A Study on the Unsteady Temperature Field of Combustion Chamber Wall in a Turbocharged Gasoline Engine

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To design and develop a turbocharged engine, it is necessary that a lot of studies should be done to find the characteristics of engine performance and thermal flow. To accomplish this purpose, turbocharger equipped to a naturally aspirated gasoline engine was utilized. A thin-film type temperature probe was made and installed onto the combustion chamber wall to measure unsteady temperature. The unsteady heat flux at combustion chamber wall was evaluated by one dimensional unsteady conduction equation.

Key Words: Thin-Film Type, Thermal Flow, Instantaneous Temperature, Hot Junction, Cold Junction, Temperature Swing, Heat Flux, Surface Temperature

1. Introduction

In the design and development of reciprocating internal combustion engines it is frequently desired to calculate the heat transfer rate at the surfaces of the combustion chamber. Often, an approximate estimate of time average heat transfer rate to the surface is sufficient, but there are many cases for which a detailed analysis of instantaneous surface heat flux during the cycle is essential (Annand, 1963).

The process of transient heat transfer in the combustion chamber of spark ignition engines has not been adequately studied experimentally, although its importance in combustion simulations and analyses was well recognized by Tabaczynski (1979) and Krieger (1980) etc.

Past studies of transient heat transfer in spark ignition engines include those of Overbye et al. (1967), and Alkidas (1980). Overbye et al. measured the heat flux at several positions on the cylinder head of a CFR engine. Tests were performed at near-stoichiometric air-fuel ratio and an engine speed of 830 rpm. Alkidas (1980) measured the transient heat flux at four locations on the cylinder head of a spark ignition engine. Kevin (1986) measured the transient heat flux at each position on the cylinder head of a spark ignition engine. The influence of engine speed and spark timing on the heat flux were also examined. This study showed that the initial rate of increase of the heat flux at each position of measurement was correlated with the calculated time of flame arrival. In addition, it was found that the measured heat flux varied considerably with the measurement positions.

Assanis and Badillo (1989) suggested an unsteady heat flux measurement by using thermocouple in metal and ceramic engines. Furuhama and Enomoto (1987) reported heat fluxes into ceramic combustion wall of internal combustion engines. Xue et al. (1994) have studied unsteady temperature field and thermal stress of the cylinder head in the gasoline engine. Although heat flux in the cylinder of spark ignition engines had been studied for a long time, it was not obvious on many aspects.

To improve the performance of turbocharged gasoline engine, we must study the occurrence of

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abnormal combustion such as knocking, and solve the problem of thermal loading, (Lee et al, 1991). However, a few experimental data on transient heat flux processes in turbocharged gasoline engines are presently avaiable.

We fabricated a temperature probe which could measure the instantaneous temperature change at combustion wall. By using self-made high response thin-film probes which was installed on the inside surface of combustion chamber, we measured the abnormal temperature excursion and thermal flow depending on the variation of engine speed and compression ratio in a turbocharged gasoline engine.

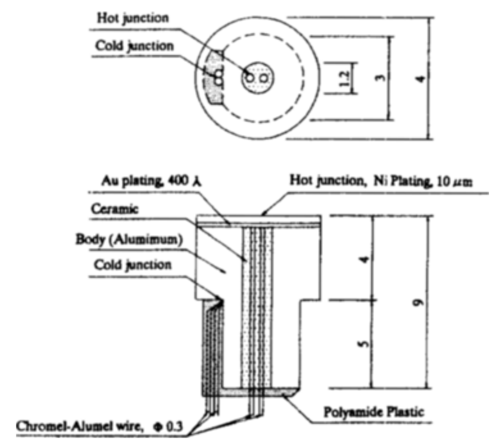

Fig. 1 Instantaneous temperature probe used K type thermocouple

2. Instantaneous Temperature Probe

To clarify the temperature swing, mean heat flux and the maximum heat flux, the unsteady temperature of combustion wall should be measured by the instantaneous temperature probe. To measure the rapidly varying instantaneous temperature of the cylinder wall, the temperature probe of a thin-film type was employed. The instantaneous temperature at the cylinder wall surface and the temperature at the position with a certain depth apart from the wall surface must be measured simultaneously to know the unsteady heat flux at each crank angle. The temperature probe for hot junction must have high responsibility because the temperature of the cylinder wall surface changes rapidly. But the temperature probe for cold junction does not require high response than the temperature probe for hot junction because the temperature at the position of cold junction does not change rapidly almost during a cycle. Because of the high response at the hot junction, it has a small diameter of wire and a thin thickness of film for rapid response.

Figure 1 shows the structure of the temperature probe. A 1.2mm hole was made in the center of adapter. K type thermocouple of 0.3mm was inserted through the hole. The adhesive between wire and adapter was the ceramic the insulation.

The cold junction was made with the point welding because it does not suffer from tempera-

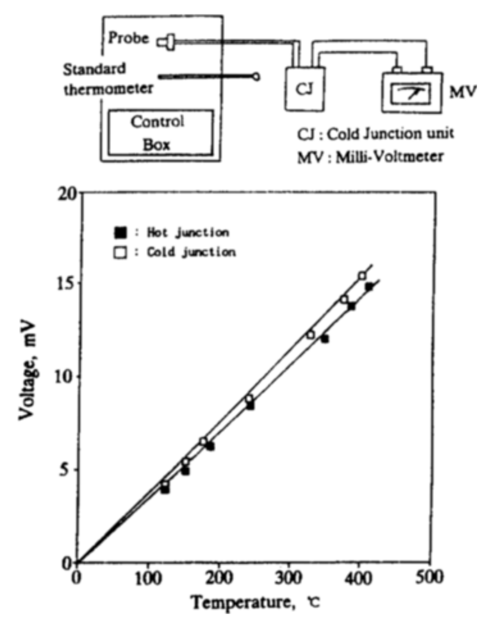

Fig. 2 Calibration diagram of surface temperature probe

ture variation. But it was very hard to wear the conductive thin film.

It was coated with nickel plating of 10μm and with gold plating of thickness of 400 Å. Through the experimental data of coating thickness parameter, high response of the temperature probe below 10μm of coating thickness was obtained. The lower end of probe was connected

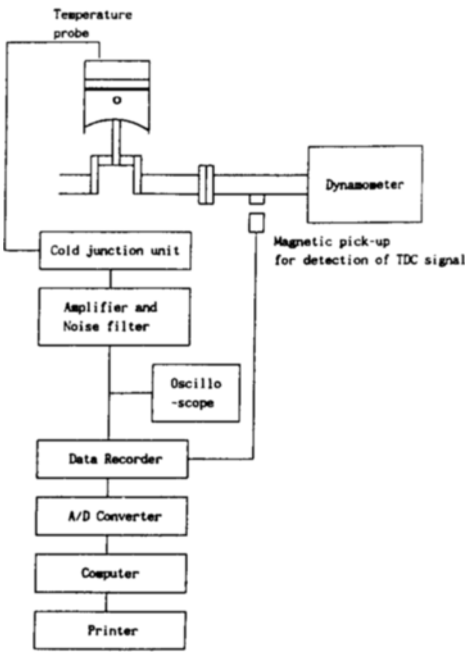

Fig. 3 Schematic diagram of temperature measuring system

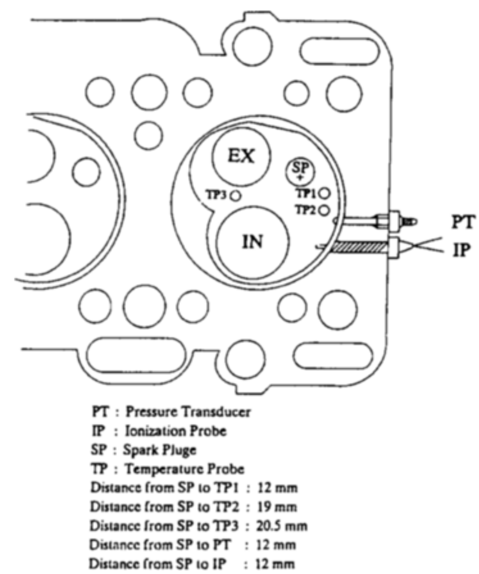

Fig. 4 Inserted locatioin of temperature probe, ionization probe and pressure transducer

Table 1 Engine operating Conditions

Air fuel ratio	15
Spark timing	for the best torque
Compression ratios	9.7, 9.2, 8.5, 7.8
Cooling water temperature	80 ℃

with cooling water. The poly-amide plastic of water-resisting and heat-resisting property can protect the percolation of water near the lower end of probe.

Figure 2 shows the calibration curve. The conversion factors of hot junction and cold junction are 26.7 ℃/mV, 25.5 ℃/mV respectively.

Figure 3 shows the schematic diagram of temperature measuring system. The output voltage from probe passes through cold junction unit, noise filter and charge amplifier and it was supplied to the data recorder. This data goes through A/D converter of direct memory type that can be analyzed the range of 0~5 voltage to 4096 division into equal parts.

3. Experimental Procedures

In this study, a four cylinder gasoline engine of carburetor type was used. Compression ratio of the engine is 9.7 and displacement volume is 1328 cc. To measure the surface temperature on the cylinder head, three thin-film thermocouple probes were installed on the cylinder head. Figure 4 shows the positions of temperature probes installed. The distances from spark plug position are 12 mm, 19 mm, and 20.5 mm respectively.

Engine was operated under the full throttle valve with variable compression ratios. Table 1 shows the engine operating conditions. For such engine operating conditions, the characteristics of the mean temperature, the maximum temperature, temperature swing, mean heat flux, and the maximum heat flux were measured and analyzed in this study.

4. The Analysis of Heat Flux Charactreistics

4.1 Heat flux equation

To calculate the heat flux through the wall of the combustion chamber, the temperature gradient in the direction of wall thickness must be known. Therefore the temperature at arbitrary position is represented by a Fourier series with time t and distance x which is the vertical direction of combustion chamber wall.

Assuming that the heat on the combustion chamber wall flows into the vertical direction only, heat conduction equation can be written as

$$\frac{\partial T}{\partial t}=\alpha\frac{\partial^2 T}{\partial x^2} \tag{1}$$

Boundary conditions are as follows:

$$T(0,\ t)=T_w(t)$$
$$T(\delta,\ t)=T(\delta)$$

where, $T_w(t)$ is instantaneous temperature of wall surface, $T(\delta)$ is steady-state temperature at the arbitrary depth δ.

Fourier conversion represents unsteady data as a series with the use of the periodic function. The surface temperature is represented by a Fourier series form as follows:

$$T_w(t)=\bar{T}_w+\sum_{n=1}^{N}(A_n\cos n\omega t+B_n\sin n\omega t) \tag{2}$$

where, $\bar{T}_w$ is the time-averaged surface temperature, A_n and B_n are the coefficients of the Fourier series such as

$$\overline{T_w}=\frac{1}{2\tau}\int_0^{2\tau}T_w(t)\,dt$$
$$An=\frac{1}{\tau}\int_0^{2\tau}T_w(t)\cos n\omega t dt$$
$$B_n=\frac{1}{\tau}\int_0^{2\tau}T_w(t)\sin n\omega t dt$$

By using boundary conditions, the temperature as a function of distance x and time t is represented by

$$T(x,t)=T_w-[\overline{T_w}-T(\delta)]\frac{x}{\delta}+\sum_{n=1}^{N}\exp[A_n\cos(n\omega t-F_x)+B_n\sin(n\omega t-F_x)] \tag{3}$$

where,

$$F=\sqrt{\frac{n\omega}{2\alpha}}$$

The corresponding surface heat flux Qw(t) is as follows:

$$Q_w(t)=-K_w\frac{\partial T}{\partial x}(0,t)=\frac{K_w}{\delta}[\overline{T_w}-T(\delta)]+K_w\sum_{n=1}^{N}\sqrt{\frac{n\omega}{2\alpha}}[A_n(\cos n\omega t-\sin n\omega t)+B_n(\sin n\omega t+\cos n\omega t)] \tag{4}$$

The first term in the surface heat flux Qw(t) is independent of time and it represents the steady-state component of heat flux. The second term is time dependent and it represents the unsteady component of heat flux.

The thermal conductivity of wall surface Kw varies according to the temperature of materials. The instantaneous temperature of combustion chamber surface and the mean temperature of arbitrary depth were measured, and the heat flux was calculated according to the thermal conductivity using this mean temperature.

To calculate the steady state component of heat flux in combustion chamber surface, the mean temperature of wall surface must be known. The measured temperature of combustion chamber surface by the temperature probe is the unsteady surface temperature, therefore the mean temperature of wall surface by using the ensemble average can be obtained as follows:

$$\overline{T_w}=(1/720)\sum_{\theta=1}^{720}[(1/N)\sum_{i=1}^{N}T(\theta,i)]$$

where, $\overline{T_w}$: time-averaged surface temperature, K

N : number of measured cycle

$T(\theta, i)$: instantaneous temperature, K

θ : crank angle, deg

i : arbitrary cycle

4.2 The characteristics of thermal flow with the variation of engine speed and compression ratio

Figure 5 shows the instantaneous temperature of wall surface and the temperature of 4mm distance from wall surface for engine speeds of 2000 rpm, 3000rpm and 4000rpm. Figure 6 shows the temperature swing and the difference of maximum and minimum instantaneous temperature as a function of engine speed at each compression ratio without intercooler. Figure 7 represents the

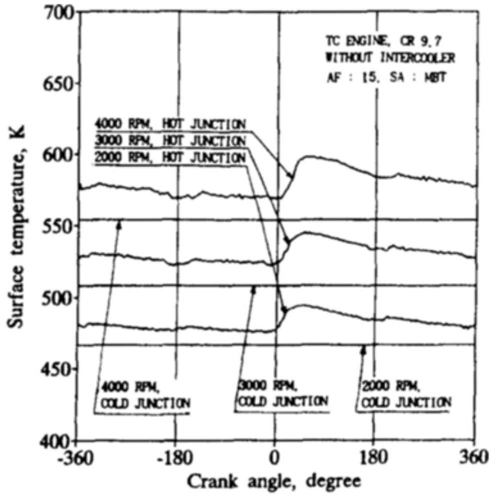

Fig. 5 Surface temperature as a function of crank angle

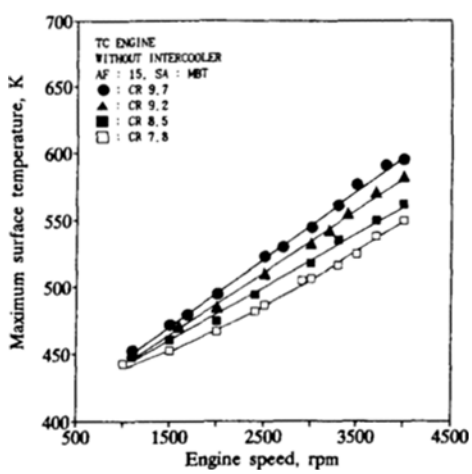

Fig. 6 Temperature swing as a function of engine speed

maximum surface temperature as a function of engine speed at each compression ratio engine without intercooler.

In Fig. 8, heat fluxes are shown as a function of crank angle and engine speed without intercooler at the compression ratio of 9.7. The heat fluxes keep nearly constant during compression, exhaust and intake stroke, but it rapidly changes during explosion and expansion stroke in which almost all the heat release.

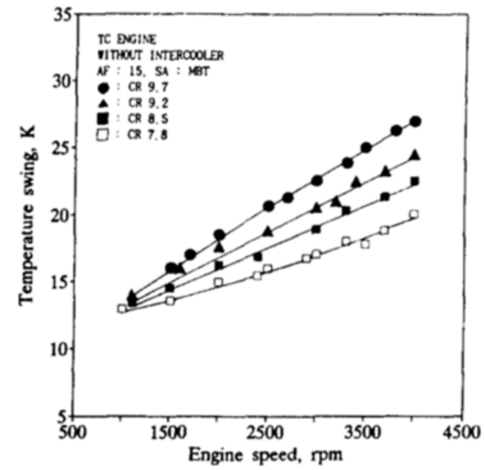

Fig. 7 Maximum surface temperature as a function of engine speed

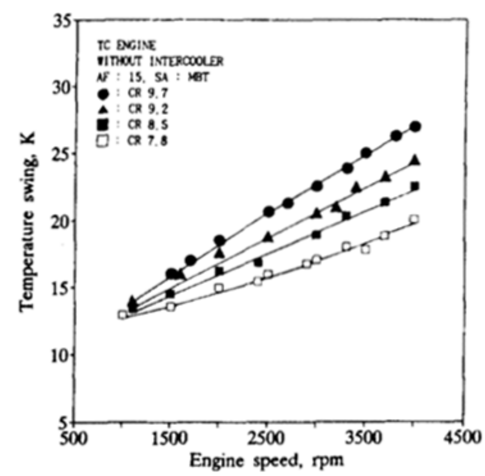

Fig. 8 Heat flux as a function of crank angle

As shown in Fig. 5 and Fig. 8, the temperature difference between hot and cold junction and the temperature swing increase as the engine speed increases. As a result, the heat fluxes increase, and the peak point of heat flux appears late as the engine speed increases.

Figure 9 shows heat loss per heat energy supplied as a function of brake mean effective pressure without intercooler at compression ratio of 9.7. The heat loss of cooling water was calculated from the entrance and exit temperatures of cooling water, and the heat loss of combustion cham-

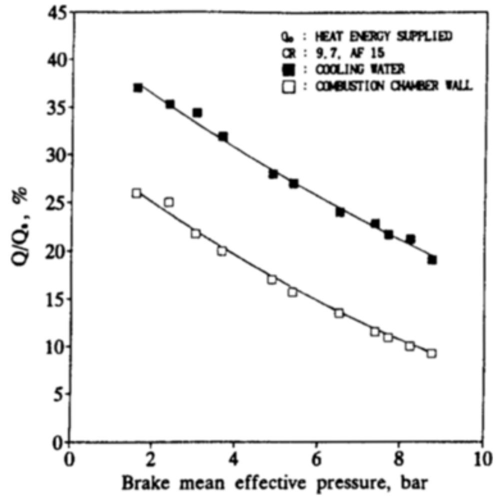

Fig. 9 Heat loss per heat energy supplied as a functioin of brake mean effecitive pressure

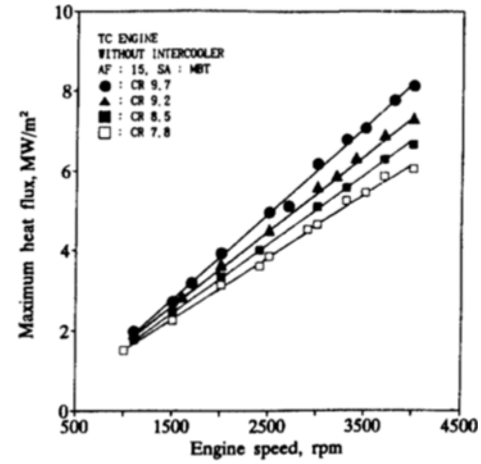

Fig. 10 Maximum heat flux as a function of engine speed

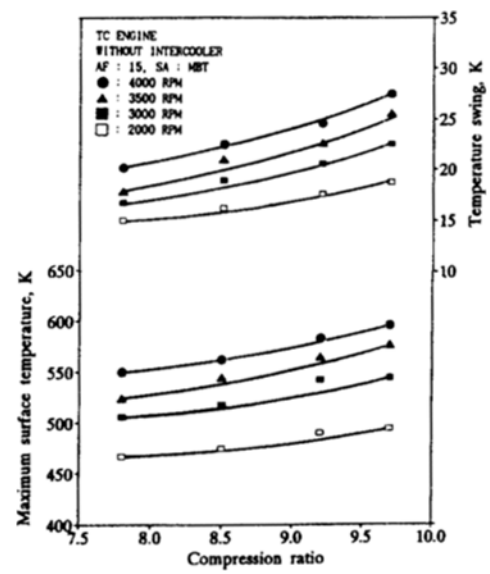

Fig. 11 Maximum surface temperature and temperature swing as a function of compression ratio

ber surface was calculated from the space mean method of mean heat flux in each point and multiplication of combustion chamber surface. The rate of heat loss of combustion chamber surface and cooling water decrease according to the increase of load. It indicates that the heat losses include the heat loss of combustion chamber surface, cylinder wall surface, and piston. The rate of heat loss of combustion chamber wall shows about 10 to 25% according to the change of load and it corresponds to about 50% of heat loss rate of cooling water. It is known that about 50% of heat loss by cooling water transfers from the combustion chamber wall to the cooling water.

Figure 10 shows maximum heat flux as a function of engine speed at each compression ratio without intercooler. In this figure, the maximum heat flux represents a maximum value among instantaneous heat fluxes during a cycle. The heat flux of combustion chamber wall is unsteady state heat flow. In the design of combustion chamber to reflect the thermal loading of combustion chamber wall, the instantaneous maximum heat flux of combustion chamber wall is an important factor. As engine speed and compression ratio increase, maximum heat flux of combustion chamber increases because heat release and combustion temperature increase. Typical maximum heat flux shows about 8.13 MW/m^2 at the compression ratio 9.7 and engine speed 4000 rpm.

Figure 11 shows the maximum surface temperature and temperature swing as a function of compression ratio at each engine speed in turbocharged engine without intercooler. As the compression ratio decreases, the maximum surface temperature and the temperature swing decrease.

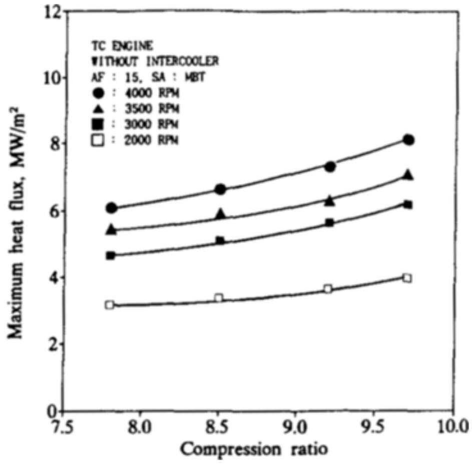

Fig. 12 Maximum heat flux as a function of compression ratio

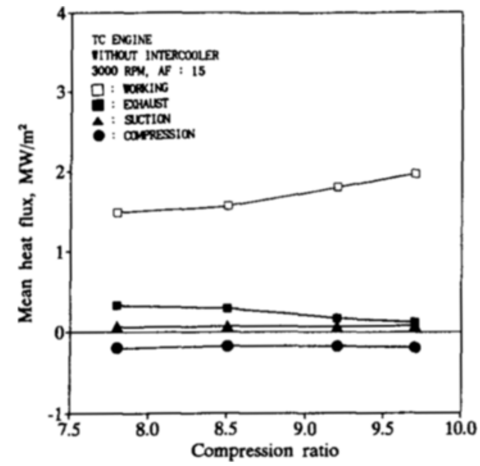

Fig. 13 Mean heat flux as a function of compression ratio

The maximum surface temperature is 595 K at the compression ratio 9.7 and engine speed 4000 rpm and the maximum surface temperature is 550 K at the compression ratio 7.8 and engine speed 4000 rpm.

Figure 12 shows the maximum heat flux as a function of compression ratio at each engine speed in turbocharged engine without intercooler. As the compression ratio decreases, maximum heat flux decreases because of the decrease of initial combustion temperature and pressure. At the engine speed 4000 rpm and compression ratio 9.7, the maximum heat flux is about 8.1 MW/m^2 and the maximum heat flux is about 6.1MW/m^2 at the compression ratio 7.8. Therefore, the maximum heat flux decreases about 25% by decreasing the compression ratio from 9.7 to 7.8.

Figure 13 shows the mean heat flux as a function of compression ratio at each stroke in turbocharged engine without intercooler. In this figure, working process represents combustion and expansion process from the initial period of heat release to the period of exhaust valve opening. The mean heat flux of wall surface in working process shows the largest value, and the mean heat fluxes in intake and compression processes show negative value. As the compression ratio increases, the combustion temperature and the mean heat flux in working process increase. The mean heat flux in exhaust process decreases according to increase

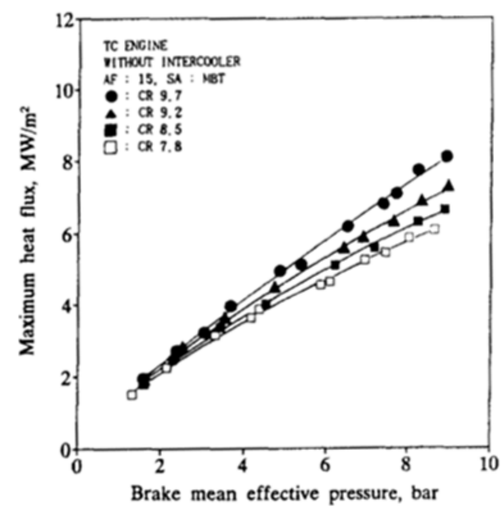

Fig. 14 Maximum heat flux as a function of brake mean effecitive pressure

of compression ratio, so that the decrease of exhaust temperature according to increase of expansion rate occurs. But compression ratio has no effect on heat fluxes of intake and compression process.

Figure 14 shows maximum heat flux as a function of brake mean effective pressure at each compression ratio in turbocharged engine without intercooler. By increasing the brake mean effective pressure, the maximum heat flux increases, and by increasing the compression ratio, the maximum heat flux increases at the same brake mean effective pressure. Therefore, it is expected that the brake mean effective pressure can be increased by decreasing the charge pressure, while the maximum heat flux of combustion chamber wall is decreased by decreasing the compression ratio of engine in order to increase the output power and to minimize the thermal loading in the turbocharged gasoline engine.

5. Conclusions

To investigate the turbocharged gasoline engine, the characteristics of the thermal flow, heat flux, thermal loading should be studied.

From this research, the characteristics of the unsteady temperature and heat flux of combustion chamber wall in a turbocharged gasoline engine, were studied.

The temperature probe of thin-film type was designed and installed onto the combustion chamber wall to measure unsteady thermal flow. The unsteady heat flux at combustion chamber wall was evaluated by one dimensional unsteady conduction equation with the wall temperature and temperature gradient.

To restrain from the knocking, the compression ratio of turbocharged gasoline engine should be decreased. To do this the wall temperature of combustion chamber and heat flux are also decreased because of decrease of combustion initial temperature and pressure.

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