Transactions of the Korean Society of Mechanical Engineers, Vol. 8, No. 1, pp. 71~78, 1984.

<**Original**>

Heat Dissipation of Cylinder Head of Reciprocating Internal Combustion Engine

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(Received June 30, 1983)

왕복동 내연기관 실린더 헤드의 방열에 관한 연구

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초 록

본 논문은 왕복동 내연기관의 방열에 관한 연구의 하나로서 디이젤 기관 실린더 헤드와 흡 배기 밸브의 온도 분포와 열유속의 분포를 구한 것이다.

방열 해석은 기관이 정상 작동된 다음의 실린더 헤드의 열부하가 일정하다고 생각하여 실린더 헤드의 밸브 시이트 양단의 온도와 연소 가스 배출 온도, 흡기 및 냉각수 온도를 측정하고 온도분포 및 열유속을 유한요소법을 적용하여 구하였다.

본 연구의 결과 실린더 헤드 및 밸브의 과부하는 밸브의 경우에는 밸브 헤드 중심과 밸브 헤드 중심 부근에서 일어나며, 실린더 헤드의 경우에는 헤드 중심부 표면에서 발생하였다.

흡 배기 밸브 및 물재킷부의 온도 분포 및 열유속의 분포를 주어진 냉각수 온도 조건에 대하여 구한 후 이들을 비교 검토 하였다.

Nomenclature

G''' : Interal heat generation
H : Heat loss at the boundary due to conduction
h : Convection heat transfer coefficient
h_C : Coolant side convection heat transfer coefficient
h_{egt} : Combustion gas side convection heat transfer coefficient
h_{egv} : Exhaust gas side or fresh air side convection heat transfer coefficient
k : Thermal conductivity
n_x, n_y : Direction cosine
p_{me} : Brake mean effective pressure
q : Heat flux
Δq : Heat flux difference between exhaust port and intake port
T : Temperature
T_1 : Coolant wall temperature
T_2 : Valve temperature
T_C : Coolant temperature
T_{egf} : Combustion gas temperature
T_{egv} : Exhaustgas or fresh air temperature
$T(x,y)$: Specified boundary temperature distribution
T_∞ : Ambient temperature

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$\Delta T = T_{me} - T_{mi}$

R : Thermal resistance

$R_1 = \dfrac{1}{h_C A}$

$R_2 = \dfrac{t}{kA}$

$R_3 = \dfrac{1}{hA}$

$R_4 = \dfrac{1}{_{egf}h_{egv} A}$

Subscripts

m : Mean value

∞ : Surrounding fluid value

e : Exhaust port

i : Intake port

1. Introduction

During the cycle of internal combustion engine, only a part of the heat of combustion of the fuel appears as mechanical work, while the remainder is rejected mainly by the coolant load and exhaust loss of engine.

The heat of combustion in cylinder is transferred by radiation, convection, and conduction from the gases to the cylinder wall and cylinder head.

In a real engine, the amount of heat radiated is a small part of total heat transfer and the rest of heat losses, due to the temperature gradient between cylinder wall and combustion gases, is transferred to cylinder head and cylider barrel through boundary layer of cylinder-circumference by the forms conduction and convection.

It is well known that the studies of heat transfer between engine cylinder and combustion gas were continued by Nusselt(1), Eichelberg(2), Overbye(3) and Taylor(4). Especially Eichelberg's empirical formula is used broadly as a heat transfer formula in engine because it is very easy to use.

According to heat balance diagram, about 50—65 % of heat losses are dissipated through cylinder head and valve seats, 27—32% through cylinder liner, 17—22% through exhaust port.

To reduce the energy losses and to better the thermal efficiency of engine have very significant meanings in view of improving the conversion from heat energy to available energy.

Several investigations of heat dissipation in four-cycle gasoline engine have been studied by Yamagishi(5) and Hirao(6). Issiki(7) tried to analyze the heat transfer in internal combustion engine applying the accordion model.

The present paper describes a study on the temperature distribution and heat flux distribution of cylinder head of four-stroke cycle diesel engine by applying the finite element methed.

2. Analysis of Heat Dissipation of Cylinder Head

2.1. Governing Equation and Boundary Condition

Heat transfer in reciprocating combustion engine is in general three dimensional and unsteady. However, the periodic variations of temperature of cylinder wall and cylinder head is very small and the depth of penetration of the temperature fluctuation inside the wall is also very limited under the constant engine speed.

From this view point, the main interests in this study are the spatial temperature distribution and heat rejection rate of cylinder head. The equation of heat dissipation of cylinder head is treated two dimensional and steady state(8).

The governing partial differential equation may be expressed in the general form as

$$\frac{\partial}{\partial x}\left(k_x \frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k_y \frac{\partial T}{\partial y}\right) + G''' = 0 \qquad (1)$$

where k_x and k_y are thermal conductivities in the x and y direction, respectively, and G''' is internal heat generation.

On the boundary of the domain cylinde head we have the following boundary conditions:

$$\left.\begin{array}{l} T = T(x,y) \text{ on the both edges of valves} \\ k_x \dfrac{\partial T}{\partial x} n_x + k_y \dfrac{\partial T}{\partial y} n_y + H + h(T - T_\infty) = 0 \text{ on} \\ \qquad \text{the coolant side or gas side} \end{array}\right\} \quad (2)$$

To predict the temperature distribution and heat flux of engine head, the exact boundary conditions as described in equation (2) must be known. But it is impossible because of cylinder head structure and the large number of factors. Therefore, it is difficult to solve the governing equation (1) with boundary conditions equation (2).

The finite element method is used in this study because of its advantages in handling complex geometry of engine cylinder, mixed boundary conditions, different properties of material.

Finite element formulation for equation (1) and (2) by Galerkin's method is given by[9]

$$\iint\left(k_x\left[\frac{\partial N}{\partial x}\right]\{T\}^{(e)}\frac{\partial N_i}{\partial x}+k_y\left[\frac{\partial N}{\partial y}\right]\{T\}^{(e)}\frac{\partial N_i}{\partial y}\right)dxdy-\iint N_iG'''dx\,dy+\int(HN_i+h[N]\{T\}^{(e)}N_i-hT_\infty N_i)d\Sigma^{(e)}=0 \quad (3)$$

where N_i is the shape function.

In matrix notation, equation (3) is expressed as

$$[K_t]^{(e)}\{T\}^{(e)}+[K_h]^{(e)}\{T\}^{(e)}=\{G'''\}^{(e)}-\{H\}^{(e)}+\{hT_\infty\}^{(e)} \quad (4)$$

From equation (4) we can obtain the temperature distribution and heat flux distribution of cylinder head of reciprocating engine.

2.2. Finite Element Model of Cylinder Head

Fig. 1 shows that the finite element model of cylinder head with overhead valve mechanism. In this work, cylinder head of test engine (Kubota diesel engine) is divided into 607 nodal points, 855 linear triangle elements.

The temperature distribution and heat flux of cylinder head can be obtained by the equation (4).

3. Experimental Apparatus and Procedure

The engine used for this test was a vertical four-stroke cycle water-cooled diesel engine, having direct injection chamber.

The bore, stroke and displacement of single-cylinder engine were, respectively, 110mm, 150mm, and 0.00143m^3. Fig. 2 shows a schematic layout of the experimental apparatus.

The engine was coupled to eddy-current dynamometer and to the temperature recorder. In this experiment, test engine was operated at 1000 rpm

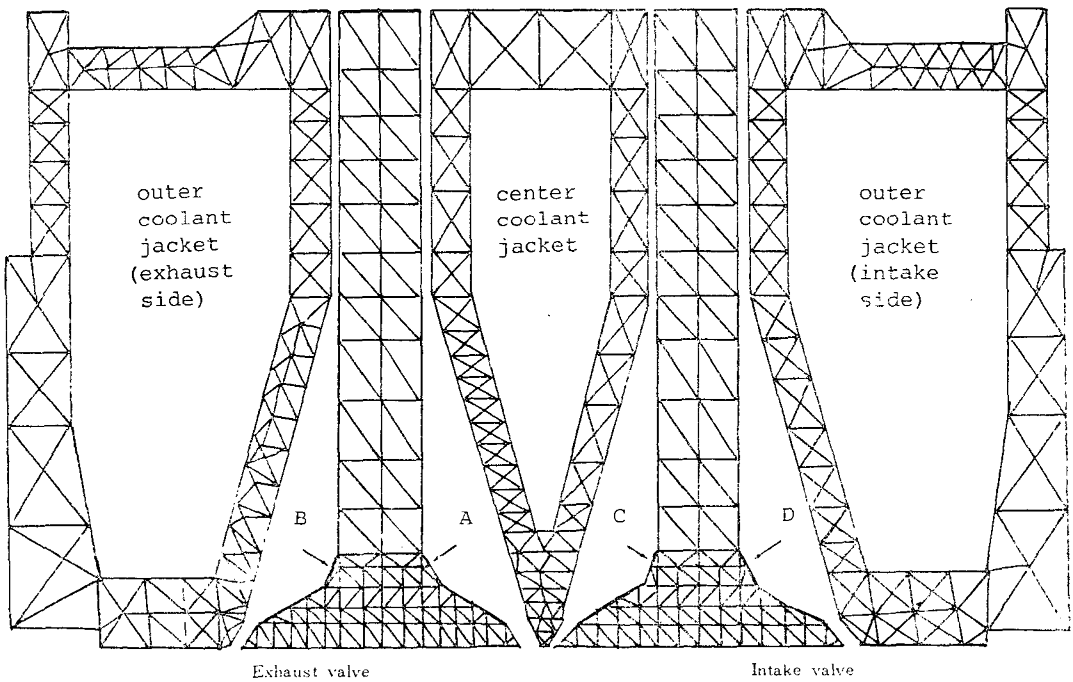
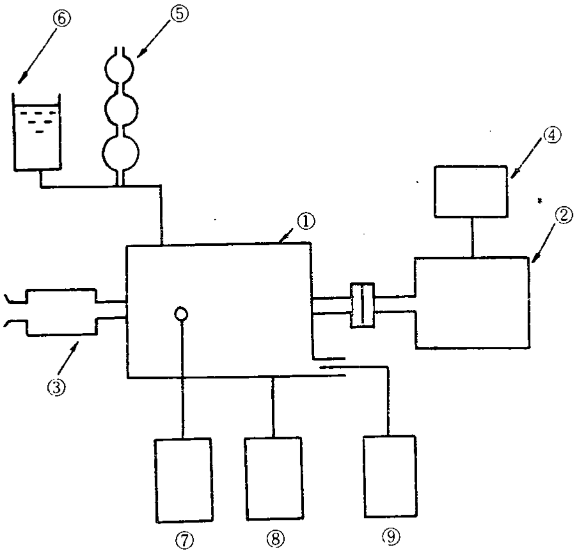


Fig. 1 Finite element model of cylinder head



① Test engine
② Dynamometer
③ Surge chamber
④ Dynamometer panel
⑤ Fuel burette
⑥ Fuel tank
⑦ Exhaust gas analyzer
⑧ Digital data recorder
⑨ Engine analyzer

Fig. 2 Experimental apparatus

and three different coolant temperatures of 50°C, 70°C and 90°C.

Test arrangements were made for measuring the temperatures of exhaust gases in the exhaust port, inlet air and both edges of cylinder head at the constant coolant exit temperature.

After the normal operating temperature, measurements of power output of engine were made on the dynamometer at constant load of engine. To investigate the effect of coolant temperature variations was studied by varying the cooling water temperatures, holding the engine speed constant and maintaining the mean effective pressure 5kg/cm².

The temperature of exhaust gas, inlet air, coolant and both sides of cylinder head was measured by digital data-logger.

In order to obtain the distribution of wall temperature and heat flux accurately, the exact boundary condition on both gas side and coolant side must be known. However, it is not possible because the engine head has the complexity of geometrical shapes and various parameters.

In this study time-average value of heat transfer coefficients and temperatures of inlet and exhaust gas are determined corresponding to engine operating conditions. Mean gas temperature and heat transfer coefficient of combustion gas in cylinder is given by[8],[10]

$$\left.\begin{aligned} &T_{egf}=616+12.7\,p_{me} \quad {}^{\circ}\mathrm{C} \\ &h_{egf}=1.167\,(160+12p_{me})\,\mathrm{W/m^{2}{}^{\circ}C} \\ &h_{c}=5700 \quad \mathrm{W/m^{2}{}^{\circ}C} \\ &h_{egv}=290 \quad \mathrm{W/m^{2}{}^{\circ}C(exhaust\ gas\ side)} \\ &h_{egv}=240 \quad \mathrm{W/m^{2}{}^{\circ}C(fresh\ air\ side)} \end{aligned}\right\} \qquad (5)$$

By using equation (4) with measured temperature and determined heat transfer Coefficients, we can obtain the temperature distribution and heat flux of cylinder head of diesel engine.

4. Results and Discussion

4.1. Temperature Distribution of Cylinder Head

Fig. 3, Fig. 4 and Fig. 5 show the isothermal lines of cylinder head under the 1000rpm with three different coolant temperatures 50°, 70° and 90°C, respectively.

Temperature distribution of exhaust and intake valves are quite different. Under the constant operating condition, exhaust valve is always contacted with high temperature combustion gas and exhaust gas in exhaust port of cylinder head. On the other hand, intake valve head is contacted with hot combustion gas in cylinder but cool fresh air flowing into cylinder affects the cooling effect of valve surface and valve stem. Therefore, thermal load on the intake valve is far smaller than that on exhaust valve.

Thus, mean temperature of exhaust valve is much higher than that of intake valve. The hot spots in the exhaust valve are at the around the center of valve head and upper part of valve stem.

The temperature distributions of center side (center coolant jacket side) and outside (outer co-

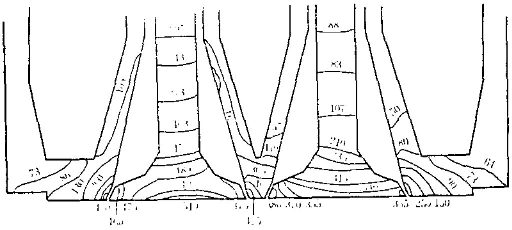


Fig. 3 Isothermal lines of cylinder head (T_c=50°C)

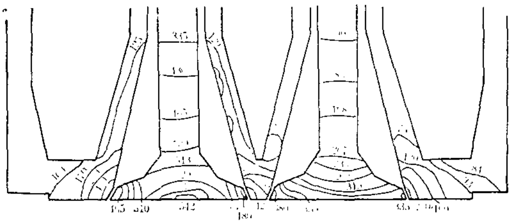


Fig. 4 Isothermal lines of cylinder head (T_c=70°C)

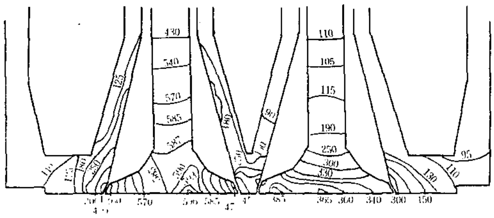


Fig. 5 Isothermal lines of cylinder head (T_c=90°C)

olant jacket side) of valves are not the same as shown Fig. 6 and Fig. 7.

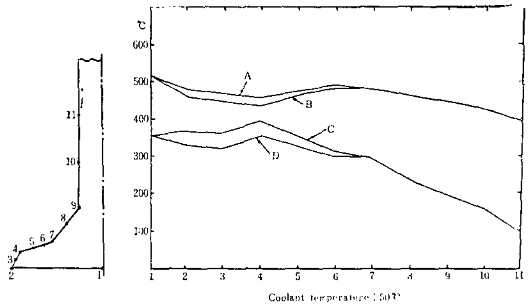
When the engine is operated at constant engine speed and load, the temperature distribution of exhaust valve is higher than that of intake valve.

The temperature distribution of coolant jacket also varies with its location; particularly coolant jacket near the exhaust port have relatively high temperature because of the exhaust gas.

The calculated mean temperature is compared with the theoretical temperature by electrical analogy as shown Fig. 8.

$$T_2=\frac{T_C+(R_1+R_2)\ (T_{egf}/R_3+T_{egv}/R_4)}{1+(R_1+R_2)\ (1/R_3+1/R_4)} \qquad (6)$$

The predicted mean temperature of valve is good agreement with relative error smaller than 0.5 percent of theoretical value using thermal resistance method.



A : Exhaust valve (center side)
B : Exhaust valve (out side)
C : Intake valve (center side)
D : Intake valve (out side)

Fig. 6 Temperature distribution of center side and out side of valves

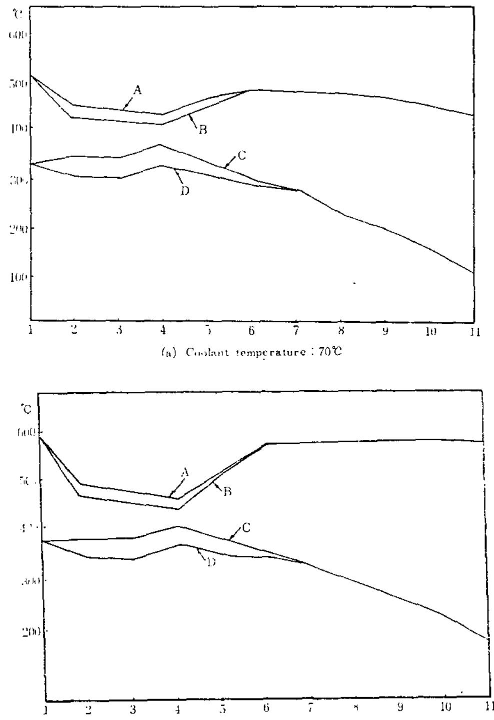


Fig. 7 Temperature distribution of center side and out side of valves

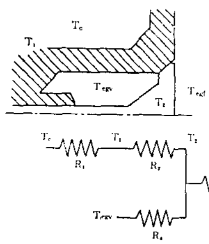


Fig. 8 Steady state heat transfer and electrical analog

4.2. Heat Flux Distributions of Cylinder Head

Heat flux distribution of valves are shown Fig. 9 and Fig. 10.

It is found that exhaust valve, being incessantly faced with hot combustion gas and exhaust gas in exhaust port, has similar temperature distribution as those gas, so the amount of heat flux is small.

Heat flows into both intake and exhaust valve *from hot gases in the conbustion chamber and part*

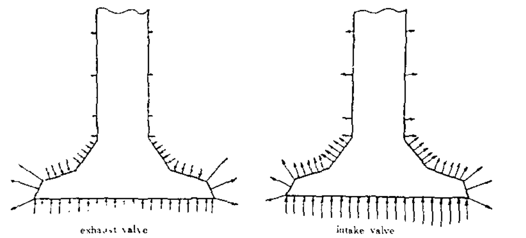


Fig. 9 Heat flux distribution of valves (T_c=50°C) −20kW/m²

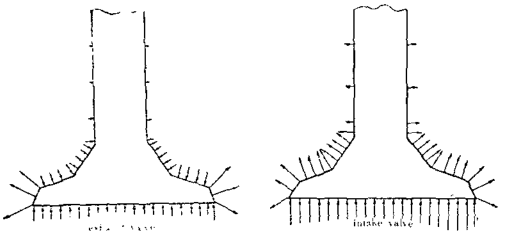


Fig. 10 Heat flux distribution of values (T_c=90°C) −20kW/m²

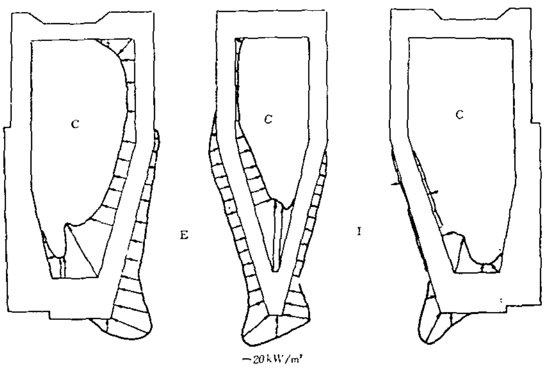


Fig. 11 Heat flux distribution of cylinder head (T_c=50°C)

culary into the exhaust valve during the exhaust process of *engine. During this process* the hot combustion gas flowing at fairly high velocity contact the valve face, valve seat, and part of valve stem.

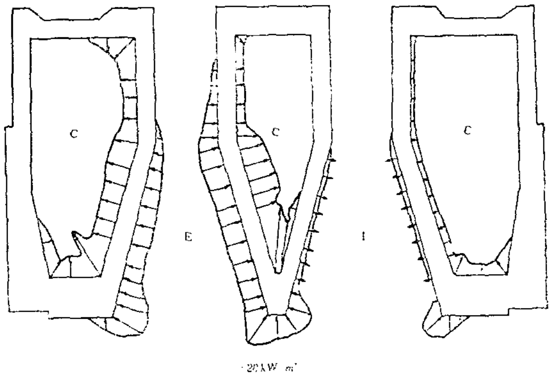


Fig. 12 Heat flux distribution of cylinder head (T_c=90°C)

On the other hand, heat flux of intake valve was relativly high under the constant test condition. Exhaust valve head is subjected to a uniform gas temperature loading, which depends upon the combustion conditions in cylinder. On the contrary, valve stem and face of intake valve is well cooled by reason of the contact of fresh air flowing into cylinder.

Heat flux distribution of coolant jacket in cylinder head are shown Fig. 11 and Fig. 12.

It is seen that large amount of heat flow goes into the exhaust valve seat of head and then goes out the coolant along the wall where the valves are contacted with lower jacket surface between intake port and exhaust port.

On the intake port, a relatively small amount of heat flow is dissipated to the coolant along the same surface of cooling jacket and a negligible amount of heat flows from the coolant into the inlet fresh air.

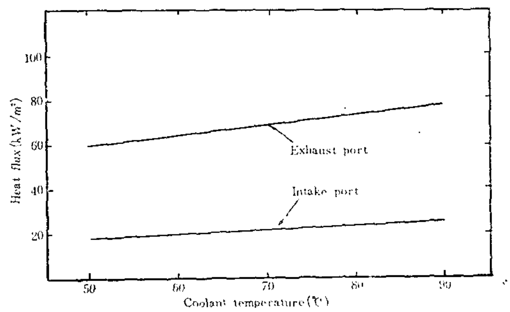


Fig. 13 Comparison of heat flux between exhaust port and intake port

Table 1 Mean heat flux of cylinder head and valves (kW/m^2)

Cylinder head and valves		Coolant temperature 50°C	70°C	90°C
Exhaust valve		25.630	27.232	33.842
Intake valve		62.865	67.811	68.013
Coolant jacket of exhaust side	inner	56.4,3	59.045	63.905
	outer	57.213	65.791	74.221
Coolant jacket of intake side	inner	18.014	18.354	19.112
	outer	17.020	17.399	18.720
Coolant jacket of center side	inner	73.776	78.685	82.260
	outer	45.479	52.340	61.189

Mean heat flux of cylinder head are summarized in Table 1

5. Conclusions

In this study, the temperature distribution and heat flux of valves and cylinder head were investigated by finite element method using measured temperature of cylinder head and gas temperature as a boundary condition.

The conclusions from the results of this study are as follow:

1) Under the constant operating conditions the hot spot of exhaust valve and cylinder head appear around the center of valve head and the valley of center in head.

2) Uhe temperature of exhaust port is higher than that of intake port by $\Delta T=64.1°C$ (coolant temperature $t_c=90°C$), $\Delta T=59.3°C$ ($t_c=70°C$), $\Delta T=56.4°C$ ($t_c=50°C$), respectively.

3) The temperature difference between exhaust valve and intake valve increase in accordance with the increase of coolant temperature in cylinder head.

4) The amount of heat flux is largest at center coolant jacket near the exhaust port and smallest at the outer coolant jacket near the intake port.

5) The amount of heat dissipation near the exhaust port is larger than that near the intake port by $\Delta q=56kW/m^2$ ($t_c=90°C$), $\Delta q=51kW/m^2$ ($t_c=70°C$), $\Delta q=42kW/m^2$ ($t_c=50°C$).

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Appendix

(1) Valve temperature T_2

In order to compare the Calculated mean temperature of valve with temperature of valve using electrical analogy, the temperature T_2 is given by equation (6).

For example, the mean temperature of exhaust valve is 407°C under the following conditions.

The condions of exhaust valve: T_c=50°C, T_{egt}= 679.5°C,

T_{egv}=404°C, h_c=0.54W/cm^2°C, h_{egt}=0.0255W/cm^2°C, h_{egv}=0.029W/cm^2°C, k=0.02W/cm°C, t= 1cm.

(2) Heat flux distribution

Heat flux distribution is obtained from following equation.

$q/A=h(T_H-T)$ W/m^2

where T_H is measured temperature and T is calculated temperature by finite element method.