Y., Inagaki, K., Sasaki, S. et al., "Mechanism of the Smokeless Rich Diesel Combustion by Reducing Temperature," SAE Technical Paper 2001-01-0655, 2001
(9) Opat, R., Ra, Y., Gonzalez D., M., Krieger, R. et al., "Investigation of Mixing and Temperature Effects on HC/CO Emissions for Highly Dilute Low Temperature Combustion in a Light Duty Diesel Engine," SAE Technical Paper 2007-01-0193, 2007, doi: 10.4271/2007-01-0193.

(10) Shimazaki, N., Tsurushima, T., and Nishimura, T., "Dual Mode Combustion Concept With Premixed Diesel Combustion by Direct Injection Near Top Dead Center," SAE Technical Paper 2003-01-0742, 2003, https://doi.org/10.4271/2003-01-0742.
(11) Stanglmaier, R., Li, J., and Matthews, R., "The Effect of In-Cylinder Wall Wetting Location on the HC Emissions from SI Engines," SAE Technical Paper 1999-01-0502, 1999, doi: 10.4271/1999-01-0502.
(12) Boyarski, N. and Reitz, R., "Premixed Compression Ignition (PCI) Combustion with Modeling-Generated Piston Bowl Geometry in a Diesel Engine," SAE Technical Paper 2006-01-0198, 2006, doi: 10.4271/2006-01-0198.
(13) Vanegas, A., Won, H., and Peters, N., "Influence of the Nozzle Spray Angle on Pollutant Formation and Combustion Efficiency for a PCCI Diesel Engine," SAE Technical Paper 2009-01-1445, 2009, doi: 10.4271/2009-01-1445.
(14) Walter, B. and Gatellier, B., "Development of the High Power NADI™ Concept Using Dual Mode Diesel Combustion to Achieve Zero NOx and Particulate Emissions," SAE Technical Paper 2002-01-1744, 2002, doi: 10.4271/2002-01-1744.
(15) Duret, P., Gatellier, B., Monteiro, L., Miche, M. et al., "Progress in Diesel HCCI Combustion Within the European SPACE LIGHT Project," SAE Technical Paper 2004-01-1904, 2004, doi: 10.4271/2004-01-1904.
(16) Hashizume, T., Miyamoto, T., Hisashi, A., and Tsujimura, K., "Combustion and Emission Characteristics of Multiple Stage Diesel Combustion," SAE Technical Paper 980505, 1998, doi: 10.4271/980505.
(17) Borman, Gary, and Kazuie Nishiwaki. "Internal-combustion engine heat transfer." Progress in energy and combustion science, 1987, 13.1: 1-46.
(18) Jankowski, Antoni. "Heat transfer in combustion chamber of piston engines." Journal of KONES, 2010, 17: 187-197.
(19) Horibe, Naoto, et al. "Effects of Fuel-injection Parameters on Performance and Exhaust Emissions in a Diesel Engine Equipped with Dual-injector System." The Proceedings of the International symposium on diagnostics and modeling of combustion in internal combustion engines 2017.9. The Japan Society of Mechanical Engineers, 2017.
(20) M.J.Murphy, J.D.Taylor, R.L.McCormick, Compendium of Experimental Cetane Number Data, NREL/SR-540-36805, (2004)
(21) Pickett, Lyle M., et al. "Transient Liquid Penetration of Early-Injection Diesel Sprays." SAE International Journal of Engines, vol. 2, no. 1, 2009, pp. 785–804.
(22) Ickes, Andrew M., Stanislav V. Bohac, and Dennis N. Assanis. "Effect of fuel cetane number on a premixed diesel combustion mode." International Journal of Engine Research, vol. 10, no. 4, Aug. 2009, pp. 251-263, doi: 10.1243/14680874JER03809.