Heat Loss into Combustion Chamber Wall
of 4-stroke Gasoline Engine*
(2nd Report, Heat Loss into Cylinder Head,
Intake and Exhaust Valves)

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Following the surface temperature variations and instantaneous heat flux in each cycle in the piston and the cylinder wall mentioned in the first report, the authors tried to verify them in the cylinder head and the suction/exhaust valve of a 4-stroke gasoline engine combustion chamber.

The authors could make clear the heat flux distribution and heat loss conditions through the cylinder head and the valve surface for each stroke.

In addition to the results as described in the first report, the heat loss on the entire combustion chamber wall against the quantity of supplied heat, namely, the ratio of heat transfer was known.

Key Words : Internal Combustion Engine, Gasoline Engine, Heat Transfer, Thin Film Thermocouple, Cylinder Head, Valve

1. Introduction

In the first report⁽¹⁾, the distribution of heat fluxes and conditions of heat losses to the combustion chamber wall surface were clarified through the use of unique and precise thin film thermocouples⁽²⁾⁽³⁾ developed by the authors et al. and by determinations of instantaneous temperatures of the piston and cylinder liner which constitute the combustion chamber wall surfaces, together with the determination of the instantaneous heat flux flowing into the combustion chamber of a 4-stroke gasoline engine during a combustion stroke.

Through this study, the distribution of heat fluxes for each stroke of a cycle and conditions of individual heat losses by the rest of cylinder head, and intake/exhaust valves exposed to the gas in the combustion chamber have been clearly determined through the application of the measuring method mentioned above, and measurements of such components.

Conditions of heat losses to the entire wall surface of the gasoline engine combustion chamber have been also determined by putting together the experimental results obtained through this study and those obtained in the first report.

2. Experimental Apparatus and Measuring Method

The engine used in the experiment was a water-cooled, 4-stroke CA-18S engine made by Nissan, the same as one used in the previous experiment.

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2.1 Thin film thermocouples

Loex-constantan thermocouples with the body material of aluminum alloy Loex were used, since the material of the cylinder head was a high silicon content aluminum alloy, similar to the material of the piston.

The intake valve of the engine was made of a heat resisting steel SUH11 (the thermal conductivity $\lambda = 25 \text{ W/mK}$ and the temperature diffusivity $a = 0.025 \text{ m}^2/\text{h}$), while the exhaust valve was made of a heat resisting steel SH36 (the thermal conductivity $\lambda = 21 \text{ W/mK}$ and the temperature diffusivity $a = 0.021 \text{ m}^2/\text{h}$). Therefore, constantan (the thermal conductivity $\lambda = 23 \text{ W/mK}$ and the temperature diffusivity $a = 0.023 \text{ m}^2/\text{h}$), with similar thermal properties to those of above mentioned materials, was used as the body material, in order to produce iron-constantan thin film thermocouples for the measurement of the intake and exhaust valves. The thermocouples were then embedded into intake and exhaust valves, to ensure a measurement of high accuracy.

The thickness of the insulation layers of the thin film thermocouples to measure valve temperatures was made slightly greater (about $25 \mu\text{m}$) than usual, since the conventional insulation material, heat-hardening resin, did not have sufficient durability, and the break down of the insulation layer could occur, as the maximum temperature of the intake valve would reach as high as 400°C or so, while that of the exhaust valve would rise as high as 800°C . Despite the thickness of the insulation layers, however, appropriate measurements were ensured by using a ceramic base adhesive to fix the thermocouples in the valves.

As for the plating of the thermocouples, a pretreatment by the application of chemical copper plating over the ceramic

base adhesive was done first. The sensitizer treatment was then carried out by applying absorption type stannous ion onto the ceramic surface that had been made hydrophilic by the chemical etching. The activator treatment was also done by the deposition of metallic palladium, with a catalytic function, onto the nonconductor in order to increase the adhesion force of the chemical copper. The desired thickness of the plating was ensured by means of electroplating.

Fig. 1 shows an iron-constantan thin film thermocouple with the constantan body material, which is embedded into a valve.

2.2 Measuring locations and methods

The locations of measurement on the cylinder head were 15 points in all as shown in Fig. 2, considering the ease in embedding thermocouples and leading out the thermocouple wires.

Fig. 3 shows locations of 7 measuring points on the intake valve and 7 others on the exhaust valve, together with the method to lead out thermocouple

Fig. 1 Iron-constantan thin film thermocouple with constantan as the body material for the measurement of valve temperatures

Fig. 2 Measuring locations on cylinder head

wires from each valve. Namely, the wire of each thermocouple was led through a 2 mm diameter hole bored along the center of each valve, then led out of the hole bored sidewise immediately below the surface contacting the rocker arm. A lateral hole of 1mm diameter was bored at a location normal to the above mentioned hole, and one end of a spring wire was fixed to this hole.

A spring wire fixed to the cylinder head bridged a loop between the cylinder head and the valve. The wire of each thermocouple was led along this spring wire to the fixing point, which could be led outside the engine. The spring wire acted also as a valve rotation stopper. By the method described above, the measurement of valve temperatures up to high engine speed becomes feasible.

Engine operating conditions were the same as those in the first paper, 8 different conditions of 2000, 3000, 4000 and 5000rpm with the full load, 1/4, 1/2 and 3/4 loads at 3000rpm, and the knocking condition when a lower octane number fuel (RON-82) than the normal fuel (RON-92) was used with the full load at 2000rpm.

During the engine operation, the cooling water temperature at the engine outlet was kept constant at 80°C, while the temperature at the lubricating oil strainer inlet was kept constant at 90°C.

Thermocouple wires led out of the engine were sealed to prevent electric noise, and the thermal electromotive force caused by a temperature difference between hot and cold junctions was so amplified by a DC amplifier that one division on oscilloscope would become 5°C, and measured values were recorded on a data recorder. At the same time, the gas pressure P in the combustion chamber and the time mark at the bottom dead center were recorded. Temperatures at cold junctions were also recorded on dot printing

Fig. 3 Measuring points on intake and exhaust valves and thermocouple wire leading method

type thermometers.

3. Results of Measurements

Fig. 4 shows results of measurements at points 2, 4 and 5 on the cylinder head surface, at 2000, 3000 and 5000rpm with full load. It is seen in the figure that the higher becomes the engine speed, the greater become the temperature difference ΔT_0 between hot and cold junctions and the heat flow rate at each measuring point. $\bar{T}_w$ represents mean surface temperature of each measuring point.

Fig. 5 shows the results obtained at the same measuring points under each load of 3000rpm.

Fig. 6 shows the measuring results under knocking condition at 2000rpm of full load. Similarly to the case of piston in the first report, the temperatures of H2, H4 and H5 rise as soon as a knocking occurs. Then the temperature differences ΔT_0 between hot and cold junctions and temperature amplitudes become larger than in the normal combustion operation in Fig. 4.

Fig. 7 shows the results with peculiar temperature variation which is measured at the point 1 around the exhaust

valve seat on the cylinder head. In addition to the occurrence of the temperature waveform peak A caused by the combustion, B and C peaks also appeared. They were due to the open-close motion of the exhaust valve shown in the figure. The B-peak started to rise immediately after the opening of the exhaust valve, presumably due to the fact that the heat transfer coefficient become greater due to a flow of the exhaust gas around the valve seat. For the C-peak, on the other hand, it is assumed that the radiation of heat from the exhaust valve which becomes hot because of the exposure to the high temperature gas, as evidenced by the fact that it starts to rise immediately before the valve seating.

Fig. 8 shows the results of measurements at the point 1 (IV-1 and EV-1) of intake and exhaust valves at 2000, 3000 and 5000rpm with the full load. It is found from the figure that the quantity of heat flowing into the intake valve is greater than that of the high temperature exhaust valve.

Fig. 4 Surface temperature variations (full load) at measuring points 2 (cold junction depth $l=4.9\text{mm}$), 4 ($l=4.65\text{mm}$) and 5 ($l=3.35\text{mm}$) on cylinder head at each engine speed; $\bar{T}_w$ represents the mean surface temperature

Fig. 5 Surface temperature variations (3000rpm) at measuring points 2 (H2), 4 (H4) and 5 (H5) on cylinder head at each load; $\bar{T}_w$ represents the mean surface temperature

Fig. 6 Example of measurement under knocking condition (2000rpm with the full load) $\bar{T}_w$: the mean surface temperature

Fig. 7 Extraordinary phenomena of transient surface temperature on cylinder head near exhaust valve seat (point 1)

Fig. 8 Surface temperature variations at point 1 on intake valve (IV-1, $l=3.00\text{mm}$) and point 1 on exhaust valve (EV-1, $l=4.00\text{mm}$) at each engine speed; $\bar{T}_w$: the mean surface temperature

4. Analytic Results of instantaneous Heat Flux and Heat Loss

The analysing method of the instantaneous heat flux was employed which is the one described in the first report. From the temperature difference at each crank angle of each measured result, the instantaneous heat flux at each crank angle was calculated. The mean heat flux q in a cycle was also determined.

Fig. 9 shows results of the analysis for the measuring point 5 of the cylinder head at 3000rpm with full load. It is found from the figure that most of the heat flows into the cylinder head during the working stroke. A peak appears at the latter half of the working stroke, due to the flow of the exhaust gas, as H5 is located near the exhaust valve seat.

The instantaneous heat flux at the measuring point 1 located at the center of the intake valve at 3000rpm with full load

Fig. 9 Example of analysis for instantaneous heat flux on cylinder head (H5) surface

Fig. 10 Example of analysis for instantaneous heat flux on intake valve(IV-1) surface

Fig. 11 Example of analysis for instantaneous heat flux on exhaust valve(EV-1) surface

Fig. 12 Distribution of heat fluxes on cylinder head and intake/exhaust valve surface at each stroke

is shown in Fig. 10, while the instantaneous heat flux at the measuring point 1 located at the center of the exhaust valve is shown in Fig. 11.

The comparison of both results reveals that the exhaust valve temperature is higher than the intake valve temperature, resulting in the loss of a considerable amount of heat during the suction and compression strokes. It is also found that the rise of the instantaneous heat flux is steep, and the peak value is also high. This is thought to be due to a sudden increase of the heat transfer coefficient. These results are quite similar to the valve at the centers of intake and exhaust valves measured by

Harigaya and others⁽⁴⁾ with a prechamber type diesel engine.

Fig. 12 shows the distribution of heat fluxes at the surfaces of the cylinder head, the intake and exhaust valves. The heat flux distribution in the working stroke is close, because a great amount of the heat flows in the stroke. Looking at the total results (e) of all strokes, the maximum heat flux is found near the plug A (1-point ignition at the plug A alone with full load), which is different from the case of the piston. This is probably due to influences of the irregular surface of the cylinder head, complex structure of the coolant passage, locations of

plug, intake and exhaust valves, etc. The above mentioned phenomenon is natural as far as intake and exhaust valves are concerned, considering the heat flux distribution on the cylinder head.

Fig. 13 shows the effects of engine speed and knocking on the quantity of heat flowing into the cylinder head at the time of full load and the mean heat flux for each stroke. What is different from the case of piston is that the heat flow rate during the exhaust stroke is nearly the same as that during the compression stroke. This is considered to be attributable to the flow of exhaust gas shown in Fig. 7. Namely, the cylinder head is more affected by the flow of exhaust gas than the piston in the

Fig. 13 Heat flow rate into cylinder head from combustion chamber gas for each stroke at full load

Fig. 14 Heat flow rate and mean heat flux into cylinder head wall surface from combustion chamber gas for each stroke at partial load

exhaust stroke. The knocking also affects the increases of the heat flow as in the case of piston. This tendency agrees very well with the experimental results obtained by Harigaya and others (4), taking account of the difference in the range of engine speeds.

Fig. 14 shows the effect of partial loads. As compared with the case of piston, the effect of rapid combustion by means of 2-point ignition at 1/2 or 3/4 load is nil. As in the case of full load, the quantity of heat flowing into the cylinder head in the exhaust stroke is nearly equal to that in the compression stroke.

Fig. 15 shows the heat flow rate and the heat flux into the intake valve for each stroke at full load. The total heat flux of all strokes is smaller by 40% than that of the cylinder head.

Fig. 16 shows the effect of partial load.

Fig. 15 Heat flow rate and mean heat flux into intake valve from combustion chamber gas for each stroke at full load

Fig. 16 Heat flow rate and mean heat flux into intake valve from combustion chamber gas for each stroke at partial load

Fig. 17 Heat flow rate and mean heat flux into exhaust valve from combustion chamber gas for each stroke at full load

Fig. 18 Heat flow rate and mean heat flux into exhaust valve from combustion chamber gas for each stroke at partial load

Fig. 17 shows the heat flow rate and the mean heat flux into the exhaust valve at full load. Since the temperature of the exhaust valve is higher than that of the intake valve, the heat radiates from the valve to the combustion chamber during suction and compression strokes. As a result, the total heat flow of all strokes (in a cycle) into the valve is smaller than that during the working stroke.

Fig. 18 shows the effect of partial loads. The comparison of heat losses between the intake and exhaust valves reveals that the total heat loss of the intake valve for all strokes is about 2 to 3 times that of the exhaust valve, at both full and partial loads. This should be a reasonable result compared with those obtained by Harigaya and others⁽⁴⁾, considering the difference in engine.

Fig. 19 shows the heat transfer ratios of the heat losses of the cylinder head, intake and exhaust valves against respective heat supply values, which are findings of this study, in addition to those of the piston and cylinder found through the first report. As for the

Fig. 19 Heat transfer ratios into piston, cylinder wall, cylinder head and valves due to engine speed under full load

total heat loss in a cycle, that of the cylinder head is 2.5 to 3.5%, which is greater than that of the piston. This results should be appropriate, considering the complex heat flow passage, such as the water cooling mechanism, intake and exhaust ports, etc., along with the fact that the surface area of the cylinder head is narrower by 12.0% or so than that of the piston. The heat loss under the knocking condition is 4.5%. The sum of heat losses of the intake and exhaust valves is in the range of 0.5 to 0.8%, while the sum at the time of knocking is 1.0%.

The total heat loss of all components including the piston and cylinder walls described in the first report amounts to 7.0 to 10.0%. The heat loss of the cylinder head unit including the intake and exhaust valves is 3.0 to 4.3%, which is considerably smaller than the results obtained by Woschni and others⁽⁵⁾⁽⁶⁾. This results should be, however, appropriate, considering the fact that the cooling loss ratio of the cylinder head unit by Woschni et al. includes heat losses through the cylinder head wall surface, intake/exhaust valves, intake/exhaust ports and intake/exhaust pipe flanges.

Fig. 20 shows the effect of partial loads. The total of the loss in cycle is 3.0 to 6.3% for the cylinder head, which is nearly the same as that of the piston. This is due to the less effect of rapid combustion by 2-point ignition

at partial loads on the cylinder head, than on the piston. The sum of heat losses through the intake and exhaust valves is in the range of 0.6 to 1.2%, and the total loss of the combustion chamber wall surface is 8.7 to 16.5%.

5. Conclusions

Measurements of surface temperature variations of different components on the combustion chamber walls have been performed with the selection of appropriate body materials to match each component whose temperature is to be measured, and with the use of unique thin film thermocouples of high measuring accuracy developed by the authors.

As a result, individual states of heat flux distributions and heat losses of the cylinder head, intake and exhaust valves of each stroke have been clearly determined. Furthermore, ratios of heat losses of different components to the entire wall surface of the combustion chamber of a 4-stroke gasoline engine have been clearly indicated, by putting together the experimental results obtained through this study.

Consequently, the following findings have been obtained.

- (1) The ratio of the heat loss to the cylinder head excluding valves is 2.5 to 3.5% at full load, while that at partial loads is in the range of 3.0 to 6.3%.
- (2) The sum of the heat losses of intake/exhaust valves is 0.5 to 0.8% at full load, while that at partial loads is 0.6 to 1.2%.
- (3) Therefore, the heat loss to the cylinder head surface including intake and exhaust valves is 3.0 to 4.3% at full load, while that at partial loads is 3.6 to 7.5%.
- (4) The total amount of the cooling loss to the entire wall surface of the combustion chamber amounts to 7.0 to 10.0% at the full load, while it is 8.7 to 16.5% at partial loads.
- (5) Under the knocking condition produced by the use of a low octane number fuel, the ratio of heat transfer increased during the working stroke the same as in cases of the piston and the cylinder walls. The ratio of the total heat loss in a cycle is 4.5% for the cylinder head, 1% for intake/exhaust valves, and 13.5% for the entire combustion chamber wall surface.
- (6) Neither the cylinder head nor intake/exhaust valves were affected by rapid combustion with the 2-point ignition at partial loads.
- (7) The heat transfer coefficient increased due to a flow of the exhaust gas, and a peak appeared in the surface temperature variation around the exhaust valve seat on the cylinder head, immediately after the exhaust valves had opened. As a result, the heat flow rate also increased. Another peak also appeared, due to the heat radiation from high temperature exhaust valve.

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Fig. 20 Heat transfer ratios into piston, cylinder wall, cylinder head and valves due to partial loads under 3000rpm