HEAT TRANSFER IN FIRE RESISTANCE TEST FURNACE WITH PARTICULAR REFERENCES TO GAS RADIATION

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ABSTRACT

A model of radiative and convective heat transfer was developed to simulate the heat transfer in fire resistance furnace. The calculated results agreed well with experimental data. By using the model, the heat flux to sample was calculated varying the absorption coefficient of gas. From the calculated results, the heat flux to sample is strongly dependent upon the absorption coefficient of gas. The apparent heat transfer coefficient between furnace and sample surface (the fraction of the overall heat flux to sample over the temperature difference between controlling thermocouple and sample surface) varied more than three times as the absorption coefficient changes from zero (simulating an electrically heated furnace) to infinite (simulating a soot-rich fire).
KEY WORDS: fire resistance test, gas radiation, reproducibility

NOTATIONS

Alphabets		unit			unit
A	area	$[m^2]$	k	absorption coefficient	[1/m]
T	temperature	[K]	q	radiative heat flux	$[W/m^2]$
c	specific heat	[J/kg·K]	q_{conv}	convective heat flux	$[W/m^2]$
d_{tc}	thickness of thermocouple sheathe	[m]	r_{ij}	distance between differential areas on surface i and j	[m]
h	heat transfer coefficient	$[W/m^2.K]$			

Greek Letters unit unit

ε	emissivity	[-]	ρ	density	[kg/m^3]
λ	thermal conductivity	[$W/m \cdot K$]	σ	Stefan-Bolzmann constant	[W/m^2K^4]

Subscripts

i,j	surface number	*g*	gas
inc	incident	*tc*	thermocouple
out	outward	*hfm*	heat flux meter
net	net		

INTRODUCTION

Even though the fire resistance test is a commonly accepted method to evaluate the fire resistance of building constructions, it can be criticized. Due to the differences in furnace size, wall lining materials, type of fuels and other conditions, the test results are widely scattered. In order to reduce the scatter of the results and to increase the reproducibility of the test results, the heating conditions should be standardized.

For this aim, several analytical and experimental studies have been carried out so far. Experimental results show that the incident heat flux to sample surface differed in more than 10% between furnaces[1]. As to the analytical approach, Sultan *et al*[2] investigated the radiative heat transfer in fire resistance furnaces by a model of inter reflection of thermal radiation between surfaces and gas. From the results they emphasized the importance of the thermal inertia of wall lining materials in order to increase the reproducibility. However, in their analysis, the gas temperature was assumed to be the same as thermocouple reading, which is not correct in most of the testing conditions. Because of the radiation heat loss, the thermocouple reading is considerably lower than the actual gas temperature. Furthermore the heat capacity of thermocouple sheath might affect the response of thermocouple especially in the early stage of heating. Similar work is carried out by Keltner *et al* [3], but the convection heat flux is not taken into account. Thus the results are not enough for the purpose of furnace harmonization.

In this study, an analytical model is developed to calculate radiative and convective heat transfer in fire resistance test. The furnace gas temperature is calculated to fit the thermocouple readings to standard time-temperature curve. In order to verify the model, the results are compared with experiments. After that, the effect of absorption coefficient upon the heat transfer is discussed.

A MODEL OF HEAT TRANSFER IN FIRE RESISTANCE FURNACE

A fire resistance test furnace is approximated by a rectangular parallelepiped as shown in figure 1. Each surface is numbered from 1 to 7. As shown in figure 2, the heat flux incident on the sample is the sum of radiative heat flux from gas and other surfaces and convective heat flux from the gas.

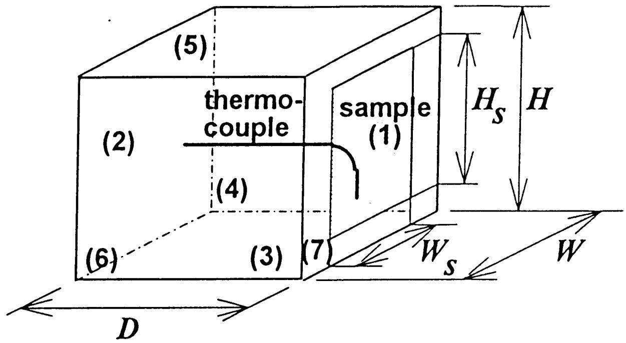

Figure 1 Configuration of fire resistance furnace

Radiative Heat Transfer

The furnace gas is approximated as a gray body for thermal radiation. Assuming that the absorption coefficient is constant, and that the scattering can be neglected, the incident thermal radiation to surfaces are obtained by solving the following simultaneous equations[4)],

$$[M]\{q_{inc}\} = \{f\} \tag{1}$$

where $[M]$ is a coefficients matrix, $\{q_{inc}\} = \{q_{inc,1}, q_{inc,2}, \ldots, q_{inc,7}\}$ is the vector of incident radiative heat flux to each surface, $\{f\}$ is a vector of constants. The explicit forms of $[M]$ and $\{f\}$ are

$$M_{ij} = \begin{cases} 1 & (i = j) \\ -(1-\varepsilon_j)\dfrac{1}{A_i}\displaystyle\iint_{A_i}\iint_{A_j} e^{-kr_{ij}} \frac{\cos\theta_i \cos\theta_j}{\pi r^2} dA_j dA_i & (i \neq j) \end{cases}, \tag{2}$$

$$\begin{aligned} f_i = & \sum_{\substack{j=1 \\ (j \neq i)}}^{7} \varepsilon_j \sigma T_j^4 \frac{1}{A_i}\iint_{A_i}\iint_{A_j} e^{-kr_{ij}} \frac{\cos\theta_i \cos\theta_j}{\pi r^2} dA_j dA_i \\ & + \sigma T_g^4 \frac{1}{A_i}\iint_{A_i}\iint_{\Sigma A_j} (1 - e^{-kr_{ij}}) \frac{\cos\theta_i \cos\theta_j}{\pi r^2} dA_j dA_i \end{aligned}, \tag{3}$$

where the subscripts i and j denote the surface number ($i, j = 1, 2, \ldots, 7$, $i = 1$ for sample surface).

After solving the equation (1), the net radiative heat flux absorbed by the surfaces are calculated by,

$$q_{net,i} = \varepsilon_i q_{inc,i} - \varepsilon_i \sigma T_i^4. \tag{4}$$

Convective Heat Transfer

Convective heat transfer from gas to surface i is calculated by

$$q_{conv,i} = h_i(T_g - T_i). \tag{5}$$

Surface Temperature of Sample and Furnace Walls

In the above equations (1) to (5), surface temperatures appear in the right hand side. Thus the heat conduction analysis must be coupled in order to close the system of equations. To reduce the computational time, heat conduction is approximated by one dimensional field as

$$\rho_i c_i \frac{\partial T_i(x_i,t)}{\partial t} = \lambda_i \frac{\partial^2 T_i(x_i,t)}{\partial x_i^{\,2}}, \qquad (i = 1,\ldots,7), \tag{6}$$

where x_i is the coordinate inward and perpendicular with surface i . This equation is solved for sample and all the furnace walls under the boundary condition shown in figure 2, namely

$$q_{net,i} + q_{conv,i} = -\lambda \frac{\partial T_i(x_i,t)}{\partial x_i}, \qquad (i = 1,\ldots,7). \tag{7}$$

Gas Temperature

Gas temperature must also be given to solve the system of equations. During the fire test, the furnace is controlled so as to fit the thermocouple reading to prescribed standard time-temperature curve. Thus the heat balance of the thermocouple sheath, shown in figure 3,

$$\varepsilon_{tc} q_{inc,tc} - \varepsilon_{tc}\sigma T_{tc}^{\,4} + h_{tc}(T_g - T_{tc}) = d_{tc}\rho_{tc}c_{tc}\frac{\partial T_{tc}}{\partial t}, \tag{8}$$

can be used to calculate gas temperature. In equation (8), the first two terms denote the net absorption of thermal radiation, third term denotes the convective heat gain from gas. The right hand side denotes the heat accumulated to the thermocouple sheath.

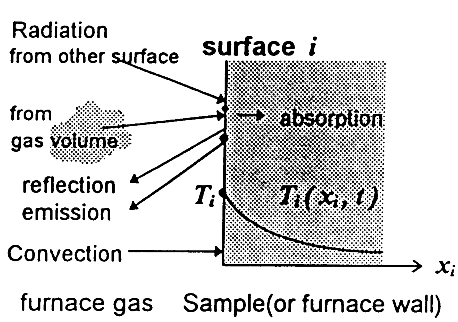

Figure 2 Heat balance at surface i

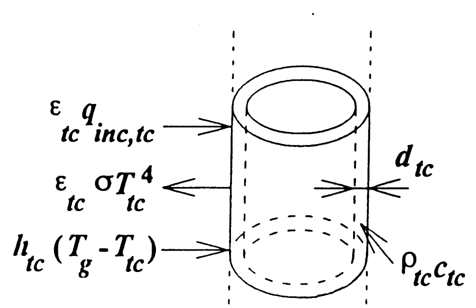

Figure 3 Heat balance of thermocouple

By combining all the equations (1) to (8) , gas temperature, radiative and convective heat flux, wall surface temperatures are calculated. Successive approximation procedure was applied to solve these equations.

VERIFICATION

Comparison with Experiments

In order to verify the theoretical model, a calculation was carried out simulating a published experimental data[1]. In the experiment, a flat wall of 80 mm thick autoclaved lightweight concrete (ALC) was heated in accordance with JIS A 1304 standard fire test method. During heating, heat flux incident to the sample surface was measured by a heat flux meter equipped at a hole of the sample as shown in figure 4.

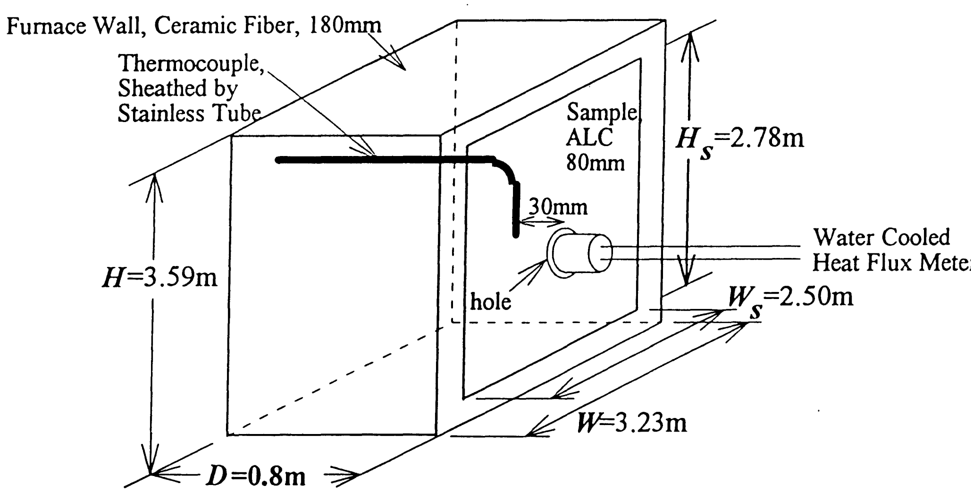

Figure 4 Experimental set-up in reference 1.

The material properties of sample and furnace wall are shown in Table 1. The furnace wall has a smaller thermal inertia than the sample. Emmisivity of the sample is larger than furnace wall.

The absorption coefficient of gas depends upon the composition of furnace gas and soot concentration. Concerning with the gas radiation, carbon dioxide and water vapor are the main absorbing-emitting components. In typical testing situations, the concentration of these spiecies would be about 9 % and 6 %. The effect of soot might be small in most of the furnace fueled by gas or oil. Neglecting the contribution from soot, we get on estimate of 0.135 [m^{-1}] for absorption coefficient.

The thermocouple was sheathed by a stainless tube. The thickness of the tube was assumed to be 1 mm. The emmisivity was assumed to be 0.1. Convective heat transfer coefficients were assumed to be 23 [$W/m^2.K$] for sample, furnace wall and thermocouple surfaces.

Table 1 Material properties[5] of sample and furnace wall

	emmisivity [-]	thermal conductivity [W/m.K]	density [kg/m^3]	heat capacity [kJ/m^3.K]	thermal inertia [J/m^2.s$^{1/2}$.K]	thickness [mm]
sample	0.94	0.15	600	653	313	80
furnace wall	0.70	0.04	160	184	86	180

The calculated gas temperature and surface temperatures of sample and furnace wall are shown in figure 5. In the early stage, the gas temperature is much higher than the standard temperature. Because of the heat capacity of thermocouple sheath, response of thermocouple reading is limited. In order to fit the thermocouple reading to standard time-temperature curve, the gas temperature must be much higher than the standard time-temperature curve especially in the beginning of heating. After 15 minutes, the difference between gas temperature and thermocouple reading tends to cease.

Calculated heat flux on sample surface is shown in figure 6. Incident radiation heat flux increases to about 55 kW/m^2 at 10 minutes, 85 kW/m^2 at 30. As the heat flux meter is water cooled, the surface temperature was kept constant (T_{hfm} =273K was assumed), it is appropriate to compare the measured results with the sum of incident radiative heat flux and convective heat flux. Therefore,

$$q_{hfm} = q_{inc,1} + h_{hfm}(T_g - T_{hfm}) \tag{9}$$

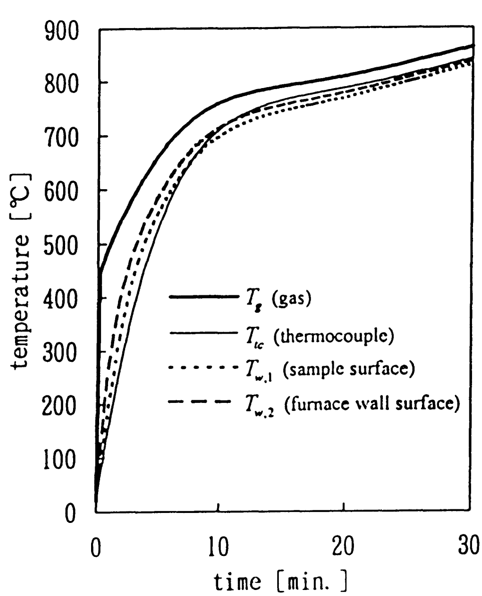

Figure 5 Calculated temperatures

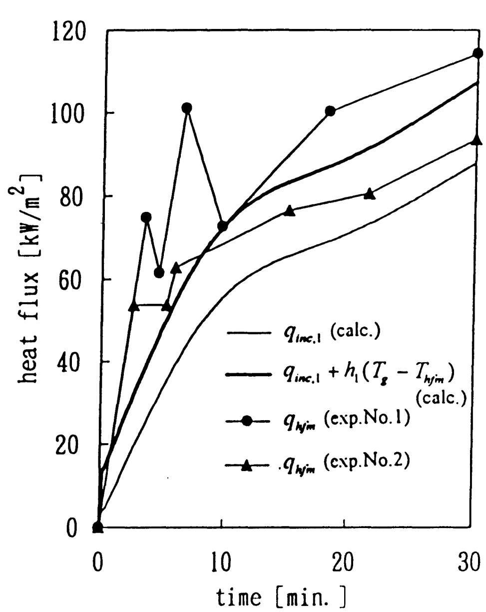

Figure 6 Comparison with experiments

was plotted in comparison with measurements. The calculated results fall between the two measurements. It can be said that the agreement is very good, except for the period between 6 and 8 minutes.

Heat Balance of Sample and Furnace Walls

In figure 7, the net heat flux on sample surface is shown. In the beginning, convective heat flux exceeds the radiative heat flux. However, the convective flux rapidly decreases. After 3 minutes, the radiative heat flux dominates the heat transfer to the sample. After 20 minutes, the fraction of convective heat flux is only 25% of the radiative heat flux. In figure 8, net radiation exchange between sample surface and the other part is shown. The radiation exchange between wall surface exceeds the exchange between gas. From figures 7 and 8, it can be said that the main part of the heat flux to sample surface came from wall surface by radiation.

In figures 9 and 10, the heat balance of furnace wall opposite to sample is shown. As shown in figure 9, convective heat flux are dominant especially in the early stage. The radiative heat flux is relatively unimportant compared with the case of sample surface because of its small emissivity. The order of heat flux is about half of that of sample surface because of the low thermal inertia of the furnace wall. As shown in figure 10, the opposite wall gets radiation from gas (as well as side walls), loses to sample surface. Consequently the opposite wall gains heat from gas by convection and radiation. The heat is used to temperature rise of itself while the rest is emitted to sample surface.

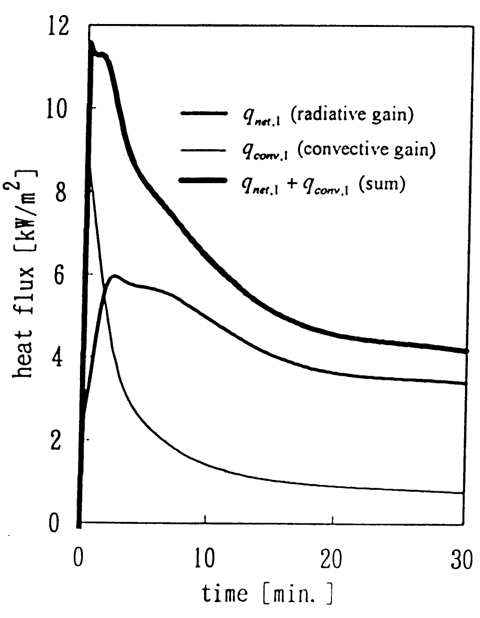

Figure 7 Net heat flux to sample surface

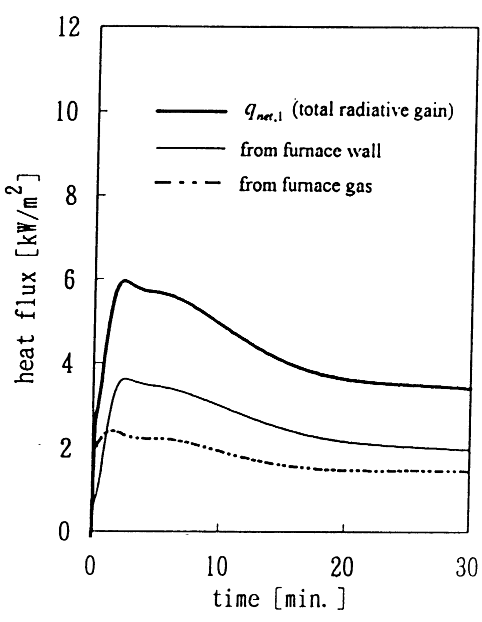

Figure 8 Net radiation exchange between sample surface and the gas or furnace wall

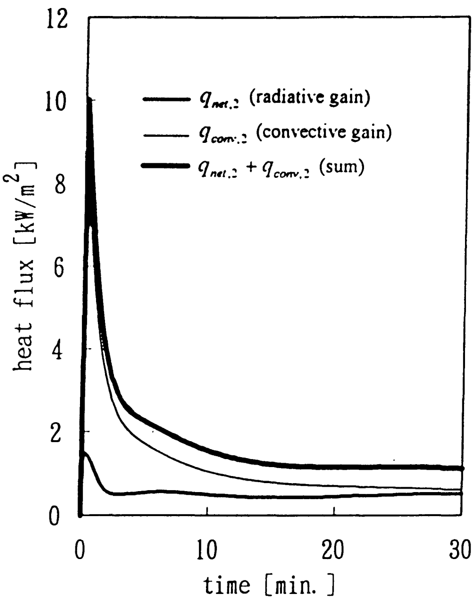

Figure 9 Net heat flux to furnace wall opposite to sample

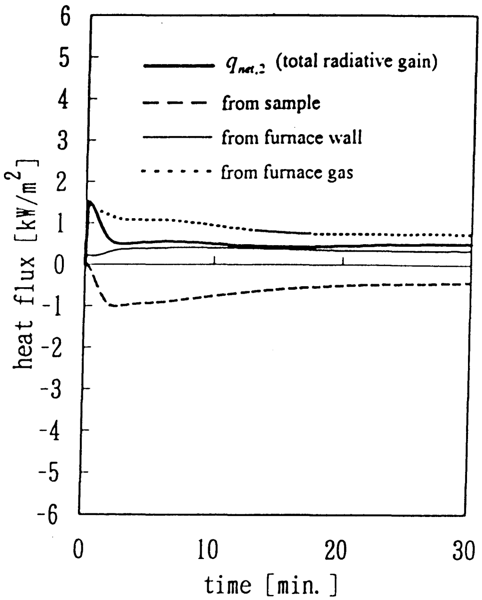

Figure 10 Net radiation exchange between furnace wall opposite to sample and the gas or sample surface

Heat Balance of Thermocouple

The heat balance of the thermocouple are shown in figures 11 and 12. As shown in figure 11, thermocouple gains heat by convection from gas. Except in the early stage, it loses heat by radiation. As shown in figure 12, the loss is mainly to sample surface. This is because the shape factor of the sample surface seen from thermocouple is large, and the sample surface temperature is low.

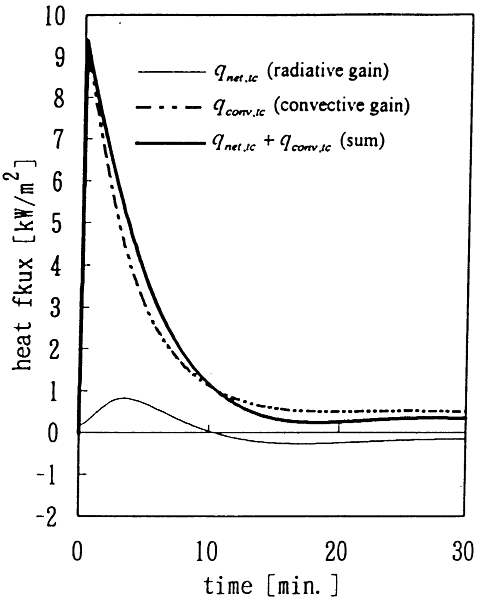

Figure 11 Net heat flux to thermocouple

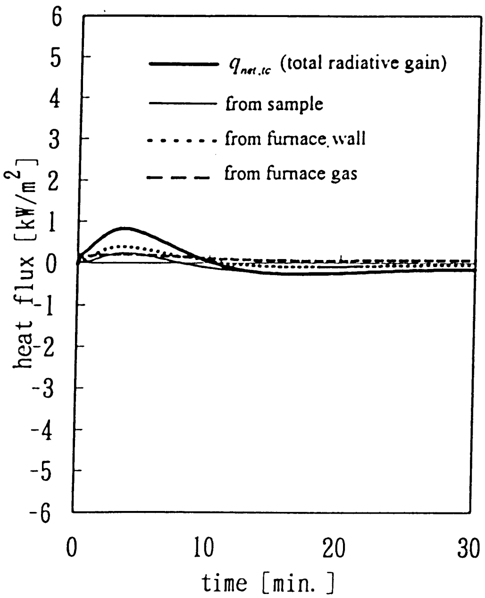

Figure 12 Net radiation exchange between thermocouple and other surfaces

EFFECT OF ABSORPTION COEFFICIENT UPON HEAT TRANSFER

The absorption coefficient of gas might change the heat transfer rate. To investigate the effect of absorption coefficient, it was changed to zero and infinite. Zero value is to simulate an electrically heated furnace, while the infinite value is to simulate a soot-rich furnace gas.

Calculated gas temperatures are shown in figure 13 together with the case of $k = 0.135$. In the three cases, the thermocouple reading is controlled to fit to the standard time-temperature curve. However the gas temperatures are far above the standard temperature especially in the case of zero absorption coefficient. When the absorption coefficient is small, the gas temperature must be increased in order to compensate the decrease of radiative heat flux from gas to thermocouple.

The net radiative heat flux changes as shown in figure 14. Comparing the two extreme cases, $k = 0$ and $\propto$, the incident heat flux changes about 40%. In figure 15 convective heat flux to sample is shown. In contrast to the radiative heat flux, convective heat flux increases as the absorption coefficient tends to zero. This is because of the increase in gas temperature as shown in figure 13.

To combine all the heat transfer rate, the apparent overall heat transfer coefficient was defined by the fraction of the sum of heat flux to sample over the temperature difference between thermocouple (standard temperature) and sample surface as

$$h_{all} = \frac{q_{net,1} + h_{conv}(T_g - T_1)}{T_{tc} - T_1}, \qquad (10)$$

and shown in figure 16. When the absorption coefficient is zero, the heat transfer coefficient is less than 200 [W/m^2.K]. However it increases by more than three times as the absorption coefficient is increased to infinity.

As the results of the changes in heat transfer rates, the surface temperature of the sample changes as shown in figure 17. As the absorption coefficient is increased from zero to infinite, the surface temperature increases as much as 25 Kelvins.

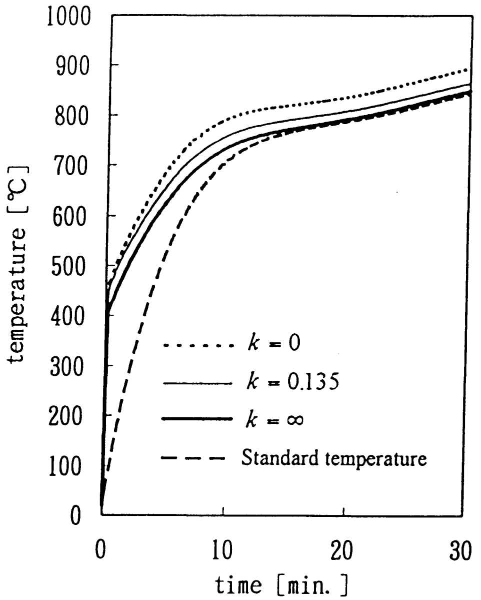

Figure 13 Change of gas temperature

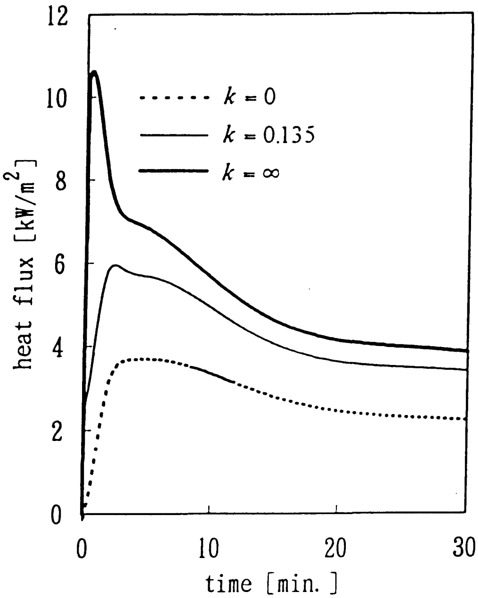

Figure 14 Change of net radiative heat flux absorbed by sample surface

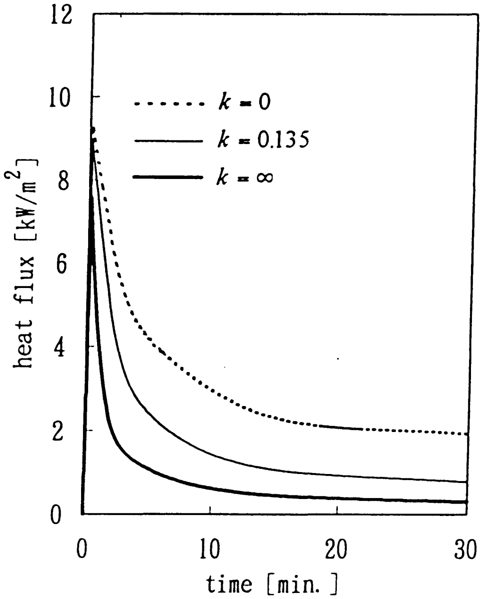

Figure 15 Change of convective heat flux to sample surface

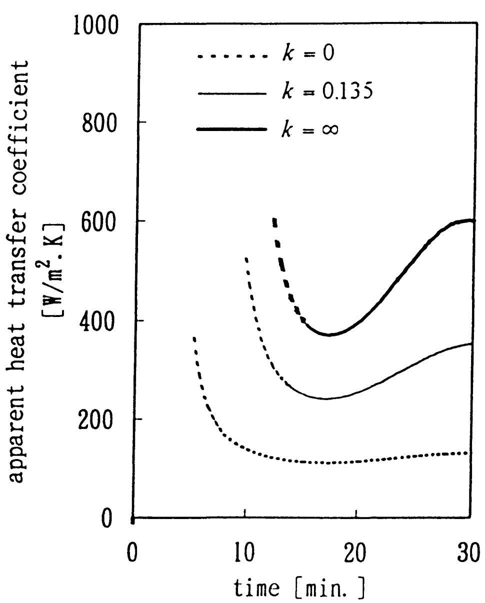

Figure 16 Change of apparent overall heat transfer coefficient

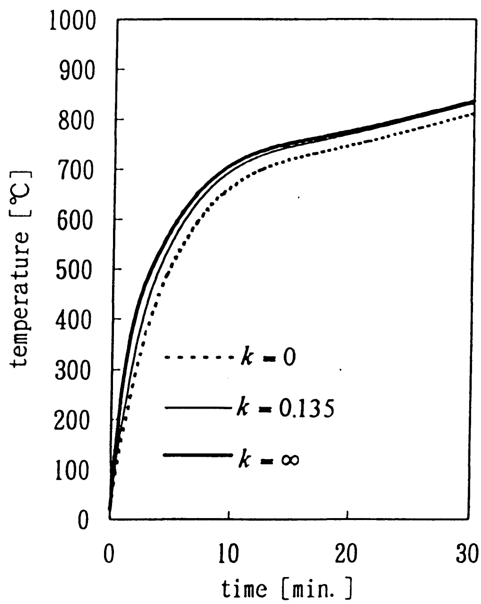

Figure 17 Change of surface temperature of sample

CONCLUSIONS

A model of radiative and convective heat transfer was developed to simulate the heat transfer in fire resistance furnace. The model takes into account the inter reflections of thermal radiation between surfaces and gas. The furnace gas was treated as gray body. The heat balance of thermocouple was also included. The gas temperature was calculated so as to fit the thermocouple reading to the standard time-temperature curve. To verify the model, a published experiment was simulated. The calculated heat flux agreed very well with experimental data.

By using the model, the effect of the absorption coefficient was analyzed. From the calculated results, the apparent heat transfer coefficient between furnace and sample surface (fraction of total heat flux over the temperature difference between sample surface and thermocouple reading) varied more than three times as the absorption coefficient changes from zero (simulating an electrically heated furnace) to infinite (simulating a soot-rich fire). Therefore there is a need to develop a method to reduce the difference in heat transfer rates in order to increase the reproducibility of fire resistance test.

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