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NUMERICAL EVALUATION OF ENERGY SYSTEMS AND THERMAL ENVIRONMENTS ON LOW ENERGY ARCHITECTURES AS A STRUCTURAL ELEMENTS OF ECO-POLIS

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Abstract : To evaluate the thermal performance of the passive solar room with PCM (phase change material) and a gypsum board as a distributed heat storage system, a numerical model to calculate the transient heat transfer in the PCM walls was developed and incorporated into a CFD code. A simulation code was used in two different ways, one is utilizing a CFD code itself developed by authors and the other is applying a modified version of the CFD code (marcro model). At first, the micro and macro models were compared through evaluating a distributed heat storage system with a passive solar room assuming a winter heating season. The thermal performance of the passive solar room, such as the required energy input or the effect of PCM as a distributed heat storage system, was investigated quantitatively through long period unsteady simulations with the macro model.

Key Words : energy saving, melting point, numerical simulation, passive solar system, PCM

INTRODUCTION

Researches on eco-polis for resource cycling society are being continued because of the limited fossil fuel and the destruction of environment. From the energy and environment point of view, how to design ecological and energy conservative architectures in urban area has become a important subjects.

The utilizing renewable resources such as solar and wind energy is preferable means to overcome these problems in urban area. To examine the performance of such low energy architectures quantitatively, experimental approaches such as using an environmental test room or measurements may not be an efficient means, because of requiring much cost, labor and time. One of more efficient approaches is to replace experimental methods with computer simulations, that is, to utilize a CFD code as a 'numerical environmental test room'.

We have developed a CFD code for evaluating indoor thermal environments[1] and have been applied to investigate the thermal performance of rooms or energy consumption estimations with various heating / cooling systems.[2,3] Radiant heat exchange between wall surfaces including solar gain from the glazing is con-

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sidered in the CFD code developed by authors. However, to utilize it as a numerical test room, some improvements should be necessary on the treatment of solar radiation in case that the room has a large glazing or that a passive solar room is supposed to be analyzed.

In the present study, the CFD code is used as a numerical test room in two different ways, one is utilizing the CFD code itself developed by authors (We call it, hereafter, 'micro model'.) and the other is applying a modified version of the CFD code (We call it, hereafter, 'macro model'). The calculating procedure of the micro and the macro model is almost same except that flow fields are not calculated in the macro model, therefore, the room temperature is same everywhere and is expressed as one value.

Firstly, outline of the micro model was described including the treatment of radiant heat transfer. Then, the micro and macro models were compared through evaluating a distributed heat storage system with a passive solar room assuming a winter heating season. Finally, the thermal performance of the passive solar room, such as the required energy input or the effect of PCM (phase change material) on a distributed heat storage. Simulations were performed with a standard weather data of Sapporo City, a cold climate district in Japan. Both models proved to be one of the useful tools to investigate low energy architecture developing.

In this study, the effect of latent heat storage in room wall materials was investigated numerically with two types of simple passive solar rooms, one is a passive solar room with a Trombe wall and the other is a direct-gain one. As the thermal storage walls, a Trombe wall made of phase change material (PCM) and a wallboard (gypsum board impregnated with PCM) were used. A numerical model to calculate the transient heat transfer in the PCM walls was developed and incorporated into a CFD code developed by authors. Thermal environments and/or required energy inputs were compared quantitatively under the condition that the thermal environment of the test room is maintained comfortable with the on-off control of a subsidiary heater operations setting the target PMV±0.5. As the outdoor conditions, a standardized weather data of Sapporo City, a cold climate district in Japan, was used. Results show that the PCM is effective both in energy savings and in peak load reduction in winter heating seasons.

ANALYSIS METHODS

CFD Code (Micro Model)

SCIENCE[1](micro model) is a CFD (Computational Fluid Dynamics) code and is applicable to calculations of room air flows and temperature distributions as well as thermal comfort index ones. Basic equations are discrete and solved with SIMPLE algorithm.[4] The energy balance equation on the wall surface is solved at each time step.

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

As a turbulence model, standard k-ε model is adopted. Boussinesq approximation is assumed for buoyant force expression and a wall function based on logarithmic law is applied as wall boundary conditions. The Radiative Heat Ray method[5] is used to estimate the longwave radiative heat exchange among inner wall surfaces. A similar method is employed for solar radiative heat transfer. It was assumed that direct solar radiation struck the solid surface through glazing reflects in two different manners, one is a direct reflection and the other is a diffusive reflection. The component of the direct reflection is traced by the Radiative Heat Ray method, on the other hand the component of the diffusive reflection is treated using Gebhart's absorption factor in the similar way of calculating the longwave radiations in the room.

To optimize the performances of the PCM heat storage systems in the practical point of view, they should be evaluated through a long period unsteady simulations. Those simulations may be possible using a CFD code, however, practically unacceptable computer efforts may be required.

Modified Code (Macro Model)

Application of the micro model with a model room and appropriate weather data can realize detail indoor environments numerically, however, it often requires fairly long term unsteady simulations to evaluate the thermal performance of the model room through a month or through a heating/cooling season. Under such situations, micro model may not be an efficient tool because of its unacceptable computer efforts required. Whereas, for the rooms in which natural convection is dominant, airflow and temperature distribution are nearly uniform except regions adjacent to wall surface (wall boundary layer). The fact indicates that flow and temperature distributions may not be important information for energy and thermal environments analysis if the convective heat transfer between room air and wall surfaces is calculated properly. The macro model is a modified version of the micro model and the modifications are as follows.

(1) Momentum equations are not calculated.
(2) Energy equation is replaced by a macro heat balance equation, therefore, room air temperature is uniform everywhere (one-air-point model).
(3) The coefficient of convective heat transfer at the wall surface is given using the results of micro model calculations.

When macro model is used, we use average convective heat transfer coefficients α_C on the each wall (including ceiling, floor) calculated by micro model. At first, simulation is performed using micro model for about 5 days. The average convective heat transfer coefficients on the each wall for about 5 days are calculated from the relation of convective heat transfer flux and the difference between reference temperature (average room temperature) and wall surface temperature.

Now, the heat transfer through the wall such as outer walls and Trombe wall including latent

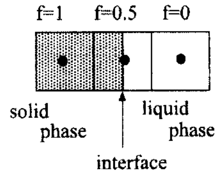


Figure 1. Description of solid fraction.

heat is described by the following one-dimensional the heat diffusion equation :

$$P c\frac{\partial T}{\partial t} = k\frac{\partial^2 T}{\partial x^2} + \rho L_a \frac{\partial f}{\partial t} \quad (2)$$

where ρ, k and c are density, thermal conductivity and specific heat, La and f are latent heat and solid fraction, respectively. Using solid fraction f, the latent heat released from solidification and moving interface between solid phase and liquid phase are simulated.

In regard to solid fraction f, it was assumed that solid fraction f is a function of temperature. For instance, in the case of temperature interval in melting-freezing is zero such as pure materials, solid fraction was assumed a linear function of temperature setting very small temperature interval in melting-freezing. In another case of temperature interval is not zero, such as mixed materials, solid fraction was assumed a linear or quadratic function. But in this simulation it was assumed that PCM is a pure material and solid fraction is a linear function of temperature.

Liner function

$$f = \frac{T_{liq} - T}{\Delta T} \quad (3)$$

Quadratic function

$$f = 1 - \frac{(T - T_{sol})^2}{\Delta T(T_{top} - T_{sol})} \quad (T_{sol} < T < T_{top}) \quad (4)$$

$$f = \frac{(T_{liq} - T)^2}{\Delta T(T_{liq} - T_{top})} \quad (T_{top} < T < T_{liq}) \quad (5)$$

where T_{sol} and T_{liq} are solidus temperature,

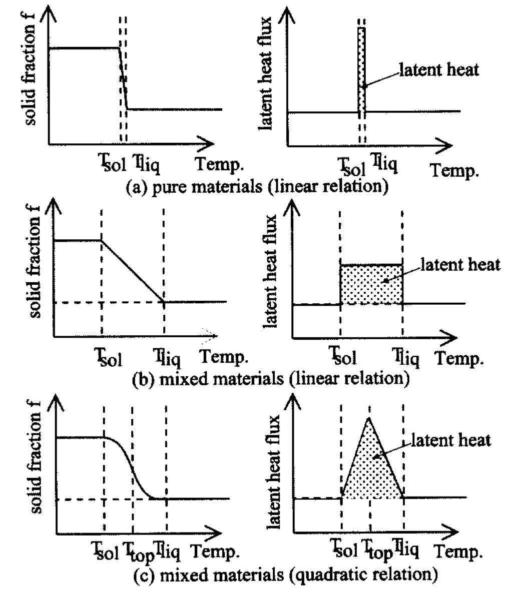


Figure 2. The relations between solid fraction and temperature, between latent heat flow and temperature.

liquidus temperature at which the melting process starts and ends, respectively. $\Delta T = T_{liq} - T_{sol}$ is the temperature interval in melting-freezing, T_{top} is the temperature at when latent heat flow reach the maximum.

Now the heat diffusion equation (Eq. (2)) including latent heat have the unstability of the implicit scheme for a nonlinear source term depending on solid fraction. Therefore source term was approximated by a linear function described before in order to enhance stability. The following source term S is obtained by substituting a linear function (Eq. (3)) into source term (Eq. (2)), where f^o is solid fraction from previous time, Δt is time step, and S_C, S_P are constant.

$$S = \underbrace{\frac{\rho L_a}{\Delta T}\left(\frac{T_{liq}}{\Delta T} - f^o\right)}_{S_C} - \underbrace{\frac{\rho L_a}{\Delta t \Delta T}}_{S_P} T \qquad (6)$$

CALCULATION CONDITIONS

Many researchers[6~9] have been studying on PCM wallboard systems which can significantly reduce energy requirements in solar houses. From this point of view, we have been studying on passive solar systems with a Trombe wall which is made of either concrete or PCM.

The test room of the direct gain system (DG) is shown in Figure 3, and the indirect gain system (Trombe wall system ; TW) is shown in Figure 4. They are consisted of a simple structure 3.6 m long by 2.7 m wide by 2.4 m high. The window area facing south is 6.48 m^2. The Trombe wall of concrete or PCM is located on the room side of a window to absorb solar energy, and forms a narrow channel between wall and window. Heat delivered to the occupied room partly by conduction through the wall and partly by a natural convection flow, through the channel and vents. In our test room of Trombe wall system, vents at the bottom and top of the Trombe wall are opened for 24 hr.

Next we used the standardized weather data of Sapporo city, located in JAPAN at around 43° North Latitude and 141° East Longitude,

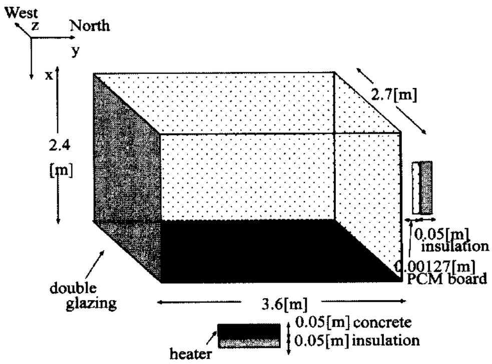


Figure 3. Direct gain system with PCM wallboard.

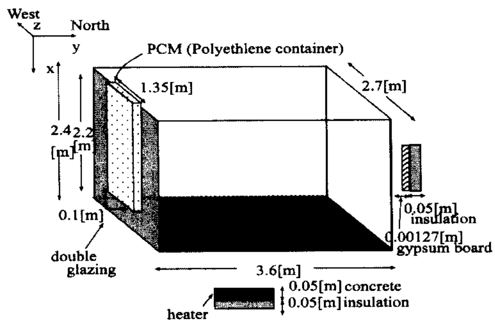


Figure 4. Trombe wall system with PCM.

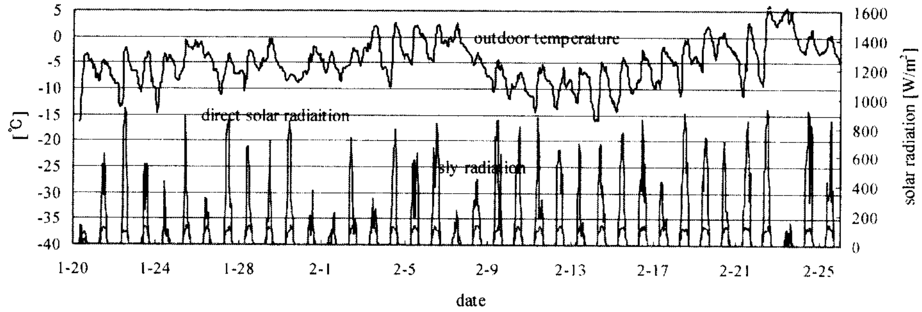


Figure 5. HASP weather data of Sapporo.

shown in Figure 5. As repetitive days in winter, January 20 ~ Feburuary 25 were chosen. Not to estimate the initial state for about first 5 days is necessary to avoid the distortions associated with the initial state, and simulation for long period of 30 days was necessary for reason that the thermal performances of PCM depend on outdoor conditions.

We used the following calculation conditions:

- The calculating mesh size for longwave and shortwave radiation is 9 x 12 x 8 = 864 cells. Time step Δt is 15 sec.
- The number of times of natural ventilation is assumed to be 0.5 time/hr.
- PMV is used as thermal comfort index. Thermal environmental conditions in the occupied zone are assumed that relative humidity is 40%, air velocity is 0.5 m/sec, clothing insulation is 1.0 clo, metabolic rate is 1.0 met.
- The solar absorptance of wallboard is assumed to be 0.4, one of concrete to be 0.6, and one of Trombe wall to be 0.9.
- To decrease the heat loss during night (11:00 pm ~ 7:00 am), aglazing is covered with insulation with a 40 mm thick insulating door.
- The physical property of glazing for the angle of incidence of direct solar beam was used.

In this study the electric floor heater (1 kW) was adopted as a heating system. The one of our aims is to calculate the energy consumption of the heater, and to estimate energy savings and peak power reduction obtained by keeping a constant level of the thermal environment in the occupied zone. PMV is a thermal comfort index empirically derived by Fanger[10] and expressed with air temperature, air velocity, mean radiant temperature and metabolic rate of a man etc.

It was assumed that the room is ventilated and cut solar radiation enter the room when the average PMV in the occupied zone exceeded 0.5, and the floor heater is switched on when the average PMV went down -0.5. And the economic performances such as energy savings and peak power reduction are estimated using power electric charge from energy consumptions of the heater. The power electric charge of type A is constant for 24 hr. On the other hand, in the One of type B the charge during peak time (16:00 ~ 18:00) is the highest, the one during nighttime is secondary highest, and the one during daytime is the lowest in winter in Sapporo.

The PCM (paraffin) Wallboard characteristics taken from[6] are shown in Table 1. It is assumed that the PCM and the gypsum matrix is a body of uniform equivalent physical and thermal properties, such as specific heat, density, thermal conductivity and latent heat. In this study the concentration of PCM in gypsum board was assumed to be 30 wt% because gypsum board can contain up to about 30 wt%. The PCM wallboard which have the same melting point were applied to all walls including ceiling. The total area of board in our test room is approximately 33 m^2. Now the melting point is important parameter, therefore we attempted to investigate the optimal melting point of it in winter in Sapporo.

Table 1. PCM Wallboard Characteristics

Wallboard	Density [kg/m^3]	Specific Heat [kJ/(kgK)]	Conductivity [W/(mK)]	Latent Heat [kJ/kg]
Conventional	696	1.089	0.173	0
30% PCM	998	1.467	0.232	58.3
20% PCM	800	1.341	0.204	38.9
10% PCM	720	1.215	0.187	16.3

Table 2. PCM Characteristics

		Solid	Liquid
Density	[kg/m^3]	1,510	1,510
Specific heat	[kJ/(kgK)]	1.43	2.31
Conductivity	[W/(mK)]	1.09	0.54
Latenet heat	[kJ/kg]	188	188

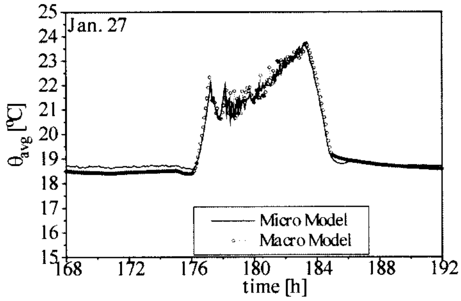
Next, Concrete is usually used as the Trombe wall, but we attempted to replace concrete by PCM. The PCM ($CaCl_2 \cdot 6H_2O$) produced by T Co., Inc., of which the melting point is 29℃ was chosen. The PCM characteristics are shown in Table 2. It was assumed that the container of PCM is polyethylene of which the thickness is 2 mm. Now, The most important parameter is the amount (thickness) of PCM, therefore the one of our aims is to investigate optimal the thickness of the PCM from the viewpoint of economic performances. As the conditions of Trombe wall, solar gain area of Trombe wall is 2.97 m^2, and the weight of PCM of which the thickness is 6, 10, 14 mm is 26.9, 44.8, 62.8 kg, respectively.

SIMLULATION RESULTS

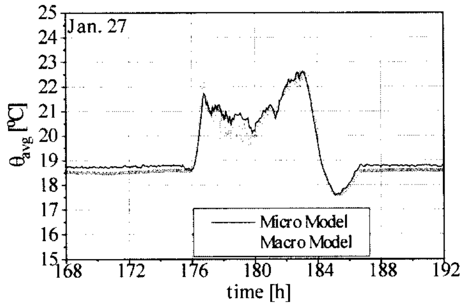
Comparisiion Micro and Macro Model

Calculated results of micro and macro model were compared through two case studies. Case 1 is a passive solar room with a Trombe wall inside it (Trombe wall system) as is shown in Figure 1. Case 2 is a passive solar room without a Trombe wall (direct gain system).

PMV was monitored to maintain the room thermal environment comfort. For two cases, unsteady simulations were performed with the weather data shown in Figure 5. Some results are shown in Figures 6, 7. As is mentioned above, although results for Jan. 25 to Jan. 27 is significant, only the data for Jan. 27 are plotted



(a) Direct gain system



(b) Trombe wall system

Figure 6. Comparison of room average tempera.

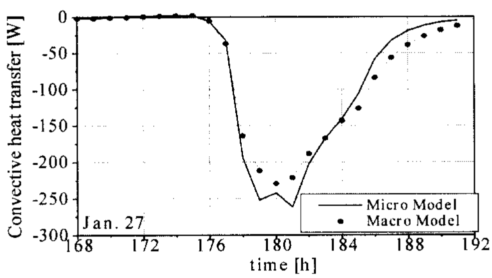


Figure 7. Comparison of convective eat flux at TW surface.

in the figures. Figure 6 is results of room average temperature θ_{avg}. The value 168 of x-axis means 0:00 o'clock of 8th day (Jan. 27) after calculation started at 0:00 on 1st day (Jan. 20). Figure 7 is comparison of convective heat flux at the Trombe wall surface. Here, negative value means that heat flows from surfaces to room air. Both in the results for θ_{avg} and for heat flux, macro model does not agree well with micro model during several hours (10:00～16:00). The reason may be explained that the ventilating fan operated during these hours to keep PMV below 0.5, so, room airflow became

Table 3. Calculating conditions

Melting point of PCM [℃]	Structure of enclosing walls	Trombe wall
gypsumboad	12 mm gypsumboad + 50 mm insulation	-
19	12 mm PCMboad + 50 mm sisulation	-
21	"	-
23	"	-
25	"	-
	12 mm gypsumboad + 50 mm insulation	200 mm concrete
29	"	6 mm thick PCM
29	"	10 mm thich PCM
29	"	12 mm thick PCM

Table 4. Energy savings, cost savings and peak power reduction of PCM wallboard in DG

		PCM19℃	PCM21℃	PCM23℃	PCM25℃
Energy Savings	[%]	12	11	7	4
Cost Savings	[%]	23	23	17	10
Peak Reduction	[%]	67	67	58	41

Table 5. Energy savings, cost savings and peak power reduction of PCM Trombe wall in TW

		Conc.20 cm	PCM6 mm	PCM10 mm	PCM14 mm
Energy Savings	[%]	21	12	18	20
Cost Savings	[%]	27	26	34	34
Peak Reduction	[%]	51	71	71	71

forced convection. The applicability of macro model may be limited for the rooms in which forced convection occur. However, these results indicate that macro model can realize the thermal environments with practically acceptable accuracy.

Another significant viewpoint is that macro model may be able to save calculating time drastically and this is the very reason why macro model is introduced and tested in this study. For the simulations presented above, the ratio of required cputime of macro model to that of micro model became about 1:50. This imply that macro model may be applicable to long term simulations, for example, to annual energy consumption analysis.

Application of Macro Model

In this section, the applicability of PCM (phase change material) in winter heating season was examined numerically using a passive solar room same as that shown in Figures 3, 4. Both direct gain (DG) systems (without Trombe wall) and Trombe wall (TW) systems were investigated. Calculating conditions are summarized in Table 3.

Simulations for 30 days were performed to investigate the effects of PCM wallboard and PCM Trombe wall as shown Table 1, the optimal melting point of PCM wallboard and the optimal thickness of PCM Trombe wall under the constant thermal environmental conditions ($-0.5 < PMV < 0.5$). In the first place, Direct gain system with gypsum board was assumed to be reference system. As the results, the energy savings, cost savings and peak power reduction of each system were calculated. The results are shown in Tables 4, 5.

In direct gain system (DG) with PCM wallboard the melting point levels of 19 or 21℃ was most effective, and energy saving was 12%, cost saving was 23%. PCM wallboard also was able to reduce the amount of energy consumption during peak time (16:00 ~ 18:00) by 67%. On the other hand, in Trombe wall system (TW) the energy saving of PCM Trombe wall of which the thickness is 14 mm was the same as the one of concrete Trombe wall of which the thickness is 20 cm. But the peak power reduction of PCM Trombe wall was 20% larger than the one of concrete Trombe wall. From these results, it can be stated that PCM is effective in energy savings, particularly in peak power reduction in Sapporo. It was also found that the thermal capacity of PCM is 14 times as large as the one of concrete, in other words it can reduce the volume of Trombe wall by 1/14.

The results of room average air temperature θ_{avg} of direct gain system with gypsum board and PCM wallboard according to calculation conditions on a sunny day (Jan. 25) are shown in Figure 8. At the peak time, the room average air temperature drop of PCM wallboard was slower than gypsum board one's. This is because the freezing process occurs and the released latent heat restricts the temperature drop of PCM wallboard. On the other hand, The air temperature drop of PCM wallboard of which melting point is 23°C was the slowest, one of which melting point is 19°C was the earliest of PCM wallboards. These room air temperature profiles is similar to the surface wallboard temperature ones (Figure 9). In short, it can be predicted that the surface wallboard temperature has the large impact on room average air temperature.

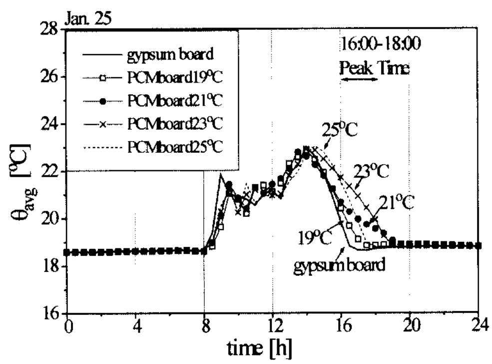


Figure 8. Room average air temperature of DG (gypsum board versus PCM wallboard).

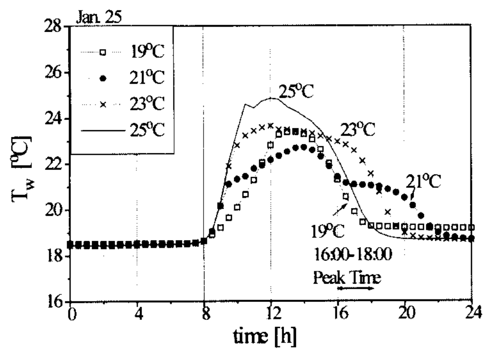


Figure 9. Surface average temperature of PCM wallboard on ceiling.

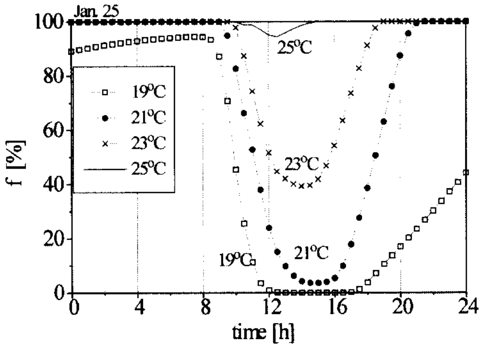


Figure 10. Solid fraction of all PCM wallboard on ceiling.

The results of surface average temperature Tw of PCM wallboard on ceiling in direct gain system are shown in Figure 9. The surface wallboard temperatures of melting point were temporarily maintained because latent heat is released at each melting point. In the case of melting point of 21°C, the feature can be seen clearly. For 4 hr (16:00 ~ 20:00) the surface wallboard temperature of melting point 21°C was maintained, therefore it was found that PCM wallboard of which melting point is 21°C is the most effective in peak power reduction under this thermal environmental conditions ($-0.5 < PMV < 0.5$). On the other hand, The surface temperature drop of other PCM wallboard is early, and it was found that they are not effective.

The results of solid fraction of PCM wallboard on ceiling are shown in Figure 10. Solid fraction f of 100% means that all PCM wallboard on ceiling is perfect solid state, on the other hand solid fraction of 0% means that all

PCM wallboard on ceiling is perfect liquid state. In the first place, it can be stated that PCM wallboard of which melting point is 25°C didn't use latent heat at all because it was solid state without transition in most time. As the results the surface wallboard temperature drop of it was early. Next, PCM wallboard of which melting point is 23°C also didn't effectively latent heat because it didn't reach maximum solid fraction of more than 40%. On the other hand, PCM wallboard of which melting point is 19°C was able to reach the liquid state perfectly but it also didn't release latent heat perfectly. Because it wasn't return solid state by night of that day, Jan. 25. After all, in the case of melting point of 21°C, latent heat was used the most effectively.

From these results, It can be stated that the thermal performances of PCM depend on melting point. Therefore, to investigate the relation between optimal melting point and thermal environmental conditions in the room is necessary for effective use of latent heat. So we thought of the optimal melting point of PCM wallboard as temperature maximizing the amount of heat storage Q_C of it, and investigated the relation between melting point T_m of each wall (including ceiling) and the surface minimum temperature $T_{w,min}$ of each wall when average PMV is -0.5. From the results of Figure 11, It was found that the optimal melting point maximizing the amount of heat storage of PCM wallboard is approximately 2°C higher than the surface minimum temperature of each wall.

Therefore, we attempted to apply the relation of optimal melting point to warmer thermal environmental conditions for 0.0 < PMV < 1.0 in the occupied zone. Under the constant thermal environmental conditions, the surface minimum

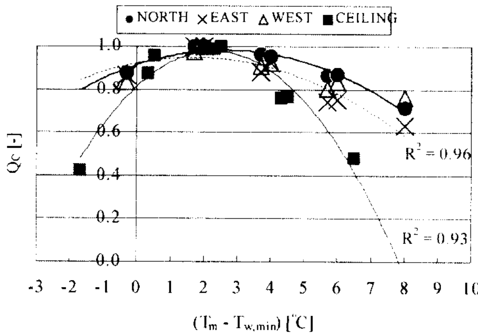


Figure 11. The relation between the amount of heat storage Q_C and melting point T_m.

temperature of ceiling was approximately 21°C, and surface minimum temperatures of other walls were approximately 19°C. The reason that the surface temperature of ceiling was approximately 2°C higher than the one of other walls is due to receive directly impact of longwave radiation from floor heater. From the relation of optimal melting point, the optimal melting point of ceiling was 23°C, the ones of other walls were 21°C. The results of energy savings, energy costs and peak power reductions are shown in Table 6. As can be seen, Energy savings and cost savings of system with optimal melting points were the largest except peak power reduction. From these results it can be stated that the relation of that the optimal melting point of PCM wallboard is approximately 2°C higher than the surface minimum temperature of each wall is valid.

CONCLUSIONS

An implicit transient finite difference model was developed to simulate the transient heat transfer in a wall with PCM. Using this model,

Table 6. Energy savings etc. of the optimal melting point

Melting point	Energy savings [%]	Energy costs [%]	Peak reduction [%]
19°C	5	9	28
21°C	10	19	48
23°C	10	21	63
25°C	6	16	54
23°C on ceiling 21°C on walls	11	23	58

an numerical investigation of the thermal performances of PCM wallboard and PCM Trombe wall was performed in a direct gain system and a Trombe wall system, respectively. The numerical results showed a significant peak power reduction due to latent heat of PCM in Sapporo. It was also found that the optimal melting point of each wall is approximately 2°C higher than the surface minimum temperature of each wall under constant thermal environmental conditions. Finally numerical simulation is effective in these examinations because it is impossible to perform them by experiments.

ACKNOWLEDGEMENT

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NOMENCLATURE

c	specific heat	[kJ/(kg·K)]
f	solid fraction (f = 1 ; solid f = 0 ; liquid)	[-]
k	thermal conductivity	[W/(m·K)]
L_a	latent heat	[kJ/kg]
q_C	convective heat flux	[W/m^2]
Q_C	amount of heat storage	[-]
q_L	conductive heat flux	[W/m^2]
q_P	heat source at the surface	[W/m^2]
q_R	radiative heat flux	[W/m^2]
t	time step	[sec]
T	temperature	[℃]
x	spatial coordinates	[m]

Greek symbols

α_C	convective heat transfer coefficient	[W/(m^2·K)]
ρ	density	[kg/m^3]
ΔT	temperature interval in transition	[℃]
θ_{avg}	room averaged temperature	[℃]
Δt	time step	[sec]

Subscripts

liq	liquid
m	melting
min	minimum
sol	solid
top	peak of latent heat
w	wall

Superscripts

o	previous time

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