INDOOR CLIMATE DESIGN BASED ON FEEDBACK CONTROL OF HVAC

COUPLED SIMULATION OF CONVECTON, RADIATION, AND HVAC CONTROL FOR ATTAINING GIVEN OPERATIVE TEMPERATURE

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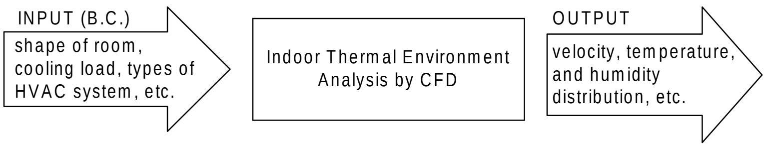
ABSTRACT

In this paper, a new CFD (Computational Fluid Dynamics) simulation method, coupled with a radiative heat transfer simulation and HVAC (Heating, Ventilating, and Air-Conditioning) control is presented. This new method can be used to assess the heating and cooling loads of different HVAC systems under the condition of same human thermal sensation (e.g. same operative temperature, etc.). To examine the performance of the new method, the thermal environment of the summer season within a semi-enclosed space which opens into an atrium space is analyzed under the steady-state condition. The most energy efficient HVAC system under the same thermal sensation can be chosen by using this method. The radiation-panel cooling system with air-curtain is found to be the most energy-efficient for cooling the semi-enclosed space in this study.

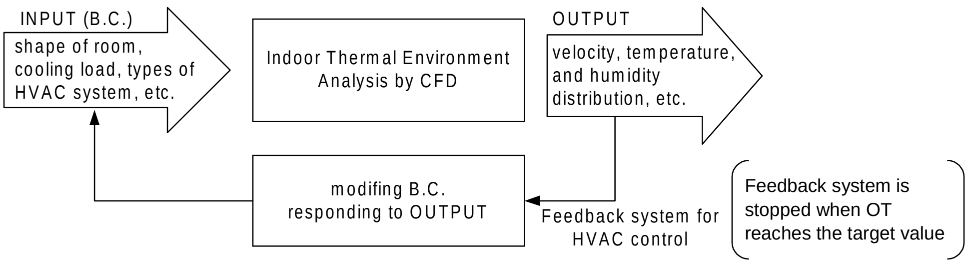
INTRODUCTION

CFD technique is used to analyze indoor thermal environments. CFD simulations (for summer season) until now have been conducted with given cooling loads in a room (Fig. 1 (a)). Since it is difficult to take into account the effects of velocity and temperature distributions in the room in advance, the B.C.s (Boundary Conditions) related to cooling load in the conventional simulation system are fixed as shown in Fig. 1 (a) and the indoor climate of the given B.C.s is analyzed. HVAC systems are, however, controlled based on temperature management of room air and the indoor climate of a targeted condition of the HVAC is obtained. Different types of HVAC system produce different thermal environments such as different vertical temperature distributions, different radiative fields, etc. Differences of thermal environment produce significant effects on the cooling loads and the indoor climate itself. In this context, CFD simulation should be able to address these problems, i.e., allow modification of the cooling load in the simulation procedure. Thus, a new simulation method that can feed back the output of the CFD to the input conditions for controlling the HVAC system is proposed as shown in Fig. 1 (b) and Fig. 2. Here the indoor climate of the targeted condition of the HVAC is obtained. This system is called here a new design method for HVAC systems with feedback [1].

This paper presents the new CFD method which is coupled with a radiative heat transfer simulation and HVAC control. This new method can provide information for modifying cooling loads which precisely



(a) Conventional indoor climate simulation with B.C. of fixed cooling load



(b) New method; Indoor climate analysis with B.C. of feedback cooling load

Fig. 1 Concept of Indoor Climate Simulation with/without Feedback System for HVAC Control
(B.C. : Boundary Condition, OT: Operative Temperature)

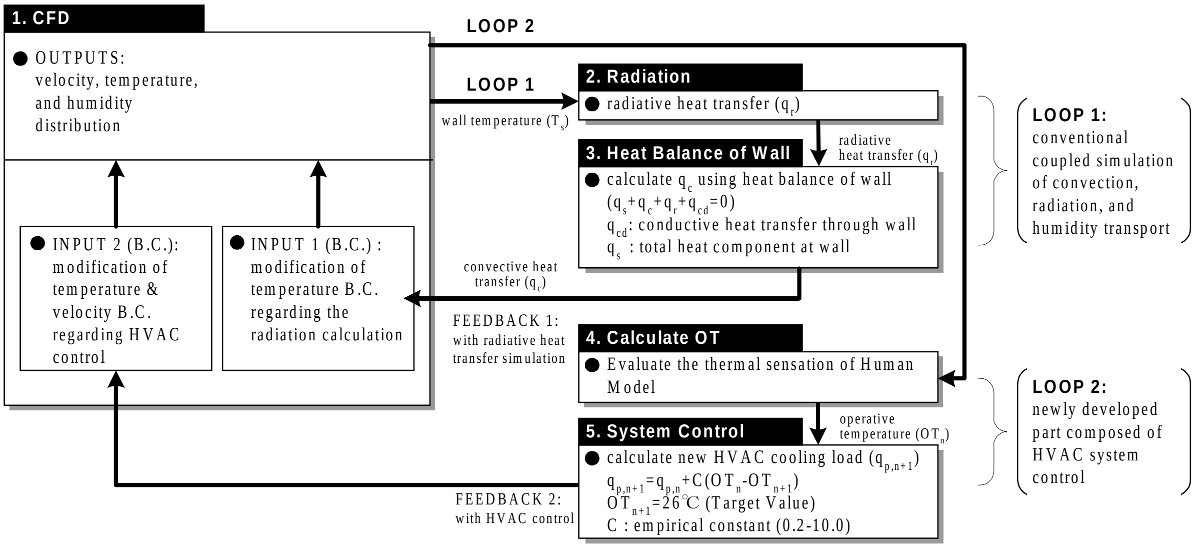


Fig. 2 Procedure of Simulation including HVAC System Control

take into account the temperature distribution, etc., and also the resultant indoor climate. In the simulation, HVAC outputs (e.g., inflow air temperature, inflow air volume, etc.) are modified by changing the input B.C.s using a feedback system of HVAC control as shown in Fig. 1 (b) and Fig. 2. These modifications are determined based on the simulations with the modified input B.C.s to keep the operative temperature of a human model at the target temperature (Fig. 1 (b), Fig. 2) [note 1].

To examine the performance of the new CFD method, the thermal environment of a semi-enclosed space (Fig. 3), which opens into an atrium space, is analyzed under the steady-state condition. In the study, the cooling condition in summer is analyzed. The atrium space is not fully air-conditioned, and its thermal environment is not the same as that of the semi-enclosed space. The opening between them causes the heat exchange between the two by air circulation and radiation. It is clear that both convective and radiative heat exchanges affect the cooling load of the semi-enclosed space. To predict the cooling load precisely, the amount of these exchanges must be accurately estimated. In this paper, two types of HVAC system are compared: one is a radiation-panel cooling system and the other is an all-air cooling system. Using the new CFD simulation with feedback system, the required cooling loads for attaining the same thermal sensation are quantitatively estimated and compared. The effects of an air-curtain at the entrance opening are also studied.

COUPLED SIMULATION OF CONVECTION, RADIATION, AND HVAC CONTROL

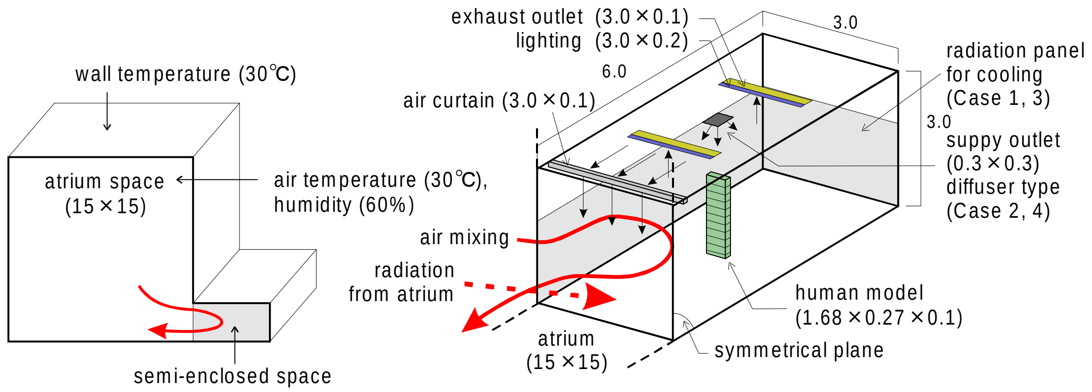
A feedback system, which returns the information of HVAC control, is added to the conventional coupled simulation of convection and radiation as shown in Fig. 2. In the procedure of the simulation, the feedback system modifies the B.C.s of CFD. In the usual HVAC system in a room, the air temperature at a specific point (e.g. exhaust outlets) is selected as the control target. However, it would be more rational for HVAC control systems to modify the HVAC outputs based on the thermal sensation of the occupants. In this paper, the thermal sensation of an occupant is evaluated based on his operative temperature (OT). The HVAC outputs are thus modified to keep the OT at the target temperature (26℃).

CFD Simulation

Indoor flow and temperature fields are calculated based on 3D CFD simulation, using the standard k-ε model. In the radiation analysis, the view factor is calculated by the Monte Carlo method [8] and the radiation heat transfer between the walls by Gebhart's absorption factor method [9]. Humidity distribution in the room is given by solving the humidity transport equation based on CFD. The authors have already confirmed by comparing the predicted results of room air flow and wall temperature with experimental results that the CFD coupled with the radiative heat transfer simulation is a sufficiently reliable tool for analyzing indoor thermal environments [5].

HVAC Control System

Using the time-dependent CFD simulation coupled with HVAC control, it is possible to control the time-dependent HVAC outputs dynamically responding to dynamic changes of various B.C.s (e.g. solar radiation, outdoor temperature, etc). In this paper, however,



(a) Semi-closed Space Opening into Atrium (b) Details of Semi-closed Space

Fig. 3 Semi-enclosed Space for Simulation (unit: m)
(half the space of the symmetrical room is illustrated)

Table 1 Cases Analyzed and Conditions Given

Case			1	2	3	4
Atrium	Wall and air temperature	[℃]	30.0	30.0	30.0	30.0
	Humidity	[%]	60.0	60.0	60.0	60.0
Semi-enclosed space	Given sensible heat load	[kW]	0.6	0.6	0.6	0.6
	Given latent heat	[kW]	0.1	0.1	0.1	0.1
	Operative temperature for human model	[℃]	26.0	26.0	26.0	26.0
Radiation-panel for cooling	Area	[m^2]	15.1	-	15.1	-
Supply outlet of all-air cooling system	Humidity	[%]	-	0.0	-	0.0
	Air flow rate	[m^3/s]		1620.0		907.0
		[h^{-1}]		30.0		17.0
Air-curtain	Air flow rate [ACH]	[h^{-1}]		-	60.0	60.0
	Outlet velocity	[m/s]			3.0	3.0
	Cooling load	[kW]			0.0	0.0

the assumption of steady-state indoor and outdoor conditions, which means that no dynamic changes of indoor/outdoor cooling loads, is introduced. In the simulation, the HVAC outputs are modified in proportion to the difference between the target value of OT and the output value i.e., simple proportional control is used here for the HVAC control.

Procedure of New Simulation Method

The procedure of the simulation is illustrated in Fig. 2. In this paper, two types of HVAC system are compared: one is a radiation-panel cooling system and the other is an all-air cooling system. In the case of the radiation-panel cooling system, the heat flux of the surface of the radiation-panel is modified based on $q_{p,n+1}$, while the temperature of the supply outlets is modified in the case of the all-air cooling system [note 2, 3]. The cooling loads of the HVAC systems, to achieve the target value of the OT of the human model (26℃), can be given through these simulations. The feedback system is stopped when this target point is achieved.

OPTIMAL DESIGN BASED ON CFD FEEDBACK MECHANISM

In this paper, only the B.C. of the heat flux of the radiation-panel and the air temperature of the supply outlet are modified in the simulation. However, with this method, other B.C.s (e.g., locations of supply outlets, number of supply outlets, etc.) can be modified so as to control the indoor climate at the target temperature. Through these simulations, the optimal design of HVAC system can be achieved. Although the description here concerns cooling system control, the same control could of course be applied to a heating system.

SEMI-ENCLOSED SPACE

To examine the performance of the new method, the thermal environment of a semi-enclosed space, which opens into an atrium space (15 m height), is analyzed (Fig. 3). For simplicity, all the walls, ceiling, and floor is assumed to be adiabatic. For this semi-enclosed space, two types of HVAC system are used for the study: one is a radiation-panel cooling system and the other is an all-air cooling system. In the CFD calculation here, the thermal environment of the atrium is modeled simply; the air and wall temperature, and humidity of the atrium are fixed at 30℃ and 60% respectively. For this reason, only radiation heat transfer is analyzed within the atrium and thus the CFD simulation within the atrium is not carried out.

Table 2　Indoor Cooling Loads (given)

	Human †	Lighting	Total
Sensible Heat (kW)	0.2	0.4	0.6
Latent Heat (kW)	0.1	0.0	0.1

† 55 W and 28 W are generated by one human model for sensible heat and latent heat respectively. The other sensible heat is assumed to be provided from the floor.

Table 3　Conditions for Calculation

Pressure B.C.	Pressure	Static pressure 0 at atrium
	Inflow	$k_{in} = 3/2(U_{in}\times0.05)^2$, $\varepsilon_{in} = C_\mu k_{in}^{3/2}/l_{in}$, l_{in} = width of the opening, T_{in} = 30℃
	Outflow	Free slip
Supply Outlet B.C.	$k_{in} = 3/2(U_{in}\times0.05)^2$, $\varepsilon_{in} = C_\mu k_{in}^{3/2}/l_{in}$, l_{in} = width of the opening, T_{in} is modified in response to the HVAC control system during CFD	
Wall, Radiation-panel, and Human Model	Velocity	Generalized log-law, free slip at symmetric plane
	Temperature	Convective heat transfer coefficient (α_c) Radiation-panel: 5.5, adiabatic surface: 3.0, others: 4.0 (W/m²℃)
	Humidity	1. Human model: emission rate is fixed (Table 2). 2. Radiation-panel: AH (Absolute Humidity) is given corresponding to the saturated vapor pressure, when the surface temperature of radiation-panel is lower than dew point temperature of the air. In other cases, Gradient of AH = 0. Humidity transfer coefficient α' is calculated based on Lewis Relation. 3. Other wall: Gradient of AH = 0.
	Emissivity of radiation	Wall, human model: 0.9 Symmetrical plane: 0.0
Mesh	CFD　　: 40 (L)×18 (W)×18 (H) Radiation: 16 (L)× 5 (W)× 8 (H)	

[symbol]

U_{in} : air velocity of supply outlet [m/s]　　k_{in} : turbulent kinetic energy of supply outlet [m²/s²]

ε_{in} : dissipation rate of turbulent kinetic energy [m²/s³]　　l_{in} : turbulence length scale [m]

T_{in} : temperature of supply outlet [℃]　　α_c : convective heat transfer coefficient [W/(m²℃)]

α' : humidity transfer coefficient [kg/(m²s(kg/kg))] ≒ $0.001\alpha_c$

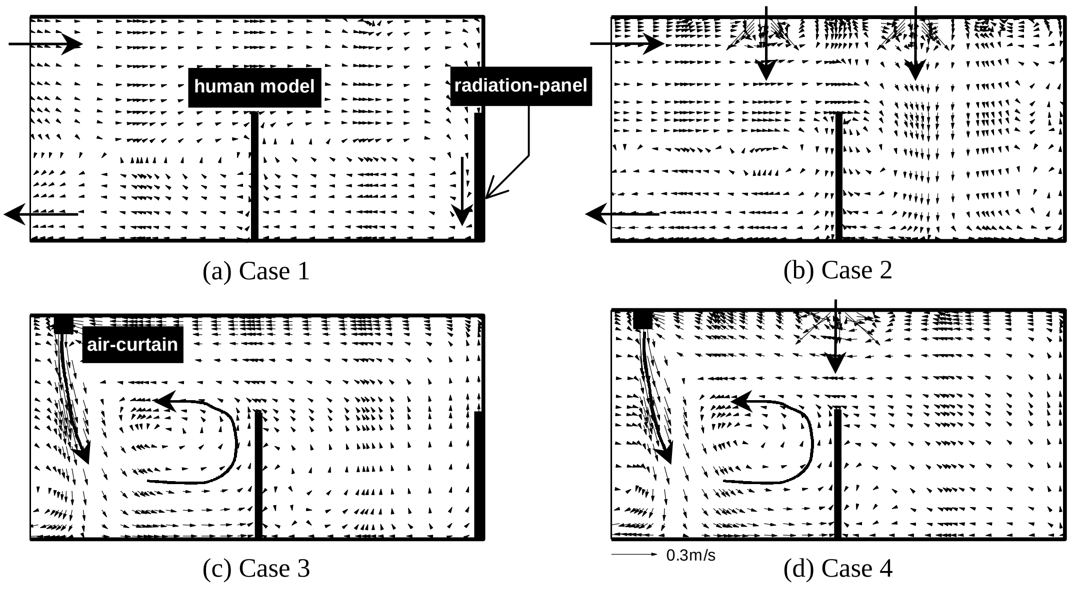
Table 4　Predicted Results of Cooling Loads (values are given for half space of semi-enclosed space)

Case			1	2	3	4
Semi-enclosed space	Sensible cooling load by air-mixing	[kW]	1.1	8.4	1.0	4.1
	Latent cooling load by air-mixing	[kW]	0.9	16.7	1.2	8.3
	Sensible cooling load by radiation	[kW]	0.2	0.05	0.2	0.09
	Total cooling load	[kW]	2.2	25.15	2.4	12.49
	Air-mixing rate	[m³/h]	931.0	5990.0	3370.0	3660.0
	Average room air temperature	[℃]	28.8	25.8	27.7	24.3
Radiation-panel for cooling	Sensible cooling load	[kW]	1.9	-	1.8	-
	Latent cooling load	[kW]	1.0		1.3	
	Surface temperature	[℃]	14.4		14.2	
Supply outlet of all-air cooling system	Temperature	[℃]	-	9.6	-	8.5
	Sensible cooling load	[kW]		9.0		4.8
	Latent cooling load	[kW]		16.8		8.4

The radiation-panels are installed in three walls (Fig. 3 (b), Case 1, 3 (Table 1)), operated below the dew-point temperature. Consequently the panels have the ability to remove the humidity of the air in the room. In the case of the all-air cooling system, cold air is supplied through the supply outlets in the ceiling (Fig. 3 (b), diffuser type, one outlet for Case 4 and two outlets for Case 2). The efficiency of the air-curtain from the ceiling is also examined. It is installed to minimize the mixing of air between the atrium and the semi-enclosed space. Half of the semi-enclosed space (3.0 m) is analyzed since the space has a symmetrical configuration. The HVAC systems are controlled so as to keep the operative temperature (i.e. thermal sensation) of the human model in the center of the room at the target value (OT = 26℃).

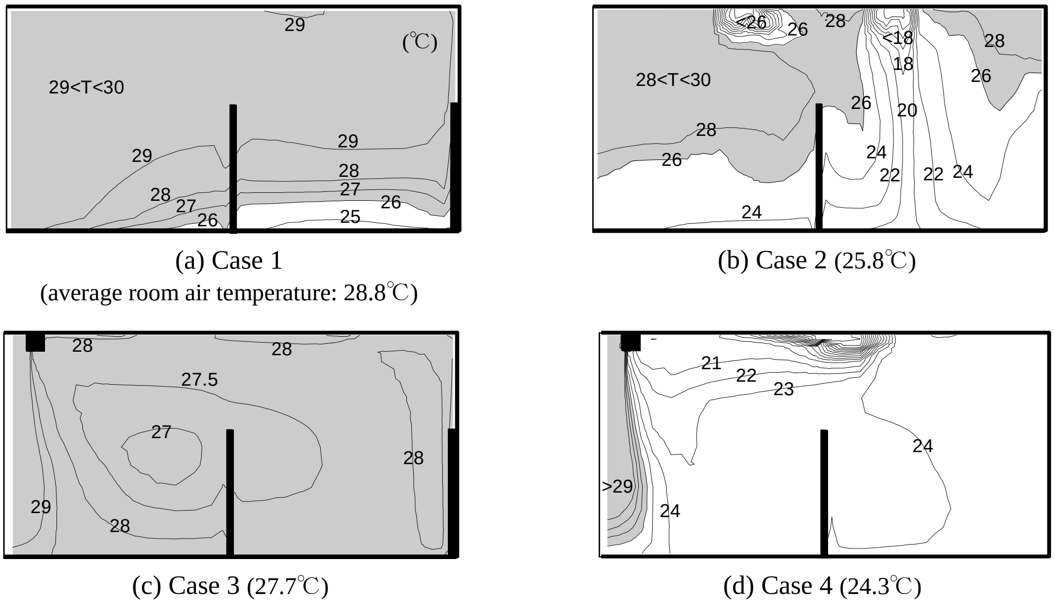
CASES ANALYZED (Table 1)

Four cases of HVAC system are analyzed in this study. The radiation-panel cooling system is used for Cases 1 and 3 and the all-air cooling system for Cases 2 and 4. In Cases 3 and 4, the air-curtain is installed at the entrance opening of the semi-enclosed space in order to minimize mixing of the air (Fig. 3). Since the temperature and relative humidity (RH) of the supply air of the air-curtain are set so as to be the same as those of the returned air itself, the air-curtain does not contribute to removing the cooling load. In case of



(a) Case 1 (b) Case 2
(c) Case 3 (d) Case 4

Fig. 4 Flow Fields (Section through center of the room)



(a) Case 1
(average room air temperature: 28.8℃)
(b) Case 2 (25.8℃)
(c) Case 3 (27.7℃)
(d) Case 4 (24.3℃)

Fig. 5 Temperature Distribution

the all-air cooling system in which the air-curtain is not installed (Case 2), the air volume from the supply outlets is increased to 1.8 times larger than that in Case 4, and two supply outlets are installed [note 4].

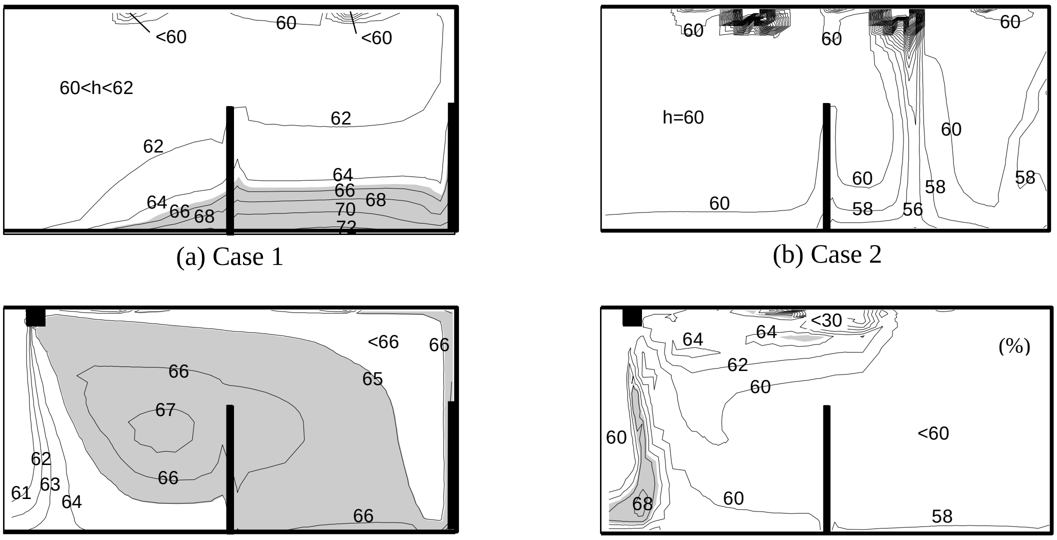
CONDITIONS FOR CALCULATION

For analyzing the air-mixing between the atrium and the semi-enclosed space, a pressure type boundary condition is applied. Detailed information about the indoor cooling loads and the CFD boundary conditions are shown in Tables 2 and 3 [note 5, 6].

RESULTS

Cooling Loads of Semi-enclosed Space (Table 4)

The sensible cooling loads of the radiation-panel cooling systems (Case 1: 1.9 kW, Case 3: 1.8 kW, see Table 4) are 60% lower than that of the all-air cooling system with the air-curtain (Case 4: 4.8 kW). In Case 2, in which the all-air cooling system with no air-curtain is applied, the sensible cooling load (9.0 kW) is five times larger than those of the radiation-panel cooling systems. The latent cooling loads for the radiation system (Case 1: 1.0 kW, Case 3: 1.3 kW) are also lower than those for the all-air cooling systems (Case 2: 16.8 kW, Case 4: 8.4 kW). Induced by the supply jet of the air-curtain, the air-mixing rate between the atrium and the semi-enclosed space in Case 3 (3370 m^3/h) becomes larger than that of Case 1 (913 m^3/h). But since the air from the atrium cannot enter deeply inside the room as it is blocked by the air-curtain, the sensible cooling load due to air-



(a) Case 1 (b) Case 2

(c) Case 3 (d) Case 4

Fig. 6 Relative Humidity Distribution

mixing in Case 3 (1.0 kW) becomes a little smaller than that of Case 1 (1.1 kW).

Flow Fields (Fig. 4)

In Case 1 (Fig. 4 (a)), the air from the atrium enters at the upper part of the entrance opening and goes out near the floor. The cold air descends near the radiation-panel due to the negative-buoyancy effect. The air-mixing rate becomes smaller than the other cases (Table 4). On the other hand, in Case 2 (Fig. 4 (b)), the total cooling load (25.15 kW) is greater than the other cases due to the large air-mixing rate. In Case 3 (Fig. 4 (c)), the descending air flow pattern near the radiation-panel observed in Case 1 disappears due to the air-circulation induced by the air-curtain. The air-mixing rate in Case 3 becomes 3.6 times as large as that in Case 1. In Case 4 (Fig. 4 (d)), the cold air from the supply outlet is mixed immediately with the room air by the effect of the air circulation. The air-mixing rate of Case 4 decreases compared to the result in Case 2. The blocking effect of the air-curtain is obvious in this case.

Temperature Fields (Fig. 5)

In Case 1 (Fig. 5 (a)), temperature stratification is formed; the temperature difference between the upper and the bottom part is about 4-5℃. The average room air temperature (28.8℃) is higher than those of other cases. But the operative temperature for the human model is the same for all cases (26.0℃). On the other hand, in Case 3 (Fig. 5 (c)), the temperature stratification disappears due to the air circulation induced by the air-curtain. The average room air temperature (27.7℃) becomes lower than that in Case 1. In Case 2 (Fig. 5 (b)) which has two supply outlets, the temperature at the inner part of the room (22-24℃) is lower than that in Case 1, due to the cold air from the supply outlet. Reflecting the lower air-mixing rate, Case 4 (Fig. 5 (d)) shows a lower average room temperature (24.3℃) than the other cases. The temperature in the room is distributed uniformly in this case.

Relative Humidity (Fig. 6)

In Case 1 (Fig. 6 (a)), the relative humidity exceeds 60% in the whole area. Because of the cold air generated by the radiation-panel, high RH is observed near the floor, and so a stratification of RH is produced. In Case 2 (Fig. 6 (b)), RH is about 60% around the human model. The lowest RH distribution is observed in Case 2 compared to the other cases, which may be attributed to the large amount of latent cooling load removed by the HVAC system (16.8 kW). The latent cooling load of Case 2 is greater than that of other cases. In Case 3 (Fig. 6 (c)), the stratification of RH, which was observed in Case 1, disappears due to the air circulation produced by the air-curtain. The RH around the human model in Case 3 is about 65%, the highest among the four cases. In Case 4 (Fig. 6 (d)), a uniform RH distribution is observed.

Distribution of Wall Surface Temperature and MRT of Human Model (Fig. 7)

MRT (24.6℃) for Case 1 is much lower than those of the all-air cooling systems (Case 2, 4), despite having a much higher average room air temperature. The average surface temperature of the radiation-panel is 14.4℃. The temperatures of the ceiling and the floor for Case 1 are lower than those of Case 2, which have large view factors between the radiation-panels so that a large amount of heat is removed by

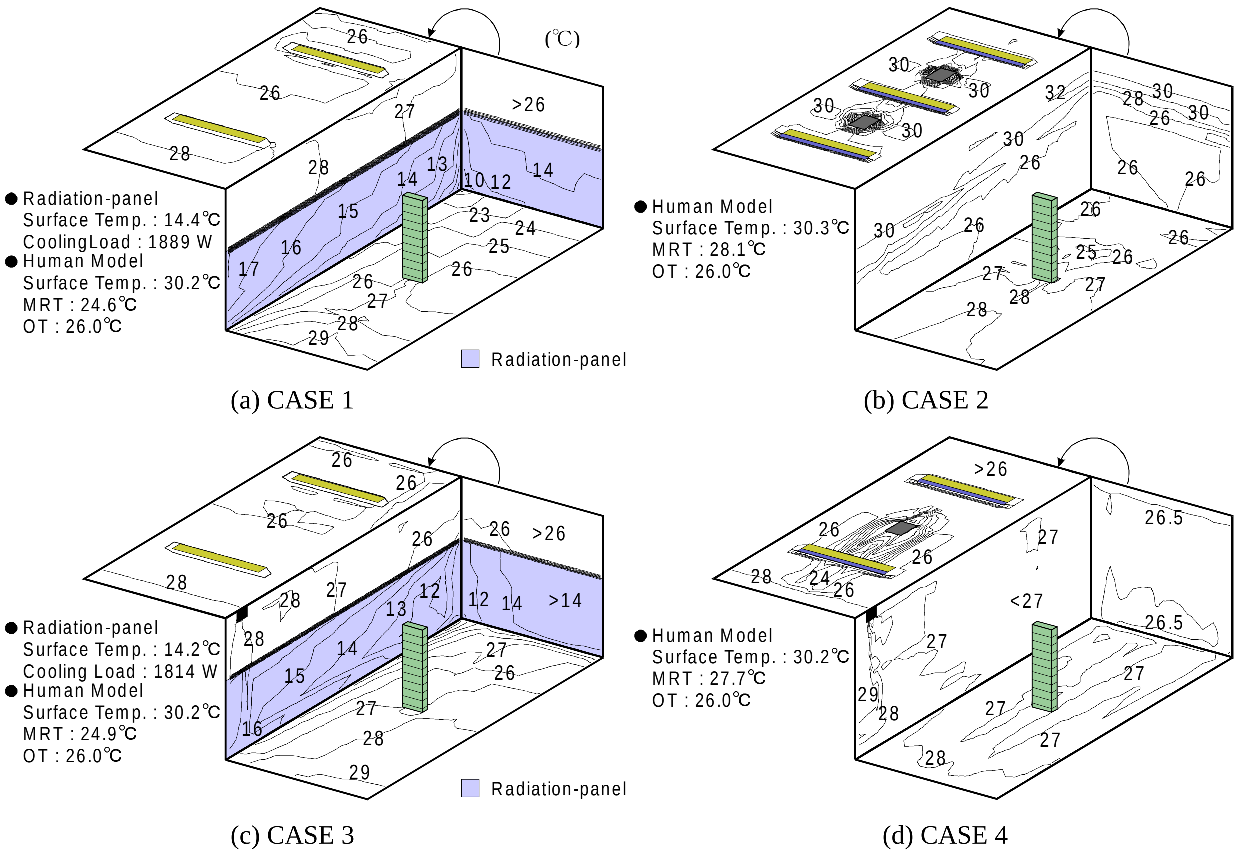


Fig. 7 Wall Temperature Distribution and MRT of Human Model

radiation. In Case 2 (Fig. 7 (b)), the temperature of the ceiling is about 30℃. On the other hand, the temperature of the floor is relatively low (20-28℃) due to the cold air from the supply outlet. MRT is 28.1℃. The difference in MRT between Cases 1 and 3 (Fig. 7 (c)) is very small (0.3℃). In Case 4 (Fig. 7 (d)), the ceiling has a lower temperature compared with Case 2. This indicates that the air-curtain effectively blocks the hot air coming from the atrium.

Characteristics of Heat Transfer in Human Model (Fig. 8)

In case of the radiation-panel systems (Case 1, 3), the amount of heat transfer by radiation from the human body is about four times larger than that by convection. Half of the total heat load from the human model (55W) is removed by the radiation-panel. But in the case of the all-air cooling systems (Case 2, 4), the amount of heat transfer by convection is two times larger than that by radiation on the contrary.

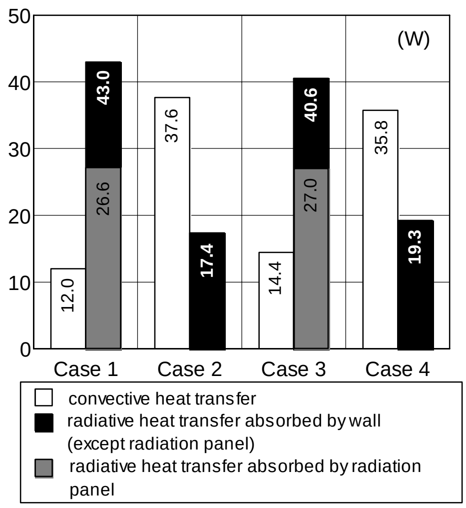


Fig. 8 Characteristics of Heat Transfer from Human Model

CONCLUSIONS

1. A new CFD technique, coupled with a radiative heat transfer simulation and HVAC control, was proposed. With this new method, the required cooling loads for attaining the same thermal sensation for occupants (OT = 26℃) were quantitatively evaluated for several HVAC systems.
2. In order to control the thermal environment of the semi-enclosed space, which opened into an atrium, the radiation-panel system was evaluated as very energy-efficient in this study.
3. In case of the all-air cooling system, the cooling load was reduced by installing an air-curtain at the entrance opening.
4. The analyses showed that the feedback design system for HVAC supported by the coupled simu-

lation is a very powerful tool for environmental design.

REFERENCES

1. S. Kato, "Application of CFD for analyzing the indoor environment (Part 7)," The Society of Heating, Air-Conditioning and Sanitary Engineers of Japan, Jan. 1998 (in Japanese)
2. Taeyeon Kim, S. Kato, and S. Murakami, "Heat Load Analysis through Coupled Simulation of Convection, Radiation and System Control," Summary of Technical Paper of The Society of Heating, Air Conditioning and Sanitary Engineering of Japan, 1998 (in Japanese)
3. Taeyeon Kim, S. Kato, and S. Murakami, "Study on Indoor Thermal Environment Controlled by Panel Cooling System with Condensation (Part 6), 3-D Coupled Simulation of Radiation, Convection, Heat Conduction and Humidity Transport within Gymnasium," Summary of Technical Paper of Architectural Institute of Japan, 1997 (in Japanese)
4. S. Murakami, S. Kato, D. Choi, H. Kobayashi, and T. Omori, "Numerical Study on Thermal Environment in Air-conditioned Room by means of Coupled Simulation of Convective and Radiative Heat Transport (Part 2)," Transactions of the Society of Heating, Air-Conditioning and Sanitary Engineers of Japan, No. 59, Oct. 1995
5. S. Murakami, S. Kato, D. Choi, H. Kobayashi, and T. Omori, "Numerical Study on Thermal Environment in Air-conditioned Room by means of Coupled Simulation of Convective and Radiative Heat Transport (Part 1)," Transactions of the Society of Heating, Air-Conditioning and Sanitary Engineers of Japan, No. 57, Feb. 1995
6. S. Murakami, S. Kato, and Jie Zeng, "Combined Simulation of Airflow, Radiation and Moisture Transport for Heat Release from Human Body," ROOMVENT '98, 1998
7. S. Murakami, "New Scales for Ventilation Efficiency and Their Application Based on Numerical Simulation of Room Airflow," ISRACE, 1992
8. Omori, T., Taniguchi, H., and Kudo, K., "Prediction method of radiant environment in a room and its application to floor heating", SHASE Trans., 42, 9-18, 1990
9. Gebhart, B., "A new method for calculating radiant exchanges", ASHRAE Transactions 65:321-332, 1959

NOTE 1 Usual HVAC systems in a room are controlled based on the air temperature at specific points of the room. But in the case where radiation has a large contribution, it is more rational to control the HVAC based on the OT at a target point.

NOTE 2 Operative Temperature (OT) for a human model is defined as:

$$OT = (\alpha_c T_a + \alpha_r T_r)/(\alpha_c + \alpha_r) \quad [℃] \quad (1)$$

α_c : convective heat transfer coefficient [W/m^2・℃]
α_r : radiative heat transfer coefficient [W/m^2 ・℃]
T_a : air temperature [℃]
T_r : radiation temperature (= MRT) [℃]

Since the convective heat transfer (q_c [W/m^2]), the radiative heat transfer (q_r [W/m^2]), the surface temperature of the human model (T_{cl} [℃]), the air temperature, and the radiation temperature are given in the process of simulation, α_c and α_r can be calculated as follows:

$$q_c = \alpha_c (T_a - T_{cl}) \quad [W/m^2] \quad (2)$$

$$q_r = \alpha_r (MRT - T_{cl}) \quad [W/m^2] \quad (3)$$

Mean Radiation Temperature (MRT) of surface i of the human model is calculated based on:

$$MRT_i = \sqrt[4]{\sum_{j=1}^{N} B_{ij} (T_j + 273)^4} - 273 \quad [℃] \quad (4)$$

B_{ij} : Gebhart absorption coefficient view from surface i to j
T_j : temperature of surface j [℃]
N : total number of surfaces

NOTE 3 In the present simulation, the latent heat removed by the radiation-panel is not controlled by the HVAC control system directly. In the case where the surface temperature of the radiation-panel becomes lower than the dew point temperature of the air near the radiation-panel, the absolute humidity (AH), which corresponds to the saturated vapor pressure of the surface of the radiation-panel, is given to the air on the surface. In other cases, it is assumed that no latent heat is removed by the radiation-panel. This means that the gradient of AH is set to be 0 on the radiation-panel.

NOTE 4 In this paper, the operative temperature that does not consider the effects of humidity is used to evaluate the thermal sensation of occupants. To compare all cases under the same thermal sensation, the RH around the human model must be made the same for all cases. In the case of the all-air cooling system, the RH at the supply outlets is fixed at 0%. 0% RH of supply air is, of course, impractical. However, with this condition, RH around the human model can be kept at about 60% in Cases 2 and 4, which is similar to that for Cases 1 and 3.

NOTE 5 A work-station of 9.24-SPECint95 and 5.75-SPECfp95 is used for this simulation. About 30,000 iterations (220 hours in CPU time) were required for the steady-state solution.

NOTE 6 In this simulation, the human thermal sensation is evaluated by operative temperature as the first step of the study; the influence of humidity (water vapor pressure) for the thermal sensation is not considered. In order to evaluate the human thermal sensation more precisely, such the effect will be included in the next step.