**Research Report**

# Combustion System with Premixture-controlled Compression Ignition

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## Abstract

The purpose of this study is to create a new combustion concept that offers a high thermal efficiency and very low NOx and soot emissions. To this end, we performed 3D-CFD simulations to identify problems with an actual PCCI that is characterized by in-cylinder mixture non-homogeneity that arises through the direct injection of diesel fuel. We compared the combustion characteristics with an ideal 'HCCI' with homogeneous mixture conditions. Then, to overcome PCCI problems such as difficulties in combustion controllability and the limited operating range, we identified the key parameters impacting the HCCI/PCCI process through experiments with a variety of paraffinic hydrocarbon fuels. Finally, based on the knowledge gained through these steps, we developed a new concept for dual-fuel PCCI combustion using high- and low-RON fuels to achieve extremely low NOx and smoke emissions. In this system, gasoline was supplied from the intake air port and diesel fuel was injected directly into the engine cylinder to act as an ignition trigger at a timing before TDC. It was found that the ignition phasing of this PCCI combustion can be controlled by changing the ratios of the two injected fuels, such that combustion proceeds very mildly, even without EGR, thanks to the spatial stratification of ignitability in the cylinder, which prevents the entire mixture from igniting instantaneously. The operable load range, where the NOx and smoke emissions were less than 10 ppm and 0.1 FSN, respectively, was extended up to an IMEP of 12 bar using an intake air boosting system together with dual fueling.

## 1. Introduction

Against a background of worldwide environmental issues, diesel engine vehicles have been gaining greater market penetration, especially in Europe and Asia, because of their good fuel economy. Unfortunately, diesel engines emit more soot and nitrogen oxide (NOx) than gasoline engines, such that reducing these emissions while maintaining a high level of efficiency presents an urgent and major challenge.

Homogeneous Charge Compression Ignition (HCCI) and/or Premixture-Controlled Compression Ignition (PCCI) have become the focus of engine research around the world because of the advantages they offer, including their high thermal efficiency and very low NOx and soot emissions.[1-6] As HCCI/PCCI involves the premixed combustion of a highly diluted mixture, the combustion process is characterized predominantly by chemical kinetics. Thus, the control of the ignition timing and the burning rate is more difficult in an HCCI/PCCI system than in a conventional diesel engine, which is governed mainly by physical processes such as the fuel injection rate and the fuel-air mixing. Consequently, a key point related to the commercialization of HCCI/PCCI is the development of a method for controlling the combustion process.

To this end, we conducted the current study in three steps. Firstly, 3D-CFD simulations were performed to identify the problems related to an actual PCCI that features in-cylinder mixture non-homogeneity that arises as a result of the direct injection of diesel fuel. This involved comparing the combustion characteristics with an ideal HCCI with homogeneous mixture conditions. Secondly, to overcome PCCI problems such as the control of the combustion and the limited operating range, the key parameters affecting the HCCI/PCCI process were identified through experiments using a variety of paraffinic hydro-carbon fuels. Finally, based on the knowledge acquired by the preceding steps, we developed a new dual-fuel controlled PCCI engine that offers a high thermal efficiency and very low NOx and soot emissions. This was achieved by utilizing the spatial temperature stratification obtained through the in-cylinder non-homogeneity

of the Research Octane Number (RON).

## 2. Combustion characteristics and problems with actual PCCI

### 2. 1 Differences in mixture formation and combustion between actual PCCI and ideal HCCI

First, the meanings of the terms 'HCCI' and 'PCCI' are defined, as follows. In this paper, Premixed Compression Ignition (PCI) that is realized by the port injection of a high-volatility fuel (like gasoline), is referred to as 'Homogeneous Charge Compression Ignition (HCCI)' because a homogeneous mixture is formed in the subsequent intake and compression strokes. On the other hand, PCI that is realized by the direct injection of a low-volatility fuel like diesel is referred to as 'Premixture-Controlled Compression Ignition (PCCI)' because there may be a spatial distribution of the fuel concentration in the cylinder, such that the term 'Homogeneous' is not applicable.

This section has two main aims. The first is to clarify the concept of HCCI/PCCI by comparing it with other combustion types, specifically, those that have already been developed. This section also aims to identify the problems associated with PCCI, and the in-cylinder mixture non-homogeneity formed through the direct injection of diesel fuel, by comparing its combustion characteristics with those of HCCI.

**Figure 1** shows the '$\phi$-T Map'[7] and illustrates the in-cylinder mixture conditions for a range of combustion types. In typical conventional diesel
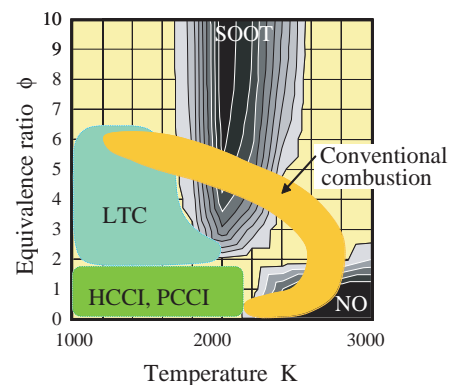
**Fig. 1** $\phi$-T map used to compare combustion characteristics for the various combustion types of diesel engine.

combustion, this involves a non-homogeneous spray-combustion process, with in-cylinder gas conditions that are widely distributed on the '$\phi$-T Map' and which cross into the soot and NOx formation areas. Therefore, considerable amounts of soot and NOx are generated during combustion.

During Low Temperature Combustion (LTC), as recently developed by Toyota Motor Corp.,[8] the in-cylinder gas conditions are shifted into the zone on the low-temperature side of the soot formation zone by relying heavily on Exhaust Gas Recirculation (EGR). Thus, the amounts of both NOx and soot are reduced drastically.

In HCCI/PCCI, as targeted by this study, the local $\phi$ is less than 0.5 and the temperature is well below the NOx formation zone, thanks to the formation of in-cylinder homogenous conditions. In this way, ultra-low levels of NOx and soot are achieved. Another attractive characteristic is that the operating range is wider than that of LTC. Unfortunately, ignition control is more difficult because the combustion process depends predominantly on

chemical kinetics and the in-cylinder homogenous conditions lead to larger combustion noise due to the rapid combustion caused by simultaneous ignition throughout the in-cylinder space.

Next, we will discuss how homogeneous in-cylinder mixtures are realized in a practical diesel engine with a direct injection of diesel fuel. A 3D-CFD simulation with KIVA2 code was conducted to predict the in-cylinder conditions. Some of the sub-models shown in **Table 1** were implemented in the original KIVA2 code to realize a more accurate prediction of the spray combustion phenomena. The engine is a typical diesel, with a common rail injection system and a narrow cone angle injector, and a compression ratio of 16. **Table 2** shows the calculation conditions in three cases. The injection timing for the PCCI case is set to -40° ATDC, so as to prevent lubricant oil dilution caused by fuel impingement on the cylinder liner. Cases 1 and 2 were intended to investigate the effects of EGR on the PCCI combustion processes, and Case 3 provides a reference to clarify the differences in the combustion characteristics between PCCI and HCCI.

**Figure 2** shows the heat release rates for the three cases. The ignition timing is advanced more for PCCI without EGR (Case 1) than for HCCI (Case 3). The combustion rate in PCCI without EGR is higher than that in HCCI. A comparison of Cases 1 and 2 shows that EGR is a very effective means of suppressing premature ignition and an excessively high combustion rate.

**Table 1** Sub-models used in modified KIVA2.

| Drop breakup and atomization | Surface-wave-growth[9] |
|---|---|
| Ignition | Shell multi-step model[10] |
| Combustion | Laminar-turbulence characteristic time model[11] |
| Turbulence | RNG k-$\varepsilon$ [11] |
| NOx | Extended Zeldovich mechanism |

**Table 2** Calculation conditions.

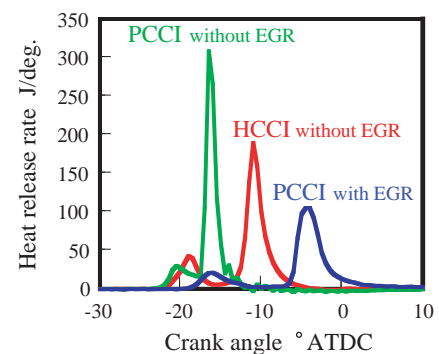| Case | 1. PCCI without EGR | 2. PCCI with EGR | 3. HCCI without EGR |
|---|---|---|---|
| Bore × Stroke | 84 mm × 100 mm | 84 mm × 100 mm | 84 mm × 100 mm |
| Engine speed | 1200 rpm | 1200 rpm | 1200 rpm |
| Injector specification Diameter × Number | $\phi$ 0.15 mm × 7 | $\phi$ 0.15 mm × 7 | Fuel vapor set to be distributed homogenously |
| Injection timing | -40° ATDC | -40° ATDC | - |
| Fuel quantity | 16.8 mg | 16.8 mg | 16.8 mg |
| EGR ratio | 0 % | 65 % | 0 % |
| A/F | 52 | 18 | 52 |



**Fig. 2** Heat release rates simulated by 3D-CFD for comparison of PCCI and HCCI.

**Figure 3** shows the calculated in-cylinder gas conditions at the end of each type of combustion, and these are overlaid on the '$\phi$-T Map'. The mixture conditions for PCCI without EGR (Case 1) are extended to cross the NOx formation area on the '$\phi$-T Map' because near-stoichiometric conditions are partially produced in the cylinder. On the other hand, PCCI with EGR (Case 2) avoids the NOx formation area because the gas temperatures are reduced as a result of the slower combustion rate. Another factor behind the reduced temperatures is that the mixture is more diluted due to there being a longer mixing time before ignition because the ignition timing is significantly delayed with EGR.

With HCCI, on the other hand (Case 3), the gas temperature is reduced sufficiently only as a result of complete dilution of the mixture, and zero NOx is realized. PCCI with a low-volatility fuel such as diesel cannot realize HCCI, such that EGR must always be employed to achieve low levels of NOx and a mild combustion rate. However, the use of EGR increases HC due to the lower $O_2$ concentration. Consequently, PCCI needs EGR for both low emissions and for suppressing premature ignition and excessively high combustion rates. For this reason, much more EGR gas is needed with PCCI than with HCCI.
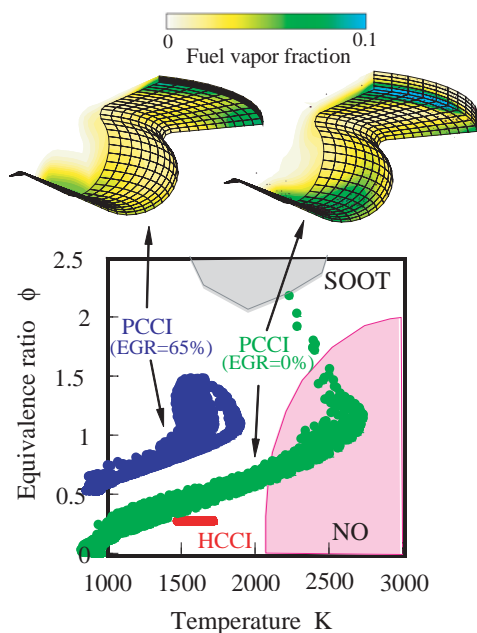
As shown above, EGR provides an effective means of controlling the HCCI/PCCI processes. Unfortunately, it incurs the following disadvantages:
(1) The thermal efficiency is reduced because the EGR gas contains tri-atomic molecules, $CO_2$ and $H_2O$, with a corresponding lower ratio of specific heats, $\kappa = C_p / C_v$.
(2) The maximum achievable engine power is low relative to the mass flow through the engine, because a higher EGR fraction is needed at higher loads.
(3) External EGR generally makes HCCI/PCCI combustion unstable during transient operation because of the delay incurred in delivering EGR to the cylinder. To solve this problem, a more sophisticated control system is necessary, such as a model-based feed-forward or feed-back control and a cylinder pressure or lambda sensor.

Given that our goal is a PCCI that is well controlled even without EGR, the following sections discuss a novel PCCI concept.

### 2. 2 Comprehensive parametric study of HCCI

As a first step in creating a new EGR-less PCCI concept, it is vital to clarify the factors that have the greatest influence on the HCCI/PCCI process. Fuel ignitability, as given by the Research Octane Number (RON), is also a key factor in controlling the combustion process. To answer these substantial questions, a comprehensive parametric study of HCCI was performed with a single-cylinder, four-stroke diesel engine with a range of fuels. **Figure 4** shows a schematic diagram of the experimental
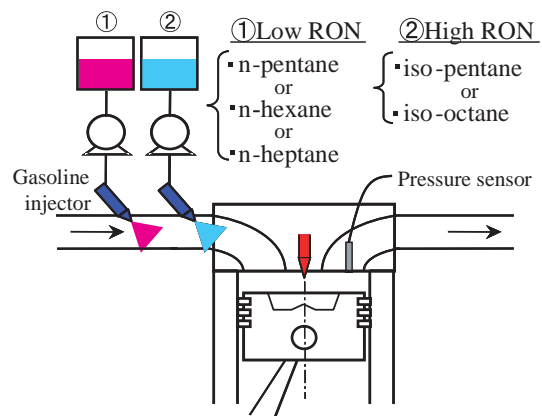


**Fig. 3**  $\phi$-T map with simulated in-cylinder gas condition plots to compare PCCI and HCCI.



**Fig. 4**  Single cylinder diesel engine for basic HCCI research with varied paraffinic hydrocarbon fuels.

setup. The engine specifications are listed in **Table 3**, and the properties of the fuels evaluated in this study are listed **Table 4**. They are all pure paraffinic hydrocarbon fuels with a different RON. Two injectors, which are commonplace in a commercial gasoline engine, were mounted in the intake port, such that a homogeneous mixture of fuel and air could be formed during the intake and compression strokes so as to realize HCCI. HCCI was used for this parametric study to eliminate the uncertainties of local in-cylinder gas mixture conditions such as the temperature and equivalence ratio, which are desirable for a more accurate understanding of the compression ignition phenomena.

The cylinder pressure was recorded with Kistler 6125B pressure transducers installed in the cylinder head. Two hundred cycles of pressure data were averaged for each condition and used to calculate the Indicated Mean Effective Pressure (IMEP), the indicated thermal efficiency and the apparent rate of heat release. Emissions of NOx, CO and THC from the engine were analyzed with a HORIBA MEXA-7100D analyzer, and the amount of smoke was measured with an AVL 415 smoke meter. The

combustion noise was measured with an AVL 450 Noise Meter.

**Figure 5** shows the apparent rate of heat release from experiments using n-pentane (n-C5), n-hexane(n-C6), and n-heptane(n-C7) as the fuel. Two-stage heat release with Low-Temperature Oxidation (LTO) and High-Temperature Oxidation (HTO) was observed for each fuel. An enlargement of the LTO range is also shown in Fig. 5. As the RON of the fuel decreases, the LTO start timing is advanced and the rate of heat release in LTO becomes larger. In addition, the HTO start timing, which is regarded as the ignition timing, is also advanced.

**Figure 6** shows the rate of heat release vs. the in-cylinder temperature obtained from the indicator analysis. It clarifies the relationship between the in-

**Table 3** Engine specifications.

| Engine type | Single cylinder, 4 cycle |
|---|---|
| Bore × Stroke | 94 mm × 100 mm |
| Displacement | 694 cc |
| Compression ratio | 14 |
| Combustion chamber shape | Shallow-dish type |
| Swirl ratio | 2.3 |
| Fueling system (intake port) | Gasoline EFI system |

**Table 4** Tested paraffinic hydrocarbon fuels.

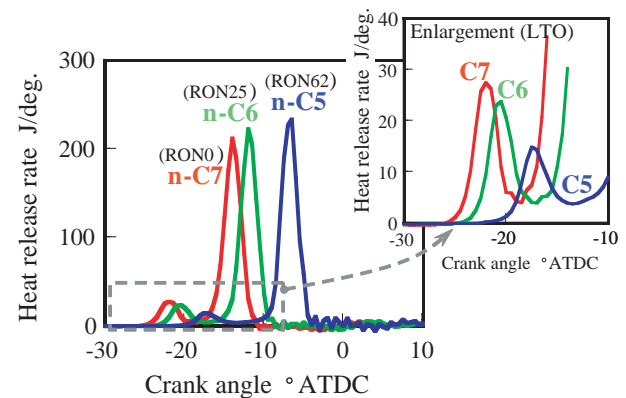| Fuel | Abbrevi-ation | RON (CN) | Boiling temp. |
|---|---|---|---|
| n-pentane | n-C5 | 62 (29) | 37 ℃ |
| n-hexane | n-C6 | 25 (48) | 68 ℃ |
| n-heptane | n-C7 | 0 (56) | 98 ℃ |
| iso-pentane | iso-C5 | 99 (11) | 31 ℃ |
| iso-octane | iso-C8 | 100 (10) | 99 ℃ |



**Fig. 5** Heat release rate of n-pentane, n-hexane and n-pentane ($\phi = 0.35$ without EGR, 1400 rpm).
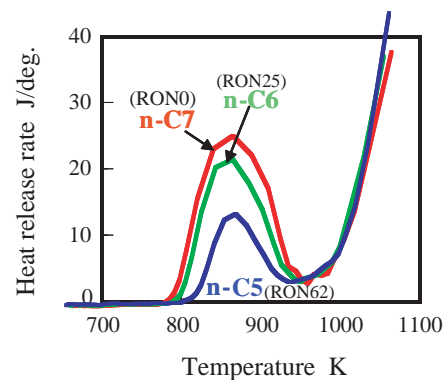


**Fig. 6** Ignition characteristics related to in-cylinder temperature for n-pentane, n-hexane and n-pentane ($\phi = 0.35$ without EGR, 1400 rpm).

cylinder temperature and ignition. As the RON falls, the LTO start timing shifts towards a lower temperature. On the other hand, the curves indicating a rapid increase in the rate of heat release during HTO are almost the same for the three fuels, regardless of their RON.

To enable a more quantitative comparison of the ignition phenomena among different RON fuels, we defined the start timings of LTO and HTO, as follows: The start timing of LTO is defined as that at which the rate of heat release, $dQ/d\theta$, exceeds 0.5 [J/deg.], as shown in **Fig. 7**. Similarly, the start timing of HTO is defined as that at which the derivative of the heat release rate, $d^2Q/d^2\theta$, exceeds 2.0 [J/deg./deg.]. **Figure 8** shows the start temperatures of LTO and HTO for different equivalence ratios, $\phi$. As the RON is lowered, the LTO start temperature decreases. On the other hand, the HTO start temperature barely changes for each of the three fuels and corresponds to approximately 980 K.
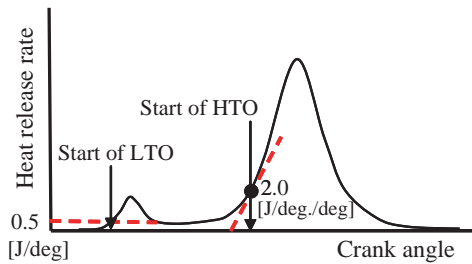
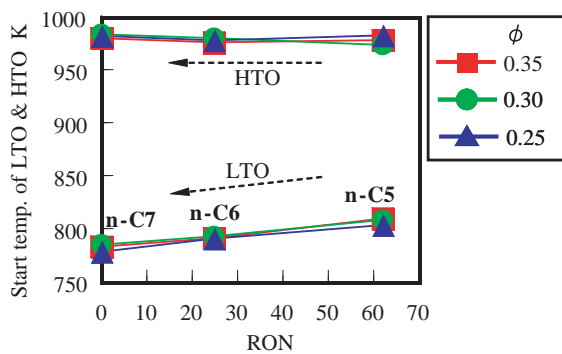**Figure 9** shows the LTO and HTO start temperatures for n-pentane at several $\phi$ values, EGR ratios, and intake air temperatures. Although the start crank angles of LTO and HTO vary depending on the conditions, the start temperatures of LTO and HTO are almost the same regardless of the conditions. These results are consistent with the phenomena obtained by Iida.[12]

## 2. 3 HCCI controlled by dual fuels with different ignitabilities

The results of the fundamental HCCI experiments discussed in the previous section can be summarized as follows:

(1) The quantity of heat released in LTO, which raises the in-cylinder temperature before HTO starts, decreases as the fuel RON increases.

(2) The ignition temperature, i.e. the HTO start temperature, is almost the same among all the paraffinic hydrocarbon fuels, even if the engine operating conditions such as $\phi$, EGR rate, and intake air temperature are changed.

Based on (1) and (2), the ignition timing can be controlled by changing the ratios of the injected amounts of the two different RON fuels.

**Figure 10** shows the rate of heat release for experiments in which the proportion of n-pentane (RON = 62) and iso-pentane (RON = 99) is changed under the same total $\phi$ of 0.35. It shows that when the ratio of iso-pentane (high RON) to n-pentane (low RON) is increased, the rate of heat release in LTO falls, causing the ignition timing, corresponding to the start timing of HTO, to be
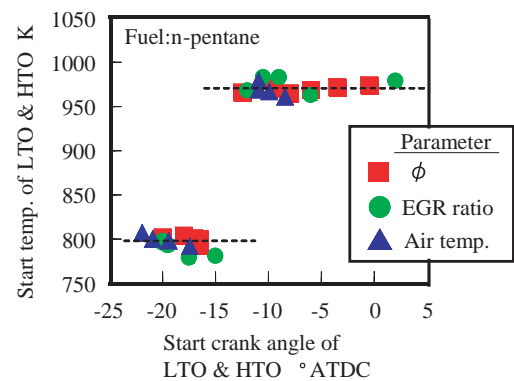


**Fig. 7**   Definitions of start of LTO and HTO.



**Fig. 8**   Temperatures of starting LTO and HTO of n-pentane, n-hexane and n-pentane related to RON with change of $\phi$.



**Fig. 9**   Temperatures of starting LTO and HTO of n-pentane with changes of $\phi$, EGR ratio and intake air temperature.

retarded. Consequently, the ignition timing can be controlled by changing the mixing ratios of the two fuels with the different RON values.

**Figure 11** shows the effect of EGR on the HCCI combustion characteristics. If we compare Fig. 10 with Fig. 11, we can see the differences in the combustion characteristics between the dual fuel strategy and the EGR approach. As the EGR rate is increased, the rate of heat release in LTO falls and the ignition timing is retarded, as shown in Fig. 11. These trends are very similar to those in dual-fuel HCCI regards changing the proportion of high-RON fuel, shown in Fig. 10. However, while EGR can also significantly reduce the rate of heat release in HTO, dual-fuel HCCI has little effect on the rate of heat release in HTO.
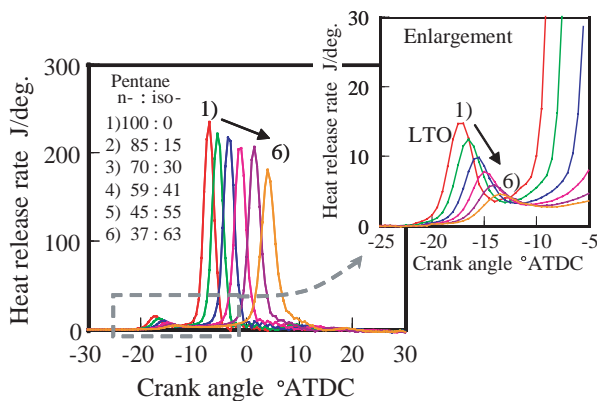
These results show that the control of the combustion rate is a critical issue that must be solved to enable a practical dual-fuel strategy. To overcome this problem, we examined the effect of in-cylinder stratification of the ignitability, i.e. the local RON distribution with a dual-fuel scenario, on the combustion rate. This is described in the following section.

## 3. Dual-fuel stratified PCCI

### 3. 1 Concept of combustion rate control by spatial temperature stratification with dual fuels

Considering (1) in the previous section, the in-cylinder stratification of RON by creating a spatial distribution of a lower RON fuel concentration in a dual-fuel approach is expected to generate a spatial temperature distribution due to the heat release during LTO. According to (2) in the previous section, ignition occurs when the in-cylinder temperature reaches a critical point. Consequently, the temperature non-homogeneity that is formed during LTO will possibly prevent the entire mixture from igniting instantaneously, leading to a slowing of the combustion rate and reducing the combustion noise. In this study, this concept is named 'Dual-fuel stratified PCCI.'

**Figure 12** is a schematic diagram of the experimental setup that we used to realize dual-fuel stratified PCCI. Iso-octane, which is used to

**Fig. 10** Heat release rate with change of blend ratio of n-pentane and iso-octane under the constant total $\phi$ of 0.35.
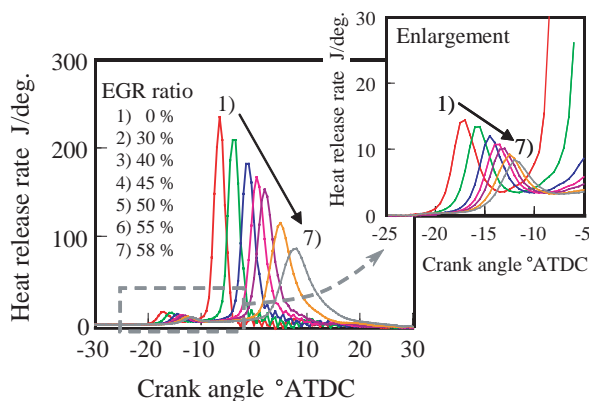
**Fig. 11** Effect of EGR ratio on heat release rate of HCCI with constant amount of injected fuel.
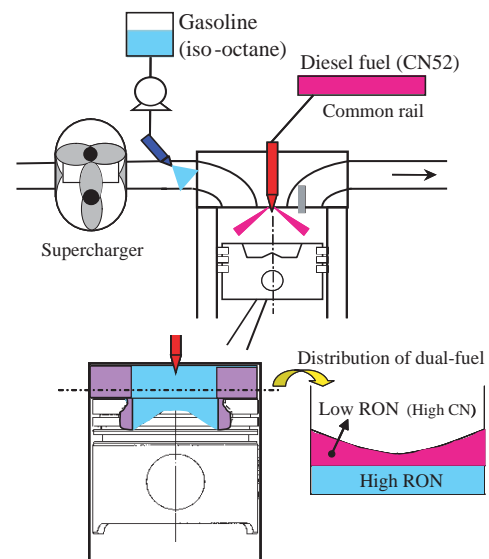
**Fig. 12** Lean-boosted dual-fuel stratified PCCI engine using gasoline and diesel fuel.

represent high-octane gasoline, is injected into the intake port through a gasoline engine injector. This engine is also equipped with a common-rail injection system, which is used to inject diesel fuel directly into the cylinder at an early timing.

The iso-octane is distributed homogenously, and then the diesel fuel migrates towards the cylinder liner while vaporizing. Consequently, diesel fuel with a low RON is concentrated in the outer regions, leading to RON stratification in the cylinder.

To reinforce our concept, 3D-CFD with KIVA2 were performed to predict the spatial distribution of the diesel fuel in a dual-fuel PCCI system. **Figure 13** shows the spatial distribution of the fuel vapor mass fraction for diesel fuel alone. The injection timing of the diesel fuel is -45$^o$ ATDC and the injection quantity is 10 mm$^3$/stroke. We believe that the diesel fuel will vaporize while migrating toward the cylinder liner. Consequently, the diesel fuel vapor is concentrated mainly in the outer regions of the cylinder. The iso-octane fuel, which is injected into the intake port, is expected to disperse almost homogenously throughout the in-cylinder region. As a result, as illustrated in Fig. 12, a moderate spatial distribution of local two-fuel $\phi$ and RON is expected to be generated in the cylinder.

## 3. 2 Engine performance of dual-duel stratified PCCI

To evaluate the potential of dual-fuel stratified PCCI, we conducted an experiment with the single-cylinder diesel engine shown in Fig. 12. The experimental conditions are listed in **Table 5**. The narrow cone angle of 132$^o$ was designed to prevent spray-impingement onto the cylinder liner even with the early injection timing of the diesel fuel. The intake air temperature was regulated to 60±1 $^o$C, regardless of the engine operating load, through the use of an electrical heater inserted into the intake port. An electrically driven supercharger system was installed to boost the intake air pressure. EGR was not applied to this engine.

The profile of the RON distribution can be changed by adjusting the injection timing of the diesel fuel and the ratio of the injected fuel quantity. An intake air boost system was employed to expand operable range to higher loads, while keeping the emissions low.

Firstly, the effect of the in-cylinder RON stratification on the combustion characteristics was examined. **Figure 14** shows a comparison of heat release rate between the dual-fuel HCCI of n-heptane/iso-octane and the dual-fuel stratified PCCI of diesel fuel/iso-octane. The dual-fuel HCCI experiment was conducted using the engine shown in Fig. 4. The quantities of iso-octane and diesel fuel were 13.7 and 9.2 mm$^3$/stroke, respectively, at a total $\phi$ of 0.35. A simple estimation showed the average RON to be 30 in this dual-fuel stratified PCCI
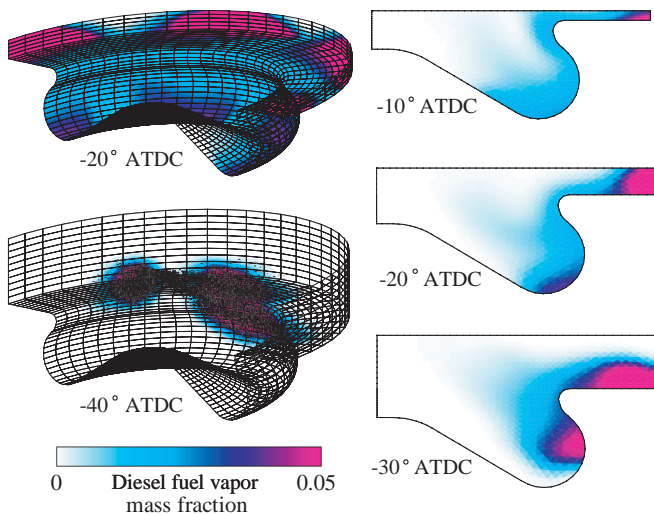


**Fig. 13** In-cylinder spatial distribution of vapor mass fraction of diesel fuel in dual-fuel stratified PCCI concept simulated by KIVA2 code.
(Injection timing of diesel fuel: -45$^o$ ATDC, Quantity of diesel fuel: 10 mm$^3$)

**Table 5** Experimental conditions.

| Common rail type | Fueling system (Direct injection) |
|---|---|
| $\phi$ 0.12 mm × 6 | Hole size × Number |
| 132 $^o$deg. | Cone angle |
| 60 MPa | Injection pressure |
| 1400 rpm | Engine speed |
| 60 ℃ | Intake air temperature |
| 0 ~ 100 kPa (Depending on load) | Boost pressure (Gauge) |

experiment. In the dual-fuel HCCI experiment using n-heptane/iso-octane, the quantity of each fuel was set so as to have the same average RON, 30, at a total $\phi$ of 0.35. As shown in Fig. 14, while the ignition timing advances earlier in the dual-fuel stratified PCCI than the dual-fuel HCCI, the combustion duration of HTO is broader in the dual-fuel stratified PCCI, and the peak heat release rate is reduced significantly. Accordingly, the dual-fuel stratified PCCI achieved a milder combustion rate at total $\phi$ of 0.35 even without EGR. The amount of NOx increases slightly with the dual-fuel stratified PCCI because locally richer mixtures are thought to be formed in the cylinder due to the non-homogenous concentration of diesel fuel. Nevertheless, the NOx level remains quite low, i.e., 9 ppm. Additionally, the dual-fuel stratified PCCI can reduce the combustion noise by 9 dB, resulting in mild combustion. It is found that the dual-fuel stratified PCCI can realize a moderate combustion rate even without EGR, while keeping the NOx and smoke at low levels.

KIVA2 simulations were conducted to examine the effect of in-cylinder RON stratification on the combustion characteristics. Our interests are in the in-cylinder spatial distribution of temperature immediately before the ignition timing or the start timing of HTO for both dual-fuel HCCI and dual-fuel stratified PCCI. The calculation conditions correspond to those shown in Fig. 14. Because a high-RON fuel such as iso-octane has no effect on the ignition process of a low-RON fuel, only the
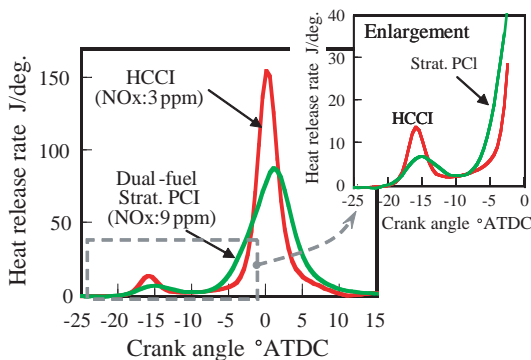
low-RON fuel was considered in the calculations. N-heptane was used as the low-RON fuels for dual-fuel HCCI, while diesel fuel was used for dual-fuel stratified PCCI. The physical properties of n-dodecane were used in the calculations, as the atomization and vaporization figures are regarded as being as representative of the diesel fuel in dual-fuel stratified PCCI. The calculations considered the process up until the end of LTO, which is -10° ATDC. The Shell model was used to predict the ignition process.

**Figure 15** compares the in-cylinder spatial distributions of the temperature and their histograms for both cases at -10° ATDC, immediately before ignition. With dual-fuel HCCI, the temperature distribution is almost uniform throughout the cylinder. With dual-fuel stratified PCCI, however, higher-temperature regions are generated in the squish area and around the cavity sidewalls. This is because, with dual-fuel stratified PCCI, in-cylinder
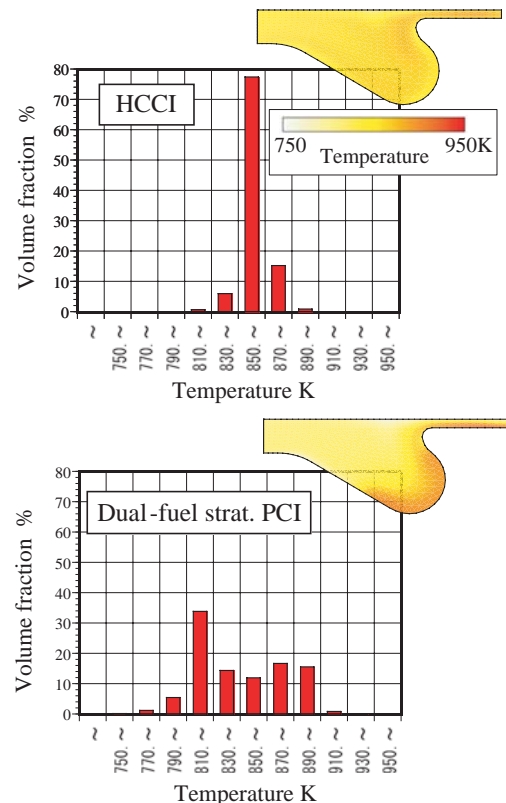
**Fig. 14** Comparison of heat release rate between HCCI and dual-fuel stratified PCCI. (Total $\phi = 0.35$, 1400 rpm)

**Fig. 15** Histograms and spatial distributions of in-cylinder local temperature of dual-fuel HCCI and dual-fuel stratified PCCI at -10° ATDC just before autoignition simulated by KIVA2 code.

stratified low-RON fuel locally releases heat during LTO and a wider range of temperature distributions is created immediately before ignition. Ignition occurs when the local in-cylinder temperature reaches a critical value, as discussed in the previous section. Therefore, this temperature non-homogeneity has the effect of scattering the local ignition timings and moderates the combustion rate.

To summarize, the mild combustion rate in dual-fuel stratified PCCI is thought to be a result of preventing the entire mixture from igniting simultaneously due to the stratification of the ignitability i.e. RON, in the cylinder. On the other hand, the low levels of NOx and smoke were achieved because the $\phi$ stratification was not sufficiently high to generate both high temperatures and rich $\phi$ regions.

As the total $\phi$ is increased, the combustion rate and NOx tend to increase even in dual-fuel stratified PCCI. In particular, when IMEP > 0.6 MPa, the combustion rate is unacceptably high. Therefore, at higher loads, the intake air was boosted using an electrically driven boost system to make the fuel-air mixture leaner.

**Figure 16** shows the engine control parameters and engine performance in dual-fuel stratified PCCI. As mentioned above, the intake air was boosted under IMEP levels greater than 0.6 MPa. When calculating IMEP or the indicated thermal efficiency, $\eta_i$, the external work provided by the externally driven boost system during the suction stroke must be removed. Therefore, pressure data for the power strokes only, i.e. the compression and expansion strokes, were used for the IMEP and $\eta_i$ calculations. The diesel fuel fraction is shown at the top of Fig. 16. At lower loads, a larger diesel fuel fraction is used to achieve stable ignition. At higher loads, a smaller diesel fuel fraction is used to prevent excessively early ignition and rapid burning.

At very low loads at which a relatively large diesel fuel fraction is needed for stable ignition, the NOx level is approximately 50 ppm. Above this region, however, the NOx level is held at the very low level of 10 ppm even at loads as high as IMEP = 1.2 MPa with intake air boosting. The smoke levels are also held at low levels of ~0.1 FSN even under loads up to IMEP = 1.2 MPa.

The HC and CO levels are higher than those of a conventional diesel engine. However, they are thought to be smaller than those of conventional PCCI as they use large amounts of external EGR gas.

The indicated thermal efficiency $\eta_i$ shown in the 4th panel of Fig. 16, is calculated based on pressure data of only power strokes as noted above. $\eta_i$ increases as IMEP increases, and exceeds 50 % in the range of IMEP values greater than 0.9 MPa. Therefore dual-fuel stratified PCCI has a great potential also of fuel economy.

To investigate the reason for the high thermal efficiency in dual-fuel stratified PCCI with a boost system, the in-cylinder pressure and the rate of heat release shown in **Fig. 17** are calculated. As shown
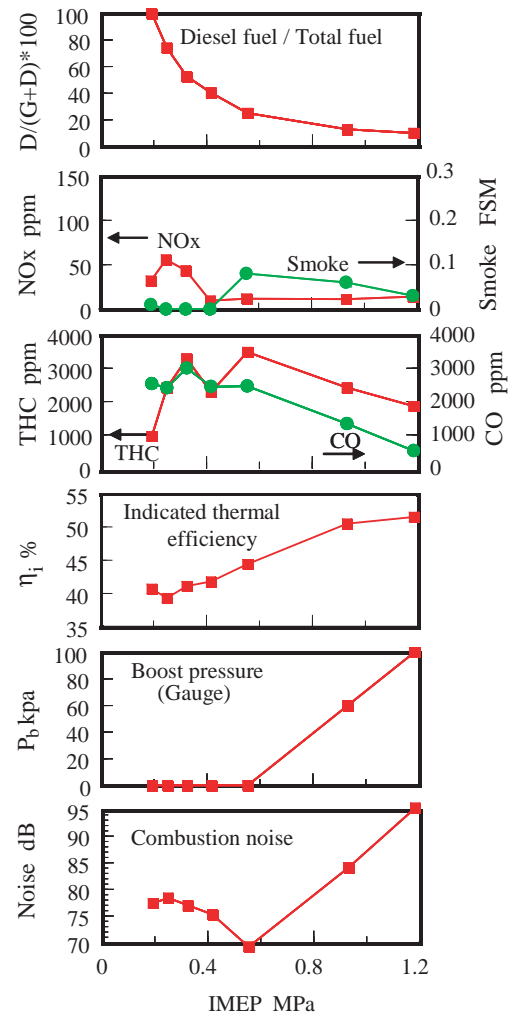


**Fig. 16** Engine parameters and performances of dual-fuel stratified PCCI related to Indicated Mean Effective Pressure (IMEP) at 1400 rpm.

in Fig. 11, the maximum heat release of HCCI is very high when EGR ratio is 0 %. In dual-fuel stratified PCCI, in spite of the high load condition, where $P_i$ is 0.93 MPa at a boost pressure of 63 kPa gauge, the maximum heat release rate is low even without EGR. Such EGR-less combustion, which is controlled by the in-cylinder stratified ignitability, improves the thermal efficiency because of the increase in the specific heat ratio, $\kappa = C_p / C_v$. The intake air boost with a leaner mixture also increases $\kappa$. Because the ignition timing is well controlled by regulating the diesel fuel fraction, the degree of constant volume, $\eta_{glt}$, reaches a high value of 0.98. The combustion temperature falls due to the larger heat capacity with intake air boosting, resulting in reduced heat losses.

The reasons for the high thermal efficiency in lean-boosted dual-fuel PCCI can be summarized as follows:

(1) Increased specific heat ratio, $\kappa$, because of the leaner mixture with intake-air boost and no EGR gas.

(2) Reduced heat loss due to lower combustion temperatures with intake air boost.

(3) High $\eta_{glt}$ combustion caused by the well-controlled ignition timing and the regulated diesel fuel fraction.

As discussed above, a dual-fuel stratified PCCI with a boost system offers great potential in terms of thermal efficiency and exhaust emissions. The operable load range while maintaining very low NOx and smoke levels was extended up to an IMEP of 1.2 MPa using a boost system, even without EGR. Furthermore, EGR-less control can improve the controllability of PCCI combustion especially under transient engine operation.

## 4. Conclusion

Firstly, 3D-CFD simulations were performed to identify problems with PCCI, which features in-cylinder mixture non-homogeneity created through the direct injection of diesel fuel, in comparison with the combustion characteristics of HCCI with homogeneous conditions.

(1) PCCI, unlike HCCI, is not clean in emissions and not low in combustion noise without EGR.

The use of EGR has some disadvantages such as the limited load and difficult controllability during transient operation. Prior to devising a new EGR-less PCCI concept, it is very important to clarify the most influential factors affecting the HCCI/PCCI process. To this end, a comprehensive parametric study of HCCI was performed with a single cylinder, four-stroke diesel engine.

(2) The rate of heat release in LTO, which affects the in-cylinder temperature prior to HTO, varies with the fuel RON.

(3) The ignition temperature, i.e. the HTO-start temperature, is almost the same among all the paraffinic hydro-carbon fuels, even if the engine operating parameters such as $\phi$, EGR rate and intake air temperature are changed.

(4) Dual-fuel HCCI with high-RON and low-RON fuels can control the ignition timing by regulating the dual-fuel ratio, but cannot control the burn rate.

Finally, we developed a concept of dual-fuel PCCI combustion controlled by the in-cylinder stratified ignitability to achieve drastically lower NOx and smoke emissions, as well as a moderate combustion rate.

(5) The ignition timing of dual-fuel stratified PCCI can be controlled by changing the ratios of the two fuels. The combustion proceeds very mildly without EGR by establishing a spatial stratification of ignitability in the cylinder, which prevents the entire mixture from igniting simultaneously.

(6) The operable load range, with NOx and smoke levels of less than 10 ppm and 0.1 FSN, respectively, is extended to a load range of up to 1.2 MPa of IMEP using an intake air boost system together with dual fueling.
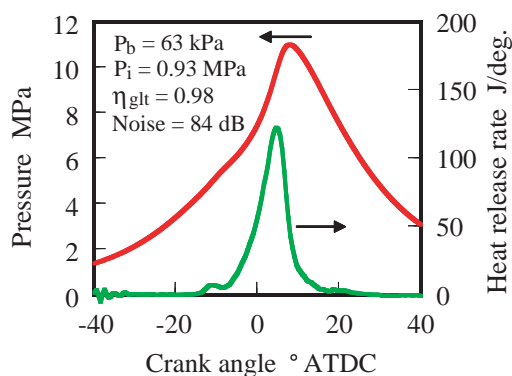


**Fig. 17** Pressure and heat release rate of lean-boosted dual-fuel stratified PCCI. ($P_b$ =63 kPa, $P_i$ =0.93 MPa, without EGR)

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