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CALCULATION METHODS FOR TRANSIENT HEAT TRANSFER THROUGH WALLS AND THEIR APPLICATIONS TO ROOM THERMAL ENVIRONMENTS ANALYSES

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SUMMARY

For the simulations of non-steady problems in the room thermal environment,it is necessary to consider transient heat transfer through room enclosures. Following three models are introduced, assuming the heat conduction through walls is one dimensional.

Model 1 : It is assumed that heat flux q_L could be divided into two parts,one is steady heat flux q_{st} and the other is non-steady component q_{tr}. Then,assuming the wall thickness be infinity, q_{tr} is calculated as indicial response excited by the wall surface temperature.
Model 2 : q_L is calculated by a convolution procedure using wall response factors. The response factors are expressed by a time series and are given for each wall.
Model 3 : To get temperature distributions inside the walls,the governing equation is solved numerically using a finite difference method. q_L is derived from the temperature distributions.

These three models are tested using a two dimensional room model. As a more practical example, transient response is simulated when heat-pump system installed in a residential room is set to work. In the simulation, the thermal environment analysis code is applied with a control program which describes controlling process of a real machine numerically.

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INTRODUCTION

In the numerical simulations of room thermal environments ,non-steady phenomena, e.g. room air temperature responses when heating (and/or cooling) systems are set to work or responses when their operating conditions are altered, are often important issues in the practical point of view.

Hitherto,a lot of studies about predictions of indoor thermal environments have been reported and practically useful results are available[1]~[7]. However,in spite of its usefulness, few researchers have reported about calculation procedures considering non-steady effects of wall heat transfer qualitatively [1][4][5]. In the analysis including room enclosures, the heat capacity of them is usually considerably large, so evaluations of heat transfer through walls should be important subjects.

Authors have developed a calculation code applicable to room thermal environments analysis and have proved it capable of predicting various problems with practically acceptable accuracy through several case studies[3][4]. In the calculation code, heat balance equation is solved at the inner surface of walls including radiative heat transfer between walls. As the code also includes analysis of radiant environment in the room, it is possible to calculate some thermal comfort indexes,such as predicted mean vote (PMV)[8] or new effective temperature (ET^*)[9]. However,non-steady heat transfer through walls is not included yet.

To evaluate the transient heat transfer, three models are introduced assuming the heat conduction through walls is one dimensional. Model 1 is a simplified method in which wall heat transfer is solved analytically with some assumptions. Model 2 is a numerical method in which a convolution technique with thermal response factors is used to calculate transient heat transfer. The response factors are expressed by a time series and should be prepared for each wall. Model 3 is also a numerical method in which the governing equations are solved by a finite difference method.

These models were applied to the calculation code described above. At first,some test runs were performed with a two dimensional room model and they proved that all results seem to be acceptable. As a more practical example, a real control process of a heat-pump was simulated. The results showed good agreement with experimental data.

MATHEMATICAL MODEL

Basic Equations

In the previous paper, a calculation procedure applicable to evaluations of heat transfer through walls or to calculations of thermal comfort index distributions was presented[4]. In the calculations,heat balance equation

$$q_C + q_R + q_L + q_P = 0 \tag{1}$$

is solved at the inner surface of walls, where q_C is convective heat flux,q_R is radiative heat flux,q_P is heat source at the surface and q_L is heat flux from the inner surface into walls.

For the simulations of non-steady problems,it is necessary to consider transient effect in the evaluation of q_L. In the present paper, following three models are introduced assuming the heat conduction through walls is one dimensional.

For the one dimensional heat transfer through wall, the basic equation is written as

$$\frac{\partial \theta}{\partial t} = a\frac{\partial^2 \theta}{\partial x^2} \tag{2}$$

where,x is distance from the inner surface into inside of the wall, θ is wall temperature and $a = \lambda/\rho c$ is thermal diffusivity. Equation 2 should be solved with following boundary conditions,

$$t = 0 \ : \ \theta = \theta_{int} \tag{3}$$

$$x = 0 \ : \ \theta = \theta_w \tag{4}$$

$$x = L \ : \ \theta = -\lambda(\frac{\partial \theta}{\partial x}) = \alpha_0(\theta|_{x=L} - \theta_0) \tag{5}$$

where L is the thickness of the wall,θ_{ini} is initial temperature, θ_w is inner surface temperature of the wall and θ_0 is outside air temperature of the wall.

Semi-Infinite Wall Model(Model 1)

To simplify the problem,some assumptions are introduced here.

(i). θ_w is constant during each time step Δt and at time $t = t^{n-1}$ it changes suddenly from θ_w^{n-1} to θ_w^n.

(ii). The transient heat transfer occurred at time $t = t^{n-1}$ approaches steady heat conduction at $t = t^n$.

(iii): The transient heat transfer at the inner surface may be regarded as that of a semi-infinite wall.

From the assumptions,the boundary condition given by eq.(5) is not used and the heat flux q_L may be divided into two parts,one is steady heat flux q_{st} and the other is non-steady component q_{tr}. The transient component q_{tr} is calculated analytically as indicial response excited by the wall surface temperature $\Delta\theta = \theta_w^n - \theta_w^{n-1}$.

The solution is given as

$$\theta = \Delta\theta_w(1 - erf\frac{x}{\sqrt{4at}}) \tag{6}$$

Then,q_{tr} at $x = 0$ becomes

$$q_{tr} = (-\lambda\frac{\partial\theta}{\partial x})_{x=0} = \frac{\lambda}{\sqrt{\pi at}}\Delta\theta_w \tag{7}$$

Finally,q_L is given as

$$q_L = -q_{st} - q_{tr} = -K_w(\theta_w^{n-1} - \theta_0) - \frac{\lambda}{\sqrt{\pi at}}\Delta\theta_w \tag{8}$$

In the calculation of non-steady conditions,q_L in the equation (1) should be replaced by equation (8).

Wall Response Factor Model(Model 2)

Thermal response factor method[10] developed for the calculation of heating/cooling load in air-conditioned rooms is a simplified procedure to evaluate non-steady heat conduction through walls provided the response factors are known for each wall. Outline of the procedure is as follows.

Let $X_j(j = 0, 1\ldots)$ be response factors for a wall and θ_w^n be wall surface temperature at time $t = t_n$. Assuming that for terms $j \geq N$, X_j is expressed as

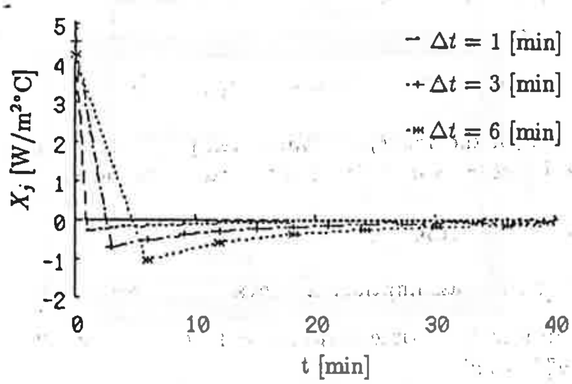


Fig. 1 Example of response factors.

$$X_{j+1} = \rho X_j, (0 < \rho < 1.0),$$

then,transient heat conduction from the inner surface into inside the wall is given as

$$q_n = \sum_{j=0}^{N-1} \theta_w^{n-j} X_j + X_N \sum_{j=N}^{\infty} \theta_w^{n-j} \rho^{j-N} \quad (9)$$

$$= \sum_{j=0}^{N-1} \theta_w^{n-j} X_j + r_n \quad (10)$$

Rearranging the second term of equation(9),r_n can be expressed in the form

$$r_n = \theta_w^{n-N} X_N + \rho r_{n-1} \quad (11)$$

As a example,figure 1 shows a response factor X_j of the wall shown in figure 2. In the calculations,q_L in equation(1) should be replaced by equation(9).

Numerical Calculation Model(Model 3)

This model is a most conventional method. To obtain temperature distributions inside the wall,the governing equations(2) $\sim$ (5) is solved numerically using finite difference method. q_L is derived from the temperature distributions.

BASIC RESULTS

The three models described above are tested using two dimensional (2-D) room model. The room model is shown in figure 2. The enclosures are constructed by two kind of walls, 6 mm thickness normal glass(windows) and 50 mm glasswool+100 mm concrete(the other walls). The surface conductance of outer surface of walls is assumed to be 17 W/m^2°C and outside air temperature to be θ_o = 0°C.

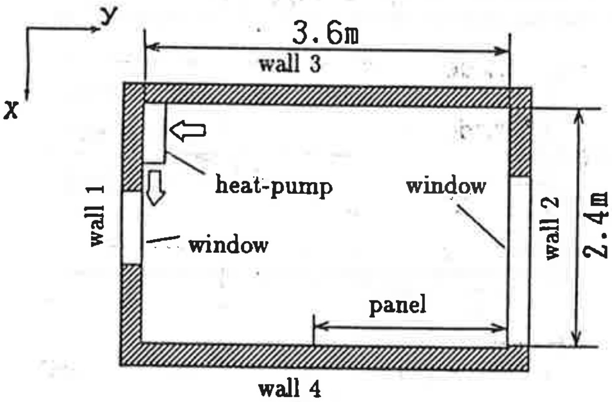


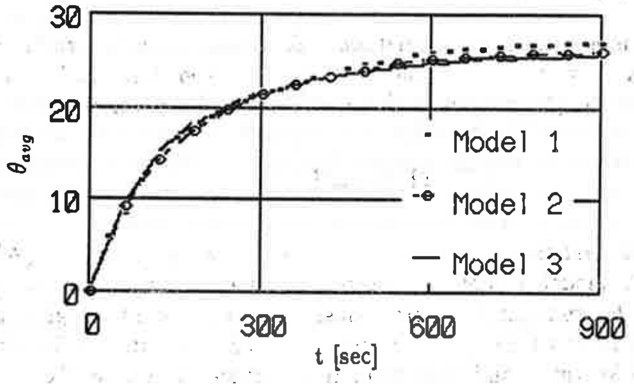
Fig. 2 2-D room model.

The test calculations are performed with two heating systems. A heat-pump system is set upper side of wall 1 and its supply air conditions are $Vin = 2.0$ m/s,$\theta_{in} = 31$ °C respectively. The outlet width of heat-pump is 0.06 m. A panel heating system is set on the floor. Its heat input Q_{in} is 558 W/m and assumed to be distributed uniformly on the panel surface.

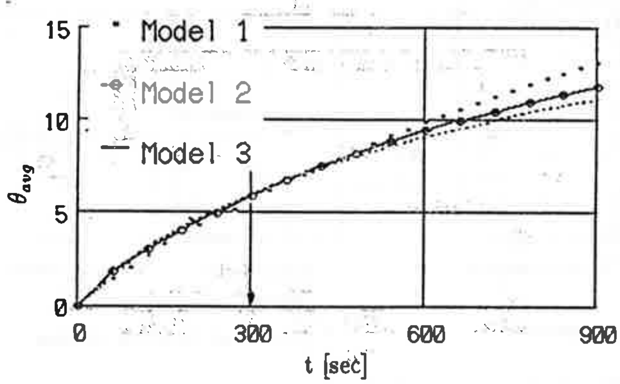
The governing equations for non-isothermal incompressible flow are solved with forward difference for time and power law difference scheme[11] for space. The number of calculating mesh is 25 × 32. Fully implicit method is adopted in the time marching. To advance calculation properly,not only convergence criteria but also overall heat balance should be satisfied at each time step. In the calculations shown below, following criteria is adopted.

$$\left.\begin{aligned} cc &= \frac{(\phi^{N+1}-\phi^{N})_{max}}{\phi^{N}_{max}} \leq 0.01 \text{ (for variable } \phi \text{)} \\ 0.98 &\leq \beta^{n} \leq 1.02 \quad \text{(for heat balance)} \end{aligned}\right\} \qquad (12)$$

where, β^n is overall heat balance at $t = t_n$ and expressed by



(a) Heat-pump system.



(b) Panel heating system.

Fig. 3 Comparison of wall models with room temperature.

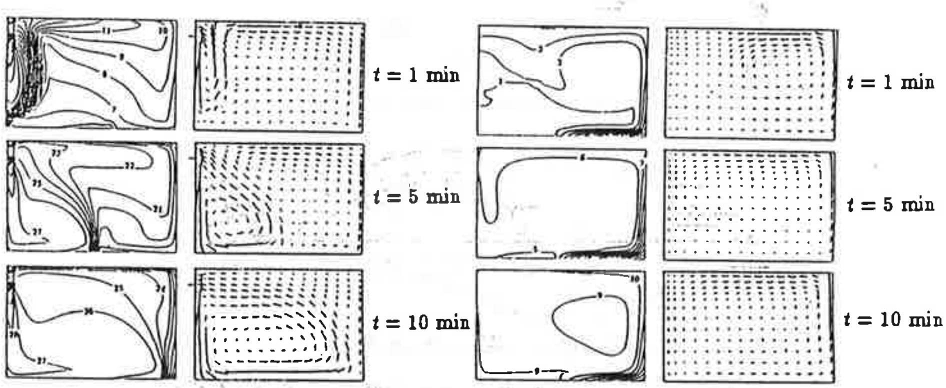
$$\beta^n = \frac{\int_0^{t_n} Q_{in} dt}{\int_0^{t_n} Q_L dt + (\text{storaged heat in the room air})} \tag{13}$$

where, $Q_L = \sum q_L$ is total heat fluxes at each time step. Other details for flow analysis are omitted here.

Three models are applied to the simulation of transient conditions when heating systems are set to work. The time step Δt is set 1 minute for all calculations and the initial condition of temperature is assumed to be 0°C everywhere.

The responses of room average temperature θ_{avg} are shown in figure 3(a) and figure 3(b). Figure 3(a) is results on the heat-pump system and figure 3(b) is on the panel heating system. For all models,the responses of θ_{avg} show almost same results until $t = 5 \sim 8$ minute ,however,as time elapses,the response for model 1 changes slightly faster than those of other two models and this tendency is a little more notable in case of panel heating than in case of heat-pump system. This indicates that the assumptions made in model 1 become invalid after heat storage inside walls increases and this effect may appear more clearly in the panel heating system the surface temperature of which is higher than that of heat-pump system.

In case of heat-pump system,the steady condition is attained at about $t = 60$ minute and θ_{avg} becomes 27.5°C, while it is attained at about $t = 120$ minute and $\theta_{avg} = 24.6$°C in the panel heating system. Model 2 and Model 3 needed 20 $\sim$ 30% more computing time than Model 1. Model 1 is a simple method and favorable in computer memories and cputime,however,for the practical applications, Model 2 or Model 3 should be preferred.



Temperature contours. Velocity vectors.

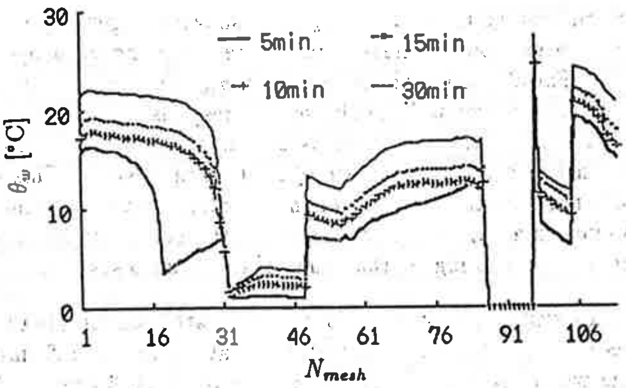
(a) Heat-pump system.

Temperature contours. Velocity vectors.

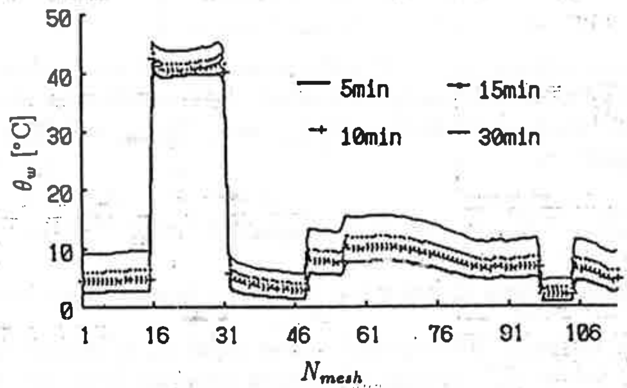
(b) Panel heating system.

Fig. 4 Response of flow pattern and temperature distribution, Model 2.

Figure 4(a) and figure 4(b) show changes of flow patterns of both systems(results for Model 2). While flow patterns are not established until time elapses about 15 minute in case of heat-pump system,they are established at not later than $t = 5$ minute and calculation advances stably at all time steps. As a reference,the distributions of wall surfase temperature θ_w are shown in figure 5(a) and figure 5(b)(results for Model 2). The abscissa N_{mesh} is the number with which calculating meshes along walls are counted in plotting θ_w. The order of the plot is wall $1(N_{mesh} = 1 \sim 32) \rightarrow$ wall $4(33 \sim 57) \rightarrow$ wall 2 $(58 \sim 89) \rightarrow$ wall $3(90 \sim 114)$.



(a) Heat-pump system.



(b) Panel heating system.

5 Response of wall surface temperature distributions, Model 2.

PRACTICAL APPLICATION

The basic examples described above prove that the calculation code with non-steady wall heat conduction models can predict non-steady problems with qualitatively reasonable accuracy. In this section, as a more practical application,it is tested to simulate real control process when a heat-pump starts to work in a residential room. The control process of the heat-pump is programed and combined with the calculation code. Outline of control process is as follows.

Outline of Control Process

In the present control process,it is assumed that the return air temperature θ_{ret} is equal to the room average temperature θ_{avg} and that the information of room thermal condition is transmitted to the control process only through θ_{ret}. Then,operating conditions of heat-pump are determined by its control system monitoring the return air temperature θ_{ret}.

Let $\theta^n_{ret,sens}$ be return air temperature at $t = t^n$ detected by a temperature sensor and S_i^n be frequency control signal sent to the compressor at $t = t^n$. The frequency control signal at the next time step S_i^{n+1} is determained using $\theta^n_{ret,sens}$,$\theta^{n-1}_{ret,sens}$ and θ_{set} as

$$\left.\begin{array}{rcl} S_i^{n+1} & = & S_i^n \Delta S_i \\ \Delta Si & = & f_{si}(E, \Delta E) \\ \text{where } E & = & \theta_{set} - \theta^n_{ret,sens} \\ \Delta E & = & \theta^n_{ret,sens} - \theta^{n-1}_{ret,sens} \end{array}\right\} \tag{14}$$

θ_{set} is a set up room temperature. $f_{si}(E, \Delta E)$ means that ΔS_i is a function of E and ΔE. The functional relation is usually determined by control theories the details of which is not described here. In deriving $\theta^{n+1}_{ret,sens}$ using $\theta^n_{ret,sens}$ and θ^{n+1}_{ret}, following relations are applied

$$\theta^{n+1}_{ret,sens} = \theta^n_{ret,sens} + (\theta^{n+1-\epsilon}_{ret} - \theta^n_{ret,sens})\{1 - \exp(-\frac{\Delta t}{\tau_s})\} \tag{15}$$

where,ϵ is dimensionless form of sensor time lag and τ_s is sensor time constant.

As the operating frequency of compressor power input f_{op} is related empirically with S_i,if S_i^{n+1} is known, f_{op}^{n+1} is easily determined using the data. f_{op}^{n+1} is used to determine electric power input of the compressor,i.e. the heating capacity Q_{in}.

Flow rate of supply air G, i.e.flow rate through condenser is proportional to fan rotating speed. It is controlled by detected condenser temperature $\theta^n_{c,sens}$ and the sign of its difference during the time step $\Delta\theta_c = \theta^n_{c,sens} - \theta^{n-1}_{c,sens}$. Where,$\theta^n_{c,sens}$ is related to θ^n_c and $\theta^{n-1}_{c,sens}$ in the same manner as equation (15). If flow rate G^{n+1} at time $t = t^{n+1}$ is determined,the supply air velocity V_{in}^{n+1} is given as

$$V_{in}^{n+1} = \frac{G^{n+1}}{A_{in}} \tag{16}$$

where, A_{in} is effective cross sectional area of air outlet.

The heating capacity Q_{in} is determined using f_{op}^{n+1}, θ_{ret}^{n} and G^{n+1} as

$$Q_{in}^{n+1} = f_Q(f_{op}^{n+1}, \theta_{ret}^{n}, G^{n+1}, \theta_o) \quad (17)$$

where, θ_o is outside air temperature. The function f_Q is empirically determined. Using Q_{in}^{n+1}, condenser temperature θ_c^{n+1} and input air temperature θ_{in}^{n+1} is derived by

$$\theta_c^{n+1} = \theta_{ret}^{n} + \alpha Q_{in}^{n+1} \quad (18)$$

$$\theta_{in}^{n+1} = \theta_{ret}^{n} + \frac{Q_{in}^{n+1}}{\rho c_p G^{n+1}} \quad (19)$$

It should be noticed that Q_{in}^{n+1}, θ_c^{n+1} or θ_{in}^{n+1} expressed by equation (17),(18) or (19) are values determined under the situation in which all control quantities maintain their values until Q_{in}^{n+1},θ_c^{n+1} or θ_{in}^{n+1} become steady conditions. Considering this situation, delayed form similar to equation (15) is assumed in the control program. Thus determining the new values of V_{in},θ_{in}, the calculation code is called. These procedure is iterated at the following time steps.

Results

To compare the simulated results with measured data, an experiment was performed using a residential room. The room configurations are shown in figure 6. A heat-pump is installed upper side of wall ①. The direction of supply air flow is inclined 30 ° to the direction of x axis. The initial condition of room air temperature and outside air temperature are 8.6 °C and 2.1 °C respectively. Thermal conductance of

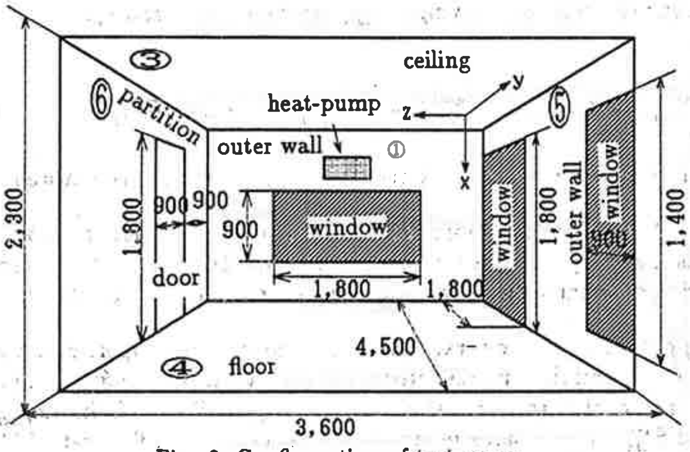


Fig. 6 Configuration of test room.

the walls are as follows,

outer wall	:	0.27 [W/m^2°C]	partition	:	0.58 [W/m^2°C]
ceiling	:	0.48 [W/m^2°C]	floor	:	0.29 [W/m^2°C]
window	:	8.8 [W/m^2°C]			

These values include outside surface conductance α_o which is assumed to be 9.3 W/m^2°C. The heat-pump is operated udner these conditions together with θ_{set} is 23 °C.

Outline of the flow field calculation is as follows.

(i). Model 3 is applied as the wall heat transfer model.

(ii). Time step Δt is set to 1 minute,and convergence criteria at each time step is used the same condition as equation (12).

(iii). To improve the accuracy,skew upwind difference scheme[12] is adopted.

(iv). Calculating mesh sizes are 21 × 24 × 19.

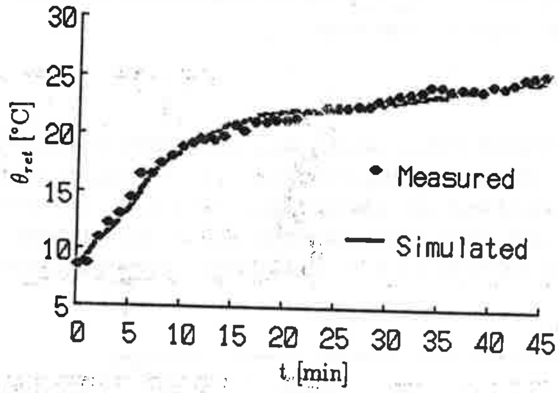


Fig. 7 Comparison of return air temperature θ_{ret}.

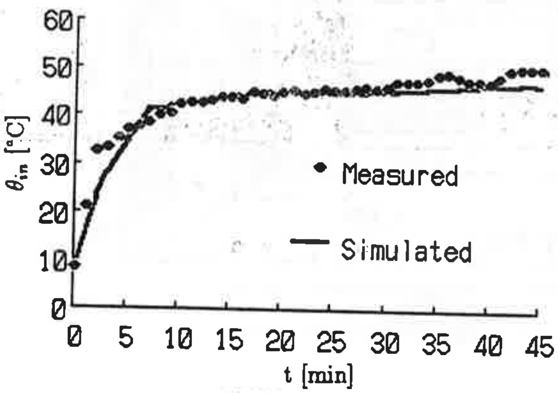


Fig. 8 Comparison of supply air empeature θ_{in}.

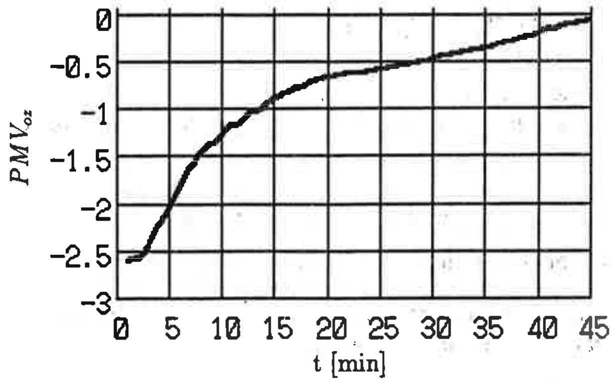


Fig. 9 Response of PMV_{oz}.

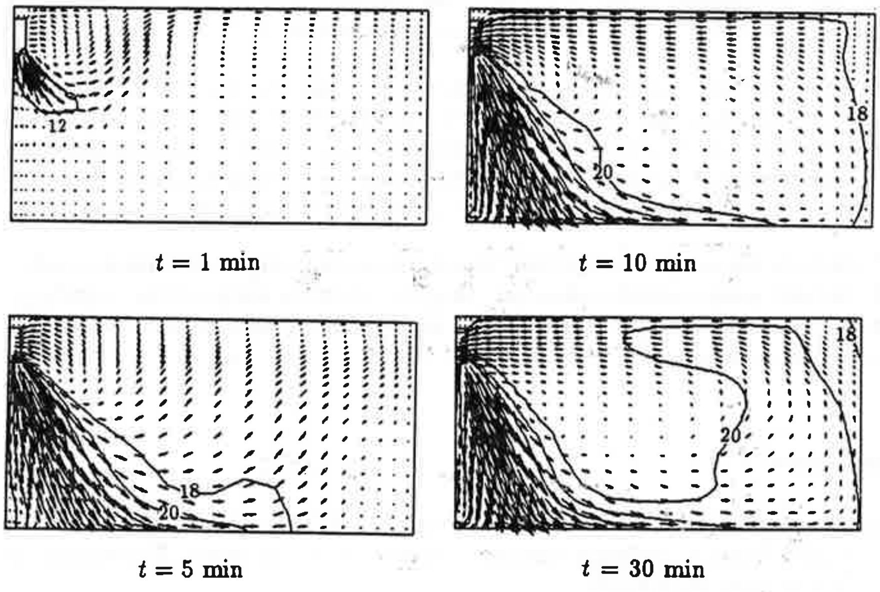


Fig. 10 Response of flow patterns and isotherms,x–y plane,z=1.8 m.

Comparisons of θ_{ret} and θ_{in} are shown in figure 7 and figure 8. The simulated values show fairly good agreement with measured values. This indicates that both the flow calculation code and the control process program are confirmed their validity and that these simulations may be useful design tools in place of experimental investigations.

Figure 9 shows the response of PMV_{oz} which is the averaged value of PMV in a occupied zone. In this calculation, it is assumed to be the whole room ,then,PMV_{oz} is same as averaged value of PMV in the room. If a desirable thermal comfort condition is regarded as the condition $|PMV_{oz}| \leq 0.5$, it is realized at about 28 minute after the operation of heat-pump starts. Examples of flow field developing are presented in figure 10 together with temperature field. The flow field is established at about t=10 minute, however, the temperature field is still growing at time t=30 minute.

CONCLUSION

Three models are introduced to evaluate non-steady wall heat transfer. Test runs using two dimensional room prove that the calculation with the models predict non-steady problems with qualitatively acceptable validity.

As a more practical example, the calculation code is applied to a simulation of real control process when a heat-pump starts to operate. A program of the heat-pump control system is used together with the calculation code. The simulated results show good agreement with measured values. The simulation method examined here may be a useful design tool for developing control processes of heating/cooling systems.

However,in the simulation described above, the room thermal condition is coupled with the heat-pump control system only through return air temperature $\theta_{ret}(\simeq\theta_{avg})$. How to couple the system control process with the room thermal environments is one of future problems.

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