

Study of Ignition Timing Control of Gasoline HCCI Engine

Toru KITAMURA*

Toshinari SHINOHARA*

Junichi TAKANASHI*

Yasuhiro URATA*

ABSTRACT

Homogeneous Charge Compression Ignition (HCCI) combustion is a method of combustion in which ignition occurs without being dependent on the ignition of a spark plug, unlike flame spread combustion. There are many parameters involved in ignition, and it is a challenge to control the ignition timing to an optimum value. Accordingly, a two-part HCCI ignition timing simulation model was constructed, with one part corresponding to the exhaust and intake strokes that determine the internal EGR and air intake amount, and the other part corresponding to the compression stroke during heat release by a chemical reaction. The results obtained using this model agreed with the performance test results. From the ignition timing simulation model, a control algorithm was developed and applied to a 4-cylinder engine fitted with an electromagnetic valve train, and the ability to control the ignition timing to the value corresponding to the demand torque in HCCI combustion was confirmed.

1. Introduction

As concerns over environmental issues have increased, the reduction of CO₂ emissions has become an important issue for the automotive industry. Homogeneous charge compression ignition (HCCI) is a combustion method that enables multi-point ignition of a homogeneous lean mixture. The rapid combustion achieved in this method ensures a high degree of constant volume, pumping loss reduction, and higher brake thermal efficiency. In addition, low temperature combustion results in a level of nitrogen oxide (NO_x) emissions of only several ppm⁽¹⁾⁻⁽⁵⁾. Given these characteristics, a considerable amount of research is presently being conducted on this combustion method.

The HCCI is a self-ignition method different from those that initiate flame spread by means of spark plug ignition. A large number of parameters are involved in achieving conditions for the initiation of self-ignition, and optimum control of ignition timing to maximize the effect of combustion in terms of generating demand torque is therefore a significant technological issue when utilizing HCCI.

2. Research Goals

The research discussed in this paper focuses on the

role of the different engine strokes in HCCI combustion. An ignition timing simulation model was formulated by the parameters for ignition timing control clarified in HCCI tests using a single-cylinder engine. The goal of the research was to control ignition timing in response to demand torque by a control algorithm based on this model.

3. Ignition Timing Simulation

Figure 1 shows the structure of the HCCI ignition timing simulation model used in this research. The model

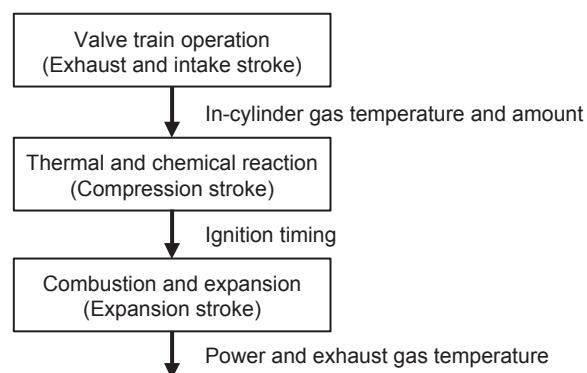


Fig. 1 Ignition timing simulation model

* Automobile R&D Center

Table 1 Specifications of single cylinder research engine

Bore × stroke (mm)	80 × 80
Displacement (cm ³)	402
Compression ratio	15
Valve diameter (mm) × number of valves	Intake 27.5 × 2 Exhaust 24.5 × 2
Fuel system	Port injection

consists of three parts: the intake and exhaust strokes, the compression stroke, and the expansion stroke. In the intake and exhaust stroke section of the model, the intake and exhaust valve train system controls the amount of internal EGR and intake air, and the in-cylinder gas temperature and inducted charge amount in the cylinders are determined. In the compression stroke section, the rise of temperature (ignition) due to the compression of the cylinder mixture (including the injection quantity) and the chemical reaction is modeled, and the ignition timing representing the initiation of the flame is determined. In the expansion stroke section, the work produced by the flame initiated by the ignition timing, as well as the exhaust temperature, are calculated. The models were determined as follows. The compression stroke model for determining the ignition timing was formulated on the basis of the results of HCCI tests conducted with intake air heating. The intake and exhaust stroke model, which determines the in-cylinder gas temperature and the charged amount of gas in the cylinders, was formulated on the basis of HCCI tests using the internal EGR and previously formulated compression stroke model.

Table 1 shows the specifications of the single cylinder engine used to confirm the characteristics of the HCCI combustion for the intake-air temperature and injection quantity. A 3-kW intake heater was positioned upstream of the intake port to enable HCCI tests with intake air heating.

3.1. HCCI with Intake Air Heating

When HCCI combustion is conducted with intake air heating, the charged gas is not heated by exhaust energy. Therefore, taking the intake conditions only into consideration, a model of the compression stroke in the HCCI combustion can be identified.

3.1.1. Results of tests with intake air heating

Figure 2 shows the heat release rate for crankshaft angles when the indicated mean effective pressure (IMEP) of the test engine was varied between 76 kPa and 327 kPa at an engine speed of 1000 rpm and an intake-air temperature of 200 °C. The heat release rate increases as the injection quantity increases. At the same time, the ignition timing advances, and the torque produced

increases. Figure 3 shows the indicated specific fuel consumption (ISFC) and NOx emission characteristics of the engine in relation to intake-air temperature and injection quantity. There is an intake-air temperature at which the ISFC for the IMEP is at its minimum, and the point at which the ISFC is at its minimum matches the point at which NOx emissions are at their minimum. From the perspective of ignition timing control, this indicates that the optimal control of in-cylinder gas temperature in response to demand torque can maximize thermal efficiency and can minimize NOx emissions at the same time.

3.1.2. HCCI combustion model

Assuming HCCI combustion modeled as a chemical reaction in a homogeneous mixture, the reaction speed can be expressed by Eq. (1) in accordance with the Arrhenius law.

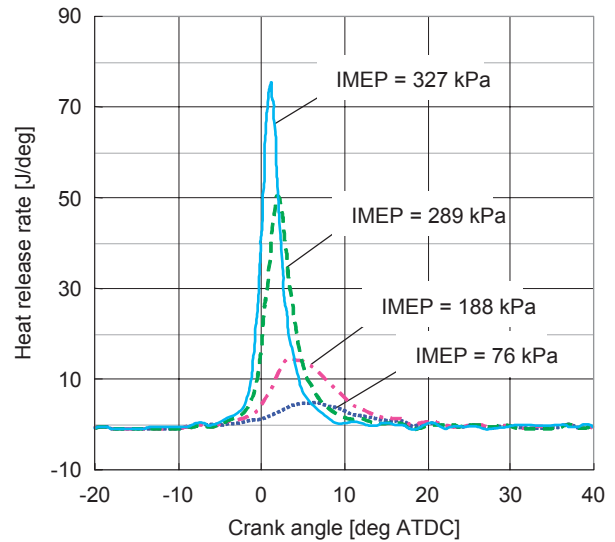


Fig. 2 Tendency of heat release pattern according to IMEP

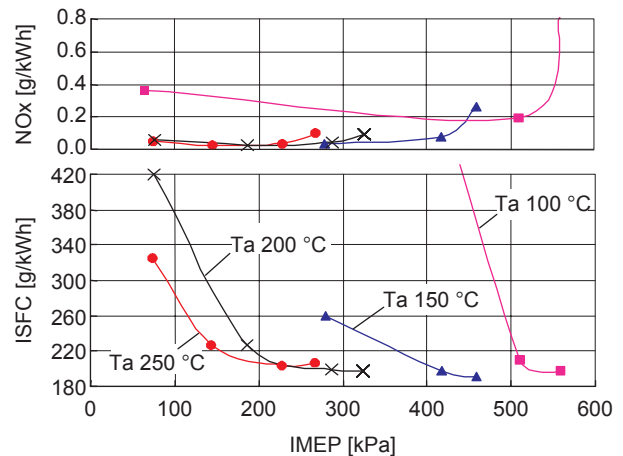


Fig. 3 ISFC and NOx characteristic of air intake heating HCCI

$$\frac{dN}{dt} = A[F][O]e^{-E^*/RT} \quad (1)$$

E^* : activation energy, A : frequency factor, $[F]$: mol number of carbon and hydrogen, $[O]$: mol number of oxygen, R : gas constant, T : mixture temperature

$[F]$ and $[O]$ are reduced in every reaction. With regard to the mixture temperature, if the reaction heat per reaction is q_0 and the cooling loss is dQ_c , the temperature change, dT , can be expressed by Eq. (2).

$$dT = \frac{q_0 dN - dQ_c}{C_v N_t} - (\kappa - 1) \frac{T}{V} dV \quad (2)$$

C_v : specific heat at constant volume, N_t : amount of gas in the cylinder, V : volume of cylinder, κ : ratio of specific heat

From Eqs. (1) and (2), the parameters controlling the HCCI combustion in an engine are the injection quantity, intake air amount, and in-cylinder gas temperature. For the initial conditions where the mol number of hydrocarbons which contributes to combustion from the injection quantity, the mol number of oxygen molecules which contributes to combustion from the intake-air amount, and the mol number of nitrogen molecules which does not contribute to combustion but only increases the temperature, the combustion model simulates the in-cylinder gas temperature by the compression and the chemical reaction. The reaction speed and reaction temperature parameters were identified to match the ignition timing output by this model and the results of engine tests. The ignition timing was defined as crankshaft angles of 2J/deg CA or above when the heat release rate clearly begins to increase.

3.1.3. Comparison with engine tests

Figure 4 shows the results for heat release rates obtained from the ignition timing simulation. The calculation conditions were identical to those used in the

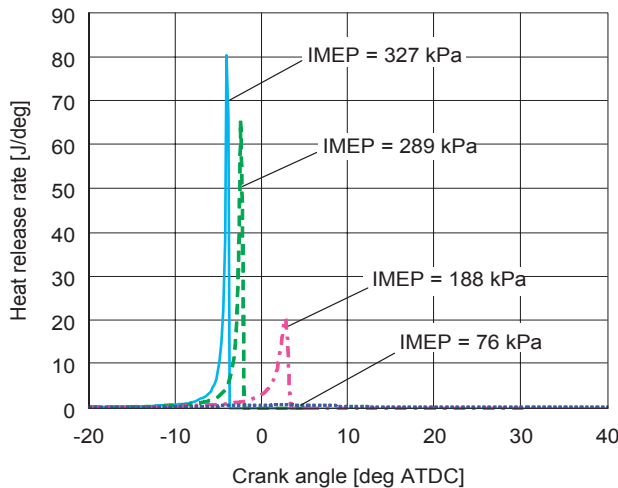


Fig. 4 Simulation result of heat release pattern

engine tests discussed previously (engine speed: 1000 rpm; intake-air temperature: 200 °C). The simulation reproduced the tendency displayed in the engine tests: as the injection quantity increased, ignition timing advanced and the amount of produced torque increased. Figure 5 shows a comparison of simulation results and engine test results when the intake-air temperature was varied. The ignition timing output by the simulation was within ± 5 deg CA of the ignition timing of the HCCI combustion in the actual engine.

3.2. Formulation of Ignition Timing Model using Internal EGR

Figure 6 shows the relation between engine control parameters and ignition timing model parameters for HCCI combustion using internal EGR. G_{fuel} represents the injection quantity, G_{air} represents the intake air amount, and T_{gas} represents the in-cylinder gas temperature. Because combustion gas remains in the cylinders when internal EGR is used, the mol number of carbon dioxide

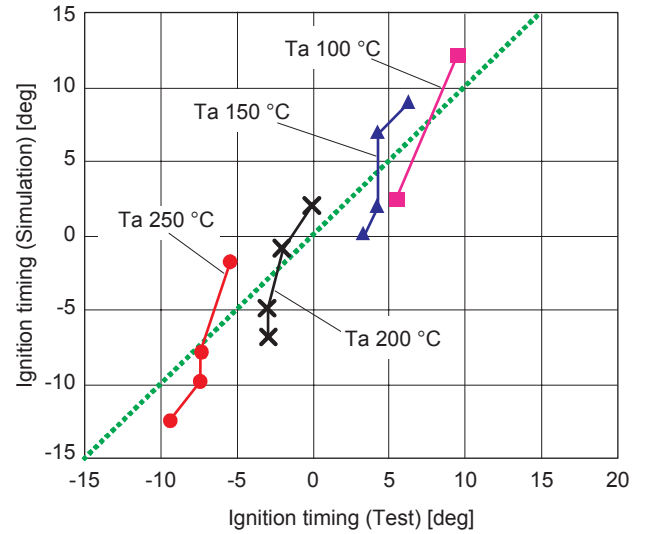


Fig. 5 Comparison of ignition timing simulation and test result

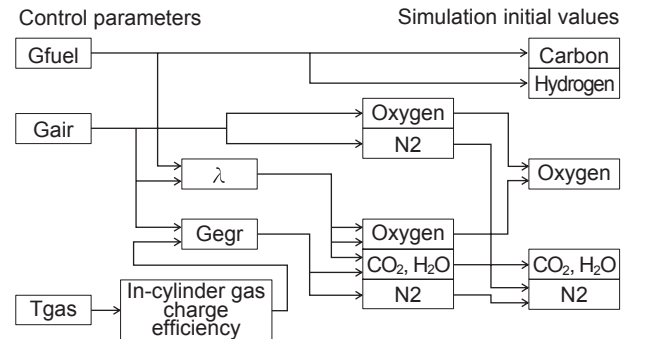


Fig. 6 Relationship between control parameters and ignition timing simulation parameters

molecules, oxygen molecules, and water vapor in the combustion gas should be considered. Because the mol number of oxygen molecules in the exhaust gas also contributes to ignition timing, an exhaust gas analyzer was used for measurement. Carbon dioxide and water vapor were treated as inert gases.

Figure 7 shows an outline of the method of determining the in-cylinder gas temperature, T_{gas} , in relation to the intake air and internal EGR. The intake air temperature, T_a , and the exhaust gas temperature, T_{ex} , are internally divided by the intake air amount, G_{air} , and the amount of internal EGR, G_{egr} , to give the in-cylinder gas temperature, which is used as the initial value of the combustion model. The in-cylinder gas temperature is determined by using Eq. (3).

$$T_{gas} = \frac{(T_{ex}G_{egr} + T_aG_{air})}{G_{egr} + G_{air}} \quad (3)$$

3.3. Comparison of Simulation and Test Results using Internal EGR

Valve timing and lift were modified for the single-

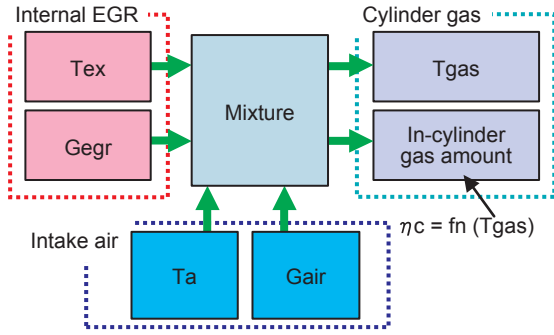


Fig. 7 Conceptual diagram of cylinder gas temperature and amount calculation

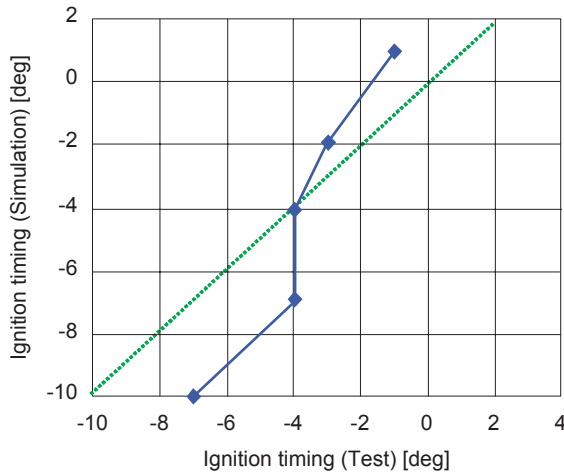


Fig. 8 Comparison of internal EGR HCCI ignition timing simulation and test results

cylinder engine mentioned above, and tests were conducted at 40% internal EGR. Various intake air temperatures at an engine speed of 1000 rpm and an IMEP of 300 kPa were studied.

As Fig. 8 shows, simulation results for ignition timing matched the test results within ± 3 deg CA or less even if internal EGR was employed.

4. Ignition Timing Control Algorithm

The ignition timing simulation model mentioned above was used to form an HCCI combustion ignition timing control algorithm. Figure 9 shows a block diagram of the control algorithm. The fundamentals of the HCCI ignition timing control system are calculating the injection quantity and the intake air amount from the demand torque, and concurrently calculating the target in-cylinder gas temperature as well as determining the amount of the internal EGR necessary to achieve the target temperature. The system also controls the valve close timing of the intake and exhaust valves to achieve the necessary intake air and internal EGR amounts.

4.1. Calculation of Target In-cylinder Gas Temperature

As indicated in Fig. 3, there are specific in-cylinder gas temperatures that produce optimum ignition timings for specific levels of demand torque. According to the Arrhenius law, when the engine is operating at high speed when a more rapid reaction speed is necessary, it will be necessary to increase the in-cylinder gas temperature, and when the engine is operating under a high load with high mol numbers of carbon and hydrogen, it will be necessary to lower the in-cylinder gas temperature to maintain the reaction speed at a fixed level. Figure 10 shows target in-cylinder gas temperatures for loads and engine speed parameters.

4.2. Method of Calculating Total Amount of Gas Charge in the Cylinders

The method of calculating the total amount of gas charge in the cylinders is as follows. The volume of the cylinder at the piston position when the intake valve is

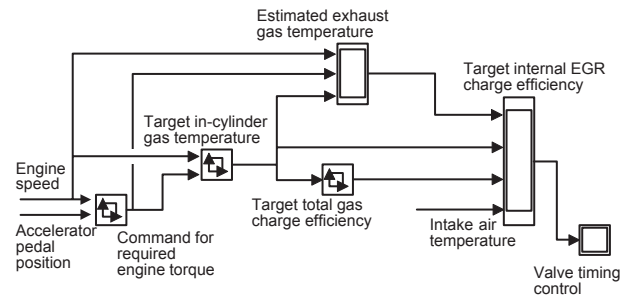


Fig. 9 Block diagram of ignition timing control for gasoline HCCI engine

closed can be measured. Assuming that the pressure in the cylinders is equivalent to the pressure in the intake manifold, and that the in-cylinder gas temperature matches the target temperature, the total amount of gas charge in the cylinders, including intake air and internal EGR, can be calculated from the gas equation of state.

Therefore, the target amount of gas charge in the cylinders, E_{tg} , can be calculated from the target in-cylinder gas temperature, T_{igas} , by using Eq. (4).

$$E_{tg} = \frac{\text{Constant}}{T_{igas}} \quad (4)$$

4.3. Exhaust Gas Temperature Estimation

As the calculation of the internal EGR temperature based on measurements of each exhaust gas temperature cycle would take considerable calculation time, this method could not be used in combustion control. Therefore, in this research, the exhaust gas temperature was estimated. Because the in-cylinder gas temperature in HCCI combustion is high at the commencement of the compression stroke, and the exhaust gas temperature is low in comparison to spark ignition combustion, the in-cylinder gas temperature has an effect on the exhaust gas temperature. Thus an exhaust gas temperature estimation logic, with consideration of the drop in temperature due to heat transfer from the walls, was used. The estimated exhaust gas temperature, T_{EX} , is calculated from the exhaust gas temperature at base engine speed T_{EX0} , cylinder wall temperature T_{EMPCYL} , and engine stroke cycle M_E , as shown in Eq. (5).

$$T_{EX} = T_{EX0} - AhM_E \frac{T_{EX0} - T_{EMPCYL}}{N_i C_p} \quad (5)$$

h : heat transfer coefficient, A : area of heat-transfer surface, N_i : amount of exhaust gas, C_p : specific heat at constant pressure

Figure 11 shows the results of tests conducted to verify the accuracy of the exhaust gas temperature estimation at different intake air temperatures. These

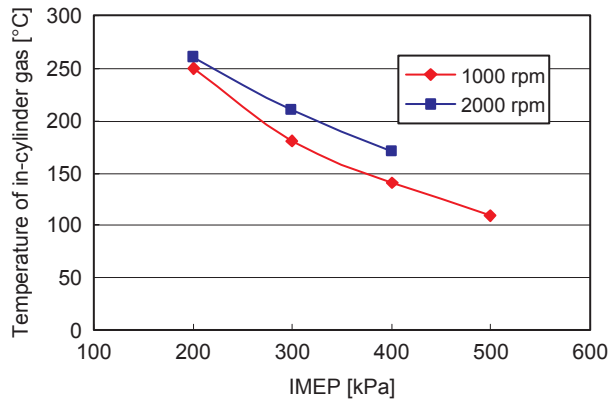


Fig. 10 Target temperature of in-cylinder gas for required torque

results show that the exhaust gas temperature could be estimated within ± 8 °C.

4.4. Method of Calculating the Necessary Amount of Internal EGR

The optimum amount of the internal EGR necessary for HCCI combustion in each engine operating state, N_{egr} , is calculated from the demanded in-cylinder gas temperature, T_{igas} , total amount of gas charge in the cylinders, E_{tg} , intake air temperature, T_a , and estimated exhaust gas temperature T_{EX} , by using Eq. (6).

$$N_{egr} = E_{tg} \frac{T_{igas} - T_a}{T_{EX} - T_a} \quad (6)$$

5. Test of Control Algorithm

Figure 12 shows the 4-cylinder HCCI engine with a variable valve timing system using an electromagnetic mechanism⁽⁶⁾, which also used the HCCI ignition timing control algorithm mentioned previously. Table 2 shows the specifications of this engine.

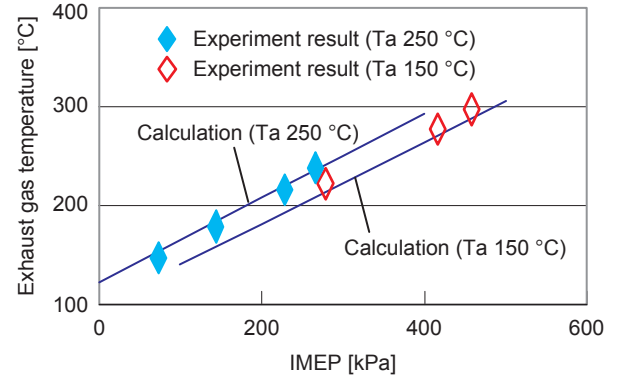


Fig. 11 Result of exhaust gas temperature estimation

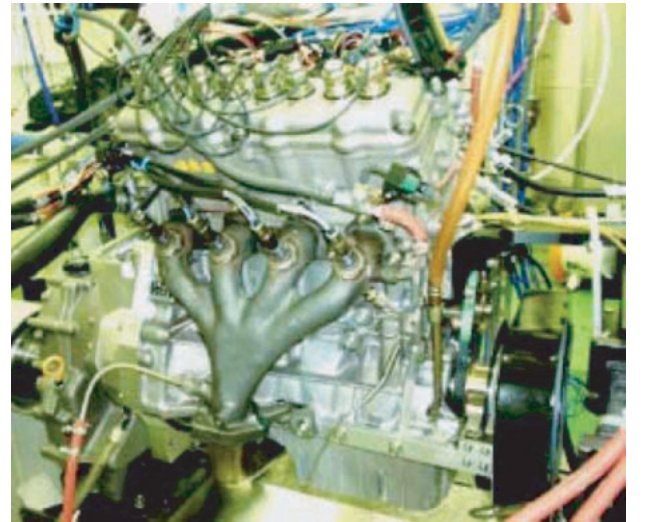


Fig. 12 Photograph of research 4-cylinder engine

Table 2 Specifications of research 4-cylinder engine

Bore × stroke (mm)	75 × 90
Displacement (cm ³)	1590
Compression ratio	15
Valve diameter (mm) × number of valves	Intake 30 × 2 (EMV)
	Exhaust 24 × 2 (EMV)
Fuel system	Port injection

5.1. Electromagnetic Valve Train Control

Figure 13 shows the outline of the electromagnetic valve train operation. As the internal EGR increases, valve control advances the close timing of the exhaust valve and delays the open timing of the intake valve at the same time.

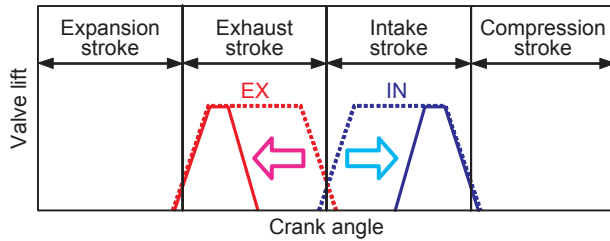


Fig. 13 Valve timing diagram of internal EGR HCCI condition

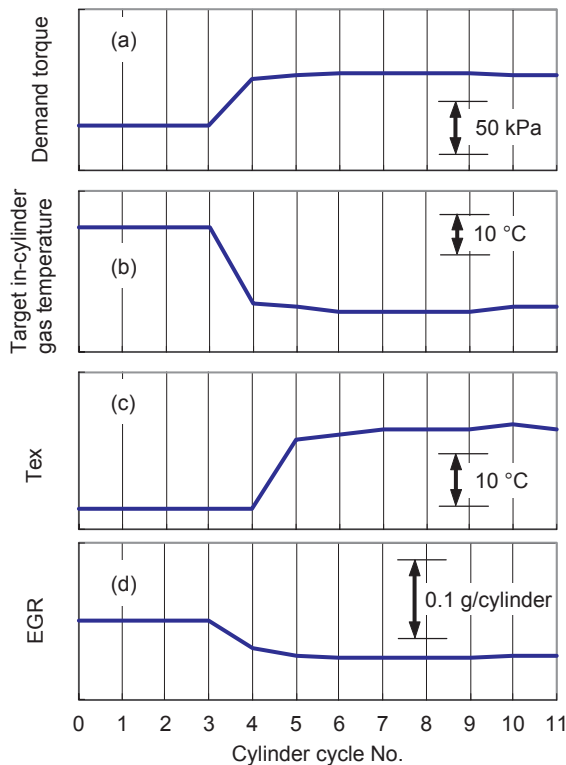


Fig. 14 Time chart of HCCI ignition timing control parameters of demand torque step

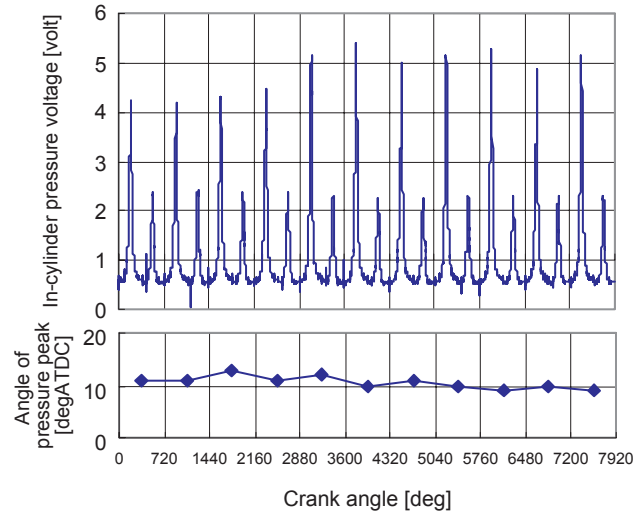


Fig. 15 Time chart of in-cylinder pressure and angle of pressure peak

5.2. Demand Torque Step Test

With the engine speed kept at 1500 rpm, torque responsiveness was tested by varying demand torque (BMEP) in steps between 300 kPa and 350 kPa.

Figure 14 shows changes in each control parameter. When demand torque is increased in Cycle No. 4 (a), the intake and exhaust valve timing functions to reduce the internal EGR (d) and to lower the in-cylinder gas temperature (b). Figure 15 shows the in-cylinder pressure waveform and the timing of pressure peaks for each cycle at this time. According to the in-cylinder pressure waveform of the fifth cycle, the peak value of the in-cylinder pressure changes in steps. The results also indicate that the timing of the in-cylinder pressure peaks can be constantly controlled to around an efficient ATDC 10 deg before and after torque changes.

6. Conclusion

This research was conducted on a gasoline HCCI ignition timing control algorithm focusing on the amount and temperature of in-cylinder gas when the intake valve is closed. The results are as follows.

- (1) An HCCI combustion ignition timing simulation model displayed an accuracy within ± 5 deg for both HCCI with intake air heating and HCCI using internal EGR, when compared to test results.
- (2) The ignition timing control algorithm of the HCCI combustion noted above was applied to a 4-cylinder engine with a variable valve timing system using electromagnetic mechanism, and a high level of responsiveness and high degree of accuracy in controlling ignition timing was confirmed for step changes in demand torque.

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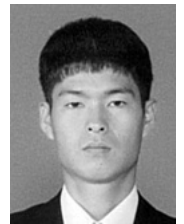
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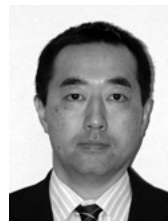
Toru KITAMURA



Toshinari SHINOHARA



Junichi TAKANASHI



Yasuhiro URATA