

Effect of Relative Positions of Air-Fuel Mixture Distribution and Ignition on Combustion Variation in Gasoline Engine

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ABSTRACT

Combustion is known to be affected by variations in the air-fuel mixture concentration, residual gas concentration, turbulent kinetic energy, ignition, etc. However, because each of these factors is related to cycle-to-cycle variations, their effects on combustion variation are unclear. The purpose of this study was to clarify the influences of the air-fuel mixture distribution near the spark plug and variation in the relative position of the ignition on the combustion variation. A 4-cylinder port injection gasoline engine was used as the test engine, and the combustion variation was investigated by measuring the cylinder pressure and air-fuel ratio (A/F) near the spark plug for each cycle using a micro-Cassegrain sensor for each cylinder. The air-fuel mixture distribution was calculated using a Reynolds averaged Navier-Stokes simulation, and the spatial region of the high ignition probability was determined from the gas flow velocity. Then, the combustion progress when ignition occurred at each point of the region was calculated, and was compared with the measured combustion variation. As a result, the frequency distribution of the A/F within the ignition position variation region was correlated with the measured A/F for each cycle. Based on the combustion calculation results when the ignition position was changed, the difference that occurred in the growth rate of the initial flame kernel was related to the difference in the air-fuel mixture concentration at the ignition timing. These results suggested that variations in the relative positions of the air-fuel mixture distribution and ignition may strongly affect combustion variation.

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INTRODUCTION

Improving the thermal efficiency of a gasoline engine is a pressing need because of rising oil prices and the need to reduce CO₂ emissions. Expanding the knocking limit is indispensable to improving the thermal efficiency of a gasoline engine. However, knocking is a probability phenomenon [1] that varies between cycles by several percent under the same operating conditions, and is closely related to combustion variation. Combustion variation is affected by changes in the ignition position from spatial variations in the air-fuel mixture concentration near the spark plug [2, 3], along with variations in the components and temperature of the residual gas [4], the in-cylinder flow field and turbulent kinetic energy [5, 6], and the spark ignition position and timing [7]. It is known that the influence of these variations will affect the following cycle. Variation of the air-fuel mixture concentration is considered to be due to the cycle-to-cycle variations of the gas flow among these factors. Zhang et al. [8] measured the gas flow field in a cylinder using particle image velocimetry. Although there were slight differences in the swirl vortex centers, the gas flow patterns for the individual cycles were similar. With similar gas flow patterns, the flow pattern for each cycle was reported to be in good agreement with the gas flow pattern of the ensemble average. Moreover, Fukano et al. [9] measured the

in-cylinder flow field using particle tracking velocimetry, and reported the following. The swirl vortex center position varied for every cycle by approximately $\pm 5\%$ of the bore diameter. These results suggest that the in-cylinder flow does not vary significantly, and the cycle-averaged flow field varies minutely for each cycle. The authors thought that the variations in these gas flow fields were related to the variation of the air-fuel mixture concentration. With respect to the variation of the air-fuel mixture, it has been reported that variations in the spark ignition also occur for every cycle. Shichida et al. [10] confirmed that the spark discharge moved in the flow direction. According to Kawahara's measurements [11], the spark discharge varied for every cycle under the same operating conditions. Ignition was also generated at a point that was approximately 8 mm from the center electrode, and the position of the flame kernel of the spark discharge was affected by the gas flow. In summary, these studies found that the relative spatial variation of the air-fuel mixture distribution and ignition by a small variation of the cycle-averaged flow field near the spark plug was one factor in the combustion variation.

On the other hand, Reynolds averaged Navier-Stokes simulation (RANS), which has high computational stability and a comparatively small calculation load, is widely utilized in the

development of gasoline engines [12, 13]. RANS treats turbulence statistically, and because the flow velocity is calculated using the time average, the solution acquired becomes the cycle-averaged flow field. When utilized for engine development, RANS is used for both the gas flow such as the air-fuel mixture and the combustion model, which can predict the in-cylinder pressure and exhaust gas components [14, 15].

Based on the background discussed above, the authors assumed that a small positional variation in the flow field similar to the cycle average in the cycle-to-cycle variations was responsible for the variation of the spatial position of the air-fuel mixture distribution, and thought that the influence of the combustion variation could be presumed by estimating the relative ignition position within the variation region. Moreover, the authors thought that an analysis focused on the variation of the ignition position in a particular concentration field was possible by using the cycle-averaged flow field calculated with RANS. Thus, as an initial stage of the investigation, this study used measurements and calculations of the relationship between the spatial relative variation of the air-fuel mixture distribution and the ignition position. The purpose of this study was to clarify the effect of the relative positional relationship between the air-fuel mixture distribution and the ignition on the combustion variation. A 4-cylinder port injection gasoline engine was used for the test engine. The in-cylinder pressure and air-fuel mixture concentration near the spark plug in each cylinder were measured. The variation range, coefficient of variation, and frequency distribution of the A/F were evaluated to find correlations with the measured A/F near the spark plug and air-fuel mixture distribution in the variation range obtained by RANS. The ignition positions were set to the variation range by using the air-fuel mixture distribution obtained by RANS, and a combustion calculation was performed. The results were compared to the variation range and standard deviation of the measured in-cylinder pressure. Based on the results, the authors determined the effect of the relative positions of the air-fuel mixture distribution and ignition on the combustion variation.

MEASUREMENT METHODS AND OPERATING CONDITIONS

Layout of Test Engine and Operating Conditions

To investigate the correlation between the air-fuel mixture concentration and ignition position in combustion variation, the air-fuel mixture concentration near the spark plug and in-cylinder pressure were measured. The test engine was a 4-cylinder port injection gasoline engine, which was based on that used in commercial vehicles and had an intake system that was specially modified for this study. Table 1 lists the specifications of the test engine and the operating conditions. The layout of the intake port of the test engine is shown in Figure 1. A tumble control valve, a plate for rectifying the gas flow in the intake port, an inlet for the external exhaust gas recirculation (EGR) gas, and two port injectors were located at each intake port. The flow rate from the external EGR gas inlet was controlled by the valve, in this experiment, this valve was closed and this test was performed. The intake system had primary and secondary lines,

with fuel injectors for each system. Figure 2 shows the valve lift curve, injection period, and fuel injection timing. The maximum lift amount of the primary valve was 0.5 mm higher than maximum lift amount of the secondary valve. The fuel injection timing was set to an injection end timing of 60° after top dead center (ATDC) during the intake stroke. The injection period was set as the period in which the air-fuel mixture of the two injectors became stoichiometric. The injection amount was set to 1/2 of the total mass in each injector of each cylinder, and the fuel was injected at the same time. These injection conditions were the same setting also in all the cylinders. In order to avoid combustion variation due to the variation of injection, the primary and secondary injectors were selected the injectors having the same injection characteristics as possible among the injectors of the same specification. To verify their similarity, the combustion variation for each injector was measured by replacing the intake systems separately, and no difference was found in the combustion variation as a result of these replacements. As for fuel, a commercial gasoline (Research Octane Number: RON=100) was used by the measurement. The engine speed was 1500 rpm, and the opening of the tumble control valve was a constant 15°. The operating condition was a low-load condition with a volumetric efficiency of approximately 23 %, and this operating condition was comparatively large condition of cycle-to-cycle variations.

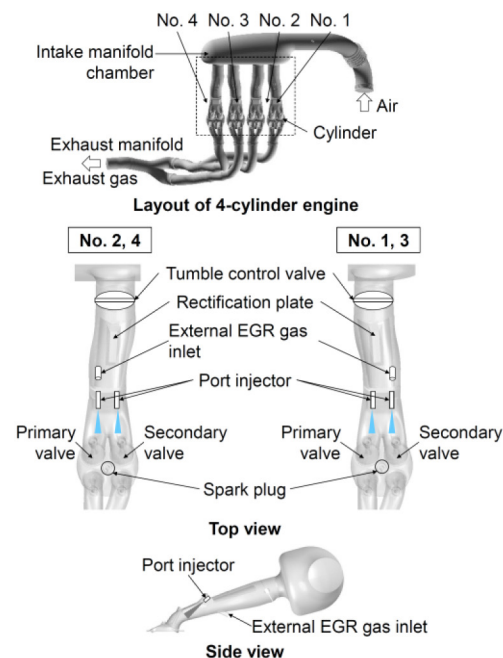


Figure 1. Intake port layout of test engine

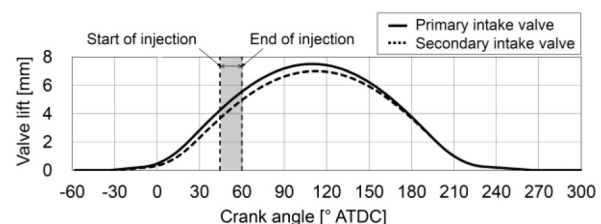


Figure 2. Intake valve lift curve of the primary and secondary valve and fuel injection timing

Table 1. Engine specifications and operation conditions

Test engine	2.2L 4-cylinder port fuel injection gasoline engine (Modified Honda F22C engine)
Bore	87.0 mm
Stroke	90.7 mm
Compression ratio	11.1:1
Engine Speed	1500 rpm
Tumble control valve opening	15°
Injection start timing	44.5° ATDC (Intake stroke)
Injection end timing	60.0° ATDC (Intake stroke)
Spark ignition timing	33.0° BTDC
Spark-discharge period	27.0° CA

Measurement Method

In-cylinder measurements were made using a spark plug integrated pressure sensor (type 6052B) made by Kistler, and the air-fuel mixture concentration near the spark plug was measured using a spark plug integrated micro-Cassegrain sensor [16, 17] made by Imagineering, Inc. Since each sensor used for measurement of the in-cylinder pressure and fuel-air mixture concentration was integral with the spark plug, each sensor were replaced, and each parameter was measured by the respectively under the same operating condition.

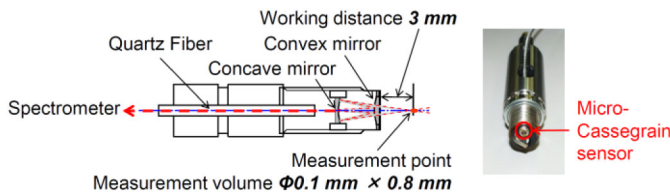


Figure 3. Schematic of the measurement principle and the appearance of the micro-Cassegrain sensor

Figure 3 shows a schematic of the measurement principle and the appearance of the micro-Cassegrain sensor. This sensor measured the radical chemiluminescence generated by combustion using concave and convex mirrors facing each other, and the signal was transmitted through an optical fiber. The radical chemiluminescence was detected by using a photomultiplier tube to determine the emission intensity at the characteristic wavelength, such as that for an OH radical, by using a narrow band pass filter and dichroic mirror. The core diameter of the optical fiber was 230 μm . As shown in Figure 4, a radical such as OH*, CH*, or C₂* was detected, and A/F was determined based on how it changed the emission intensity ratio [18]. In this study, A/F was calculated from the intensity ratio of CH*/C₂*. Figure 5 shows a schematic of the relationship between the measurement position and the flame propagation around the spark plug in a cylinder. A flame kernel formed from the mixture gas was ignited by the spark plug, and emission radicals were detected when the flame passed through the measurement point in the cylinder. The detection timing of the emission radicals was the time obtained by detecting the maximum value of the emission intensity as

shown in figure 4. The measurement area was a $6.28 \times 10^{-3} \text{ mm}^3$ space approximately 5 mm from the center electrode of the spark plug in the opposite direction of the ground electrode.

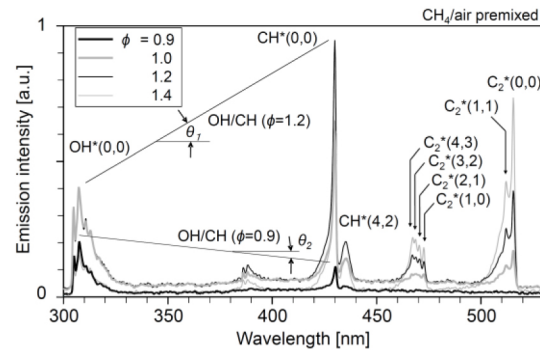


Figure 4. Flame spectra for different equivalence ratio

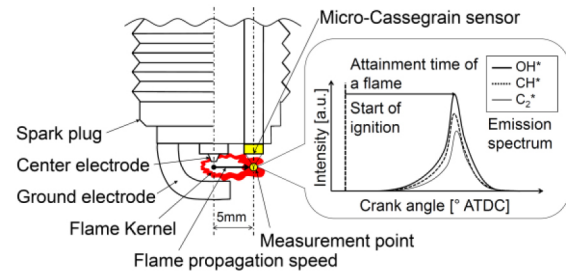


Figure 5. Schematic of the measurement position and parameter of micro-Cassegrain sensor

MEASUREMENT RESULTS

The in-cylinder pressure measurement results for 100 consecutive cycles of each cylinder under the operating conditions listed in Table 1 are shown in Figure 6. In the case of the operating conditions for this test engine, the variation in the in-cylinder pressure for each cycle could be confirmed for any cylinder, and there were differences in the variations for the cylinders. In particular, the variation of No. 3 was larger than that of the other cylinders. Figure 7 shows the measurement results for the air-fuel mixture concentration near the spark plug under the same operating conditions. The horizontal axis represents the time that the sensor detected the light emission. The spark-discharge period was -33° ATDC to -6° ATDC , but the ignition was detected on the continued discharge after the end. When the traveling direction of the flame and the ignition positions were the counter direction toward the sensor, the arrival of the flame to the sensor were delayed. Thereby, the detection time of the radical emission was delayed. The air-fuel mixture concentration varied from cycle to cycle for each cylinder. The variation range of A/F for the air-fuel mixture concentration of No. 3 was larger than those for the other cylinders, just as with the measurement results for the in-cylinder pressure.

Figure 8 shows a comparison between the indicated mean effective pressure (IMEP) in each cylinder and the cycle-averaged equivalence ratio of the mixture for each cylinder using micro-Cassegrain sensor. The IMEP of No. 3 was higher

than the values for the other cylinders, with a value that was 4.7 % higher than the lowest IMEP (No. 1). The IMEP values for No. 2 and No. 4 were almost the same at 1.2 % and 1.6 % higher than that for No. 1, respectively. Against these trends, the mean equivalence ratio of No. 3 was the closest to the stoichiometric equivalence ratio, with a value that was 5.6 % higher than the lowest value (No. 1). The values for No. 2 and No. 4 were almost the same at 1.1 % and 1.5 % higher than that of No. 1, respectively. Based on this result, the average equivalence ratio of No. 3, which had the highest IMEP, was the closest to stoichiometric equivalence ratio, and the average equivalence ratio of No. 1, which had the lowest IMEP, was the smallest compared to the stoichiometric equivalence ratio. Thus, IMEP was strongly influenced by the mean concentration of the air-fuel mixture.

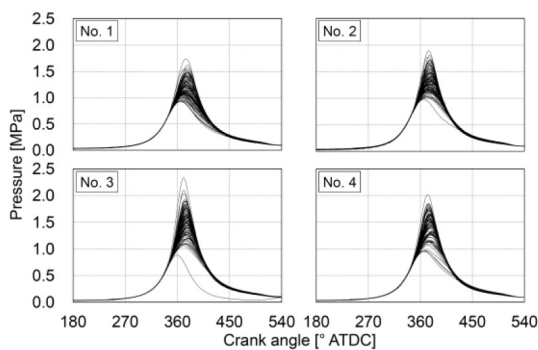


Figure 6. Measurement results of in-cylinder pressure of 100 cycles for each cylinder

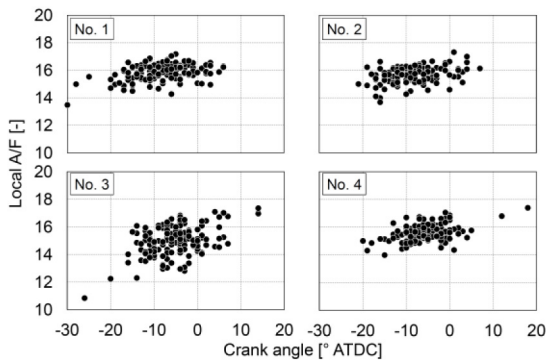


Figure 7. Measurement results of A/F near the spark plug of 100 cycles for each cylinder by using micro-Cassegrain sensor

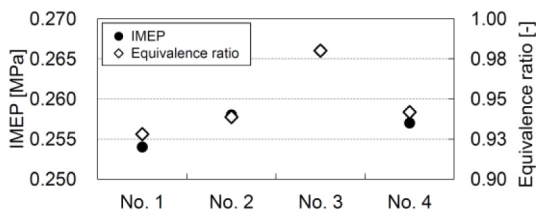


Figure 8. Comparison of cycle-averaged value between IMEP and equivalence ratio in each cylinder

In order to compare the cycle-to-cycle variations tendencies, the coefficient of variation for IMEP (COV_{IMEP}) was calculated. The equation is shown below.

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{\text{meanIMEP}} \times 100 \quad (1)$$

COV_{IMEP} is the value obtained by dividing the standard deviation of IMEP by the cycle-averaged IMEP [1]. The coefficient of variation for the equivalence ratio (COV_{ϕ}) was calculated using equation (2) in a calculation similar to equation (1).

$$COV_{\phi} = \frac{\sigma_{\phi}}{\bar{\phi}} \times 100 \quad (2)$$

Here, σ_{ϕ} represents the standard deviation of the equivalence ratio, and $\bar{\phi}$ represents the cycle-averaged equivalence ratio. Figure 9 shows the result of a comparison between COV_{IMEP} and COV_{ϕ} . A comparison of the COV_{IMEP} values for the cylinders shows that No. 3 was the highest, with a value that exceeded 10 %. In general, when COV_{IMEP} exceeded 10 %, it was determined that the level of noise and vibration in the engine was a problem. The COV_{IMEP} values of the other cylinders were almost the same, with values ranging from 6.7 % to 7.3 %. On the other hand, the high COV_{ϕ} of No. 3 varied from cycle to cycle by approximately 10 %. Other than No. 3, the coefficient of variation values for the other cylinders was almost the same (3.7 %-4.1 %). In addition to the results of the IMEP and cycle-averaged equivalence ratio comparisons for the cylinders, because the tendencies for the COV_{IMEP} and COV_{ϕ} comparisons were also the same, it was found that the combustion variation was strongly affected by the cycle-averaged air-fuel mixture concentration and its variation.

Based on these results, the authors considered the following factors for the variation of A/F near the spark plug. Because the measurement position of the micro-Cassegrain sensor was the same for each cycle, the air-fuel mixture varied spatially. Two phenomena were considered in relation to this variation of the air-fuel mixture. The first was the variation of the air-fuel mixture for each cycle. It was considered that there was variation of the slight position of a flow pattern similar to cycle-averaged flow field. The second was the movement of the air-fuel mixture during the spark-discharge period. The air-fuel mixture was moved by the flow in the cylinder, and was ignited by the spark discharge when the ignitable conditions such as the A/F and residual gas concentration were satisfied.

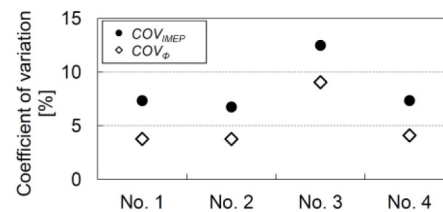


Figure 9. Comparison of coefficient of variation between IMEP and equivalence ratio in each cylinder

