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Applicability and limitations of a numerical method in thermal environment analysis for air-conditioned rooms

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APPLICABILITY AND LIMITATIONS OF A NUMERICAL METHOD IN THERMAL ENVIRONMENT ANALYSIS FOR AIR-CONDITIONED ROOMS

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SUMMARY

This paper describes a numerical simulation method to predict thermal environments in air-conditioned rooms. The method is capable of treating radiative interactions between the room wall surfaces. Basic equations are solved with a finite difference method together with a heat balance equations at the inner surface of walls. Some case studies are attempted supposing three types of heating systems:(1)Forced convection hating systems. (2)Combined convective and panel heating systems. (3)Panel heating systems. In the case studies,the computer code are examined with two simple room model focusing on the applicabilities of the method to the room thermal environments analysis. Room thermal levels are determined with the averaged value of PMV in the occupied zone. Reasonable results are shown in all case studies and present method will be applicable to practical designing in various heating or cooling systems. However,detailed validation of the method should be discussed together with some reliable experimental data.

Introduction

Recently many researchers have applied the numerical analysis to evaluate room air and temperature distributions and practically useful results are presented(1,2). However, not so many works were reported about the predictions of room thermal environments including evaluations of heat losses through enclosing walls. For the precise prediction of thermal qualities in residential rooms installed various heating or cooling systems, the computer code should be so arranged to have following features:

1. The accuracy of predicted flow fields is acceptable.
2. The accuracy of predicted levels of room air temperature is acceptable.
3. It is possible to set thermal boundary conditions outside the room.
4. It is possible to treat radiative heat transfer.
5. It is possible to calculate some thermal comfort indexes distributions.

Although,predicted flow fields are influenced by many factors,such as calculating sizes of mesh systems,difference schemes,inlet flow conditions, adopted turbulent models,etc., detailed discussions for each factor were not presented here.

One of the most important conditions for predicting thermal environments in air-conditioned rooms is that the predicted temperatures are acceptable levels. Although the room temperature levels are not so sensitive to slight change of flow fields, change of heat transfer rate through enclosing walls may directly influences them. Therefore,it is important to evaluate precisely the heat transfer through the enclosing walls to the room air.

In designing heating/cooling systems, some thermal comfort indexes are often used as a measure of thermal sensation in the occupied zone. In the ordinary residential rooms,the distributions of thermal sensation are not uniform mainly because of non-uniform surface temperature of enclosing walls. Under such conditions, radiative heat exchange should be considered in order to predict thermal comfort indexes properly.

The main subject of present paper is to investigate applicabilities and limitations of a computer code through a few case studies. The computer code developed by authors was equipping the properties described above and reasonable results have been obtained in cases of forced convection heating systems(3,4). In the case studies, three types of heating systems were examined:

1. Forced convection heating systems.

2. Panel heating systems.

3. Combined convective and panel heating systems.

Predicted results for each system were compared under nearly same thermal environments which were controlled by the average values of predicted mean vote(PMV) in the occupied zones. At first,fundamental characteristics of the computer code were examined using a two-dimensional(2-D) room model. Then,as a more practical case, it was applied to a three-dimensional(3-D) room model. In both cases, the computer code worked fairly well, and was proved to be applicable for various heating/cooling systems.

Outline of Calculating Procedure

Flow Analysis

To calculate room air flows,basic equations with Boussinesq approximation were solved using a finite difference method with the power law scheme(5). Two equation turbulence model(6)(turbulent kinetic energy k and its dissipation rate ϵ) was used to express the eddy viscosity. The values of empirical constants in it are as follows: $C_\mu = 0.09$, $\sigma_k = 1.0$,$\sigma_\epsilon = 1.3$,$C_1 = 1.44$, $C_\mu = 1.92$(6).

At each near wall grid point P,following wall functions(7) were used:
For the calculation of velocity components u,v, momentum flux at wall surface τ_w was given by

$$\tau_w = \frac{u_P}{u^* \ln(E y_P^+)}$$

(1)

where

$y_P^+ = C_\mu^{0.25} k_P^{0.5} y_P/\nu$
$u^* = (\tau_w/\rho)^{\frac{1}{2}}$
y_P:distance between grid point P and wall surface [m]
ρ:density of air [N/m^3]
ν:kinetic viscosity [m^2/s]

For the turbulent kinetic energy k,wall flux was set equal to 0 and

$$\int_0^{y_P} \epsilon dy$$

(2)

was used in the dissipation term.
Dissipation rate of turbulent kinetic energy was calculated by

$$\epsilon_P = \frac{C_\mu^{0.75} k_P^{1.5}}{\kappa y_P}$$

(3)

Calculations were performed with 27×34 non-uniform mesh system for 2-D cases and 24×26×22 non-uniform mesh system for 3-D case studies.

Calculations of Heat Flux at Fluid Solid Interfaces

To evaluate the heat loss through walls precisely, the effect of radiative heat transfer should be considered not only in case of panel heating systems but also in case of some forced convective heating systems. In the present calculation, a heat balance equation is solved at the wall surface to determine the thermal boundary conditions for the flow calculations. Assuming one dimensional heat transfer through walls,the steady state heat balance equation is expressed as:

$$q_C + q_R + q_T + q_P = 0$$

(4)

and

$$q_C = \alpha_C(T_f - T_w)$$

(5)

$$q_R = \varepsilon_R \sigma (T_{mrt}^4 - T_w^4)$$

(6)

$$q_L = K_w(T_O - T_w)$$

(7)

where

T_w:inner surface temperature of wall[K]
T_{mrt}:mean radiant temperature at wall surface[K]
T_O:outside temperature[K]
q_C:convective heat flux from wall to room air[W/m^2]
q_L:heat loss through wall[W/m^2]
q_R:radiative heat flux at the wall surface[W/m^2]
q_p:heat flux at heating panel surface(heat source at the wall surface)[W/m^2]
α_C:inner surface conductance of wall[W/m^{2}K]
K_W:thermal transmittance of wall [W/m^{2}K]
σ:Stefan-Boltzmann constant [W/m^2/K^4]
ε_R:emissivity of wall surface[–]

Applying Reynolds' analogy to equation (1), α_C in equation (4) is expressed as

$$\alpha_C = \frac{\rho C_p C_\mu^{0.25} k_P^{0.5}}{(\sigma_t/\kappa)\ln(Ey_P^+) + P(Pr, \sigma_t)}$$

(8)

where

Pr:Prandtl number [–]
σ_t:turbulent Prandtl number [–]

P(Pr,σ_t) accounts for the resistance to heat transfer across the viscous sublayer and is known as P-function(8).

In the calculation of radiation interaction between walls,each wall was divided into small elements. In the 2-D cases,the sizes of the elements were equal to the calculating mesh sizes,however,in 3-D cases, all walls were divided into 10×10 elements to reduce the computer effort in the calculation of angle factors between them. Equation (4) was solved iteratively using Gebhart's absorption factor B_{ij}. B_{ij} is the absorption factor between wall element i and j and defined as(9)

$$B_{ij} = \varepsilon_i f_{ij} + \sum_{k=1}^{n} f_{ik}(1 - \varepsilon_k) B_{kj}$$

(9)

where,f_{ij} is angle factor between wall element i and j.

Thermal Sensation Index

To evaluate thermal sensation or to control thermal environments in rooms, not only air temperature and air velocity but also radiant temperature are important factors,especially,in case of non-uniform enclosing wall surface temperatures. Under such conditions,some thermal sensation indexes may be useful standards to compare heating/cooling systems. In the following case studies, the predicted mean vote(PMV)(10) was calculated at every calculation cells. Then,all calculations were controlled with their average values in occupied zone, PMV_{oz} to fulfill nearly same thermal conditions. Here,occupied zone is defined that the positions in it satisfy $0.4m \leq \bar{x} \leq 1.5m$,where,$\bar{x}$ is vertical distance from the floor.

In the calculation of PMV, following facts were assumed:

1. Metabolic rate is constant and equal to 58 [W/m^2].
2. Clo value is equal to 0.6.
3. Humidity ratio χ is constant and equal to 0.0103[Kg/Kg$'$]

In the calculation of mean radiant temperature,human body is assumed to be a small sphere. As a reference,the New Effective Temperature ET^*(11) was also calculated.

Calculating Conditions

In this report,three heating systems were introduced to investigate the basic applicabilities of the computer code:

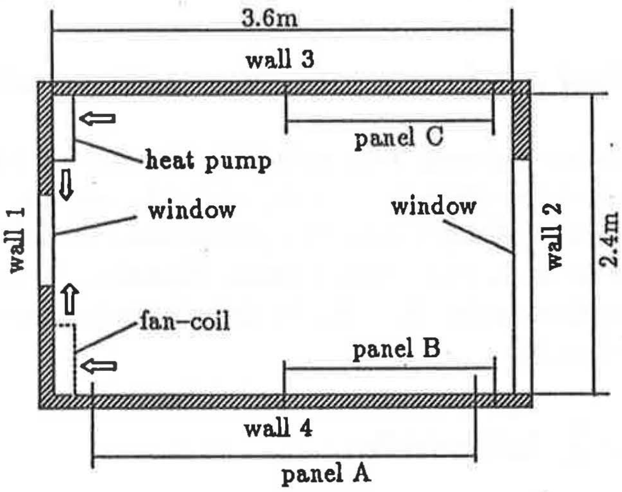


Fig.1 2–D room model

Table 1 Wall conditions (2–D)

wall *No.*	wall type	K_W [W/m^2°C]	α_O [W/m^2°C]	T_O [°C]
1	outer wall	0.33	23.0	0.0
	window	136.0	23.0	0.0
2	outer wall	0.33	23.0	0.0
	window	136.0	23.0	0.0
3	ceiling	45.5	17.0	14.0
4	floor	0.67	17.0	8.4

Table 2 Calculating conditions (2–D)

case *No.*	heating system	panel	T_{in} [°C]	V_{in} [m/s]	Q_P [W]
1–1	heat pump	—	33.0	2.0	—
1–2	fan–coil	—	33.3	2.0	—
1–3	heat pump	—	33.0	2.0	—
1–4	fan–coil	—	32.5	2.0	—
2–1	heat pump + panel	A	30.0	1.8	317
2–2	heat pump + panel	B	30.0	2.0	318
2–3	heat pump + panel	C	30.0	1.5	318
2–4	fan–coil + panel	A	30.0	2.0	317
3–1	panel	A	—	—	636
3–2	panel	B	—	—	615

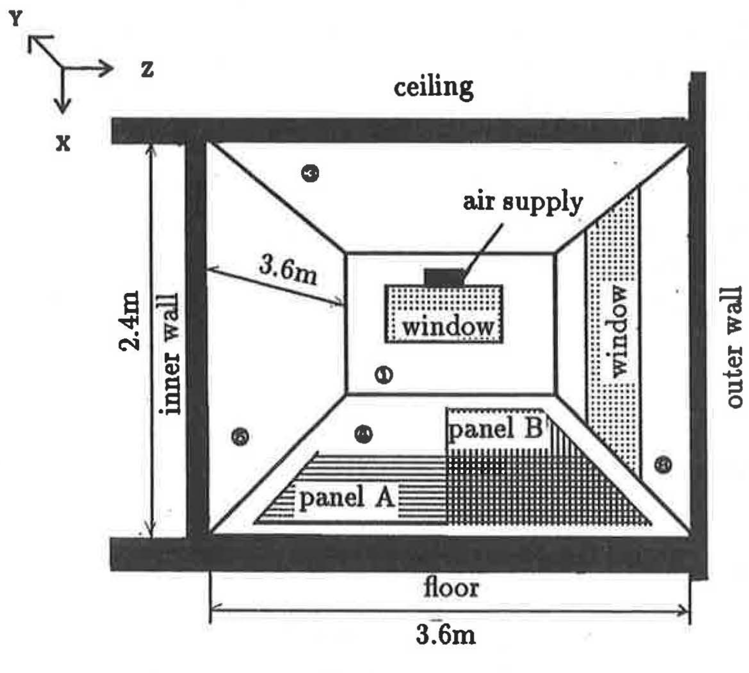


Fig.3 3–D room model

Table 3 Wall conditions (3–D).

wall *No.*	wall type	K_W [W/m²°C]	$T_{w,o}$ [°C]
1	outer wall	0.34	3.0
	window	137	9.0
2	partition	0.47	7.0
3	ceiling	0	–
4	floor	0	–
5	partition	0.47	5.0
6	outer wall	0.34	3.0
	window	137	9.0

1. Heat pump or fan-coil unit heating systems.
2. Combined heat pump or fun-coil unit and panel heating systems.
3. Panel heating systems.

The room configurations of 2-D room model is shown in Figure 1. Heat pump is set upper side of wall 1,and supply air is discharged downward. Fan-coil unit is set lower end of wall 1 and supply air is discharged upward. In both units, the width of air outlet is 6 cm and that of return air is 24 cm.

In the case studies for 2-D room,following facts were assumed:

1. Supply air temperature T_{in} and air velocity V_{in} are constant.
2. Panel input Q_P is kept constant and uniformly distributed on its surface area.
3. Outside air temperature T_o is used as the thermal boundary conditions.
4. Infiltration rate is constant and equal to 0.5[1/h].
5. Heat source inside the room is not considered.
6. Effect of solar radiation is not considered.

The wall conditions and calculation conditions for 2-D cases are shown in Table 1 and Table 2 respectively. α_o is outside surface conductance of walls. In Table 2, cases 1-3 and 1-4 are the same conditions as cases 1-1 and 1-2 except that wall 1 is replaced with wall 2.

Figure 2 is a 3–D room model used for the 3–D case studies. The air outlet size of the air supply unit is 12cmx65cm and that of return air is 24cmx65cm respectively. All calculating conditions are nearly same as the 2–D cases except that the outside wall surface temperature $T_{w,o}$ is used in stead of T_o. Wall conditions for 3–D room model is shown in Table 3.

Results and Discussions

Results for 2-D room model

Main quantities obtained by calculations are presented in Table 4. Q_{in} is the heat input of the heat pump or fan-coil unit. T_m is averaged air temperature weighted by calculating cell volumes. Q_{INF} is heat sink caused by infiltration. In the flow analysis,Q_{INF} was assumed to be distributed uniformly throughout the room. Q_L is the summation of q_L along walls. PMV_{oz} and ET^*_{oz} are averaged values of PMV and ET^*in the occupied zone. Overall heat balance in these cases are expressed as

$$Q_{in} + Q_P = Q_L + Q_{INF} \tag{10}$$

For the precise estimation of total heat inputs, it should be necessary to consider heat losses from back side of the panel,$Q_{P,L}$ in cases 2 or cases 3. In the present calculations,$Q_{P,L}$ was about 10 % of Q_P. Even if $Q_{P,L}$ is taken into account, panel heating system(cases 3) is most profitable on the basis of energy consumption. However,as is shown below,panel heating system is not necessarily preferable on the basis of thermal sensation distributions.

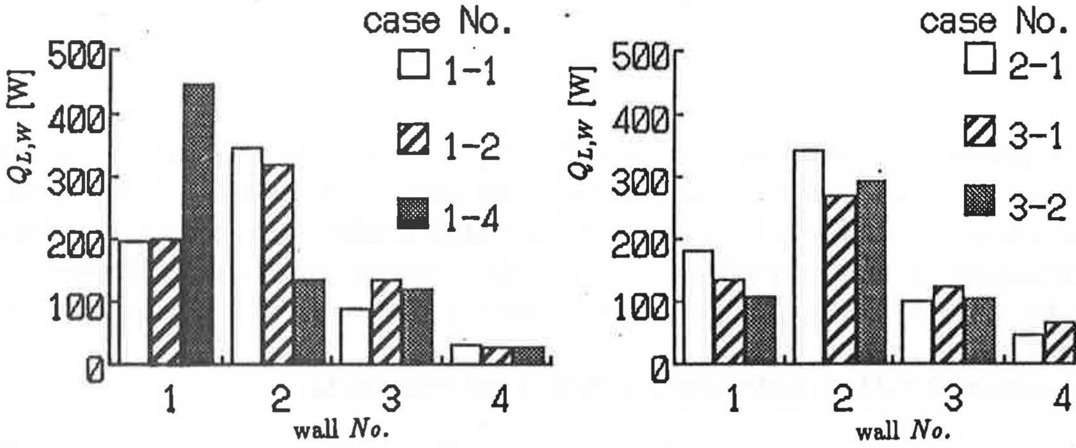
Inspection of Table 1 and Table 2 indicates following facts:

1. Change of wall material or wall construction affects the heat losses Q_L appreciably.
2. It is possible to lower the supply air temperature about 2.0 °C in cases 2 than in cases 1.
3. Panel heating system(cases 3) can fulfill same thermal environments with about 4 °C lower room temperatures than those of convective heating system(cases 1).
4. New effective temperature ET^* may be also useful for the controlling room thermal sensation.

The results described above may suggest that the present simulation method will be applicable to practical designing in various heating/cooling systems.

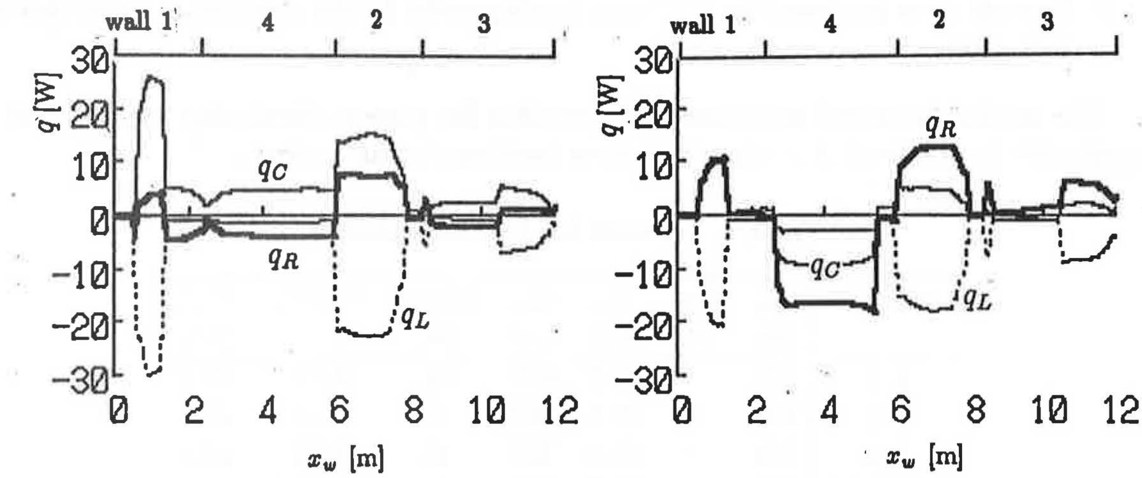
Table 4 Results for 2–D case studies.

case $No.$	Q_{in} [W]	Q_P [W]	T_m [°C]	Q_L [W]	Q_{INF} [W]	PMV_{oz} [-]	ET^*_{oz} [°C]
1–1	722	—	29.2	681	41	0.74	25.2
1–2	738	—	29.4	697	41	0.72	25.1
1–3	764	—	29.0	723	41	0.72	25.1
1–4	795	—	29.3	754	41	0.71	25.1
2–1	411	317	27.6	689	39	0.71	25.2
2–2	454	318	27.6	733	39	0.73	25.3
2–3	368	318	27.0	648	38	0.70	25.0
2–4	441	317	27.7	719	39	0.70	25.1
3–1	—	636	24.8	605	31	0.71	24.8
3–2	—	615	25.1	581	34	0.69	24.5



(a) (b)

Fig.3 Comparison of heat loss at each wall.

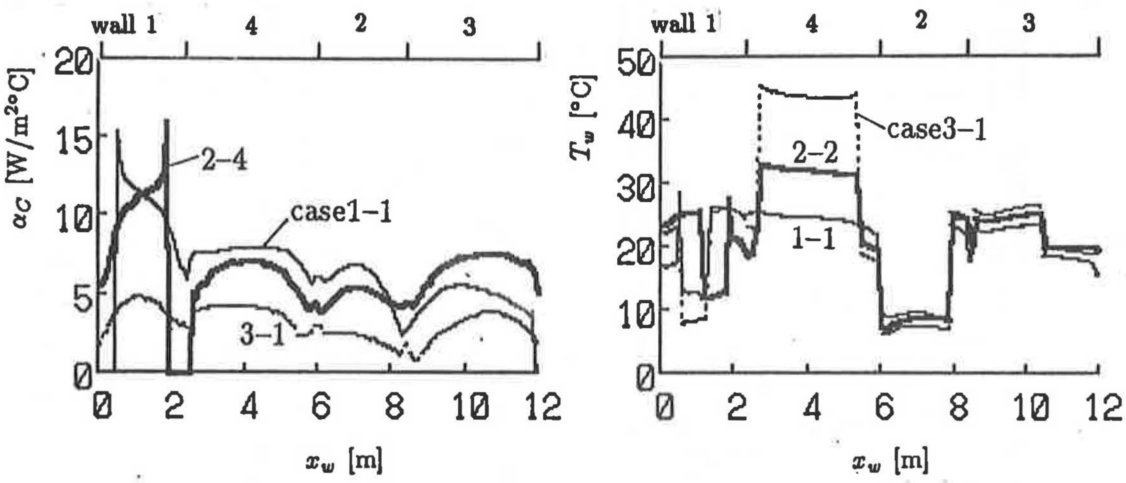


(a)case1-1 (b) case3-1

Fig.4 Distribution of heat fluxes along walls.

Figure 3 present heat losses at each wall. $Q_{L,W}$ is the summation of q_L along each wall. Figure 3 shows that the difference of heat losses between case 1-1,1-2 and case 1-4 is caused mainly by the difference of $Q_{L,W}$ of wall 1 and wall 2. This result indicates that the position of air supply unit influences total wall heat losses appreciably. Figure 3 (b) is comparisons for case 2-1,3-1,and 3-2. In these cases, similar difference of $Q_{L,W}$ is shown,but the amount is not so much as the cases in Figure 3 (a).

As a reference,some quantities along walls are shown in Figure 4 and Figure 5. Comparisons of wall heat fluxes q_C,q_R,q_L are shown in Figure 4. The abscissa $x_w$is a distance measured along walls. The order of plot is wall 1→ wall 4 → wall 2 → wall 3. Figure 5 (a) is distributions of calculated surface conductance α_C and Figure 5 (b) is wall surface temperatures respectively. In general,magnitude of α_C is proportional to the velocity near wall. Figure 5 (a) express the property well and predicted α_C values seem to be acceptable. However,detailed inspection should be discussed together with some reliable experimental data.

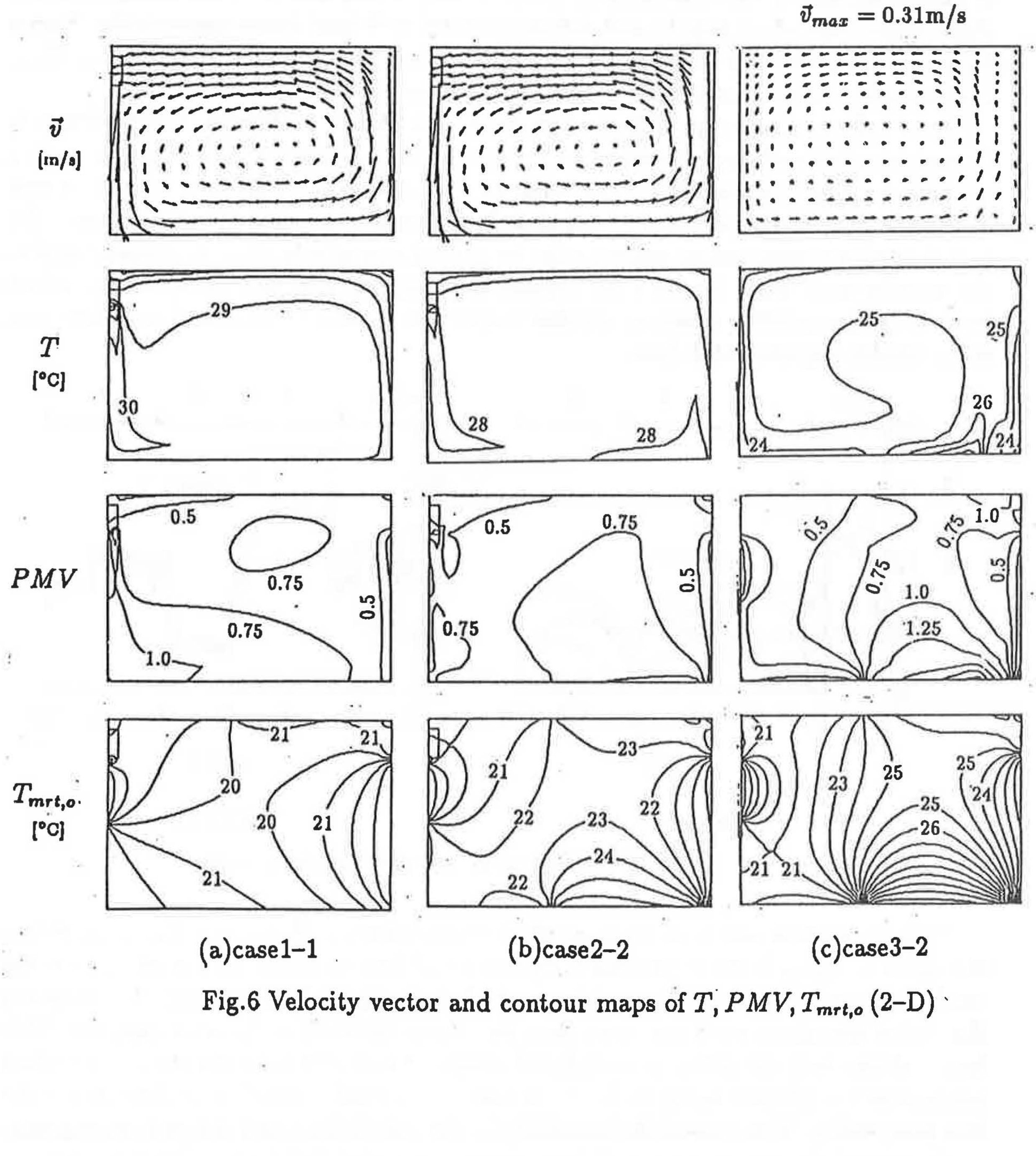


(a)α_C (b) T_W

Fig.5 Distribution of α_C and T_W along walls.

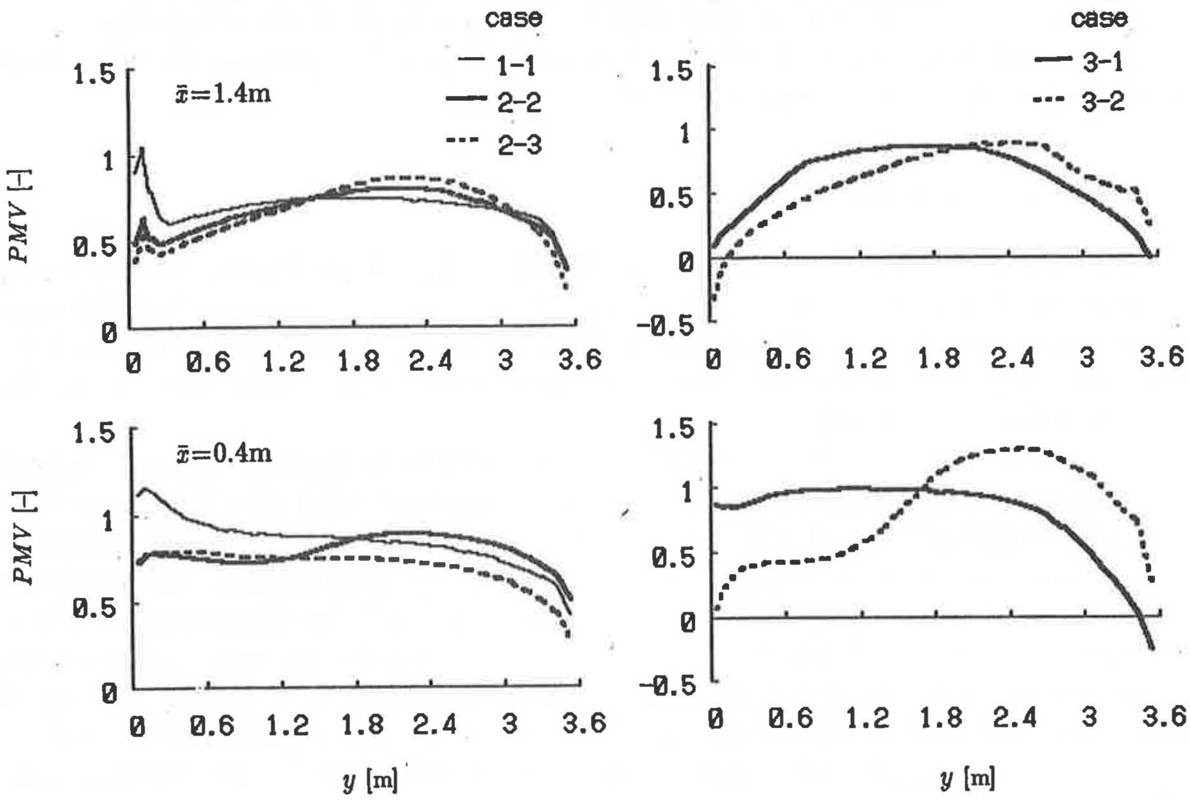
Velocity vectors and contour maps of air temperature T,PMV and $T_{mrt,o}$ are shown in Figure 6. $T_{mrt,o}$ is mean radiant temperature of human body. In Figure 6,(a) is the result for case1-1,(b) is case2-2 and (c) is case3-2. In Figure 6 (a) and (b), it is observed that velocities along walls are more than two times faster than those of case 3-2. This fact explains well the differences of distributions of wall flux components q_C or surface conductance α_c(Figure 4,Figure 5(a)). In case 3-2, a weak down flow is observed in the flow near wall 2. This caused the instability in the calculations and delayed convergence.

In spite of the non-uniform wall temperatures(Figure 5(b)),heating systems with air supply unit produce fairly uniform thermal environments. On the other hand, distributions of thermal sensation of the panel heating system become highly non-uniform, especially,the regions above the panel and near windows(case 3-2). Figure 6 also in-



(a)case1–1 (b)case2–2 (c)case3–2

Fig.6 Velocity vector and contour maps of T, PMV, $T_{mrt,o}$ (2–D)



(a) case1 and case2 (b) case3

Fig.7 Comparison of horizontal distribution of PMV (2–D).

Table 5: Results for 3-D cases.

case $No.$	heating system	T_{in} [°C]	V_{in} [m/s]	Q_{in} [W]	Q_P [W]	Q_{INF} [W]	T_m [°C]	PMV_{oz} [-]
1	air supply unit	39.0	2.0	1094	—	52	25.2	–0.22
2	air supply unit + panel A	35.0	1.5	610	343	50	24.0	–0.35
3	air supply unit + panel B	35.0	1.5	625	344	50	24.3	–0.27
4	panel A	—	—	—	825	46	22.3	0.0
5	panel B	—	—	—	825	48	22.8	0.16

dicates that the distributions of PMV show similar patterns with $T_{mrt,o}$. These facts suggests that $T_{mrt,o}$ may be the most important factors in the thermal sensation.

Horizontal distributions of PMV are shown in Figure 7. It presents the differences of uniformity described above more clearly.

Results for 3-D Room Model

Main results for 3-D case studies are shown in Table 5. Case 1 is convective heating system, case 2 and case 3 are combined convective and panel heating system and case 4,case 5 are panel heating systems. In case1,case2,and case3, the angle θ between inlet flow direction and wall 1 is 30°. The assumptions used in the calculations are nearly same as those in 2-D cases.

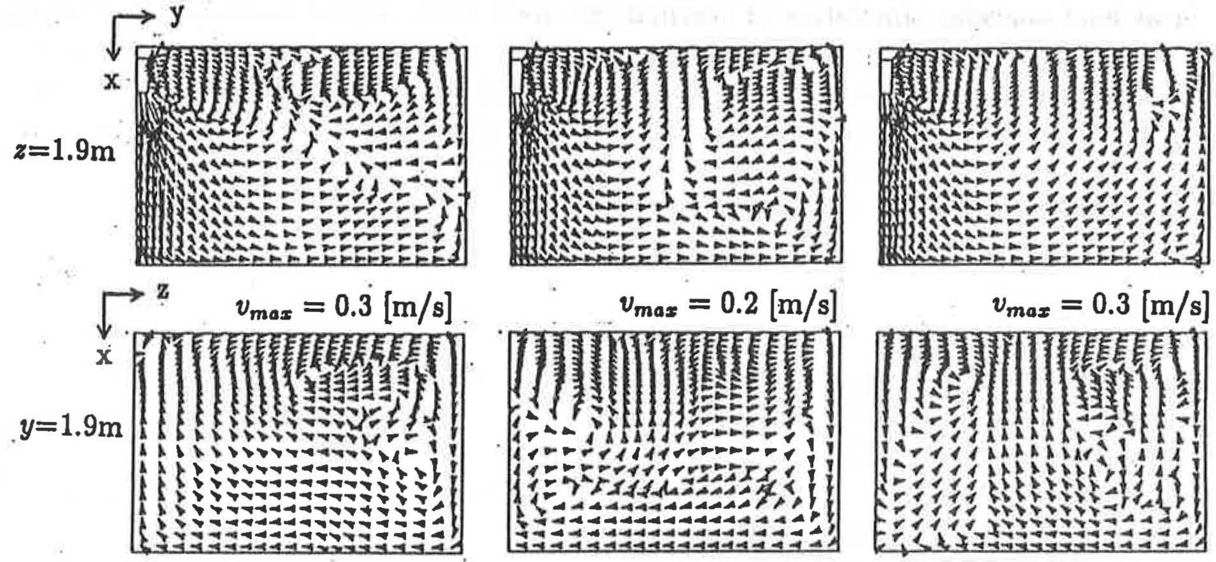
In 3-D case studies, wall 3 and wall 4 were assumed to be insulated walls, then,heat losses from back side of the panels $Q_{P,L}$ were not calculated. Even though $Q_{P,L}$ are supposed to be 10% of Q_P and added to Q_P values in Table 5, panel heating system(case 4 and case 5) is most profitable in the basis of energy consumption. All quantities show similar tendency qualitatively as 2-D cases shown in Table 4,however,the uniformity of PMV_{oz} is not sufficient, care is necessary to compare the results qualitatively. Flow patterns of x–y plain and x–z plain are shown in Figure 8. Both plains are cross sections positioned near the room center. As the supply air is discharged at slow velocity and ,moreover,the ceiling and the floor are insulated,flow patterns become complicated,especially, in case2 and case3. In 3-D case studies, instability of calculations occurred in cases of panel heating systems(case4 and case5) and relaxation factors could not set larger than 0.2.

Distributions of PMV are shown in Fig.9. $\bar{x}$ is the distance of y–z plain from the floor.

Flows in 3-D rooms show generally very complicated flow patterns even in isothermal cases,therefore, detailed inspection is not easy. However,through these case studies, all results are qualitatively acceptable and the simulation method described above may be applicable to practical designing in various problems.

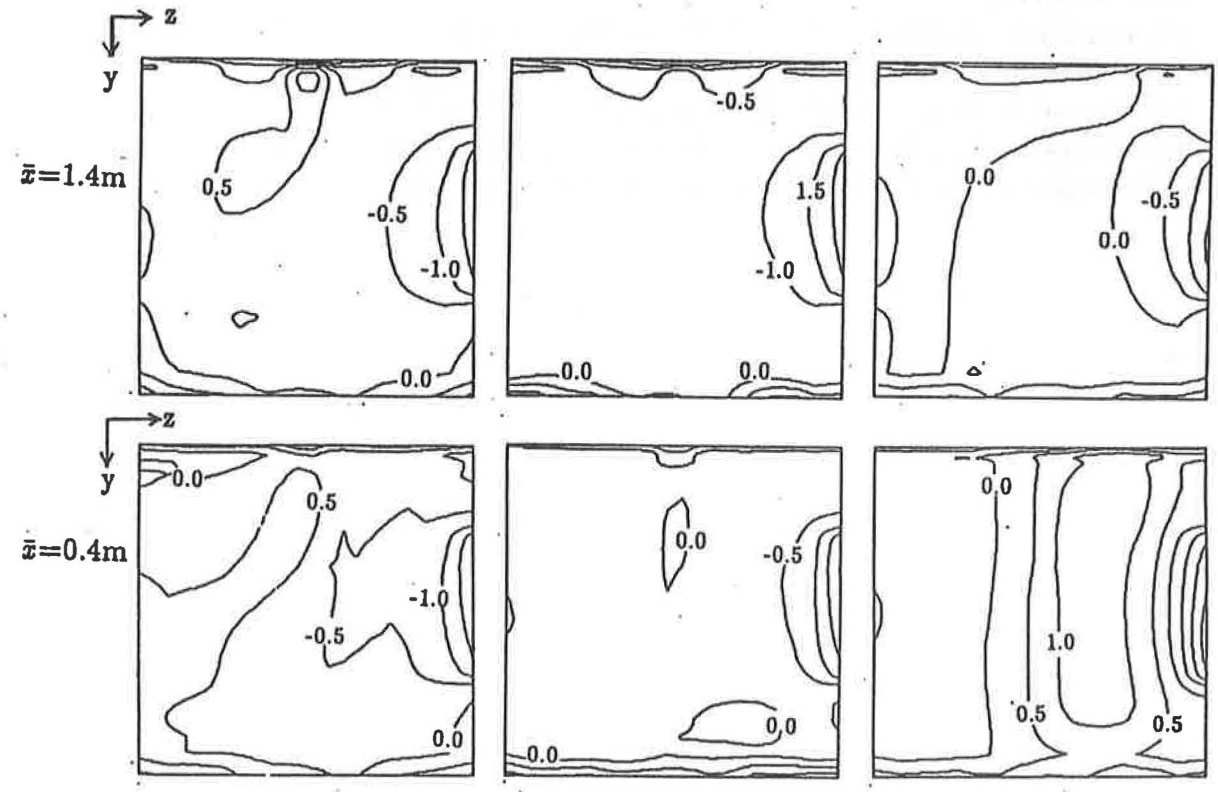
Conclusion

A numerical simulation method capable of treating radiative heat transfer between walls are investigated through some case studies. In the case studies, our primary concern has been focused on the accurate estimation of heat transfer problems in air-conditioned rooms. Although some calculations in 3-D cases are numerically instable, as for thermal environments analysis,the computer code works well in all cases and qualitatively acceptable results are obtained. The simulation method developed in this paper may be applicable to practical problems,such as designing air supply units, exact



(a) case1 (b) case2 (c) case3

Fig.8 Comparison of flow patterns (3–D).



(a) case1 (b) case3 (c) case5

Fig.9 Comparison of contour maps for PMV (3–D).

heat load analysis, simulation of thermal sensation basis control of heating or cooling systems,etc.. In the three dimensional room airflow analyses, we often confront the situations that essentially instable phenomena should be analyzed numerically. Those situations may be the serious limitations of our computer code. To validate the method strictly,the results described above should be compared with the reliable experimental data and that is one of our future problems.

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