ANALYSIS OF THERMAL SENSATION IN A RADIANT COOLED ROOM BY MODIFIED PMV INDEX

Toshiyuki Miyanaga[1], Yukio Nakano[2]

[1&2]Customer Systems Department,
Komae Laboratory,
Central Research Institute of Electric Power Industry,
2-11-1, Iwato-kita, Komae-shi, Tokyo 201-8511 JAPAN

AIVC 12064

ABSTRACT

The objective of this study is to develop a method for analyzing the thermal environment and the thermal sensation in the radiant cooled room.

In this paper, detailed three-dimensional models of the room and the indoor occupant were constructed and the steady-state thermal environment was analyzed by the conjugate heat transfer analysis of thermal radiation and convection. The modified PMV index was proposed to evaluate the thermal sensation of the occupant in the asymmetrical thermal radiant environment such as the radiant cooled room. The modified PMV index was simply introduced by replacing the heat exchange by convection and radiation around the real human body, which approximately calculated in the Fanger's equations for the PMV index, with the analyzed results around the body surface model.

The analyzed values of air temperature and wall temperature distributions in the room were in good agreement with the measured values. The modified PMV index was closer to the actual thermal sensation by real subjects than PMV indices. The validity of the analysis method and models used were examined.

KEYWORDS

Radiant cooled room, Modified PMV index, Conjugate heat transfer analysis of radiation and convection

1. INTRODUCTION

A radiant cooling system is a kind of air-conditioning method that removes thermal load from a room mainly by radiation heat exchange between a cooled radiant panel and objects. A more satisfactory thermal sensation can be obtained in the radiant-cooled space than in spaces air-conditioned by other methods. Various objects and heat sources are included in the living space. Therefore, the complicated heat transfer such as radiation and convection will occur in it. In some cases, occupants as heat sources have a thermal effect upon their living space. Analysis of complicated heat transfer in the radiant-cooled room is required to systematically design the radiant cooled room for obtaining satisfactory thermal sensation. In addition, the method for evaluating the thermal sensation of the occupant in the asymmetrical thermal radiant environment is required. The PMV index introduced by Fanger [1] is generally useful for evaluating the thermal sensation of the occupant. However, the accuracy of the PMV index is not guaranteed in the asymmetrical thermal condition. We assume the reason is that the heat exchange by radiation around the human body calculated in Fanger's equation will be especially different from the real value of that in the asymmetrical thermal radiant environment.

The objective of this study is to develop a method for analyzing the thermal environment and the thermal sensation in the radiant cooled room.

In this paper, detailed three-dimensional models of the room and the indoor occupant were constructed and the steady-state thermal environment was analyzed by the conjugate heat transfer analysis of thermal radiation and convection. The modified PMV index was proposed to evaluate the thermal sensation of the occupant in the asymmetrical thermal radiant environment such as the radiant cooled room. The modified PMV index was simply introduced by replacing the heat exchange by convection and radiation around the real human body, which approximately calculated in the Fanger's comfort equation for the PMV index, with the analyzed results around the body surface model. The validity of the analysis method and models used were examined.

Nomenclatures are listed in the end of the paper.

Table 3 Thermal and optical properties of solid surfaces in the room

Surface	Surface temp. (℃)	Over all Heat transfer coefficient (W/m²/℃)	Reflectivity (%)
Radiant panel	20.0	-	5
Wall (E,W,S,N)	unknown	0.96	5
Floor	unknown	1.21	5
Fluorescent light	45.0	-	5

4. RADIANT COOLING EXPERIMENT

The schematic view of the experimental room is shown in Figure 4. The radiant panel cooled by water is equipped at the ceiling. The entire room is almost coated with a thick layer of heat insulating material (glass wool). The inlet and outlet of air are equipped at the lower side of the east and west walls, respectively. The inlet air velocity was always kept at 0.67 m/sec, parallel to the floor.

The experimental condition was listed in Table 4. Two occupants were always present in the room during the course of experiment. The indoor air temperature, humidity, temperature of the radiant panel and the outdoor air temperature were controlled according to the target value all the time by the specially designed air handling unit.

Various values of air, wall and mean radiant temperature, air velocity and humidity at several points in the room were recorded at regular 1 minute intervals.

Table 4 Experimental condition

Surface temp. of panel (℃)	Air temp. (℃)	Relative humidity (%)	Inlet air velocity (m/sec)	Occupants
20	26~29	40~50	0.67	2

5. EVALUATION TEST OF THERMAL SENSATION BY REAL SUBJECTS

Evaluation tests of the thermal sensation were performed in the room by real subjects. A total of 72 real subjects of which 36 were female, participated in the evaluation test. The mean age was 26.4 years. The mean clothing value was 0.5 clo. All participants sat quietly and their activity level was expected as 1.0 met. They were regarded to get accustomed to the indoor thermal environment within 30 minutes. Their thermal sensation votes were obtained by questionnaires at regular 10 minute intervals 30 minutes after they entered the room. The thermal sensation scale was the same as that of the PMV index.

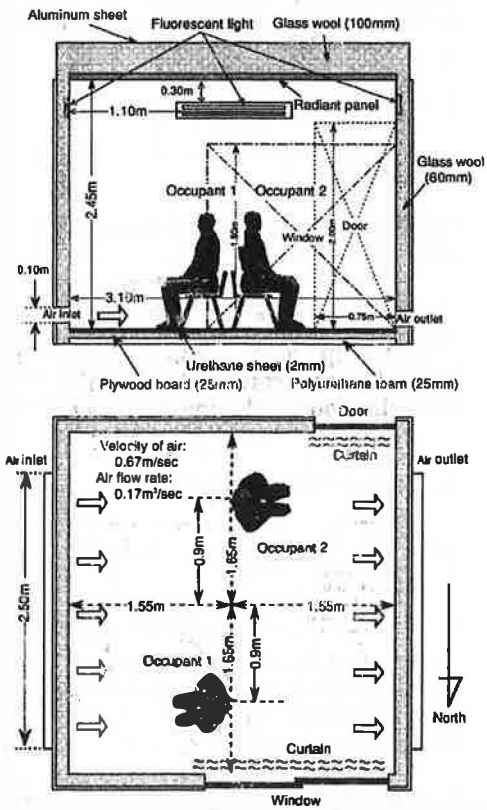

Figure 4 Schematic view of experimental room

6. ANALYSIS RESULTS AND DISCUSSION

6.1 Temperature of indoor air and solid surface

The analyzed values of vertical air temperatures at the center of the floor was compared with the measured ones in the case of T_p=20℃, $T_{typical\ air}$=28℃ (Figure 5).

Analyzed values were good agreement with the measured ones.

The analyzed values of temperature on the center of each solid surface was also compared with the measured one (Figure 6). The horizontal axis of Figure 6 is the operative temperature (T_{op}). T_{op} is determined to be a mean value of $T_{typical\ air}$ and T_r in this paper. Analyzed values were in good agreement with the measured ones except for the floor. Errors remain less than 1.5℃.

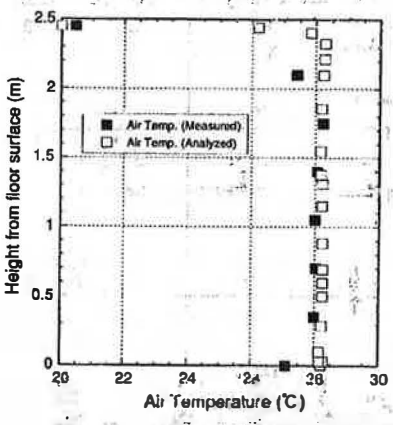

Figure 5 Comparison of analyzed vertical air temperature at the center of the floor with measured one (T_p=20℃ $T_{typical\ air}$=28℃)

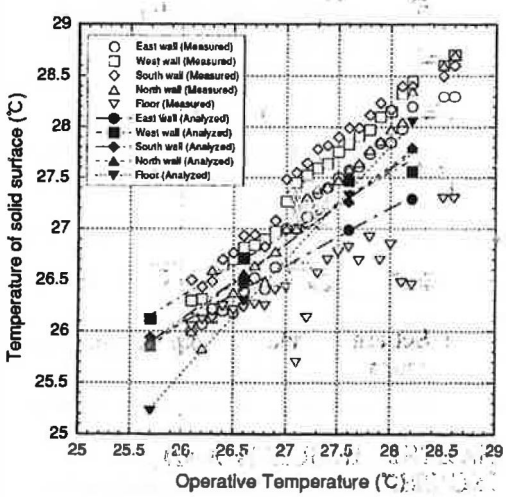

Figure 6 Comparison of analyzed surface temperature on the the center of each solid surface with measured one

6.2 Comparison of PMV index and modified one with thermal sensation vote by real subjects

PMV and modified PMV indices were compared with mean values of thermal sensation votes by real subjects (Figure 7). The horizontal axis of Figure 7 is T_{op}. It is clearly found that modified PMV indices were closer to thermal sensation by real subjects than PMV indices.

As mentioned in the introduction of this paper, we assume one of the reason is that the heat exchange by radiation around the human body calculated in Fanger's equation will be especially different from the real value of that in the asymmetrical thermal radiant environment. Values of Q_c (Eq.(4)) and Q_r (Eq.(5)) calculated by Fanger's equations were compared with those of Q_c' (Eq.(6)) and Q_r' (Eq.(7)) analyzed by the conjugate analysis in order to examine the above assumption (Figure 8). Values of Q_c were similar to those of Q_c'. On the other hand, values of Q_r were noticeable different from those of Q_r'. Values of Q_r' exceeded those of Q_r. It was found that the difference between Q_r and Q_r' caused the difference between PMV and modified PMV. The three-dimensional body surface model used in the conjugate analysis has the advantage of simulating the radiation heat transfer similar to the real human body. Therefore, we consider that values of Q_r' are appropriate and our assumption was correct in this case.

We considered our analysis method and models to be sufficiently valid for simulating the real thermal environment in the radiant cooled room.

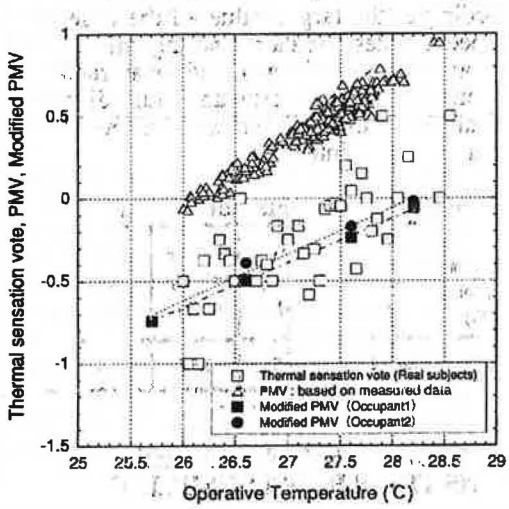

Figure 7 Comparison of PMV index and modified one with thermal sensation vote by real subject

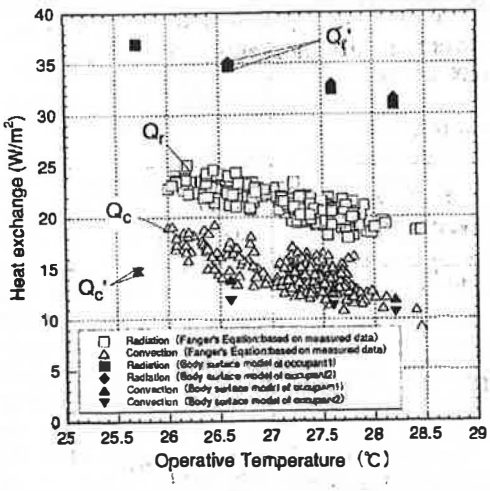

Figure 8 Comparison of heat exchange by convection and radiation calculated by Fanger's equations with that analyzed by the conjugate analysis

7. CONCLUSION

(1) Detailed three-dimensional models of the radiant cooled room and occupants were constructed and the steady-state thermal environment was analyzed by the conjugate heat transfer analysis method of thermal radiation and convection.

(2) The analyzed values of air temperature and temperature on the center of each solid surface were compared with the measured ones. Analyzed values were in good agreement with the measured ones.

(3) The modified PMV index was proposed to evaluate the thermal sensation of the occupant in the asymmetrical thermal radiant environment such as the radiant cooled room. The modified PMV index was simply introduced by replacing the heat exchange by convection and radiation around the real human body, which approximately calculated in the Fanger's comfort equation for the PMV index, with the analyzed results around the body surface model.

(4) PMV and modified PMV indices are compared with mean values of thermal sensation votes by real subjects. Modified PMV indices are closer to thermal sensation by real subjects than PMV indices.

(5) We considered our analysis method and models to be sufficiently valid for simulating the real thermal environment in the radiant cooled room.

It has not been completely clarified as to what type of effect the asymmetrical radiant environment has on the thermal sensation. Further physiological investigations of a human body will be required for the accurate prediction of thermal sensation in such an environment.

ACKNOWLEDGEMENT

Many thanks to Mr. Koushirou Maeda (Denryoku Computing Center Co.), Mr. Masakuni Horiguchi and Mr. Shinji Kanamitsu (Science University of Tokyo) for their technical support.

NOMENCLATURES

A	Area of the solid surface (m²)
A_{Body}	Surface area of the body surface model (m²)
F	View factor between small surfaces
G	Gebhart's absorption factor
f_{cl}	The ratio of the surface area of the clothed body to the surface area of the nude body
h	Convective heat transfer coefficient calculated by the conjugate analysis (W/m²/℃)
h_c	Convective heat transfer coefficient introduced by Fanger's equation (W/m²/℃)
I_{cl}	Thermal resistance of clothing (clo) (1clo = 0.155m²℃/W)
K	Overall heat transfer coefficient (W/m²/℃)
M	Metabolism (W/m²)
N	Total number of small element
P_a	Vapor pressure of air (pa)
Q_c	Heat exchange by convection calculated by Fanger's equation (W/m²)
Q_c'	Heat exchange by convection calculated by the conjugate analysis (W/m²)
Q_{evp}	Heat loss by evaporation (W/m²)
Q_r	Heat exchange by radiation calculated by Fanger's equation (W/m²)
Q_r'	Heat exchange by radiation calculated by the conjugate analysis (W/m²)
Q_{res}	Heat loss by respiration (W/m²)
T_a	Air temperature of the first cell near a solid surface (℃)
T_{ext}	External temperature of the room (℃)
T_{cl}	Temperature of the clothing surface (℃)

T_{op}	Operating Temperature (℃)
T_p	Temperature of the radiant panel (℃)
T_r	Mean radiant temperature (℃)
$T_{typical\ air}$	Typical Air temperature around the human body (℃)
T_w	Temperature of the solid surface (℃)
V_{air}	Air velocity (m/sec)
ε	Diffuse emissivity (=1- ρ)
ρ	Diffuse reflectivity
σ	Stefan-Boltzmann constant (=5.67×10⁻⁸) (W/m²/℃⁴)

Subscripts

i,j,k Number of small element

REFERENCES

[1] P.O.Fanger ; Thermal Comfort, McGraw-Hill Book Company, New York, 1972.

[2] G.P.Mitalas, D.G.Stephenson ; FORTRAN IV Program to Calculate Radiant Interchange Factors, National Research Council of Canada, Division of Building Research, Ottawa, Canada, DBR-25, 1966.

NOTE 1

Calculation method of view factor considering obstacles of thermal radiation

Five points for judging obstacles (p_t) are placed at four vertices and at the center point of each quadrilateral small element as shown in Figure A-1. Two weighting coefficients (w) are determined, one at the vertices and another at the center point. The sum total of w is equal to 1. The value of w is equal to 1/6 at each vertex. The value of w is equal to 1/3 at the center point. These values were previously determined to be sufficiently valid for the accurate judgment of obstacles between small elements.

The obstacle level between small elements i and j (V_{ij}) is determined quantitatively by the intersection test between the lines p_ip_j and other small elements. A shadow function (S) is introduced. V_{ij} and S are expressed as follows.

$$V_{ij} = \sum_{p_j=1}^{5}\left[\omega(p_i)\sum_{p_j=1}^{5} S(p_i,p_j)\,\omega(p_j)\right] \qquad \text{(A-1)}$$

$$\begin{cases} S(p_i,p_j)=1 \\ \quad : \textit{obstacles don't exist between } p_i \textit{ and } p_j. \\ S(p_i,p_j)=0 \\ \quad : \textit{obstacles exist between } p_i \textit{ and } p_j. \end{cases} \qquad \text{(A-2)}$$

The view factor, considering obstacles between small elements i and j (F_{ij}), is expressed as

$$F_{ij} = V_{ij}F'_{ij} \qquad \text{(A-3)}$$

The view factor, disregarding obstacles between small elements i and j (F'_{ij}), is calculated by Mitalas and Stephenson's method [2].

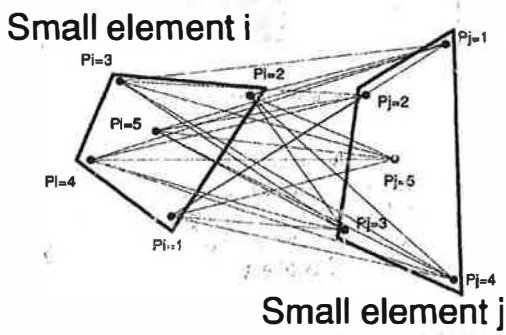

Figure A-1 Judgment of obstacle between small elements

NOTE 2

Equations for Q_{Res}, Q_{Evp}, f_{cl}, h_c, T_{cl}

Equations cited from the literature [1] are expressed as follows.

$$Q_{Res} = 1.7\times10^{-5}M(5867-P_a) + 0.0014M(34-T_{typical\ air}) \qquad \text{(A-4)}$$

$$Q_{Evp} = 3.05\times10^{-3}\{5733-6.99M-P_a\} \qquad \text{(A-5)}$$

$$f_{cl} = \begin{cases} 1.00+0.2I_{cl} & \textit{for } I_{cl}<0.5 \\ 1.05+0.1I_{cl} & \textit{for } I_{cl}>0.5 \end{cases} \qquad \text{(A-6)}$$

$$h_c = \begin{cases} 2.38(T_{cl}-T_{typical\ air})^{0.25} \\ \quad \textit{for } 2.38(T_{cl}-T_{typical\ air})^{0.25} > 12.1\sqrt{V_{air}} \\ 12.1\sqrt{V_{air}} \\ \quad \textit{for } 2.38(T_{cl}-T_{typical\ air})^{0.25} < 12.1\sqrt{V_{air}} \end{cases} \qquad \text{(A-7)}$$

$$T_{cl} = 35.7-0.028M-0.155I_{cl}\times \left[3.96\times10^{-8}\times f_{cl}\times\left\{(T_{cl}+273.15)^4 - (T_r+273.15)^4\right\} - f_{cl}h_c(T_{cl}-T_{typical\ air})\right] \qquad \text{(A-8)}$$