

Improvement of Natural-gas HCCI Combustion by Internal EGR by Means of Exhaust Valve Re-opening

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ABSTRACT

To control natural-gas HCCI combustion, internal exhaust gas recirculation (EGR) by exhaust valve re-opening (EVRO) during the induction stroke was applied to a single-cylinder test engine. The results demonstrate that combustion phasing can be controlled successfully by adjusting the EGR ratio, and so improvement of thermal efficiency and reduction in unburned exhaust emissions are feasible. In addition, the results of the EVRO method were compared to those of intake-valve pilot opening (IVPO) during the exhaust stroke. It was shown that EVRO is more useful than IVPO as a heat-recovery method for HCCI combustion.

INTRODUCTION

The homogeneous charge compression ignition (HCCI) engine is attracting great attention because of its potential to produce high thermal efficiency with low pollutant emissions. Especially, a natural-gas HCCI engine is expected to be a new power source for small-scale co-generation systems. However, because the primary component of natural gas is methane, it reaches a higher autoignition temperature of approximately 1000 K compared to that of other liquid fuels. Consequently, in addition to a diesel-like compression ratio of approximately 20, the intake air has to be heated to initiate successful heat release [1-5]. Furthermore, to produce high thermal efficiency without a knock or misfire, the in-cylinder temperature has to be precisely controlled so that combustion starts at the optimum crank-angle timing near the top dead center. Therefore, to produce a feasible engine system, the ignition control technique must not consume additional energy.

In a natural-gas HCCI engine, internal exhaust gas recirculation (EGR), also referred to as burned gas trapping, is effective in lowering the charge heating requirement [6-10]. Our previous study [11] showed that internal EGR based on the intake-valve pilot opening (IVPO) improves HCCI combustion and control; in this technique, the intake valve is opened together with the exhaust valve at the beginning of the

exhaust stroke. Then, part of the hot burned gas is introduced backward into a suction pipe [12], resulting in an increase of the intake gas temperature.

In the present study, we apply internal EGR by exhaust valve re-opening (EVRO) in a natural-gas HCCI engine. In this technique, the exhaust valve is opened during the induction stroke to re-breathe burnt gas from the exhaust pipe. Then, mixing of the hot EGR gas as the fuel-air mixture begins in the cylinder, so that it is possible that heterogeneity remains in the gas temperature at the end of the compression stroke. The existence of local hot-spots with large burnt-gas concentrations may bring a beneficial effect on initiating combustion. The purpose of this study is to clarify the effects of internal EGR by EVRO on engine performance and exhaust emission by comparing the results of the proposed EVRO method and the IVPO method.

EXPERIMENTAL APPARATUS AND METHOD

TEST ENGINE AND TEST CONDITIONS

The experimental layout is shown in Figure 1. A four-cycle single-cylinder diesel engine (YANMAR NFAD-5) was modified for HCCI operation. The main specifications of the test engine are shown in Table 1.

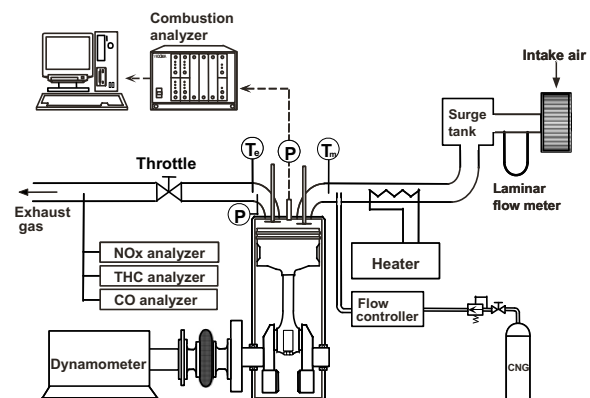


Fig. 1 Experimental layout

Table 1 Engine specifications

Engine type	4 cycle/water cooled/N.A.
Rated speed	2600 rpm
Bore	70 mm
Stroke	72 mm
Comp. ratio	20:1
Combustion chamber	Dog-dish type

The fuel, which was natural gas for commercial use, was composed of approximately CH₄ (88%), C₂H₆ (6%), C₃H₈ (4%) and C₄H₁₀ (2%). The fuel was introduced continuously into the suction pipe through the nozzle, mounted 155 mm in front of the cylinder head. An electric heater (max. power 3 kW) was installed to control the temperature of the fuel-air mixture. To control the EGR ratio, as described later, an exhaust throttle valve was installed with an exhaust pressure sensor (KISTLER 4047A0).

In the present study, the EGR ratio was given by the intake air flow rate based on the following equation.

$$f_{EGR} = \frac{Q_0 - Q}{Q_0} \quad (1)$$

Q and Q_0 denote the intake air flow rate with and without EGR, respectively, for the same engine revolution speed.

To evaluate the emission characteristics, nitrogen oxides (NO_x), carbon monoxide (CO) and total unburned hydrocarbons (THC) were measured.

The in-cylinder pressure was recorded using a piezoelectric transducer and charge amplifier (KISTLER 6053C/5011) coupled with a data-acquisition unit (YOKOGAWA WE7000). The heat release rate, gas temperature and some combustion characteristics were calculated from the pressure indicators. In addition, to detect the knock caused by the abnormally large combustion rate, the high-pass filtered pressure was monitored. The knock limit was defined as the detection of high-frequency pressure vibration above 4 kHz.

The cooling water temperature was held at approximately 80°C and the oil at 70°C. All experiments were conducted with an engine revolution speed of 1200rpm

CONTROL OF EGR RATIO

Hot exhaust gas was introduced into the cylinder by re-opening the exhaust valve during the induction stroke. Figure 2 shows the valve-lift profile for EVRO together with that for the intake-valve pilot opening (IVPO). In the case of EVRO, the exhaust valve is mechanically re-opened by the cam, which can lift the exhaust valve up during the interval of 380° ~ 460° After Top Dead Center (ATDC). As shown also in Figure 2, re-opening duration $\Delta\theta$, indicated by the dotted line, is increased by using an additional

hydraulic device to increase the EGR ratio. The maximum valve-lift height, which is dependent on the cam, is maintained by using the hydraulic piston as long as hydraulic fluid is kept in the hydraulic cylinder. In the case of IVPO, the intake valve is lifted up at 130°ATDC by the cam, and its closure timing is also delayed by the hydraulic device.

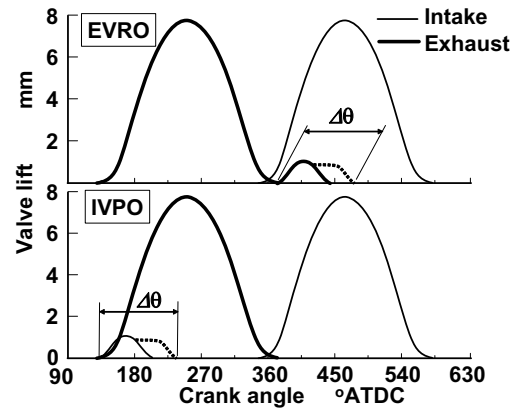


Fig. 2 Valve-lift profile of EVRO and IVPO

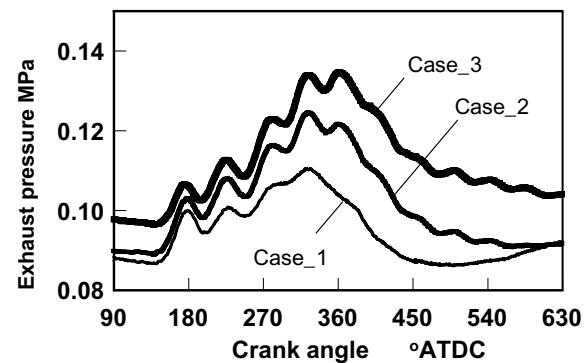


Fig. 3 Pressure variation in exhaust pipe for different degrees of opening of exhaust throttle

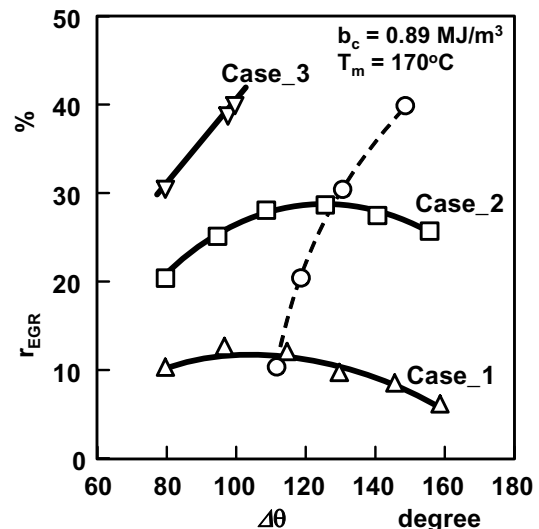


Fig. 4 Variation of EGR ratio with re-opening duration for different degrees of opening of exhaust throttle

Figure 3 shows the pressure vibration measured in the exhaust pipe for different degrees of exhaust throttling. Case_1 has no throttling, and the degree of throttling becomes larger in this order: Case_2, Case_3. In Case_1, we can see that the exhaust pressure falls below atmospheric pressure during the induction stroke due to the pulsation effect in the long exhaust pipe. This result means that when $\Delta\theta$ is increased above 110°CA without exhaust throttling (Case_1), part of the fresh mixture blows through the combustion chamber into the exhaust pipe. As the result, the EGR ratio is decreased, as shown in Figure 4, and a large amount of unburned fuel is emitted. So, when a large EGR ratio is required in addition to the increase in re-opening duration, the exhaust gas flow is throttled to increase the back pressure, as shown in Case_2 and Case_3. In the present experiment, we adjust the re-opening duration so that the EGR ratio is maximized for each throttling degree to avoid an increase in the pumping loss, as indicated by the open circles and dotted line in Figure 4.

RESULT AND DISCUSSION

EFFECT OF INTERNAL EGR BASED ON EVRO

First, the effect of internal EGR by EVRO on engine performance and exhaust emissions are investigated. The heat consumption for one stroke divided by the stroke volume b_c is 0.89 MJ/m^3 . Our HCCI engine with the compression ratio of 20:1 cannot run without intake air heating. So, the temperature of air-fuel mixture T_m is kept at 170°C by an electric heater. Figure 5 shows exhaust gas temperature T_e and average in-cylinder temperature T_a at the beginning of the compression stroke for various EGR ratios. T_a is determined by T_m , T_e and the EGR ratio r_{EGR} based on the following equation.

$$T_a = T_m (1 - r_{\text{EGR}}) + T_e r_{\text{EGR}} \quad (2)$$

Note that in the case of $b_c = 0.89$ and $r_{\text{EGR}} = 0\%$, the equivalence ratio is 0.4, and it increases as the EGR ratio increases. By mixing the EGR gas at the temperature of 280°C into a fresh mixture at 170°C , average temperature T_a linearly increases with the EGR ratio, as shown in Figure 5.

Figure 6 shows the in-cylinder pressure and rate of heat release for different EGR ratios. The gas temperature before compression increases by re-breathing the hot burnt gas, and the auto-ignition timing advances. Then, the peak pressure increases due to the increase in the degree of constant volume of combustion, even though the heat capacity of the in-cylinder gas is increases by EGR. In addition, the increase in exhaust gas temperature, which is the result of improvement in combustion efficiency due to EGR, as mentioned later, brings a recursive effect on the in-cylinder temperature at the beginning of the compression stroke.

Figure 7 illustrates the variation of the indicated mean effective pressure (IMEP) with EGR ratio, which is calculated from the p - V diagram and includes an

intake and exhaust stroke. In the case of the largest heat-consumption condition of $b_c = 0.98 \text{ MJ/m}^3$, IMEP is lowered by increasing the EGR ratio because the heat release timing advances much beyond the top dead center. On the contrary, in the case of $b_c = 0.84$ or 0.89 MJ/m^3 , IMEP increases with the increasing

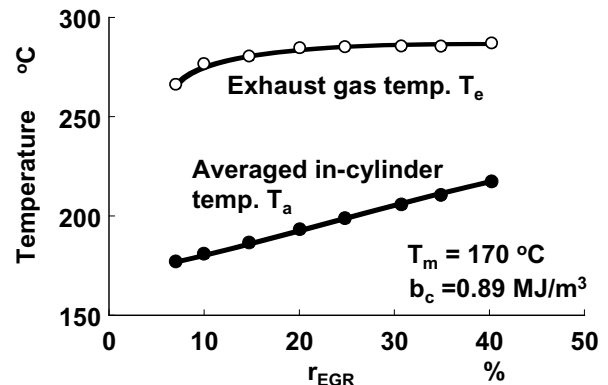


Fig. 5 Variations of exhaust gas temperature and average in-cylinder temperature with EGR ratio

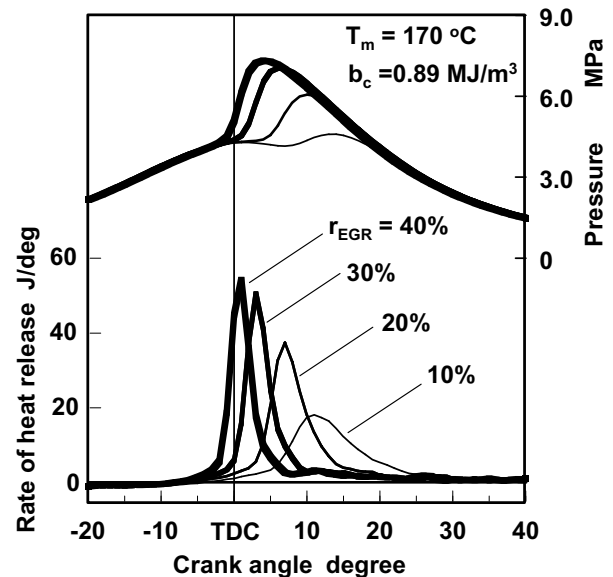


Fig. 6 In-cylinder pressure and rate of heat release for different EGR ratios

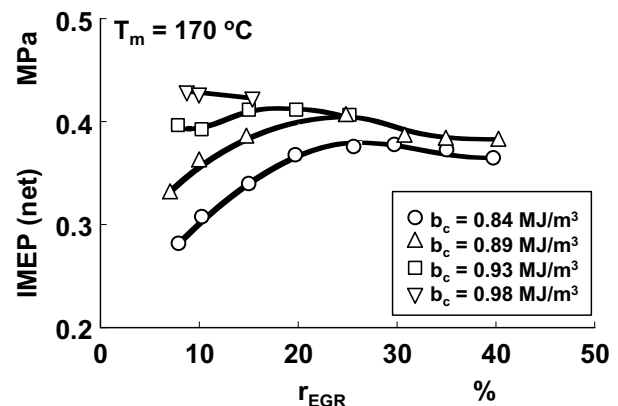


Fig. 7 Variations of net IMEP with EGR ratio