PREDICTION OF SENSORY INDEX UNDER NON-UNIFORM THERMAL ENVIRONMENT BASED ON HEAT AND MOISTURE TRANSFER AND AIRFLOW OF WHOLE BUILDINGS

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ABSTRACT

Many simulation software to predict thermal environment of buildings, such as temperature, humidity, heating and cooling load of building spaces, have been developed. However, most of them do not take into account moisture transfer in wall assemblies. Then, sensory index such as standard new effective temperature is even excluded from calculation. A Heat, Air and Moisture (HAM) simulation software called THERB for HAM has been developed for the purpose of estimating the hygrothermal environment within buildings. This software has complete HAM features including principles of moisture transfer within walls and can estimate temperature, humidity, sensory index 'COMSET*' based on hygrothermal balance of human body, and heating/cooling load for multiple zone buildings. In this paper, the prominent features of THERB are highlighted. The calculation accuracy is verified through the comparison with monitoring results of a test house. Furthermore, thermal comfort under non-uniform thermal environment such as floor heating is indicated as the sensory index based on hygrothermal balance of human body in combination with heat and moisture transfer and airflow of whole buildings. Then energy simulation for space heating and floor heating is performed to evaluate energy conservation of both system on the basis of sensory index 'COMSET*'.

INTRODUCTION

THERB is dynamic simulation software, which can estimate temperature, humidity, sensory index, and heating/cooling load for multiple zone buildings and wall assemblies. The heat and moisture transfer models used in THERB such as conduction, convection, radiation and ventilation (or air leakage) are based upon the detailed phenomena describing actual building physics, and can be applied to all forms of building design, structure or occupant schedules, etc. All the phenomena are calculated without simplification of the heat and moisture transfer principles of any building component or element. The moisture transfer model using the water potential, which is defined as thermodynamic energy, is a progressive feature, which incorporates moisture transfer including moisture sorption and desorption of walls. Thus THERB can predict the hygrothermal environment of the whole building taking into consideration the complex relationship between heat and moisture transfer and airflow. This paper explains prominent features of the calculation models, and the accuracy of THERB through the comparison with a test house equipped with a hydronic floor heating system. Then sensory index 'COMSET*' based on hygrothermal balance of various parts of human body is calculated under non-uniform thermal environment of floor heating with a combination of THERB.

THEORETICAL FEATURE OF THERB

The followings outline the algorithms for heat and moisture transfer and airflow used in THERB, which are derived from building physics principles (Ozaki et al. 2006). Numerical models such as the fin efficiency and the convective and radiative heat transfer from floor relating to hydronic floor heating system, which is newly incorporated in THERB, are particularly highlighted.

Conductive Heat and Moisture Transfer

The finite difference method is applied to the model of one-dimensional transient hygrothermal conduction of multi-layer walls. Regarding thermal conduction to the ground, the finite difference method of two or three dimensions is applied to the previous calculation of the ground temperature and then the results are used as the input excitation for conductive calculation of the earthen floor and basement walls.

Water Potential which is derived by applying the chemical potential of thermodynamics to moisture diffusion is used as the driving force of moisture transfer (Ozaki et al. 2001). This approach is proposed to be more theoretical and accurate than other models based on physical properties such as vapor pressure (Trechsel 2001). The model called P-model using water potential makes it possible to combine moisture transfer with heat transfer perfectly as thermodynamic system, and take into account internal energy and external forces such as pressure increment and gravity.

The fin efficiency, which explains the ratio of actual heat transfer from the fin surfaces to hypothetical heat transfer assuming the fin temperature is equal to hot-water temperature in a tube, is applied to

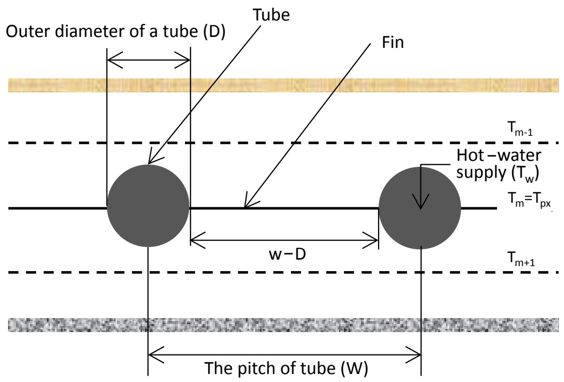


Figure 1 Hydronic floor heating system

hydronic floor heating system. Figure 1 illustrates the hydronic floor heating system. Eqs.(1) to (8) show the calculating formula of the fin efficiency.

- Fin efficiency

$$\eta_f = \frac{1}{w}\left[D + (w-D)\frac{\tanh mD}{mD}\right] \quad (1)$$

$$mD = \sqrt{\frac{C_f \cdot P}{\lambda_f \cdot t}}D \quad (2)$$

- Heat transmission coefficient from hot-water to tube surface

$$K_p = \frac{A_f}{L_f \cdot R_b} \quad (3)$$

$$R_b = \frac{D}{\lambda_w \cdot Nu} \quad (4)$$

$$Nu = \frac{0.0395 \cdot \mathrm{Re}^{0.75} \cdot \mathrm{Pr}}{1. + (1.99 \cdot \mathrm{Re}^{-0.125} \cdot (\mathrm{Pr} - 1.0))} \quad (5)$$

- Heat balance of hot-water

$$C_w \cdot \rho_w \cdot V_w \frac{\partial T}{\partial t} = \eta_f \cdot K_p \cdot (T_m - T_w) \cdot L_f + Q_s \quad (6)$$

$$Q_s = q_f \cdot C_w \cdot \rho_w \cdot (T_s - T_w) \quad (7)$$

$$-\frac{\partial T}{\partial t} = \eta_f \cdot K_p \cdot (T_m - T_w) = \frac{1}{R_m} \cdot (T_m - T_{m-1}) + \frac{1}{R_{m+1}} \cdot (T_m - T_{m+1}) \quad (8)$$

Convective Heat and Moisture Transfer

The convective heat transfer coefficients are recalculated at every time step on all surfaces of the exterior, interior and cavities of buildings using dimensionless equations as shown in Table 1 which are derived from either the profile method for boundary layer (based on the energy equation, the momentum equation and the fluid friction) or defined from the experimental findings according to natural or forced convection (Fujii et al. 1972, Ozaki et al. 1990). Furthermore the natural convective heat transfer coefficients are classified into either vertical or horizontal surfaces. It is possible to use the functional equations of the wind direction and velocity for the exterior convective heat transfer coefficients and the functional equations of the temperature difference between surface and room for the interior convective heat transfer coefficients. It is also possible to set constant heat transfer coefficients all day long or modify the coefficients to take into consideration space conditioning time for all parts of the building. The convective moisture transfer coefficients on all surfaces of the exterior, interior and cavities of buildings are calculated on the basis of the analogy between heat and mass transfer.

Table 1 Convective Heat Transfer Coefficient

Part of Buildings	Dimensionless Number
Exterior	$Nu = 0.037\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3}$
Interior (Vertical Plane)	$Nu = 0.241(Gr_i \cdot \mathrm{Pr})^{0.4}$ $Gr_i = g\beta_{air}\Delta T_a l^3 / \upsilon^2$
Interior (Horizontal Plane)	$Nu = C \cdot Ra_f^m$ $Ra_f = Gr_i \cdot \mathrm{Pr}$ $f = (T_s + T_\infty)/2$
Upward	C:0.58, m:1/5
Downward	C:0.54, m:1/4 (Ra_f: 2E4 to 8E6) C:0.15, m:1/3 (Ra_f: 8E6 to 1E11)
Cavity (ventilated)	$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}$
Cavity (closed)	$Nu = 0.035(Gr_c \cdot \mathrm{Pr})^{0.38}$ $Gr_c = g\Delta T_s l^3 / T_M \upsilon^2$

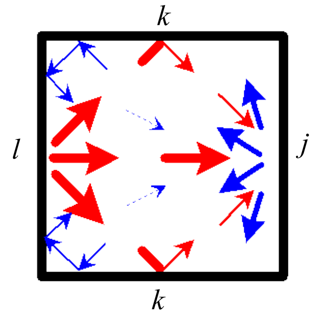


Figure 2 Multiple reflection of long-wave radiation

Radiant Heat Transfer

On the exterior surfaces of the buildings, the general method of using the radiant heat transfer coefficients and atmospheric radiation is applied in default. Interrelated radiation between both surfaces of building and the ground can be also calculated with temperature calculation of the ground. On the interior of buildings, the use of the long-wave absorption coefficient makes it possible to simulate a net absorption of radiant heat as a consequence of multiplex reflection among interior surfaces. Mutual radiation between the surfaces of cavities in walls and windows can be also calculated.

The multiplex reflection of long-wave radiation between interior surfaces is simulated on the basis of Gebhart's absorption coefficient (Gebhart 1959). The long-wave absorption coefficient $\beta_{l,j}$ is defined as the net ratio of absorbed radiant heat on surface j from the surface l.

$$\beta_{l,j} = F_{l,j}\varepsilon_j + \sum_{k=1}^{J} F_{l,k}(1-\varepsilon_k)\beta_{k,j} \tag{9}$$

where $F_{l,j}$ is the view factor from the surface l to the surface j and ε_j is emissivity of the surface j. Thus the net radiant heat emitted from the surface j (see Figure 2) is expressed by Eq.(10).

$$ELR_j = \varepsilon_j \sigma T_j^4 - \sum_{k=1}^{J} \beta_{k,j}\varepsilon_k \sigma T_k^4\, S_k / S_j \tag{10}$$

Eq.(10) can be re-arranged into Eq.(13) by using the principle of the conservation of energy and the reciprocal theorem expressed by Eqs.(11), (12).

$$\sum_{k=1}^{J} \beta_{j,k} = 1 \tag{11}$$

$$\varepsilon_k S_k \beta_{k,j} = \varepsilon_j S_j \beta_{j,k} \tag{12}$$

$$\begin{aligned} ELR_j &= \varepsilon_j \sigma \sum_{k=1}^{J} \beta_{j,k}\left(T_j^4 - T_k^4\right) \\ &= \sum_{k=1}^{J} \beta_{j,k}\alpha_{r,jk}\left(T_j - T_k\right) \end{aligned} \tag{13}$$

where S_j is the area of surface j. $\alpha_{r,jk}$ is the radiant heat transfer coefficient from surface j to surface k and can be approximated by Eq.(14).

$$\alpha_{r,jk} \cong 4\varepsilon_j \sigma \left\{\left(T_j + T_k\right)/2\right\}^3 \tag{14}$$

The long-wave absorption coefficient can be applied to long-wave radiant heat emitted from lights and appliances and human bodies by assuming that such radiant heat is equally emitted from the ceiling or floor. The absorption amount ALR_j on the surface j can be described by Eq.(15).

$$\begin{aligned} ALR_j &= \sum_{c=1}^{C} \beta_{c,j}\, LI_c S_c / S_j + \sum_{f=1}^{F} \beta_{f,j}\, LH_f S_f / S_j \\ &= \varepsilon_j \left(\sum_{c=1}^{C} \beta_{j,c}\, LI_c / \varepsilon_c + \sum_{f=1}^{F} \beta_{j,f}\, LH_f / \varepsilon_f \right) \end{aligned} \tag{15}$$

where LI_c and LH_f are the radiant heat from lights and appliances and human bodies, respectively. Thus the net radiant heat emitted from the surface j can be calculated by Eq.(16).

$$NLR_j = ELR_j - ALR_j \tag{16}$$

Incident Solar Radiation

Incident solar radiation on the exterior and into the interior of buildings is divided into direct and diffuse solar radiation and calculated for all parts of the building in all directions using accurate geometric calculations of shaded and unshaded portions of the building by considering the influence of overhangs and side walls. Transmitted solar radiation is calculated by the multi-layer window model which considers multiplex reflection (depending on an incidence angle of solar radiation) between not only the glazing layers but also between the window and interior shade at every time step. The multiplex reflection of both transmitted direct and diffuse solar radiation among interior surfaces including re-transmission of solar radiation from the inside to the outside through the windows is calculated by using the short-wave absorption coefficient. In addition the absorption coefficients of short wave are applied to radiant heat emitted from lights and appliances.

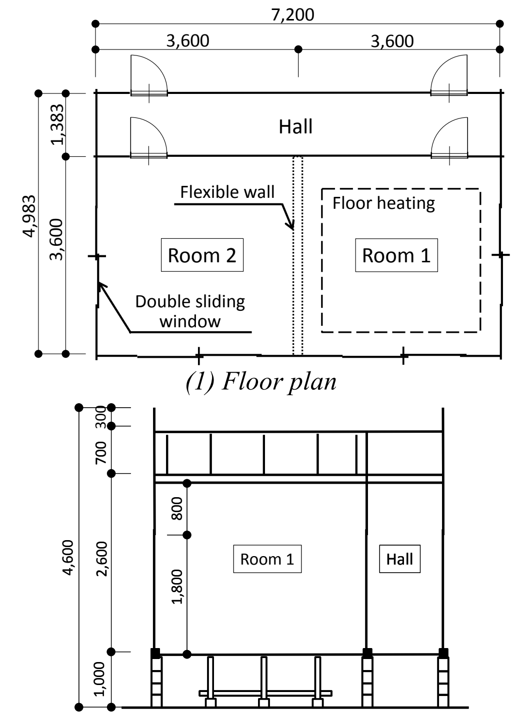
Ventilation

The network airflow model integrating a thermal model with a plant model estimates natural and forced ventilation quantities of each zone (rooms and cavities) caused by air leakage, infiltration and mechanical ventilation. As for independent cavities naturally ventilated in the walls, it is possible to estimate airflow quantities by hydrodynamic analysis as the solution to the equations of motion, energy and continuity. Constant ventilation quantities can be also set for all zones every hour.

Space Conditioning

Indoor air temperature and humidity can be calculated from heat and moisture balance of a space based on convection, ventilation, internal generation of heat and moisture. Indoor humidity is interrelated with moisture sorption and desorption of walls by applying the P-model of heat and moisture transfer. General humidity calculation that is just affected by ventilation is also available.

Sensible and latent heat load are obtained from the equations of heat and moisture balance, in which unknown quantities are space heating and cooling load, on condition that temperature and humidity are set at reference ones. Control methods for space conditioning are classified into three types; heating,



(1) Floor plan

(2) Cross section

Figure 3 Plan of the experiment model

cooling, and simultaneous heating and cooling. By default, humidity control and temperature control are linked. Temperature and humidity set-point and ranges can be optionally set every hour. Moreover the control of temperature and humidity is automatically performed in the case when the sensory index or sensible temperature such as PMV, SET* is set as the set-point of space conditioning.

Table 2 Calculation conditions

Heat loss coefficient [W/(m^2 • K)]		Ceiling	1.6
		Ex wall	1.2
		Floor	1.5
Floor area of room 1		12.96m^2 (3.6 m×3.6 m)	
Floor heating area		2.62 m×3.12 m (ratio of 70% of floor area)	
Diameter of tube		0.0098 m	
Pitch between tubes		0.075 m	
Input data	Temperature	Measured values per one minute(hall, room 2,crawl space, hot-water supply)	
	Flow rate of hot-water supply	1L/min	
Control method of hydronic floor heating system	Case 1	Room air temperature is kept at 21degrees C constant by on-off control of hot-water circulation (Manual operation)	
	Case 2	The hot-water is circulated for 12 minutes at intervals of 8 minutes (Intermittent operation)	
	Case 3	The hot-water is always circulated (Continuous operation)	

CALCULATION ACCURACY

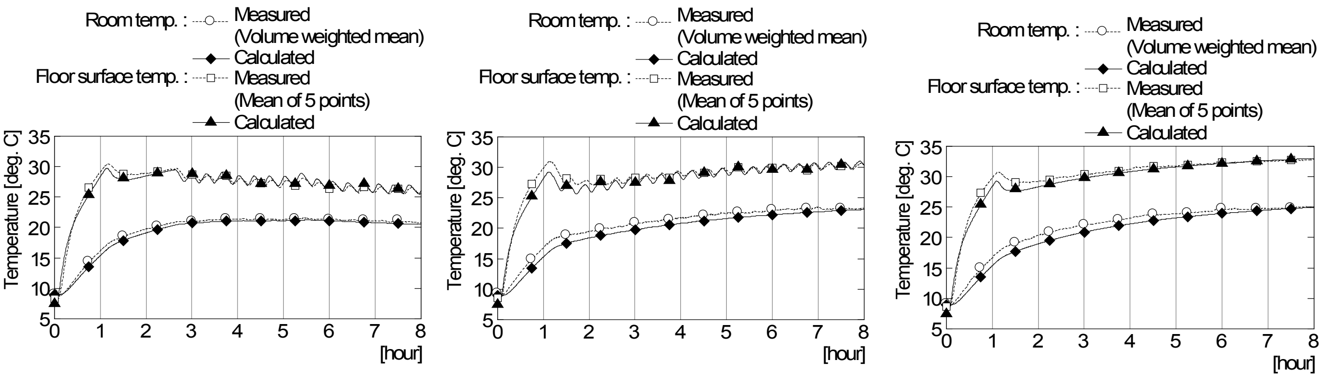
Test House and Experimental Conditions

Figure 3 illustrates the test house equipped with the hydronic floor heating system. The test house is an actual-size structure and built in an environmental test laboratory. The hydronic floor heating system is constructed in room 1 (3.6*4.98*2.6m) at a rate of 70% to floor area. Experimental conditions of floor heating are the following 3 cases. Room air temperature is kept at 21degrees C constant by on-off control of hot-water circulation in Case 1. The hot-water is circulated for 12 minutes at intervals of 8 minutes in Case 2. The hot-water is always circulated in Case 3. Initial temperature of the test house is about 7 degrees C with the same temperature as the environmental test laboratory. The hot-water at 80 degrees C is supplied for the first one hour in each case. After that, the hot-water at 60 dgrees C is supplied during turn-on circulation time.

Table 2 shows the calculation conditions. The calculation interval is one minutes and the measured temperature of next room, hall, crawlspace and hot-water supply in each experiment are used as input conditions.

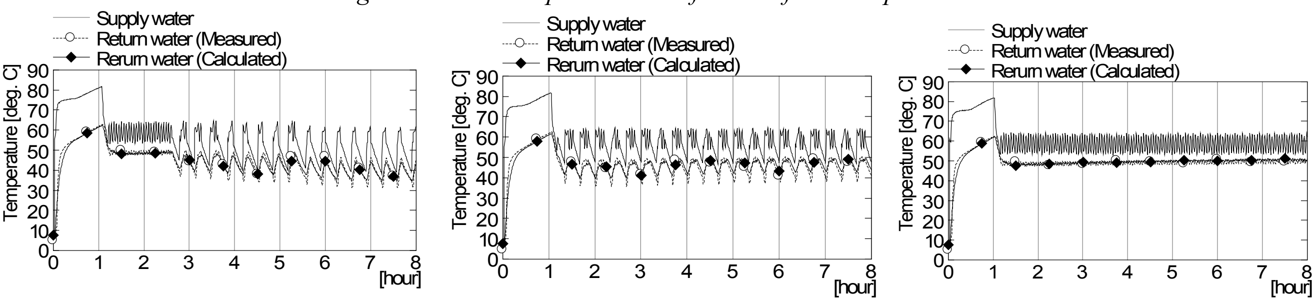
Comparison of calculated and measured results

Figure 4 and 5 show the temperature of room air, floor surface and return hot-water of the hydronic floor heating system. The measured air temperature is bulk temperature calculated as volume-weighted average of 150 points in the room. The measured floor temperature is average value of diagonal 5 points on the floor surface. Calculated values agree



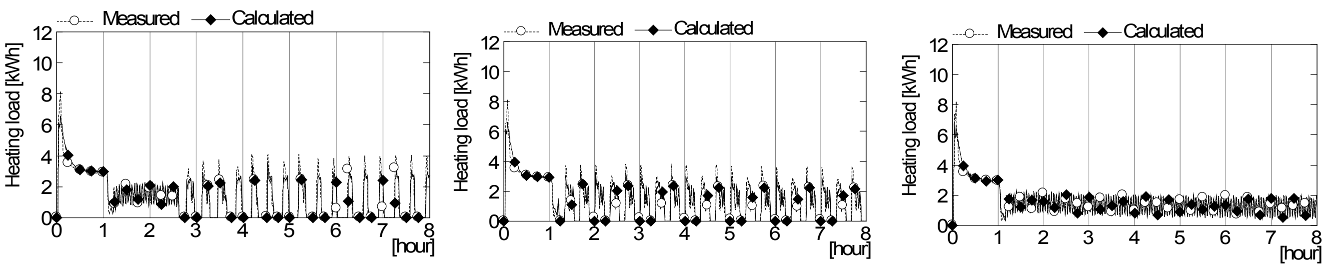
(1) Case 1 *(2) Case 2* *(3) Case 3*

Figure 4 Room temperature and floor surfaces temperature



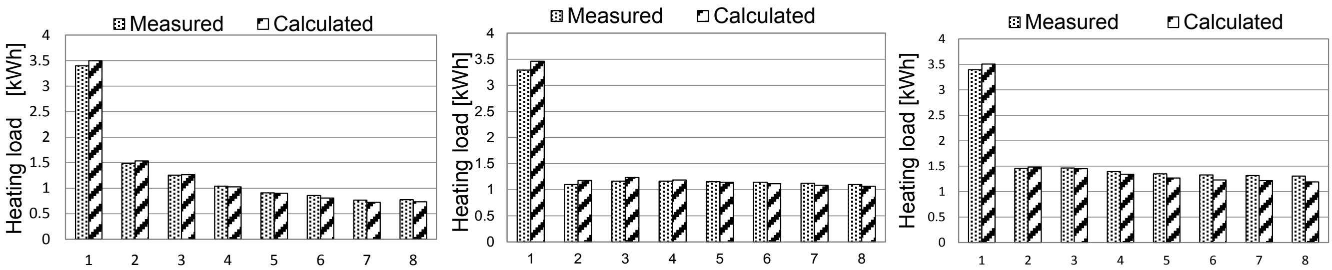
(1) Case 1 *(2) Case 2* *(3) Case 3*

Figure 5 Supply water temperature and return water temperature



(1) Case 1 *(2) Case 2* *(3) Case 3*

Figure 6 Changes of Heating load



(1) Case 1 *(2) Case 2* *(3) Case 3*

Figure 7 Total heating load per hour

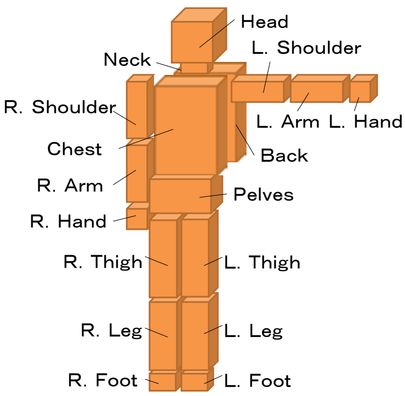
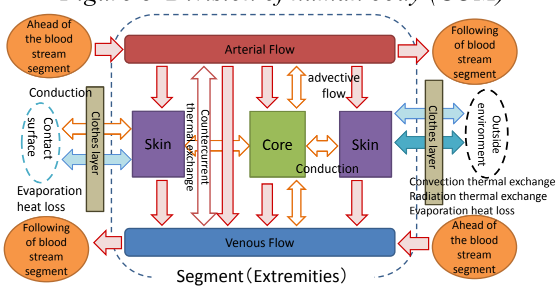
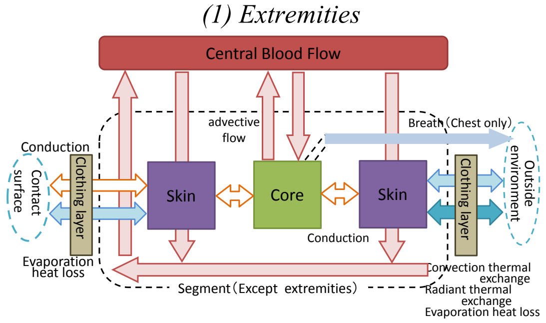


Figure 8 Division of human body (COM)



(1) Extremities



(2) Except extremities

Figure 9 Thermal exchange among core, skin, and blood

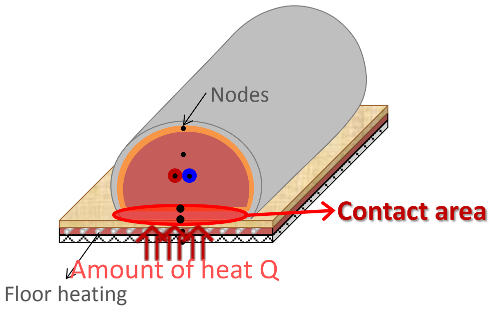


Figure 10 Heat balance of human body with contact area (building flame)

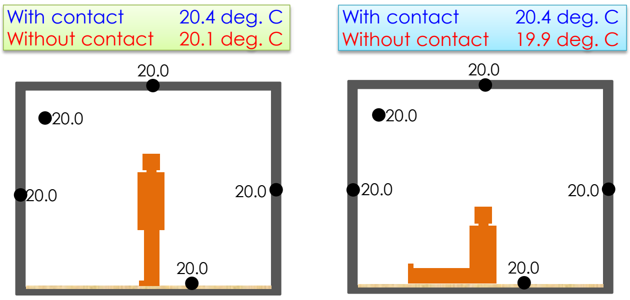
rather well with measured values in each experiment. Figure 6 and 7 show the variation with time and the integration value each for hour of heating load (heating amount of hot-water). The estimate has a smaller margin of error. THERB can predict thermal environment of room equipped with the hydronic floor heating system with absolute accuracy.

PREDICTION OF SENSORY INDEX

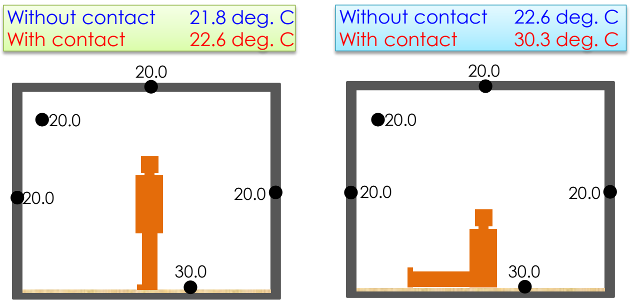
The influence of non-uniform thermal environment such as floor heating on occupant comfort is evaluated by using the sensory index 'COMSET*' which is derived from the heat balance of body parts in combination with THERB (Ozaki et al. 2011).

COMSET*

COMSET* is a mathematical model of sensory index (Tanabe et al. 2006), such as standard new effective temperature 'SET*', derived from the detailed heat balance of body parts with consideration of blood circulation (arterial and venous flow) throughout the body involving the extremities. Figure 8 to 10 illustrate the conceptual diagram of COMSET*. By breaking whole body into 17 segments with each skin and core layer, it can predict temperature distribution



(1) Uniform thermal environment



(2) Non-uniform thermal environment

Figure 11 Influence of contact sensation

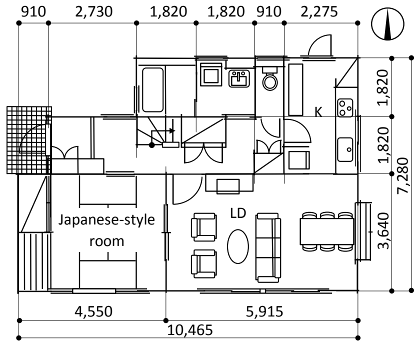
of skin and blood at 59 parts all over the body. Then COMSET* can calculate a generalized sensory index under the condition of non-uniform thermal environment by setting up the boundary conditions of surrounding air temperature and humidity, airflow velocity (convective heat flux), radiant heat flux, clothing amount and contact area on floor for each body segment and metabolic energy.

The original COMSET* does not take into account heat conduction in area contacted by floor. In this study, the influence of contact thermal conductance on sensation of warmth is considered by coupling heat balance of buildings and human body.

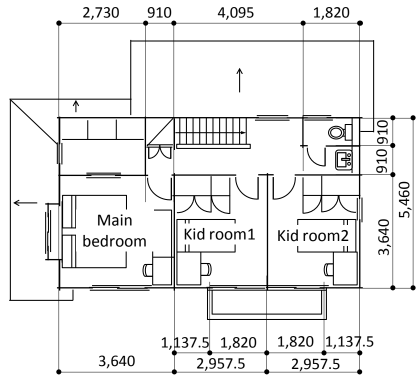
Influence of contact thermal conductance on sensation of warmth

As an example, Figure 11 shows the calculated values of COMSET* in standing and sitting position on the floor under uniform and non-uniform thermal environment. The every temperature of floor, walls and ceiling are the same as 20 degrees C in the uniform condition. The floor temperature is 30 degrees C and the temperature of walls and ceiling are 20 degrees C in the non-uniform condition. The influences of contact thermal conductance on COMSET* is clarified under the different conditions.

Calculated COMSET* in standing and sitting position are about the same with or without contact thermal conductance in the uniform condition. In contrast, COMSET* with contact thermal conductance in the non-uniform condition becomes higher than that without contact thermal conductance, particularly in sitting position which has larger contact area than standing position. Temperature difference of 7.7 degrees C is occurs on whether contact thermal conductance is incorporated.



(1) First floor



(2) Second floor

Figure 12 Floor plan of the building model

Table 3 Outline of buildig model

Gross floor area	[m^2]	120.07
Floor height	[m]	2.83
Opening ratio	[%]	26.80
Opening area	[m^2]	32.20
Heat loss coefficient Q	[W/(K･m^2)]	2.70
Solar acquisition coefficient μ	[-]	0.07

Table 4 Calculation conditions

Calculation region	Okayama (IVb region)	
Calculation date	January	
Air conditioning space	LDK	Space conditioning or Floor heating
	Other rooms	Space conditioning
Family structure	2 adult and 2 children	
Posture	Standing position and Sitting position	
Floor heating area	14.9 m^2 (ratio of 70% of floor area, LDK)	
Set temperature	20 deg. C	
Heating	All day within whole building	
Control of floor heating	On-Off control of hot-water supply	

EVALUATION OF THERMAL ENVIRONMENT BY COMSET*

Thermal environment and energy conservation of single-family house equipped with space conditioning system or hydronic floor heating system

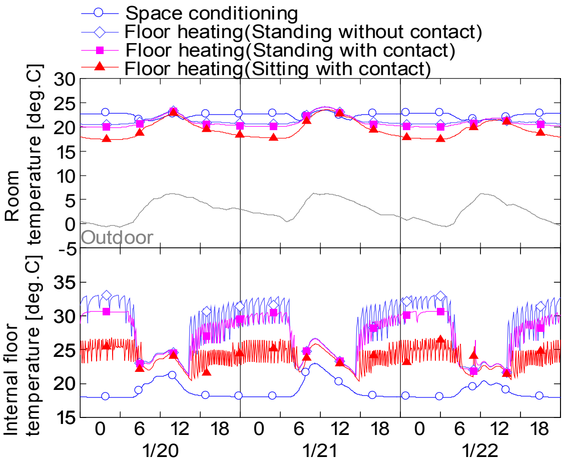


Figure 13 Room air temperature and internal floor temperature

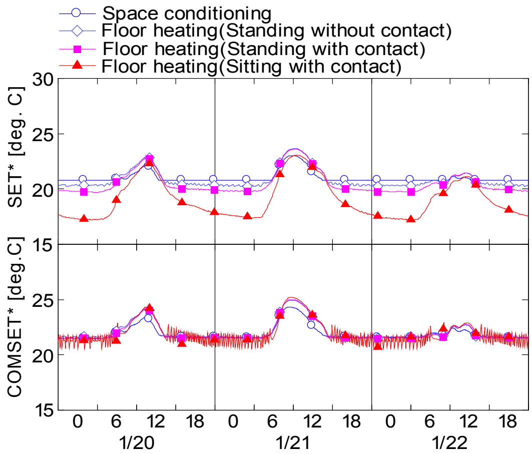


Figure 14 SET and COMSET**

are evaluated on the basis of COMSET* predicted by coupling simulation of both the building and human body, namely, THERB and COM.

Figure 12 and Table 3 shows the building model and Table 4 explains calculation conditions. The hydronic floor heating system is constructed in the living room at a rate of 70% to floor area. The standing and sitting position in the centre of the living room are supposed as the physical posture to calculate COMSET*. The space conditioning system in the living room is not used when the hydronic floor heating system is operated. The living room is controlled by COMSET* at comfortable thermal environment (around 21 degrees C of COMSET*).

Figure 13 and 14 show the time variation of air and floor temperature, SET* and COMSET* of living room from January 20 to 22. Although COMSET* of each heating condition are almost the same, room air temperature and SET* become higher in order of space conditioning, floor heating in standing position and floor heating in sitting position. In comparison with COMSET*, SET* is easily affected by air temperature. The sitting position that is significantly affected by radiative heat and contact thermal conductance from floor can lower air temperature under the condition of floor heating, if the sensory

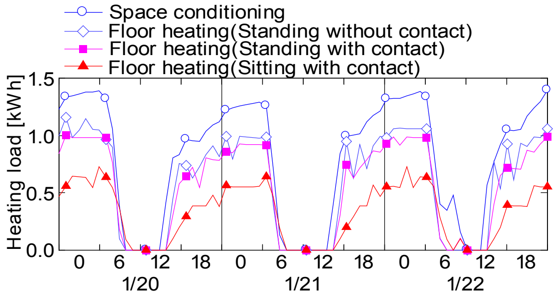


Figure 14 Heating load

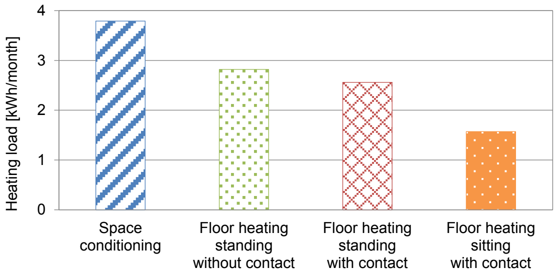


Figure 15 Term Heating load per one month

index of COMSET* is used as control requirement in keeping with the reality of human behavior.

Figure 15 and 16 show the time variation of heating load and the term heating load for a month (January) in the living room. In comparison with space conditioning system, the heating loads of hydronic floor heating system are decreased, particularly in the sitting position.

It is necessary to consider the radiative heat and contact thermal conductance from floor in each part of the human body in the floor heating system because the sensation of warmth in the floor heating is obviously different from the general space conditioning. In case that the sensory index of COMSET* is used as control requirement of heating, depending on the physical posture, the floor heating system has possibilities to dramatically decrease the heating load.

CONCLUSION

The simulation software 'THERB' incorporating complete features on heat, moisture and airflow has been developed to predict hygrothermal environment and sensory index within whole buildings. Hygrothermal theories on conduction, convection, radiation and ventilation are outlined, particularly numerical models relating to hydronic floor heating system, which is newly incorporated in THERB. The calculation precision in the condition of floor heating is verified through the comparison with monitoring results. THERB can predict thermal environment of room equipped with the hydronic floor heating system with absolute accuracy.

Furthermore, sensitive analyses on the heating system and the sensory index get the following results; 1) COMSET* derived from the detailed heat balance of body parts differs substantially on whether contact thermal conductance is incorporated, particularly in sitting position under the condition of floor heating, 2) even if the values of COMSET* are the same, room air temperature becomes higher in order of space conditioning, floor heating in standing position and floor heating in sitting position to cause a sense of warmth, 3) the floor heating system has possibilities to dramatically decrease the heating load, depending on the physical posture, if the radiative heat and contact thermal conductance from floor in each part of the human body are realistically considered as control requirement of heating.

NOMENCLATURE

A_f : internal area of tube [m^2]
C_f : thermal conductance from fin surface [W/(m^2.K)]
C_w : specific heat of water [J/(kg.K)]
D : diameter of tube [m]
F : view factor [-]
Gr : Grashof number [-]
K_p : apparent turbulent heat transfer coefficient in tube [W/(m^2.K)]
L_f : length of tube [m]
LI_c, LH_f : radiant heat from lights and appliances and human bodies [-]
Nu : Nusselt number [-]
P : circumferential length of tube [m]
Pr : Prandtle number [-]
Q_s : amount of heat [W]
R_b : thermal resistance per unit length from inner surface to outer surface of tube [m^2.K/W]
R_m : thermal resistance of material [m^2.K/W]
Ra : Rayleigh number [-]
Re : Reynolds number [-]
T_M : mean temperature of surfaces [K]
T_m : temperature in floor [K]
T_s : supply hot-water temperature [K]
T_w : temperature in tube [K]
ΔT_a: temperature difference between surface and air [K]
ΔT_s: temperature difference between surfaces [K]
V_w : amount of water [m^3]
g : gravitational constant [=9.8m/s^2]
l : characteristic length [m]
q_f : flow rate of hot-water [m^3/s]
t : thickness of tube [m]
ν : kinematic viscosity [m^2/s]
w : pitch of tube [m]
α_r : radiant heat transfer coefficient [-]
β : long-wave absorption coefficient [-]
β_{air} : expansion coefficient [1/K]
ε : emissivity [-]
η_f : fin efficiency [-]
λ_f : thermal conductivity of tube [W/(m.K)]
λ_w : thermal conductivity of water [W/(m.K)]
ρ_w : specific weight [kg/m^3]

Subscript

j,k,l: number of surface

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