#3490

Trans. SHASE
No. 33, Feb., 1987

A Numerical Calculation on the Distribution of Surface Temperature and Thermal Comfort Index caused by Radiation Interaction in a Heated Room

by Masamitsu KAIZUKA*1 and Shizuo IWAMOTO*2

Key Words: Numerica Analysis, Thermal Environment, Radiation Interaction, Thermal Comfort, PMV, MRT, Simulation, Floor Panel Heating, Forced Convective Heating

Synopsis: As a part of studies to establish a numerical prediction method of thermal environments, this paper presents a numerical calculation on the steady state distributions of surface temperature and thermal comfort index caused by radiation interaction in a heated room with floor panel heating or forced convective heating. Results of the calculation of the room air temperature, surface temperature, vector radiant temperature, mean radiant temperature, Predicted Mean Vote and heat balance illustrate well the characteristics of radiation interaction with the heating systems. This calculation procedure can be combined with the calculation of air convection as shown previously by one of the authors for the two-dimensional case.

Introduction

Indoor thermal comfort is evaluated by physical conditions like air temperature, air humidity, air movement and mean radiant temperature, and by human body conditions like clothing and activity. Combining these factors together, PMV (Predicted Mean Vote) was proposed by Fanger[1] as a thermal sensation index. In ordinary air conditioned rooms, such physical conditions and PMV are not uniform causing discomfort to the occupants and an inferior air conditioning efficiency. For example, a cold draft, significant vertical temperature difference, serious lateral radiant cooling or heating may be induced. In order to evaluate a thermal environment more precisely, therefore, the distribution of such physical conditions and PMV should fully be known. For this purpose, the numerical prediction has been recognized as an influential method.

Many studies on the numerical prediction method for indoor air movement or temperature distribution have been conducted, as reviewed by Kaizuka[2] for example, and they are proving to be of practical use. The calculation method of indoor radiation interaction to be discussed in this paper has already been formulized by Gebhart[3] and Fanger[1]. In addition, the calculation for a model indoor space with heated wall has been conducted through a combination of air convection and radiation interaction by Sakamoto[4], and the distributions

*1 Department of Architecture, Faculty of Engineering, Meiji University, Member
*2 Doctoral Course, Graduate School of Engineering, Meiji University, Student Member

of vector radiant temperature and operative temperature inside a two-dimensional heated room have been indicated by Hiramatsu and Kaizuka[5].

As one of the studies to establish the numerical prediction method for indoor thermal environments, this paper aims to formulize the calculation method of wall surface temperature, vector radiant temperature, mean radiant temperature and PMV by taking the distribution of thermal environment caused by radiation interaction into our main consideration. Also this paper aims to apply this method to a room adapted with floor panel heating (FPH) and forced convective heating (FCH) to display one of the aspects of the characteristics of both heating systems.

In the study of Hiramatsu and Kaizuka[5], the numerical prediction was conducted through a combination of the calculations for air convection and radiation interaction by making the distribution of air temperature and convective heat transfer coefficient unknown values. In this paper, however, the calculation of air convection is excluded to examine the calculation method of radiation interaction, therefore the air temperature is assumed to be uniform, and appropriate values are employed as known values for the air velocity and convective heat transfer coefficients.

Such calculation method is basically similar to the method already shown by Fanger[1]. Differences include the following;

1) The indoor surface is subdivided into small surface elements to facilitate the combination of the calculation of air movement with that of radiation interaction.
2) The vector radiant temperature [proposed by Nakamura[6]] representing the directivity of radiation field is calculated.
3) The method [also proposed by Nakamura[6]] of approximating the shape factor of a human body by using an infinitely small cube is employed.
4) The radiosity used by Fanger[1] is replaced by the absorption factor of Gebhart[3].
5) The excellent algorithm by Yamazaki[7] is used in calculating the shape factor.

Note that a part of this paper was already reported by the authors in the references 8), 9), 10) and 11).

Nomenclature

B_{ij} : Gebhart's absorption factor [−]
C_i : Overall wall conductance from inside surface element i to outside air [kcal/m²·h·K]
c_p : Specific heat of air at constant pressure [kcal/kg·K]
ds : Area of infinitely small cube's surface [m²]
FCH : Forced convective heating
FPH : Floor panel heating
F_{ij} : Shape factor from surface element i to j [−]
F_{pi} : Shape factor from human body to surface element i [−]
f_{bi} : Shape factor from cube's surface b to surface element i [−]
f_{ib} : Shape factor from surface element i to cube's surface b [−]
f_{cl} : Clothing area factor [−]
h_c : Convective heat transfer coefficient between human body and room air [kcal/m²·h·K]
h_{ci} : Convective heat transfer coefficient between surface element i and room air [kcal/m²·h·K]
I_{cl} : Thermal resistance from skin to outer surface of clothed body [clo]
M : Metabolic rate [kcal/m²·h]
N : The number of surface element
n : Air change rate [1/h]
P_a : Partial pressure of water vapour in room air [mmHg]
PMV : Predicted Mean Vote
Q_S : Supply heat to room air [kcal/h]
Q_L : Heat loss from room to outside air [kcal/h]
S_i : Area of surface element i [m²]
T_a : Room air temperature [K]
T_i : Temperature of surface element i [K]
T_{mrt} : Mean radiant temperature of cube [K]
T_{oi} : Outside air temperature for surface element i [K]
T_{out} : Outside air temperature [K]
$T_{p\,mrt}$: Mean radiant temperature of human body [K]

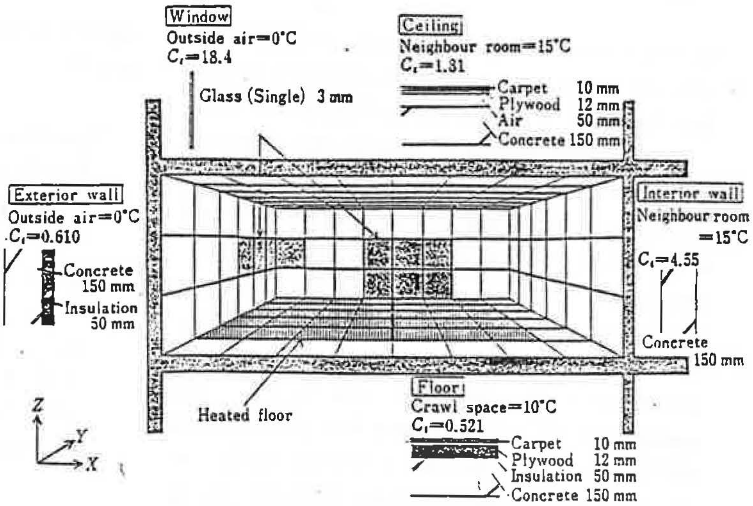


Fig. 1 Room model for calculation

T_{rb} : Mean radiant temperature of cube's surface b [K]
T_V : Vector radiant temperature [K]
t_{cl} : Mean temperature of outer surface of clothed body [°C]
V : Volume of room [m³]
v : Relative air velocity [m/s]
W : External mechanical power performed by human body [kcal/m²·h]
ε_i : Emittance of surface element i [—]
ρ : Air density [kg/m³]
σ : Stefan-Boltzmann constant [kcal/m²·h·K⁴]

1. Room model for calculation

Calculation is conducted on a room assumed to be used as a living room in a rectangular parallelepiped with a width of 6.4 m, depth of 4.8 m and height of 2.4 m as shown in **Fig. 1**. On the left and inner front exterior walls, glass windows are provided, and at the right and this side in front, interior walls are installed. Under the floor is a crawl space, and over the ceiling is another room.

The materials and dimensions of each wall, the wall conductances from the inside surface to the outside air, and the outside air temperatures are shown in Fig. 1. The air change rate caused by infiltration is assumed to be 0.1 1/h, namely, 7.4 m³/h. The indoor surface is assumed as the perfect diffusing surface under the Lambert's cosine law, and the emittance ε_i for long wave radiation is assumed to be 0.95 without taking the radiation transmission and short wave radiation into consideration.

After dividing the indoor surface into surface elements, S_i of 0.8 m×0.8 m, the surface temperature of each element, T_i, is calculated. The total number of surface elements is 180. In addition, the indoor air space is divided into cubic cells of 0.8 m×0.8 m×0.8 m. In the center of each an infinitely small cube is placed. This infinitely small cube was employed, according to Nakamura[6], as an imaginary detector to calculate vector radiant and mean radiant temperatures, and to convert the shape factor for a human body. The total number of cubes is 144.

2. Indoor surface and room air temperatures

2.1 Shape factor between surface elements

The shape factor of surface element j against that of i can be obtained by the equation below.

$$F_{ij}=\frac{1}{S_i}\int_{S_j}\int_{S_i}\frac{\cos\theta_i\cos\theta_j}{\pi r^2}\,dS_i\,dS_j \qquad \cdots\cdots(1)$$

This integral equation was theoretically calculated by Yamazaki[7] using the law of solid-angle projection and the contour integration method which results in an excellent program. Under the circumstances, all calculations of F_{ij} in this study are conducted by using the program by Yamazaki[7].

2.2 Gebhart's absorption factor

For the calculation of the radiation interaction between the surface elements, the absorption factor of Gebhart[3] is used. As the reflectance of the surface element i is $1-\varepsilon_i$ according to the law of Kirchhoff, the absorption factor of Gebhart can be calculated by the equation below.

for $i=1$ to N and $j=1$ to N

$$F_{ij}\varepsilon_j+\sum_{k=1}^{N}B_{kj}(1-\varepsilon_k)F_{ik}=B_{ij} \quad \cdots\cdots(2)$$

2.3 Heat balance of a surface element

For the surface element i (unheated surface element i in floor panel heating), the following heat balance equation can be established under steady state condition.

$$Q_{R+}-Q_{R-}+Q_{CV}+Q_{CD}=0 \quad \text{[kcal/h]} \quad \cdots\cdots(3)$$

where,

Q_{R+} : Heat flow rate absorbed by surface i caused by the radiation from the other surface elements $\left(=\sigma\sum_{j=1}^{N}\varepsilon_j B_{ji}T_j^4 S_j\right)$

Q_{R-} : Heat flow rate emitted from surface element i $(=\sigma\varepsilon_i T_i^4 S_i)$

Q_{CV} : Heat flow rate transferred from room air by convection $[=h_{c_i}(T_a-T_i)S_i]$

Q_{CD} : Heat flow rate conducted to surface element i from outside air through a wall $[=C_i(T_{o_i}-T_i)S_i]$

By substituting these values, T_i can be arranged as follows.

$$\sigma\varepsilon_i(1-B_{ii})T_i^4+(h_{c_i}+C_i)T_i -\left(\sigma\sum_{j=1(j\neq i)}^{N}\varepsilon_j B_{ji}T_j^4\frac{S_j}{S_i}+h_{c_i}T_a+C_iT_{o_i}\right)=0 \quad \cdots\cdots(3')$$

2.4 Heat balance of room air

The following heat balance equation can be established for the room air.

$$Q_{CA}+Q_V+Q_S=0 \quad \text{[kcal/h]} \quad \cdots\cdots(4)$$

where,

Q_{CA} : Heat flow rate transferred to room air from surface elements by convection $\left[=\sum_{i=1}^{N}h_{c_i}(T_i-T_a)S_i\right]$

Q_V : Heat flow rate entering by infiltration $[=c_p\rho nV(T_{out}-T_a)]$

Q_S : Heat flow rate supplied to room air by forced convective heating (0 for floor panel heating)

By substituting these values, T_a can be arranged as follows.

$$T_a=\frac{\sum_{i=1}^{N}h_{c_i}T_iS_i+c_p\rho nV\cdot T_{out}+Q_S}{\sum_{i=1}^{N}h_{c_i}S_i+c_p\rho nV} \quad \cdots\cdots(4')$$

2.5 Calculation procedure

T_i and T_a can be obtained from the equations (3') and (4') through the procedure below.

1) Substitute proper initial values for T_i and T_a.
2) Obtain the approximate value of T_i by solving eqution (3') with the Newton-Raphson method.
3) Obtain the approximate value of T_a with the equation (4') based on T_i obtained in 2).
4) Repeat the procedure of 2) and 3) until T_a converges.

2.6 Heat loss

After obtaining the surface temperature T_i and the room air temperature T_a, the heat loss Q_L [kcal/h] from indoor to outdoor can be calculated by the equation below.

$$Q_L=\sum_{i=1}^{N}C_i(T_i-T_{o_i})S_i+c_p\rho nV(T_a-T_{out}) \quad \cdots\cdots(5)$$

(at unheated floor in FPH)

In addition, the heat loss from the floor, wall and ceiling can be calculated individually.

3. Calculation of the radiant field

3.1 Shape factor between a cube surface and a surface element

The shape factor of the surface element i for the cube surface b can be calculated by the equation below.

$$f_{bi}=\int_{S_i}\frac{\cos\theta_b\cos\theta_i}{\pi r^2}dS_i \quad \cdots\cdots(6)$$

Same as F_{ij}, the above equation is calculated with the program by Yamazaki[7]. However, f_{bi} is calculated as the shape factor between a point and a rectangle by assuming that the size of each surface of a cube can be neglected when compared to the surrounding space.

3.2 Mean radiant temperature of a cube surface

The heat flow rate emitted from the surface element i is $\sigma\varepsilon_iT_i^4S_i$ [kcal/h], and the heat flow rate reflected by the surface i is $(1-\varepsilon_i)H_i$ when

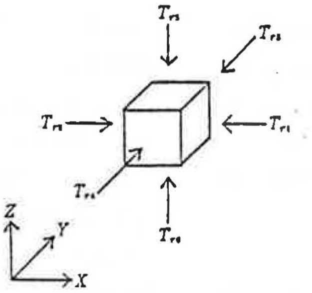


Fig. 2 Infinitely small cube

assuming the total sum of the radiant heat flow rate entering from the other surface elements is H_i. Therefore, the heart flow rate substantially emitted from the surface element i is a sum of both values above. The portion of heat flow rate substantially emitted that enters the cube surface b is $f_{ib}\{\sigma \varepsilon_i T_i^4 S_i+(1-\varepsilon_i)H_i\}$. On the other hand, when the radiation heat flow rate absorbed by the surface element i is expressed in terms H_i, $\varepsilon_i H_i$, then this value $\sum_{j=1}^{N}\sigma\varepsilon_j B_{ji} T_j^4 S_j$ can be set equal to the absorbed heat flow rate. And therefore,

$$H_i=\frac{1}{\varepsilon_i}\sum_{j=1}^{N}\sigma\varepsilon_j B_{ji} T_j^4 S_j$$

Therefore, the radiant heat flow rate substantially entering the cube surface b from the surface element i is,

$$f_{ib}\left(\sigma\varepsilon_i T_i^4 S_i+\frac{1-\varepsilon_i}{\varepsilon_i}\sum_{j=1}^{N}\sigma\varepsilon_j B_{ji} T_j^4 S_j\right)$$

And the heat flow rate entering from all surface elements is equal to the summation, $i=1$ to N, of the equation above. When making the mean radiant temperature of the surface b of cube T_{rb}, then

$$\sigma T_{rb}^4\,ds=\sum_{i=1}^{N} f_{ib}\left(\sigma\varepsilon_i T_i^4 S_i+\frac{1-\varepsilon_i}{\varepsilon_i}\sum_{j=1}^{N}\sigma\varepsilon_j B_{ji} T_j^4 S_j\right)$$

On the other hand, from the reciprocity theorem of shape factor,

$$f_{ib}S_i=f_{bi}\,ds$$

Therefore,

$$T_{rb}=\left\{\sum_{i=1}^{N} f_{bi}\left(\varepsilon_i T_i^4+\frac{1-\varepsilon_i}{\varepsilon_i}\sum_{j=1}^{N}\varepsilon_j B_{ji} T_j^4\frac{S_j}{S_i}\right)\right\}^{1/4} \quad \cdots(7)$$

3.3 Mean radiant temperature of a cube

Based on the mean radiant temperature of a cube surface, the mean radiant temperature of a cube is approximated by the equation below.

Table 1 Weighting factor of shape factor for each surface of infinitely small cube

Weighting factor	$\alpha_1\sim\alpha_4$	$\alpha_5\sim\alpha_6$
Seated position	0.199	0.102
Standing position	0.238	0.024

$$T_{mrt}=\frac{1}{6}\sum_{b=1}^{6}T_{rb} \quad \cdots\cdots(8)$$

3.4 Vector radiant temperature

As shown in Fig. 2, when T_{rb} calculated by equation (7) is used for $T_{r1}\sim T_{r6}$ and the unit vector on each coordinate axis is expressed as $\boldsymbol{i}, \boldsymbol{j}$ and $\boldsymbol{k}$ respectively, the vector radiant temperature $\boldsymbol{T}_v$ representing the directivity of radiant heat flow can be calculated by the equation below.

$$\boldsymbol{T}_v=(T_{r2}-T_{r1})\boldsymbol{i}+(T_{r4}-T_{r3})\boldsymbol{j}+(T_{r6}-T_{r5})\boldsymbol{k} \quad \cdots\cdots(9)$$

4. Calculation of thermal sensation index PMV

4.1 Shape factor between a surface element and a human body

Nakamura[8] proposed a method to approximate the shape factor of a surface element for a human body by applying weighting factors to the shape factors of cube surfaces. Namely, by using the weighting factors listed in **Table 1**, the shape factor between a surface element and a human body, F_{pi}, can be approximated by using the equation below.

$$F_{pi}=\sum_{b=1}^{6}\alpha_b f_{bi} \quad \cdots\cdots(10)$$

Where, a cube located 0.6 m above the floor represents a seated position, and a cube located 1.0 m above the floor represents a standing position.

4.2 Mean radiant temperature for a human body

Using the same form of equation (7), the mean radiant temperature for a human body, T_{pmrt}, can be calculated by the equation below.

$$T_{pmrt}=\left\{\sum_{i=1}^{N} F_{pi}\left(\varepsilon_i T_i^4+\frac{1-\varepsilon_i}{\varepsilon_i}\sum_{j=1}^{N}\varepsilon_j B_{ji} T_j^4\frac{S_j}{S_i}\right)\right\}^{1/4} \quad \cdots\cdots(11)$$

4.3 Calculation of PMV

The thermal sensation index PMV proposed by Fanger[1] is a function of room air temperature (T_a[K]), Partial pressure of water vapour in room air (P_a[mmHg]), mean radiant temperature

(T_{pmrt}[K]), relative air velocity (v[m/s]), metabolic rate (M[kcal/m²·h]), external power performed by the human body (W[kcal/m²·h]) and thermal resistance of clothing (I_{cl}[clo]), and can be defined by the equation below.

$$PMV=(0.352e^{-0.042M}+0.032)[(M-W)-0.35\{43-0.061(M-W)-P_a\}-0.42(M-W-50)-0.0023M(44-P_a)-0.0014M(34-T_a+273)-3.4\times10^{-8}f_{cl}\{(t_{cl}+273)^4-T_{pmrt}^4\}-f_{cl}h_c(t_{cl}-T_a+273)] \quad \cdots\cdots(12)$$

Where, the mean temperature of outer surface of clothed body, t_{cl}, can be determined by the equation below.

$$t_{cl}=35.7-0.032(M-W)-0.18I_{cl}[3.4\times10^{-8}f_{cl}\times\{(t_{cl}+273)^4-T_{pmrt}^4\}+f_{cl}h_c(t_{cl}-T_a+273)]$$

And the convective heat transfer coefficient for a human body, h_c[kcal/m²·h·K], and the clothing area factor for a human body, f_{cl}, can be determined by the equations below.

$$h_c=\begin{cases}2.05(t_{cl}-T_a+273)^{0.25} \\ 10.4\sqrt{v}\end{cases} \quad \text{greater value}$$

$$f_{cl}=\begin{cases}1.00+0.2I_{cl} & \text{for } I_{cl}<0.5\text{ clo} \\ 1.05+0.1I_{cl} & \text{for } I_{cl}>0.5\text{ clo}\end{cases}$$

The thermal sensation scale is defined corresponding to PMV as shown in Table 2.

Table 2 PMV and sensation scale

PMV	Sensation scale
3	Hot
2	Warm
1	Slightly warm
0	Neutral
−1	Slightly cool
−2	Cool
−3	Cold

5. Results and discussion

Results obtained by applying the calculation method being formulized in Sections 2. to 4. for the room shown in Fig. 1 for either floor panel heating (FPH) or forced convective heating (FCH) are described below.

5.1 Setting condition

The setting conditions for the calculations in Table 3 are as follows. The heated floor temperature for FPH and the heat supply for FCH were set so that the mean PMV values at a seated position inside the occupied zone are approximately 0 (neutral) after conducting several preliminary calculations. The occupied zone above is defined as a space located less than 1.6 m above the floor and more than 0.8 m from the wall surface.

In the case of FPH, 0.1 m/s was set for the air velocity of still air, and the values commonly used for the convective heat transfer coefficients were set according to the direction of heat flow, namely, upward, horizontal and downward directions respectively.

In the case of FCH, a comparatively large air velocity (0.5 m/s) was set, imagining a fully mixed state to obtain uniform room air temperature, and the values 1.5 times that of FPH was set for the convective heat transfer coefficients.

For the air humidity, clothing and metabolic rate, the values based on an imaginary room in ordinary winter condition were adapted.

5.2 Room air temperature and heat loss

Calculation results on the room air temperature,

Table 3 Setting condition for the calculation

Floor panel heating		Forced convective heating	
Heated floor 32.5°C		Supply heat 2 590 kcal/h	
v=0.1 m/s		v=0.5 m/s	
h_{ci}= Heated floor, Ceiling	4.0	h_{ci}= Ceiling	6.0
Wall, Window	3.0	Wall, Window	4.5
Unheated floor	1.0	Floor	1.5
[kcal/m²·h·K]		[kcal/m²·h·K]	

I_{cl}=1.0 clo, M=50 kcal/m²·h, W=0 kcal/m²·h
Relative humidity=40%
σ=4.88×10⁻⁸ kcal/m²·h·K⁴, $c_p\rho$=0.3 kcal/m³·K

heat loss and the mean PMV inside occupied zone are given in **Table 4**. Although slight errors were found, the mean PMV values inside the occupied zone were near to 0 (neutral). Naturally, the seated position was warmer than the standing position in the case of FPH, and no difference between the seated and the standing positions was found in the case of FCH.

The room air temperature in the case of FPH was significantly lower than that in the case of FCH by 3.4°C. This may prove the benefit of FPH enabling occupants to feel the freshness of the room air because of lower air temperature while providing warmth in a similar degree to FCH.

The heat loss (=heat supply) in FPH was smaller than that in FCH by approximately 15% illustrating the excellent efficiency of FPH. However, the value of heat loss is only considering indoor heat loss not indicating loss from the entire heating system. Especially in the case of FPH, heat loss to the crawl space tends to be enlarged hindering the heating efficiency unless proper insulation is applied.

5.3 Room surface temperature

The room surface temperature distribution of FPH and FCH is shown with contour lines on the perspective drawing in **Fig. 3**. Despite the 3.4°C lower room air temperature of FPH, the surface temperatures of the ceiling, walls and windows in both heating systems show the similar values although that of FPH is slightly lower.

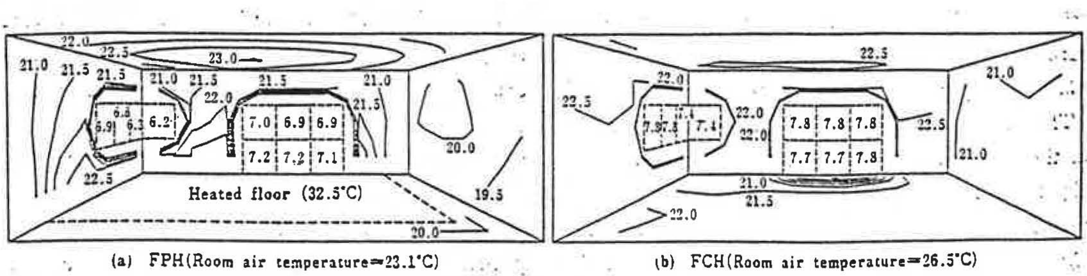
The temperature of the floor surface near the windows is significantly low due to radiant cooling by the windows. The surface temperature of the exterior wall (at the left side) is slightly higher than that of the interior wall (at the right side) due to the fact that the exterior wall is insulated while the interior wall is not.

5.4 Mean radiant temperature of a cube

The mean radiant temperature distribution of the cubes for FPH and FCH is shown with contour lines on the four cross sections of the perspective drawing in **Fig. 4**. In both heating systems, the mean radiant temperature is low near the windows clearly indicating the effect of radiant cooling. The mean radiant temperature of the rest of the

Table 4 Calculation results on the room air temperature, heat loss and the mean PMV

		FPH		FCH	
Room air temperature		23.1°C		26.5°C	
Heat loss [kcal/h] (%)	Floor	44 (2.0)		184 (7.1)	
	Interior wall	591 (27.3)		729 (28.1)	
	Exterior wall	272 (12.6)		280 (10.8)	
	Ceiling	401 (18.5)		430 (16.6)	
	Window	806 (37.2)		908 (35.1)	
	Air change	51 (2.4)		59 (2.3)	
	Total	2165		2590	
The mean PMV inside occupied zone		Seated 0.021	Standing −0.129	Seated −0.0007	Standing −0.0108



(a) FPH(Room air temperature=23.1°C) (b) FCH(Room air temperature=26.5°C)

Fig. 3 Room surface temperature

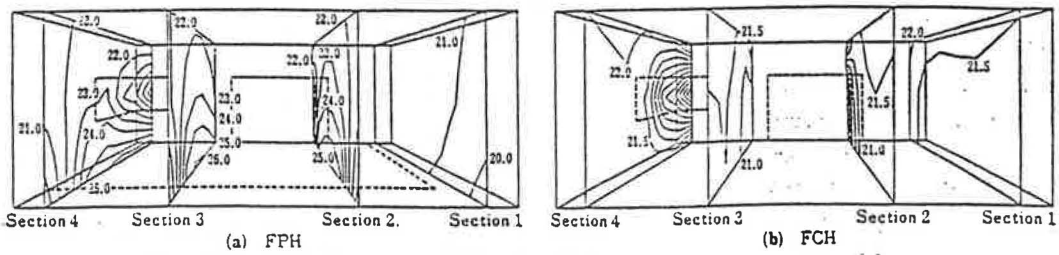


Fig. 4 Mean radiant temperature of cube

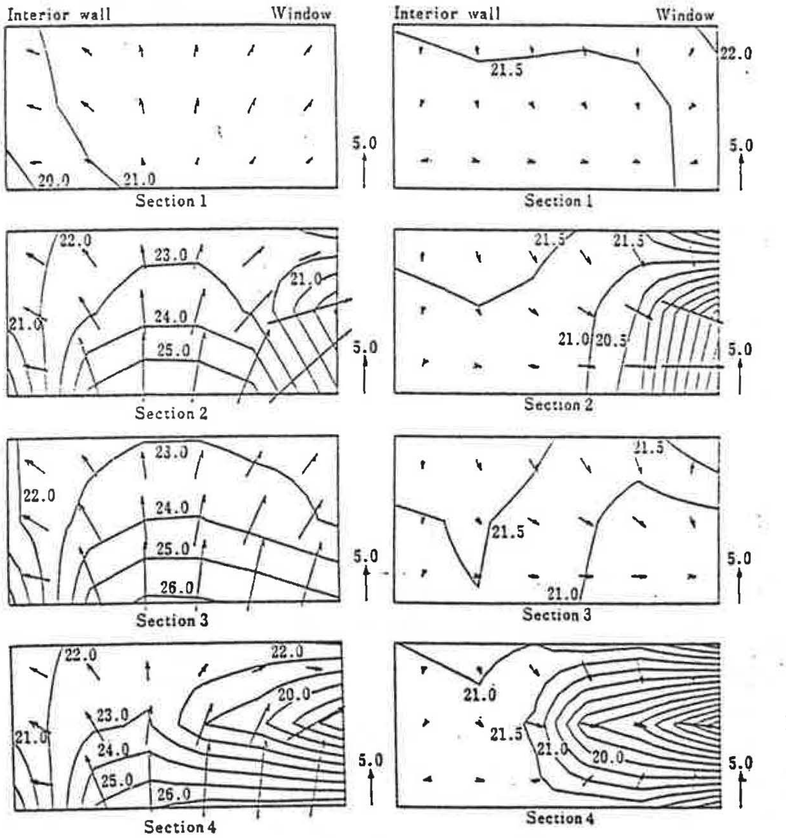


Fig. 5 Vector radiant temperature on the four cross sections

room is uniform in FCH, while it is higher than FCH from the floor surface to the ceiling surface by 5~0°C in FPH showing the comfort characteristic so called "Keeping the head cool and the feet warm".

5.5 Vector radiant temperature

The vector radiant temperature being projected on the four cross sections for FPH and FCH in **Fig. 4** is indicated by overlapping it with the contour lines of the mean radiant temperature in **Fig. 5**. In FPH, an intensive directivity toward the ceiling, interior wall and windows from the heated floor is observed. While in FCH, no significant directivity is found except that toward the windows.

Fig. 6 shows the distribution of the vector radiant temperature viewed from the top with a perspective drawing. As described, the directivity of the radiation field can be expressed precisely by introducing the concept of the vector radiant temperature.

5.6 Predicted mean vote

The contour lines of PMV at seated and standing positions for FPH and FCH are shown on a plan in Fig. 7. In both cases, it is low near the windows

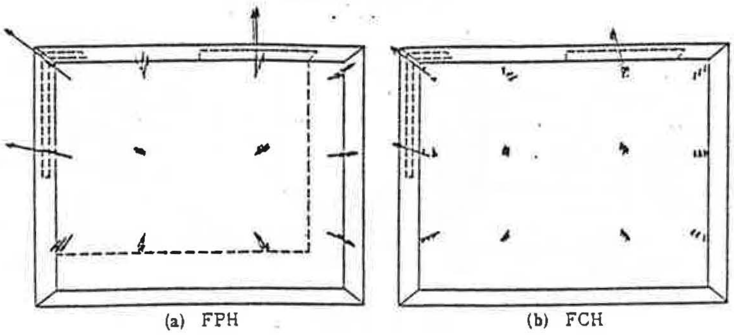


Fig. 6 Vector radiant temperature viewed from the top

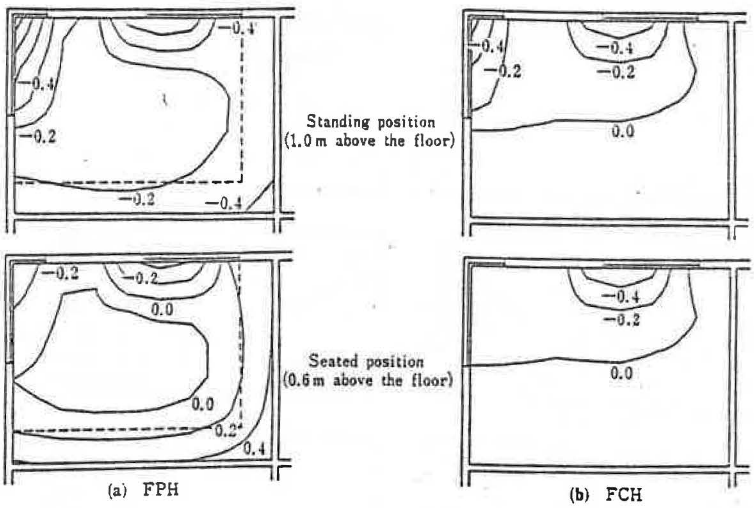


Fig. 7 The contour lines of the PMV

due to radiant cooling. In FPH, it is lower at a standing position than that at a seated position, and in FCH, no difference between the seated and standing positions is found except at a place near the windows.

Through calculations of PMV, the evaluation index of thermal environments (Predicted Percentage of Dissatisfied) and the thermal non-uniformity index (Lowest Possible Percentage of Dissatisfied) both proposed by Fanger[1] can easily be obtained.

These calculations were conducted by the HITAC M-280H at the Computer Center of Tokyo University. The CPU time for all calculations was less than 1 minute, and a calculation of the room surface temperature required only about 10 seconds.

6. Summary and conclusions

Taking a rectangular parallelepiped room into our consideration, the calculation method to obtain the room surface temperature and the radiation field caused by radiation interaction, and the distribution of the thermal sensation index for a human body was developed. Some of the characteristics of FPH and FCH systems caused by radiation interaction were quantitatively clarified by applying this calculation method.

Since this calculation method is based on the assumption of uniform room air temperature and air movement, the serious influences due to forced convection or natural convection can not be evaluated. However, the authors believe that this calculation method is thoroughly effective as a simplified prediction method for first estimation.

Needless to say, this calculation method can easily be combined with that of air convection or wall heat conduction under non-steady state conditions through which a more realistic prediction of themal environments can be performed.

Acknowledgement

In conducting this study, the authors wish to thank for their assistance and efforts to Messrs. R. Kajiya, T. Hiramatsu, T. Karasawa, Y. Hirahara, N. Sawai, H. Ishido, and Mmes. M. Hattori and K. Akutsu.

References

1) P. O. Fanger : Thermal Comfort, (1972), McGraw-Hill Book Company
2) M. Kaizuka : Numerical Prediction of Room Air Distribution (in Japanese), Journal of SHASE of Japan, 58-3 (1984-3) "Numerical Calculation of Air Currents in a Room", translated by the National Bureau of Standards from the above.
3) B. Gebhart : A New Method for Calculating Radiant Exchanges, ASHRAE Trans., Vol. 65 (1959).
4) Y. Sakamoto : Study on the Numerical Analysis of Room Air Distribution (in Japanese), Thesis for PhD. (1977), Tokyo University
5) T. Hiramatsu and M. Kaizuka : Numerical Predictions of Room Air Distribution in Long Wave Radiation Heatings (in Japanese), Summaries of Technical Papers of Annual Meeting, Architectural Institute of Japan (1981), pp. 253~254
6) Y. Nakamura : A Basic Study on the Method of Bio-meteorological Architecture (in Japanese), Thesis for PhD. (1979), Kyoto University
7) H. Yamazaki : Shape Factor Calculation and Computer Graphics, (1983), Proceedings of the Fourth International Symposium on the Use of Computers for Environmental Engineering Related to Buildings, pp. 559~564
8) M. Kaizuka and S. Iwamoto : A Calculation of the Thermal Environment and Heat Balance in the Heated Room (in Japanese), Proceedings of Annual Meeting of Kantou-Shibu of Architectural Institute of Japan, (1985), pp. 25~28
9) M. Kaizuka and S. Iwamoto : A Calculation of the Radiant Temperature Distribution and the Heat Balance in the Room (in Japanese), Summaries of Technical Papers of Annual Meeting of Architectural Institute of Japan, (1985), pp. 399~400
10) M. Kaizuka and S. Iwamoto : A Calculation of the Thermal Comfort Index Distribution in the Heated Room (in Japanese), Proceedings of Annual Meeting of the Society of Heating, Air-Conditioning and Sanitary Engineers of Japan, (1985), pp. 581~584
11) M. Kaizuka and S. Iwamoto : A Numerical Calculation on the Distribution of Surface Temperature and Thermal Comfort Index Caused by Radiation Interaction in a Heated Room, NBS/CBT Building Technology Symposium, Room Convection and Indoor Air Quality Simulation Modeling, National Bureau of Standards, Gaithersburg, Maryland, (March 11, 1986)

(Received May 28, 1986)

放射授受による暖房室内の表面温度と温冷感指標の分布の数値計算

貝塚正光*1
岩本静男*2

キーワード：数値解析・熱環境・放射授受・熱的快適性・PMV・平均放射温度・シミュレーション・床暖房・強制対流暖房

室内の熱環境は，気温・湿度・気流・放射などの物理的条件と，着衣・代謝などの人体側条件とによって評価される．これらを総合した温冷感指標の一つとしてPMVがP.O. Fanger[1]によって提唱されている．

通常の空調室内では，これらの物理的条件やPMVは一様ではなく，その不均一さは在室者の不快や空調効率の悪化を招く．例えば，コールドドラフト，大きな上下温度差，大きな片側放射冷却や加熱などの障害を生じたりする．したがって，熱環境をより的確に評価するためには，このような物理的条件やPMVなどの分布をも把握する必要があり，数値予測法はそのための有力な方法となりつつある．

室内気流や気温の分布の数値予測法については，例えば貝塚[2]に概説されているように，多くの研究がなされており，実用的にも用いられる段階に至っている．また，本論文で取り扱う室内放射授受の計算法についてはB. Gebhart[3]，P.O. Fanger[1]などがすでに定式化している．さらに，坂本[4]は加熱面のある模型室内に対して気流流動と放射授受を組み合わせた計算を行っており，平松・貝塚[5]は二次元暖房室内のベクトル放射温度や作用温度の分布の計算をも示している．

本論文は，室内熱環境の数値予測法を確立するための研究の一環として，特に放射授受による熱環境の分布のみに着目し，壁面温度・ベクトル放射温度・平均放射温度・PMVなどの算法を定式化し，床暖房または強制対流暖房を想定した室内に適用し，両暖房方式の特性の一面が把握できることを示すものである．

二次元室内を対象とした平松・貝塚[5]では，気流流動と放射授受の計算を組み合わせ，気温や対流熱伝達率の分布をも未知量として数値予測を行ったが，本報では放射授受の計算法を吟味するために気流流動の計算は組み込まず，気温と気流は一様なものと仮定し，対流熱伝達率は既知なものとして適切な値を用いた．

このような計算法は，基本的にはP.O. Fanger[1]がすでに示したものと同様である．異なる点は，気流計算と組み合わせることを考慮して室内表面を細かく分割したこと，放射場の方向性を表す中村[6]によるベクトル放射温度を計算したこと，人体の形態係数を微小立方体の形態係数から近似する中村[6]の方法を用いたこと，射度(radiosity)の代わりにB. Gebhart[3]の吸収係数を用いたこと，形態係数の計算に山崎[7]による優れた算法を用いたこと，などである．

なお，本論文の一部は筆者らによって，参考文献8)，9)，10)，11)にすでに報告したものであることを付記する．

*1 明治大学工学部建築学科　正会員
*2 明治大学大学院工学研究科博士後期課程　学生会員

(昭和 61. 5. 28 原稿受付)