

# Advanced Ignition Control Technology for HCCI Combustion

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## ABSTRACT

Spark-Assisted HCCI combustion, which assists ignition by igniting direct injection spray with a spark, has been proposed to help enable the practical realization of homogeneous charge compression ignition combustion, thereby achieving high thermal efficiency. This method increases the controllability of ignition, an issue of homogeneous charge compression ignition, and expands the range of ignition combustion by a maximum of three times compared to the conventional method.

The fuel efficiency of Spark-Assisted HCCI was verified, with results indicating a decrease of 16% in steady-state fuel consumption against cooled EGR combustion technology and a JC08 mode fuel efficiency simulation showing an increase of 4.5% in fuel efficiency.

## 1. Introduction

Given today's high level of concern towards global environmental issues, the automotive industry is being called on to develop technologies that will further reduce CO<sub>2</sub> emissions in order to help to mitigate climate change and address the issue of depletion of fossil fuels. Because the realization of increased thermal efficiency can contribute to reducing CO<sub>2</sub> emissions, research towards this end is being continuously conducted in a wide variety of fields by companies and university and other research institutes<sup>(1), (2)</sup>. Homogeneous-Charge Compression Ignition (HCCI) technology makes it possible to stably combust lean air-fuel mixtures through auto-ignition of a homogeneous mixture. This reconciles a high theoretical thermal efficiency with low NO<sub>x</sub> emissions, and HCCI thus presents good prospects as a mode of combustion in future engines<sup>(3)-(6)</sup>.

Figure 1 shows details of Honda's main HCCI initiatives up to the present. In 1999, Honda commenced fundamental research towards the realization of HCCI by increasing the compression temperature via Exhaust Gas Recirculation (EGR) realized by means of Negative

valve Overlap (NOL) in an engine with an electromagnetic variable valve timing system. This research increased efficiency and realized low NO<sub>x</sub> emissions in the low-load range. Using this technology in combination with inter-cylinder EGR boost, which introduces EGR using exhaust blowdown pressure from other cylinders, made it possible to extend the operating range into the medium-load range. Figure 2 shows examples of the operating range for HCCI when using NOL EGR and the expansion in the operating range realized by inter-cylinder EGR boost. Honda has proceeded with numerous other initiatives in this area of research, for example increasing the controllability of combustion through the use of dual fuels (introducing the use of alcohol fuels)<sup>(7)-(9)</sup>.

However, this research is still at the concept engine level, and has not yet been practically realized. One of the main factors standing in the way of the practical realization of HCCI is the challenge represented by control of ignition timing to help enable continuous stable combustion. The authors therefore proposed Spark-Assisted HCCI (SA-HCCI) as a method of accurate ignition timing control, verified the combustion concept in the NOL range, and developed the necessary hardware.

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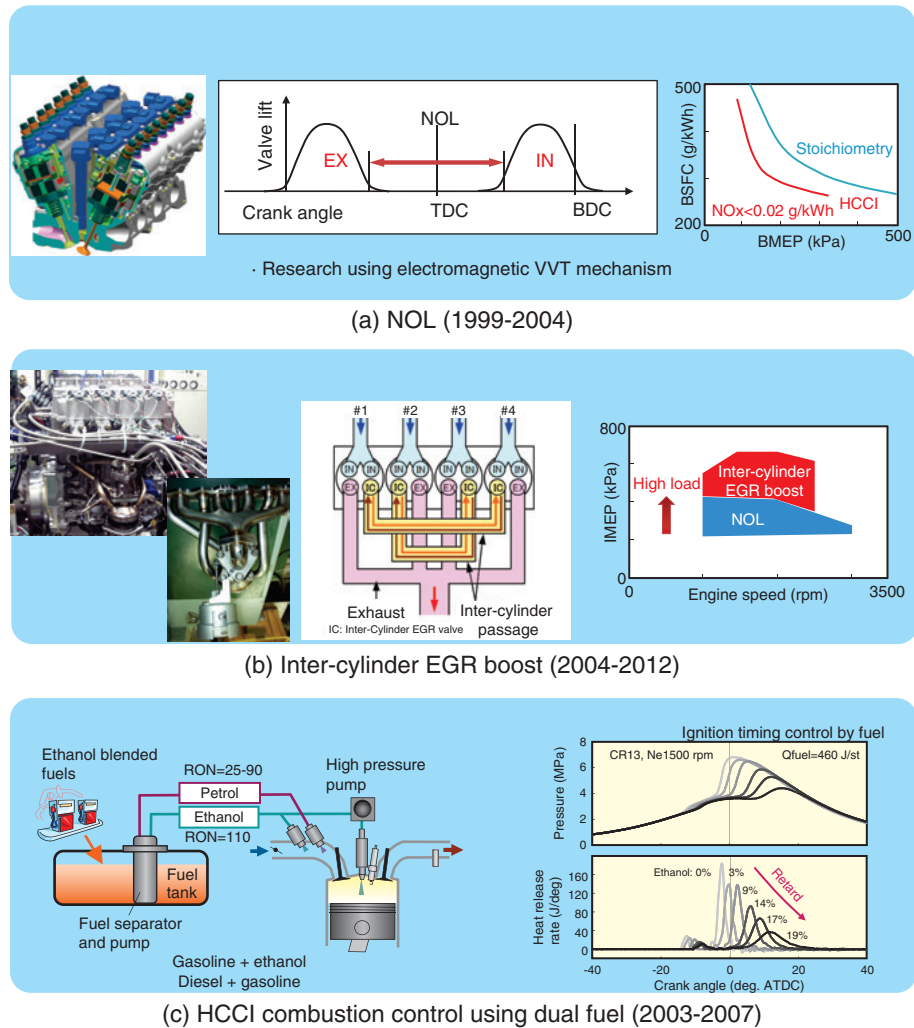


Fig. 1 Research on HCCI at Honda

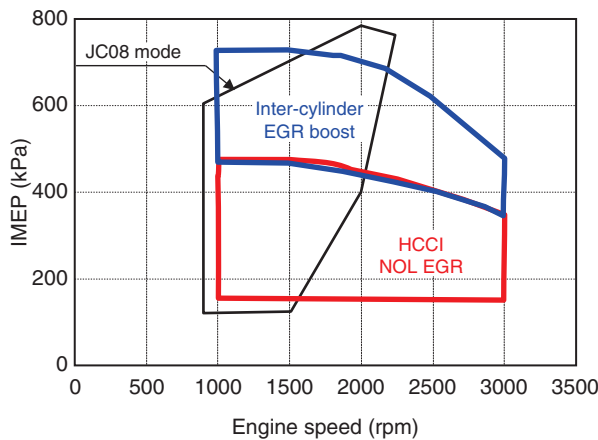


Fig. 2 HCCI operating range

## 2. SA-HCCI Combustion Concept

### 2.1. Merits of HCCI

HCCI is a type of homogeneous premixed combustion

in which high compression ratios and internal EGR realized using the methods shown in Fig. 1 increase the temperature of the mixture in the cylinders to approximately 1000 K at close to compression top dead center, causing a reaction in the entire mixture in the combustion chamber resulting in auto-ignition. By initiating combustion through auto-ignition, it becomes possible to employ lean combustion at the air-fuel ratio range of A/F25 and above, which represents a challenge in terms of the realization of combustion stability when spark ignition (SI) combustion, in which combustion is initiated by spark ignition, is used.

Making possible leaner combustion than SI by means of auto-ignition, HCCI displays high theoretical thermal efficiency and low pumping loss and time loss. In addition, because the combustion temperature is low, heat loss is low and NO<sub>x</sub> emissions are minimal. Given these merits, this combustion method can be expected to increase fuel efficiency by around 20% against stoichiometric SI, and at the same time will also be able to realize low emissions (EM).

Figure 3 shows P-V diagrams for SI and HCCI with internal EGR realized by NOL in the same engine. HCCI is clearly able to minimize pumping loss and time loss.

However, HCCI also presents a number of issues that are barriers to its practical realization.

## 2.2. Issues of HCCI

Three major issues of HCCI can be recognized. These are:

- (1) The operating range is narrow;
- (2) Combustion ignition control represents a challenge; and
- (3) Exhaust gas purification performance is poor in the lean environment.

With regard to Issue (1), the HCCI operating range, despite the fact that the expansion of the range via inter-cylinder HCCI, as shown in Fig. 2, is able to extend it to largely cover JC08 mode, HCCI operation is not possible in the high-load, high-engine-speed range outside the mode. At present, therefore, it is not possible to cover the entire necessary engine operating range using HCCI alone.

Issue (2), ignition control, is an important issue for vehicles used in a wide range of environments.

Figure 4 shows a diagrammatic representation of the heat generation history for HCCI from the compression stroke to auto-ignition, including the mixture oxidation reaction process.

In HCCI combustion, a low-temperature oxidation reaction that produces a cool flame commences from a mixture temperature of around 600 K. The accumulation of formaldehyde and the proliferation of CO and OH radicals generates a blue flame, until finally a hot flame that generates  $\text{CO}_2$  from CO is produced. When employing SI combustion, in which the spark plugs advance the reaction rapidly and a hot flame is produced, ignition timing can be easily controlled by control of the spark timing of the plugs. In the case of HCCI, because changes in temperature and pressure during the compression stroke change the oxidation reaction process, control of ignition timing represents a challenge. To help ensure stable combustion in a variety of environments, it will be necessary to accurately control ignition timing, and attempt to reliably prevent misfires,

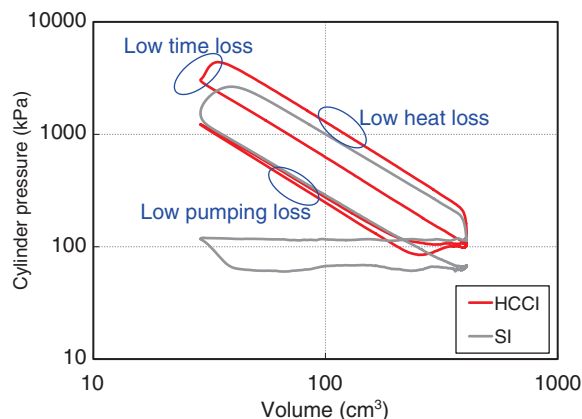


Fig. 3 Comparison of cylinder pressure

torque fluctuations, and increases in emissions.

With regard to after-treatment, Issue (3), the temperature of exhaust gas generated by HCCI is lower at the exhaust manifold outlet than stoichiometric SI exhaust gas. At an indicated mean effective pressure (IMEP) of 400 kPa, HCCI exhaust gas is around 320°C, compared to 500°C for SI. In addition, because combustion is lean, clearing EM regulations via the conventional method of employing a three-way catalyst (TWC) also represents a challenge. This makes a new after-treatment system essential.

Of these issues, it is the controllability of ignition, Issue (2), that it is particularly necessary to address, or it will not be possible to employ HCCI in automotive engines. The research discussed in this paper focused on this issue of controllability of ignition.

## 2.3. Concept of SA-HCCI Combustion

As discussed above, in order to realize stable ignition timing, which represents one of the issues of HCCI, a method has been tested in which auto-ignition is assisted by a miniscule combustion event (trigger flame) generated by igniting a miniscule amount of directly injected fuel via the spark plug. Figure 5 shows the differences in the method of increasing temperature in the cylinders and the heat release history for conventional HCCI and SA-HCCI. In the majority of cases, standard HCCI increases temperature by means of internal EGR and pressure in order to induce auto-ignition of the lean, homogeneous mixture. By contrast, in SA-HCCI, a small amount of fuel is directly injected towards the spark plug during the compression stroke, and the propagation of a flame by the spark plug increases temperature, inducing auto-ignition. The heat release history for SA-HCCI shows that by contrast with conventional HCCI, a trigger flame releases heat prior to auto-ignition. Because the position of the combustion center of gravity for the realization of increased thermal efficiency is close to 10 deg ATDC, as in the case of SI and conventional HCCI, the aim was to realize combustion with a heat release history in

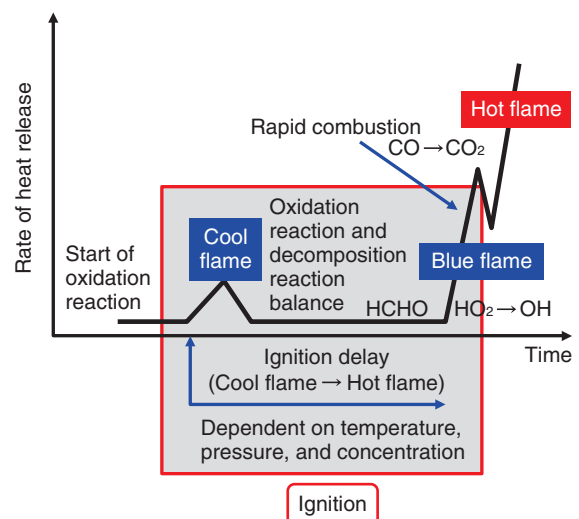


Fig. 4 Oxidation reaction process for HCCI

which auto-ignition occurred at close to 2 deg ATDC.

At the same time, because the trigger flame is propagated by spark ignition, it is necessary to set the conditions for the trigger flame fuel injection at close to stoichiometric conditions. Therefore, the trigger flame combustion temperature is high, and NO<sub>x</sub> is locally produced. Figure 6 shows the results of shadowgraph visualization of the

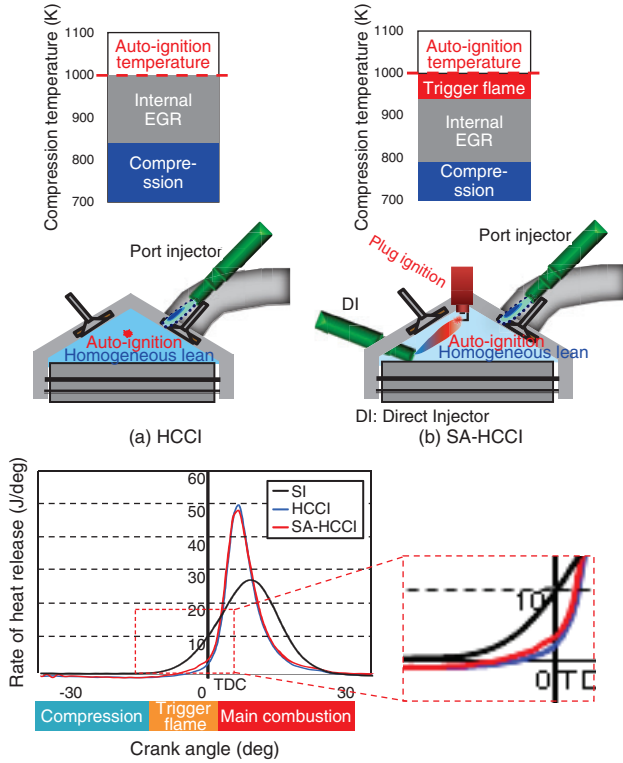


Fig. 5 SA-HCCI

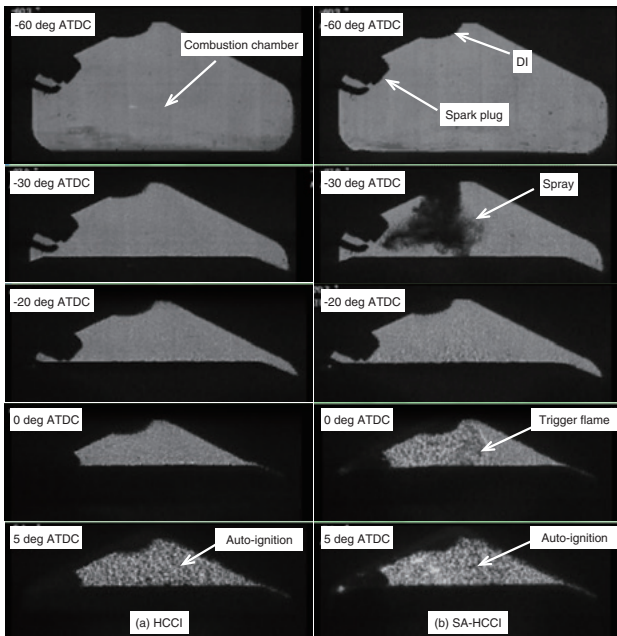


Fig. 6 Visualization of combustion chamber by shadowgraph method

ignition process for HCCI and SA-HCCI. This shows the process of induction of auto-ignition by flame propagation via direct injection in SA-HCCI.

### 3. Technologies Required for Realization of SA-HCCI

#### 3.1. Setting of Trigger Flame

The setting of the trigger flame is an important factor for SA-HCCI. It was necessary to set the trigger flame with consideration of the following:

- (1) Required calorific value for trigger flame;
- (2) Spray distance for arrival at plug (penetration); and
- (3) Restriction of NO<sub>x</sub> emissions.

These factors will each be discussed in detail below.

##### 3.1.1. Required calorific value for trigger flame

It is necessary to realize a calorific value for the trigger flame that will cause the mixture to burst into a hot flame.

The necessary calorific value was back-calculated from the target heat release history, and was set with consideration of realization of a flame propagation speed that made it possible to achieve the temperature for auto-ignition at close to 2 deg ATDC. Figure 7 shows an in-cylinder temperature history at an engine speed ( $N_e$ ) of 1500 rpm and an IMEP of 350 kPa. The necessary gas temperature for ignition of the mixture was approximately 1070 K, and it was not possible to realize this temperature via the temperature rise due to internal EGR and piston compression. Because of this it would be necessary to realize the target temperature following a temperature increase of approximately 50 K, realized by combustion of a directly injected mixture. With 375 cm<sup>3</sup> as the precondition for each cylinder, the necessary level of trigger flame heat for a temperature increase of 50 K would be approximately 10 J per cylinder.

##### 3.1.2. Penetration to reach plug

Because the stratified mixture that would generate the trigger flame would be ignited by the spark plug, it was essential for the DI spray to arrive at the plug. If the fuel flow rate for the trigger flame was insufficient, the spray

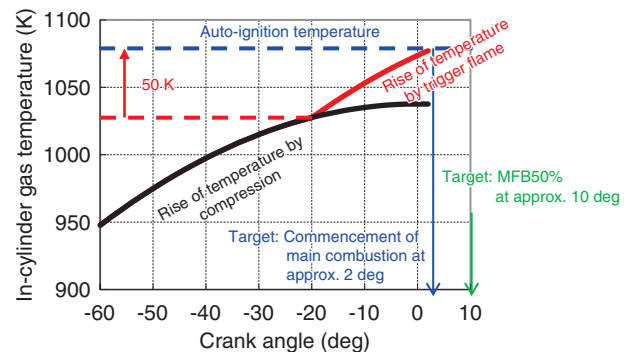


Fig. 7 Temperature rise process for SA-HCCI

would possess inadequate penetration, and depending on the distance between the plug and the direct injector, the spray would not arrive at the plug.

Figure 8 shows results for the relationship between the distance between the direct injector and the plug and DI fuel energy enabling ignition.

The calorific value of the trigger flame to help ensure ignition (the flow rate of the DI spray) is determined by the distance between the direct injector and the spark plug. However, when the calorific value of the trigger flame increases, NO<sub>x</sub> emissions also increase. When the distance between the direct injector and the spark plug is set at 30 mm, the level of heat generation by the trigger flame to help ensure penetration is 20 J per cylinder.

### 3.1.3. Restriction of NO<sub>x</sub> emissions

Increasing the calorific value of the trigger flame would increase the controllability of ignition, but NO<sub>x</sub> emissions would also increase, and the benefit of lean burn would be lost. Figure 9 shows the relationship between the calorific value of the trigger flame and NO<sub>x</sub> emissions.

The allowable level of NO<sub>x</sub> emissions varies based on the engine type and EM regulations. Assuming an engine displacement of 1.5 L, in order to realize a reduction in NO<sub>x</sub> emissions of 75%, as specified by Japan's Post New Long-Term Regulation, it would be necessary to control

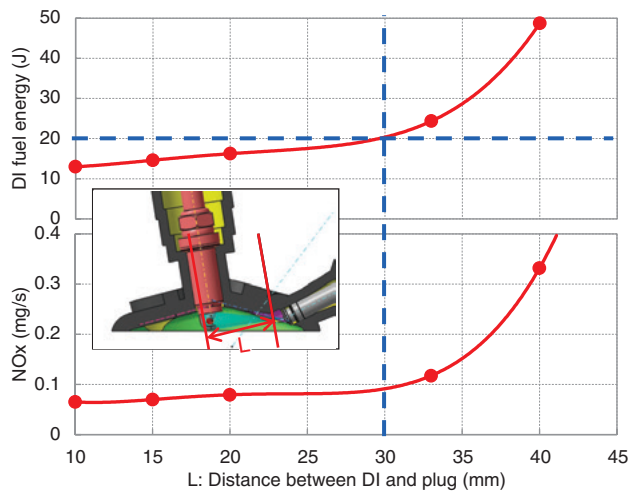


Fig. 8 Relationship between DI flow and DI penetration

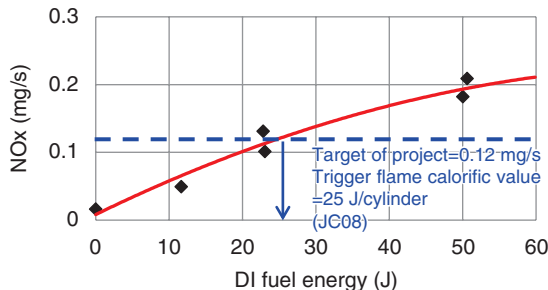


Fig. 9 Relationship between DI fuel energy and NO<sub>x</sub>

emissions to an upper limit of approximately 0.12 mg/s. At an Ne of 1500 rpm and an IMEP of 350 kPa, the upper limit for the calorific value of the trigger flame would therefore be 25 J per cylinder.

### 3.1.4. Summary of setting of trigger flame

Figure 10 shows the method of setting the trigger flame, with the temperature of the gas in the cylinders as the benchmark.

10 J was set as the minimum calorific value necessary for the supplied homogeneous lean air-fuel mixture to auto-ignite and produce a hot flame. A further 20 J was added as corresponding to the flow rate necessary to help ensure that the DI spray would arrive at the spark plug. The range from 20 J to 25 J, the upper limit value for NO<sub>x</sub> emissions, was therefore established as the range of setting values for the trigger flame. The calorific value above the 10 J necessary for the trigger flame represents a temperature margin, and is available to make it possible to absorb variations in environmental conditions and amount of control.

### 3.2. SA-HCCI System Configuration

Figure 11 shows the necessary system configuration for the practical realization of the SA-HCCI concept.

For (1) in the figure, internal EGR control, a variable

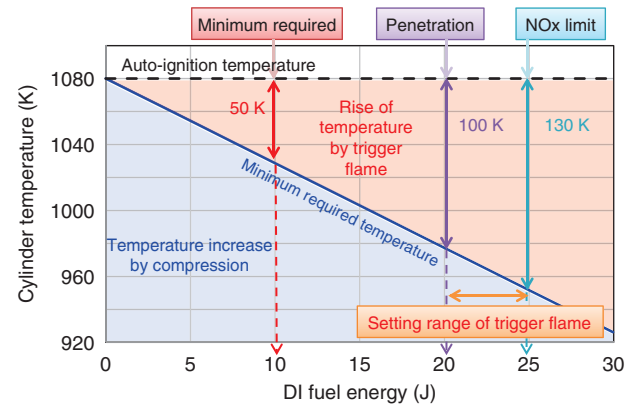


Fig. 10 Method of setting trigger flame

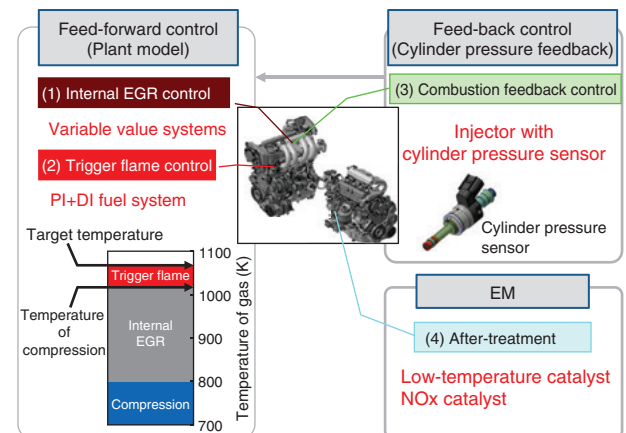


Fig. 11 SA-HCCI system configuration

valve system making it possible to change the intake (IN) and exhaust (EX) valve timing is necessary. As Fig. 12 shows, the amount of internal EGR can be appropriately controlled in response to load by varying NOL via Valve Timing Control (VTC) employing IN and EX cams.

For (2) in the figure, trigger flame control, two injection systems are necessary: A low flow-rate direct injector to produce the trigger flame, and a port injector for control of load. Precise specification of the flow rate at a low level is necessary for the direct injector for the trigger flame. The reason for this is that because the injector cannot provide the dynamic range necessary to cover fuel injection amounts across the entire load range, and injection is highly oriented towards the direction of the spark plug, there are concerns that injection of too much fuel would cause combustion to become unstable.

In-cylinder pressure sensors are necessary for feedback combustion control, (3) in the figure. The state of the mixture is estimated based on the values measured by the in-cylinder pressure sensors, and ignition timing control is precisely applied with the amount of internal EGR calculated from operating conditions and the calorific value and injection timing necessary for the trigger flame as parameters. The use of in-cylinder pressure sensors built into the direct injectors<sup>(10)</sup> was made a prerequisite.

For (4) in the figure, after-treatment, an HC purification catalyst to respond to the low exhaust temperatures of lean combustion and an NO<sub>x</sub> purification catalyst to make it possible to broaden the HCCI combustion range are necessary.

This paper will not discuss details of after-treatment and combustion control.

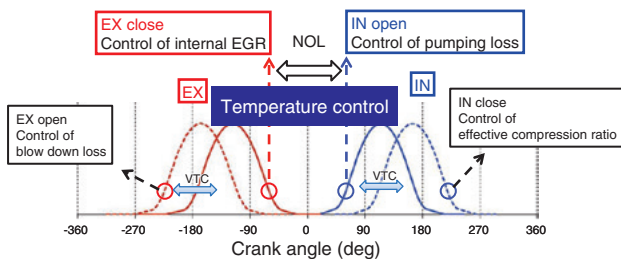


Fig. 12 Variable valve systems

## 4. Results of Verification of Concept

The SA-HCCI system discussed above was tested in a 4-cylinder engine. Table 1 shows the specifications of the engine used in verification.

### 4.1. Verification of Controllability of Ignition Timing

Figure 13 shows the results of comparison of the controllability of ignition in HCCI and SA-HCCI at an Ne of 1500 rpm and an IMEP of 420 kPa. The verification compared the combustion range when the in-cylinder

Table 1 Engine specifications

|                                 |                    |
|---------------------------------|--------------------|
| Number of cylinders             | 4                  |
| Displacement [cm <sup>3</sup> ] | 1500               |
| Bore × Stroke [mm]              | φ 73 × 89.4        |
| Compression ratio               | 14                 |
| Valve train                     | IN EX VTEC and VTC |
| Gasoline                        | 91RON              |

temperature was varied by varying the amount of internal EGR. The combustion range was defined as a combustion fluctuation rate of 4% or less and a pressure rise rate ( $dP/d\theta$ ) (an index of combustion noise) of 400 kPa and below. The combustion range within which standard HCCI satisfied the conditions above was a narrow range between EGR rates of 50% and around 53% (the difference in in-cylinder gas temperature at this time was around 6 K). In the case of SA-HCCI, despite the fact that the EGR rate was lowered and the in-cylinder temperature was reduced, the advancement of the trigger flame ignition timing and the increase in the combustion amount for the trigger flame made it possible to realize continuous combustion in the range above an EGR rate of 10% (in-cylinder temperatures of 20 K and above).

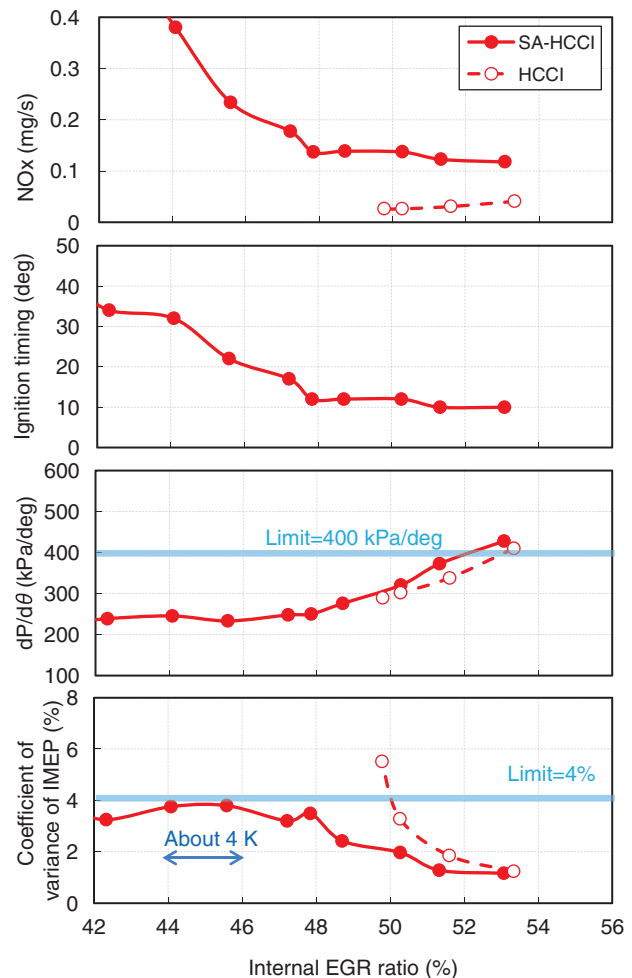


Fig. 13 Concept verification

At the same time, an increase in NO<sub>x</sub> emissions was also observed as a result of the increase in the combustion amount for the trigger flame. The status of SA-HCCI can also be expressed by the waveforms of the heat release rate,  $dQ$ , as shown in Fig. 14. The results show that a trigger flame combusts prior to auto-ignition and that the advancement of ignition timing has expanded combustion of the trigger flame in comparison to HCCI.

Figure 15 shows the results of verification of the combustion range under various load conditions. The results show that SA-HCCI has a greater effect than HCCI at all loads.

As the results discussed above indicate, by increasing the controllability of ignition, SA-HCCI has increased the combustion range in relation to the amount of EGR by two to three times against HCCI.

#### 4.2. Verification of Steady-state Fuel Consumption

Figure 16 shows a comparison of the results of fuel consumption and NO<sub>x</sub> measurements at an Ne of 1500 rpm for SA-HCCI and stoichiometric SI combustion. Table 2 shows the main specifications of the technologies under comparison. Based on indexes for commercial appeal, the upper limit for the operating range of SA-HCCI was a brake

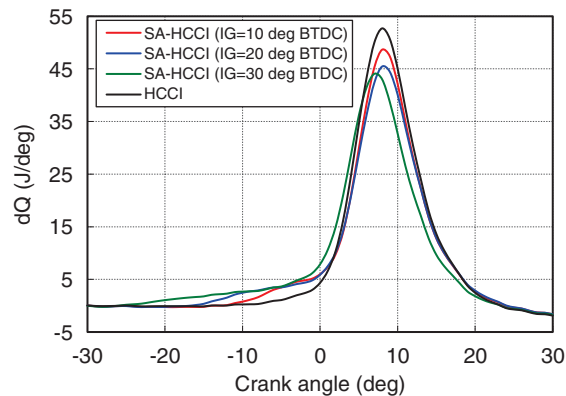


Fig. 14 Comparison of heat release rates

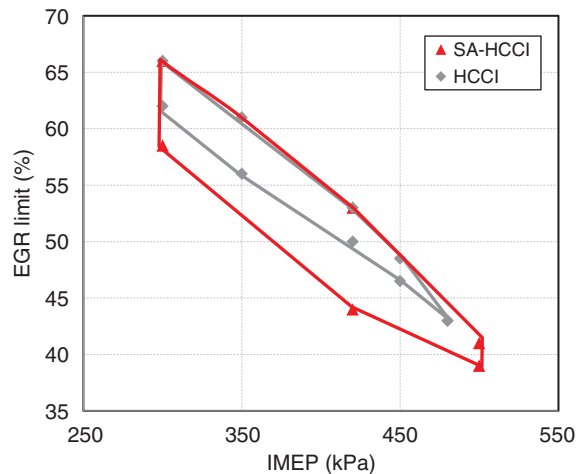


Fig. 15 Comparison of combustion range

Table 2 Engine specifications

|                                 | Stoichiometry | Atkinson cycle | Cooled EGR | SA-HCCI         |
|---------------------------------|---------------|----------------|------------|-----------------|
| Number of cylinders             | 4             | 4              | 4          | 4               |
| Displacement [cm <sup>3</sup> ] | 1.8           | 1.8            | 1.5        | 1.5             |
| Compression ratio               | 11            | 13             | 13         | 14              |
| EGR                             | None          | Hot 15%        | Cooled 25% | Internal 30-60% |
| Gasoline                        | 91RON         | 91RON          | 91RON      | 91RON           |

mean effective pressure (BMEP) of 400 kPa, and brake specific fuel consumption (BSFC) was lowest at this figure. Figure 17 shows a comparison of the difference in fuel consumption at the BMEP 400 kPa point between SA-HCCI and SI combustion using cooled EGR, the condition under which fuel efficiency is best, in addition to a breakdown of a thermal analysis. SA-HCCI reduces fuel consumption by approximately 16% against cooled EGR combustion.

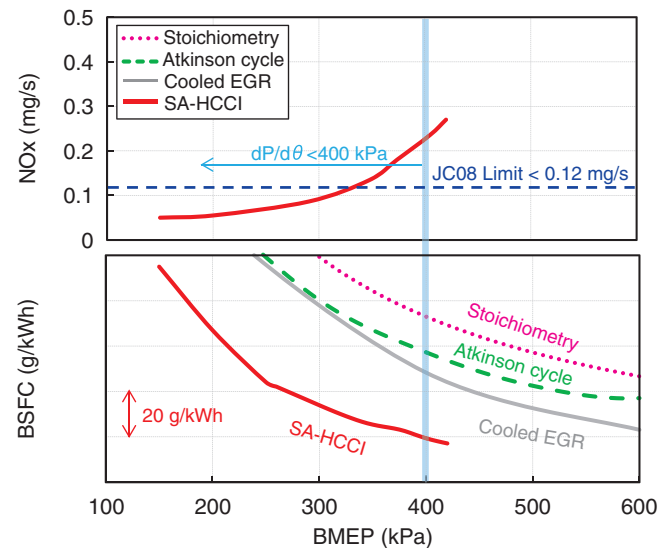


Fig. 16 Results for 4-cylinder engine

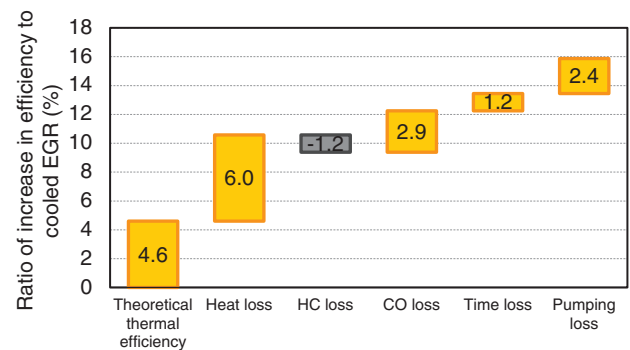


Fig. 17 Ratio of increase in efficiency to cooled EGR

Table 3 Specifications of simulated car

| Car                             | FIT  |
|---------------------------------|------|
| Weight [kg]                     | 968  |
| Displacement [cm <sup>3</sup> ] | 1500 |
| Transmission                    | CVT  |

Among the main factors in this are an increase in theoretical thermal efficiency (as a result of an increase in the ratio of specific heat due to increased leanness and the dilution of EGR, and the expansion of the compression ratio) and a reduction in heat loss. In the future, further increases in fuel efficiency can be expected as a result of efforts to reduce the pressure rise rate and expand the range of operating loads.

With regard to NO<sub>x</sub> emissions, figures are low in the area below 300 kPa, but as load increases emissions increase rapidly. In the low-load area, because the gas temperature of the main combustion area is low, NO<sub>x</sub> is only produced by the trigger flame; however, as the gas temperature of the main combustion area increases with increasing load, it also produces NO<sub>x</sub>. At present, emissions exceed the level that would clear the established target of a 75% reduction in NO<sub>x</sub> emissions, as specified by Japan's Post New Long-Term Regulation, with operation up to a maximum load point of BMEP 400 kPa as a prerequisite. The use of lean NO<sub>x</sub> catalysts in order to maximize the increase in fuel efficiency offered by SA-HCCI will be considered, and operation up to a BMEP of 400 kPa is projected.

#### 4.3. Results of Verification using JC08 Mode Fuel Efficiency Simulation

A verification of the system was conducted using a JC08 mode fuel efficiency simulation, based on the current SA-HCCI operating range and results for steady-state fuel consumption performance. Table 3 shows the main specifications of the vehicle used in the simulation. The mode fuel efficiency of cooled EGR combustion and cooled EGR combustion with SA-HCCI combustion were compared.

Figure 18 shows the simulation results. The results show that the addition of SA-HCCI to the cooled EGR technology increases fuel efficiency by approximately 4.5%.

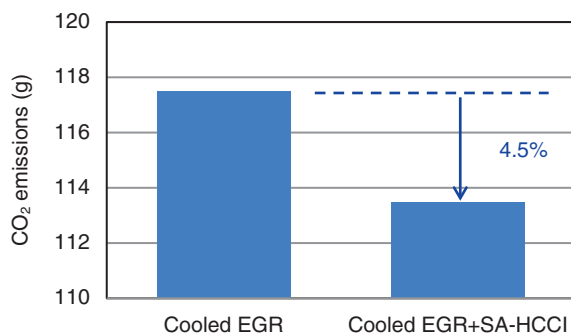


Fig. 18 Simulation results

## 5. Conclusion

A development with a focus on the controllability of ignition was conducted towards the practical realization of HCCI, with the following results.

- (1) SA-HCCI was developed to assist HCCI ignition using direct injection in an attempt to increase the controllability of ignition. Verification of the concept showed that it expanded the combustion range by two to three times against conventional HCCI.
- (2) The fuel efficiency of SA-HCCI was verified, demonstrating that the system reduced steady-state fuel consumption by approximately 16% against cooled EGR. Simulation verification of JC08 mode fuel efficiency indicated that an increase of approximately 4.5% in fuel efficiency could be expected.

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