SIMULATION MACRO MODEL FOR PREDICTING MIXING ENERGY LOSS

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ABSTRACT

The purpose of this study is to develop the technique that mixing energy loss is predicted at the time of the air conditioning system design by the simulation. If air conditioning load, which contains mixing energy loss, can be calculated easily at the time of air conditioning system design, we can design most suitable air conditioning system which mixing energy loss doesn't occur in. In this paper we would like to examine simulation macro model which has following characteristics.

(1) The room is divided into more than one zone in the vertical and horizontal direction.

(2) Heat balance equations of the wall surface and the room air are calculated based on general idea of successive state transition.

(3) Airflow between each zone is calculated by the calculation method of ventilation.

INTRODUCTION

It is general to separate an air conditioning system corresponding to the heat load such as perimeter zone and the interior zone at the time of the air conditioning system design. In interior zone, cooling load occurs all the year round. In perimeter zone, cooling load occurs in summer though heating load occurs in winter. Thus one room is cooled and heated simultaneously in winter and mixing energy loss often occurs because of mixing of cooled and heated jets [1, 2].

The previous studies include ours that present prevention method of mixing energy loss about the typical air conditioning system by experiment and simulation [3,4,5]. We also examined prevention method of mixing energy loss by measurements and simulation. The conclusions of these studies are that mixing energy loss is the phenomenon which indoor air temperature distribution, airflow and air conditioner control influence complicatedly and which happens. And it is important for prevention of mixing energy loss to plan to make air jets of air conditioners and air conditioning control most suitable.

On simulation macro-model of our past study, a space is divided into more than one of zone in the vertical and the horizontal direction. And heat exchange and airflow between each zone are being taken into consideration. But this macro model has following problems.

(1) Airflow between each zone is considered, but that is not based on calculation method of ventilation.

(2) Diffusion of air jets from air conditioners to each zone is decided by heat distribution coefficients that are assumed according to operation of the air conditioner. Heat distribution coefficients are the values found from the actual measurement data, and they are characteristic values under the actual measurement condition.

Consequently application of this simulation program was limited under only comparable conditions to the measurements. The simulation method that mixing energy loss is predicted on the optional conditions should be developed.

In this paper we would like to examine macro model which considered airflow between zones using the calculation method of ventilation. And we show the air conditioning load calculation results, which contains the mixing energy loss prediction about some cases of air conditioning system. Whereas we recognize that air conditioning control influences mixing energy loss strongly, we are not concerned here with that.

MACRO MODEL

It is necessary for the simulation of mixing energy loss that the temperature distirbution of the depth direction and the vertical direction of the simulation

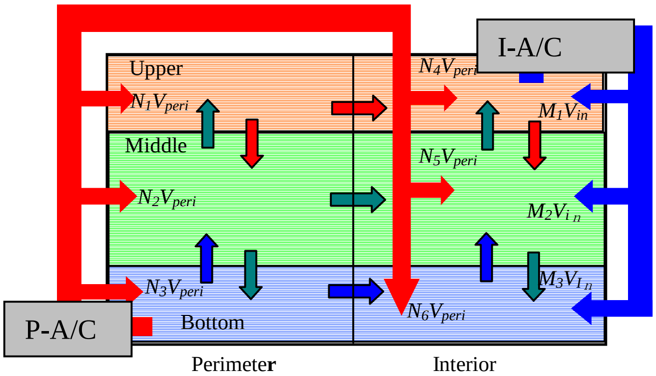

Fig. 1 Macro model (6 zones model)

$$\sum_{j=1}^{N} \rho_{i,j,n} V_{i,j,n} + N_i \rho_{peri,n} V_{peri,n} + M_i \rho_{in,n} V_{in,n} = 0 \quad (1)$$

$$V_{i,j,n} = A_{i,j} \sqrt{2\Delta p_{i,j,n} / (c\rho_{j,n})} \quad (2)$$

$$\Delta p_{i,j,n} = p_{j,n} - p_{i,n} + gh(\rho_{j,n} - \rho_{i,n}) \quad (3)$$

room model can be taken into consideration. Because the simulation of mixing energy loss which caused by airflow in the horizontal direction can't be done if the room model takes only the temperature distribution of the vertical direction into consideration. In this simulation the room is divided into the perimeter and the interior zone. And both of the zones are further divided into three zones of the upper, middle and bottom. Therefore, the room is devided Into 6 zones.

AIR BALANCE BETWEEN THE ZONES

Airflow between the zones in the 6 zones model is reperesented in the **Fig. 1**. Then the airflow balance between the zones is shown in the **equation (1)-(3)**. It is decided fundamentally in this model that there is airflow only with the neighboring zone through the boundary surface of the zone. As for the airflow direction between the zone of this model, though the airflow of the horizontal direction is only 1 direction, the airflow of the vertical direction is mutual direction.

The airflow of the horizontal direction is temperature difference ventilation of the zones. The pressure difference of two zones is shown the **equation (3)**. The volume of airflow is calculated based on the **equation(2)** from the pressure difference between zones. As the mention above, the airflow of the horizontal direction is only 1 deirction.

The airflow of the vertical direction is gravitational ventilation fundamentally. But, the airflow in the vertical direction ventilation between the zones is mutual direction. Moreover, the airflow of the

$$T_{i,n} / q_{i,n} - \sum_k \alpha_{c,k,n} S_{k,n} T_{k,n}$$
$$- \sum_l c\rho_{l,n} V_{l,n} T_{l,n} - \sum_j c\rho_{j,n} V_{j,n} T_{j,n}$$
$$= TA_{i,n-1} + c\rho_{o,n} V_{o,n} T_{o,n} + CH_{i,n}$$
$$+ N_{i,n} c\rho_{peri,n} V_{peri,n} T_{peri,n}$$
$$+ M_{i,n} c\rho_{in,n} V_{in,n} T_{in,n} \quad (4)$$

$$TA_{i,n-1} = \Phi_{i,n} T_{i,n-1} / q_{i,n}$$
$$+ P_{i,n} / q_{i,n} (c\rho_{o,n} V_{o,n} T_{o,n-1} + \sum_l c\rho_{l,n} V_{l,n} T_{l,n-1}$$
$$+ \sum_j c\rho_{j,n} V_{j,n} T_{j,n-1} + CH_{i,n-1}$$
$$+ N_{i,n} c\rho_{peri,n} V_{peri,n} T_{peri,n-1} + M_{i,n} c\rho_{in,n} V_{in,n} T_{in,n-1}$$
$$+ \sum_j \alpha_{c,k,n} S_{k,n} T_{k,n-1}) \quad (5)$$

$$\Phi_{i,n} = \exp\left(- B_{i,n} \Delta t / Q_{i,n} \right) \quad (6)$$

$$B_{i,n} = \sum_k \alpha_{c,k,n} S_{k,n} + \sum_l c\rho_{l,n} V_{l,n}$$
$$+ \sum_j c\rho_{j,n} V_{j,n} + c\rho_{o,n} V_{o,n} + N_{i,n} c\rho_{peri,n} V_{peri,n}$$
$$+ M_{i,n} c\rho_{in,n} V_{in,n} \quad (7)$$

$$P_{i,n} = -\left(\Phi_{i,n} - Q_{i,n} \frac{1 - \Phi_{i,n}}{B_{i,n} \Delta t} \right) \Big/ B_{i,n} \quad (8)$$

$$q_{i,n} = -P_{i,n} + \frac{1 - \Phi_{i,n}}{B_{i,n}} \quad (9)$$

vertical direction is calculated by adding diffused air from the air conditioners based on the heat distribution coefficient to the airflow of the gravitational ventilation.

HEAT BALANCE BETWEEN THE ZONES

Heat balance of the wall surface and the room air are calculated based on the general idea of successive state transition [6]. The successive state transition formula of the heat balance of the air used in this paper is shown in **equation(4)-(9)**.

Unknown terms are on left hand side of **Equation (4)**. The second term on the left hand side of **Equation (4)** is the convection heat transfer with the wall surface *k* in the zone. The third term on the left hand side of **Equation (4)** is the draft from the neighboring room which it flows from the wall *l* inside the zone. The fourth term on the left hand side of **Equation (4)** is the airflow from the neighboring zone *j*. Known terms are on the right hand of **Equation (4)**.

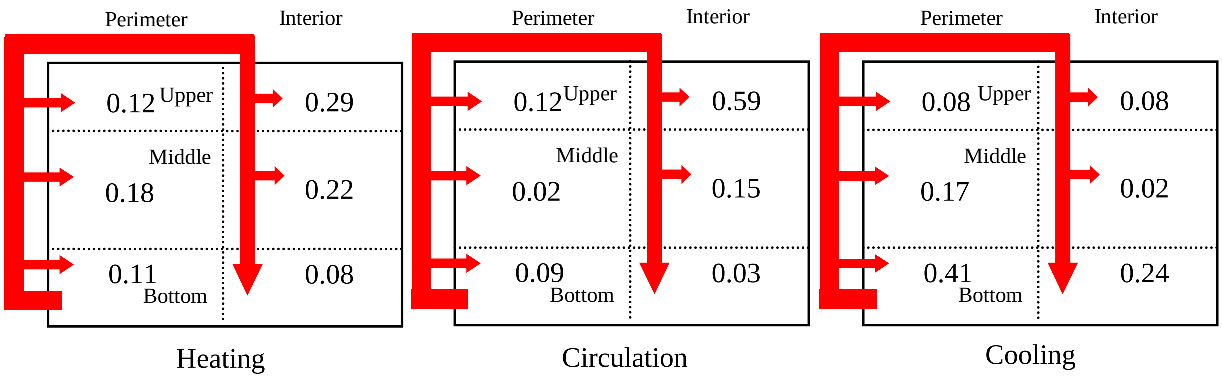

Fig. 3 Heat distribution coefficient of perimeter airconditoner

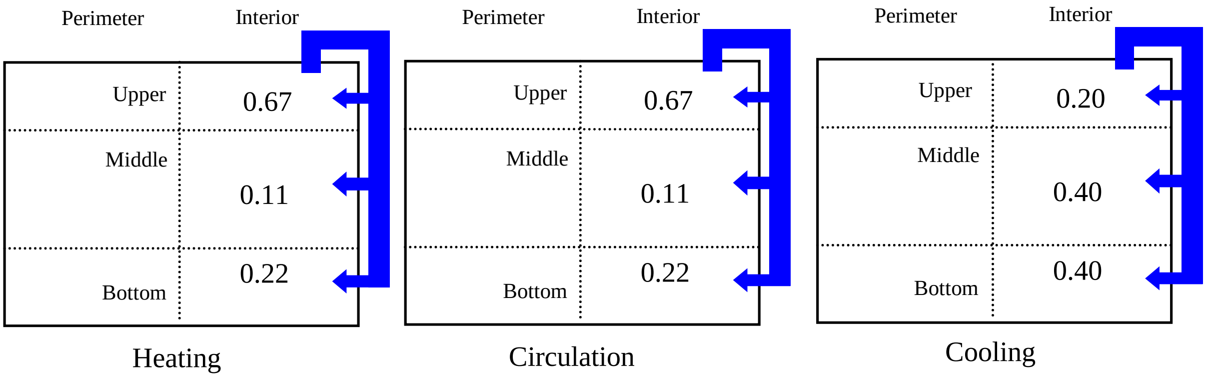

Fig. 4 Heat distribution coefficient of interior airconditoner

HEAT DISTRIBUTION COEFFICIENT

As for the actual airflow field under the air conditioning, the air conditioner blowoff air passes through some zones, and flows into the zones where they left the air conditioner outlet. And, because it is supposed that the air and the heat which flowed into the zone diffuse momentarily in the zone, it is greatly different from the actual airflow field if the distribution of the heat and air isn't decided suitably in this macro model. In this paper we deal with the distribution to each zone of the air conditioner supply quantity of heat in accordance with the heat distribution coefficient of Ishida and Udagawa [7]. And heat distribution coefficient in this simulation is the value estimated from the actual measurement data of the office building.

As for the airjet from the perimeter air conditioner, it was found out that it reached until not only each zone of the perimeter but also interior deep zone by the analysis of the actual measurement data. The quantity of heat supplied by perimeter air conditioner was decided to be distributed in the whole zone of the room. On the other hand, because interior was deep in comparison with the perimeter depth, the supply quantity of heat by the interior air conditioner was never decided to be distributed directly in each zone of the perimeter.

In this paper, the heat distribution coefficient was estimated based on the actual measurement detailed

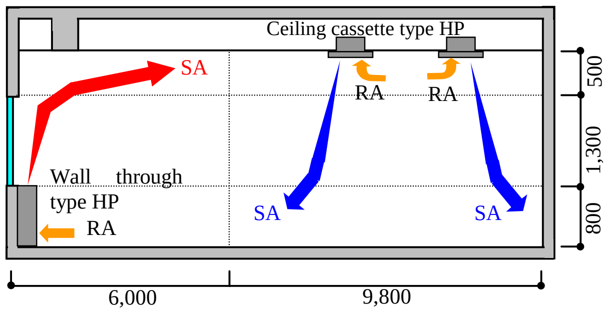

Fig. 2 Air-conditioning system

data of indoor thermal environment. The air conditioning system are represented in the **Fig. 2**. As a result of analyzing room air temperature distribution under the air conditioning based on the actual measurement data, it was found out that there were the heat boundary of perimeter and interior between 4m and 8m of the perimeter depth. Therefore a perimeter and interior were divided in the position of 6m of the intermediate point of the perimeter depth 4m and 8m in this simulation. Moreover the space of vertical direction was divided by 3 corresponding to the height of the wall, the window, peri-counter as shown in the **Fig. 2**.

Fig. 3 and **Fig. 4** shows the heat distribution coefficient of the perimeter and the interior air conditioner. Each Fig. shows the heat distribution coefficient by three air conditioner operation appearance of heating, cooling and circulation . The heat distribution coefficient which showed it in the **Fig. 3** and **Fig. 4** are the mean of the heat distribution coefficient of each estimated time. But as for the heat distribution coefficient in interior heating, it was made the value which was the same as the heat distribution coefficient in the circulation with the interior, because there was no heating in interior at the time of the actual measurement. Moreover, the heat distribution coefficient was greatly different from the value when a perimeter air conditioner start up and other heating times. Therefore, when air conditioner operation appearance in the start up was heating, it was decided that the heat distribution coefficient of the perimeter air conditioner used additional value with others. And the total value of the heat distribution coefficient is 1.0 in each air conditioner.

METHOD OF SIMULATION

In this simulation, the room temperature fluctuation of each zone and the heat extraction rate are calculateted from input items such as the standard weather data , the building specification and air conditoner setting conditions. Though it was calculated by the six zones model at the air conditioning time, it was calculated by the two zones model that one room was divided into the perimeter and the interior zone at the non-air conditing time. Indoor air temperature gradient isn't taken into consideration in this two zones model. The control action of the air conditioner in this simulation is on/off control which was the same as that in the actual measurement. We examine the influence which outer wall directions and the air temperature sensor position give to the mixing energy loss by this simulation.

SIMULATION CONDITIONS

The criterion calculation conditions appear in the **Table 1**. A calculation period is one year. The calculation term is 10 minutes. The internal heat generation is divided into the radiation component with the convection component as shown in the **Table 1**. The wall composition appear in the **Table 2**. We calculate the mixing energy loss(*ML*) by using the **equation(10)**. The quantity of heat transfer from the zone is calculated from the air temperature difference between the case of heat transfer to other zone and the case of no heat transfer. If the heating and cooling simultaneously, the total sum of the absolute value is considered the whole mixing energy loss. And, when there is no heat transfer from the zone to other zones, the supply heat from the perimeter air conditioner is distributed to only the perimeter zone. As for the interior as well, it is the same. In this case it was decided that the heat distribution coefficient to each zone depended on the capacity ratio of the zones.

Table 1 Calculation conditons

Weather data	Standard weather data Fukuoka Japan
Air-conditioning time	Mon. – Fri. 8:00–18:00
Internal heat generation (Convection)	Top : 7.5W/m^2 Middle: 10.6W/m^2 Bottom: 4.6W/m^2
Internal heat generation (Radiation)	Ceiling : 10.2W/m^2 Floor: 8.8W/m^2
Setting temperature	24℃
Sensor position	Perimeter: bottom Interior: middle
Diffused air volume	Perimeter: 1330m^3/h Interior: 800m^3/h

Table 2 Wall composition

	Material	Thickness
Outer wall	Steel	1mm
	Air space	-
	Polystyrene form	10mm
	PC concrete	150mm
	Tile	10mm
Floor	Carpet	4mm
	Plastic plate	12mm
	Air space	-
	PC concrete	175mm
Ceiling	Rock wool	12mm
	Gypsum	9mm
Window	Heat absorbing glass	10mm

$$H_{i,n} = V_i C\rho_{i,n} T_{i,n} - C\rho_{i',n} T_{i',n}$$

$$ML_n = \sum_i |H_{i,n}| \qquad (10)$$

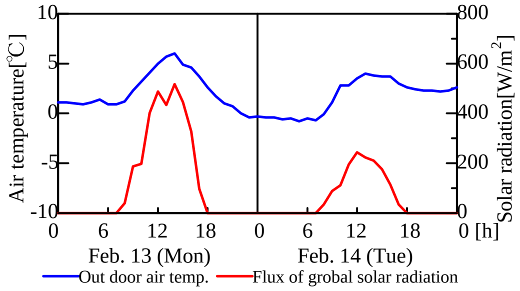

Fig. 5 Weather data

RESULTS BY OUTER WALL AZIMUTH

Firstly, we examined the mixing energy loss by the outer wall azimuth. As for the position of the air temperature sensor, perimeter sensor is in the bottom zone, interior sensor is in the middle zone. In this paper we show the calculation results on February 13 and 14 in all calculation period. The outdoor air

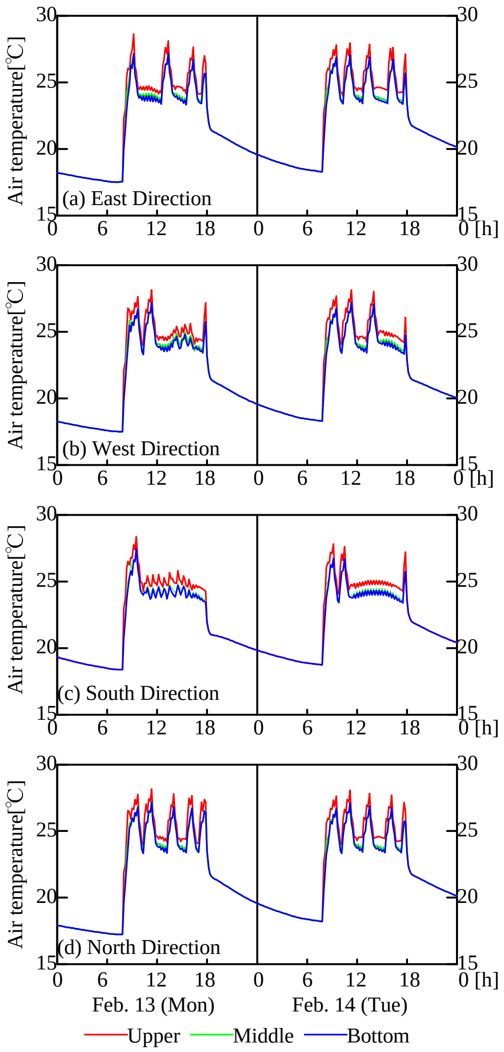

Fig. 6 Zone air temperature by outer wall directions

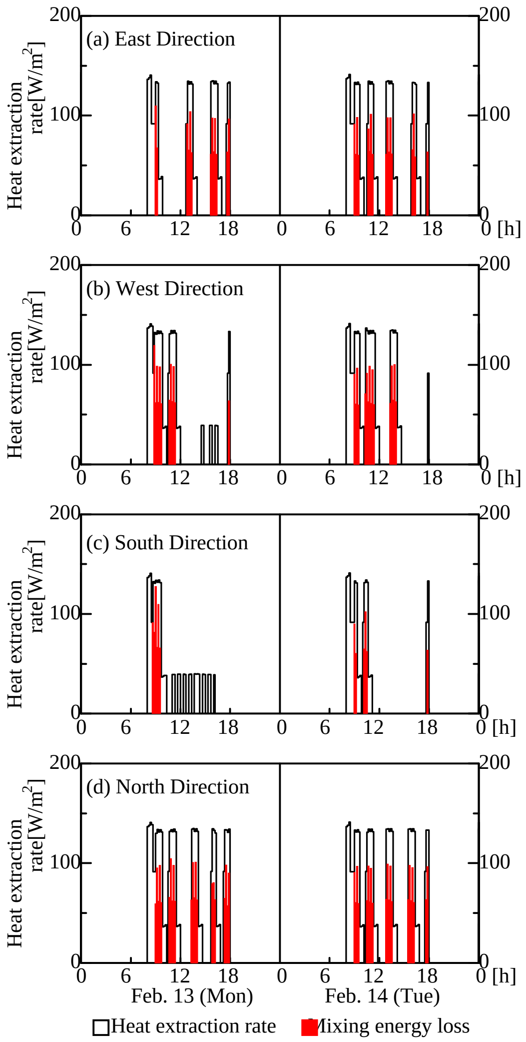

Fig. 7 Mixing energy loss by outer wall directions

temperature and flux of grobal solar radiation are plotted in the **Fig. 5**. The amount of the mixing energy loss(ML) and mixing energy loss ratio(MLR) in February 13, 14 and the amount of the mixing energy loss ratio(MLR) in February by the outer wall azimuth appear in the **Table 3**. Each zone air temperature fluctuation on February 13 and 14 are plotted in the **Fig. 6**. And the fluctuation of the heat extraction rate and mixing energy loss are plotted in the **Fig. 7**. And, as for the mixing energy loss ratio, it is the rate which the quantity of mixing energy loss occupies in the heat extraction rate.

When the air conditioners started, all air conditioners became heating operation by the influence of thermal storage load at all the azimuth. In the case that the outer wall azimuth is for the north, there is most heat extraction rate. In the case of the outer wall for the south, mixing energy loss occurs most low.

MLR in February is about 30%. Moreover, there is most the mixing energy loss and MLR in the case that outer wall azimuth is for the north. The long heating

Table 3 Mixing energy loss by outer wall directions

Direction		ML	HER	MLR
East	2/13	0.668	1.516	44%
	2/14	0.668	1.894	35%
	Feb.	5.098	16.271	31%
West	2/13	0.492	1.484	33%
	2/14	0.637	1.676	38%
	Feb.	6.136	19.564	31%
South	2/13	0.280	1.096	26%
	2/14	0.229	0.976	23%
	Feb.	3.721	17.482	21%
North	2/13	0.874	2.259	39%
	2/14	0.830	2.122	39%
	Feb.	7.417	20.493	36%

ML: Mixing energy Loss [MJ/day] or [MJ/month]
HER: Heat Extraction Rate [MJ/day] or [MJ/month]
MLR: Mixing energy Loss Ratio

Table 4 Mixing energy loss by sensor position

Perimeter	Interior	ML	HER	MLR
Upper	Upper	2.066	8.225	25%
	Middle	45.437	59.174	77%
	Bottom	47.568	61.727	77%
Middle	Upper	4.100	11.792	35%
	Middle	2.392	11.370	21%
	Bottom	6.876	15.413	45%
Bottom	Upper	12.095	25.245	48%
	Middle	7.417	20.493	36%
	Bottom	4.874	16.011	30%

ML: Mixing energy Loss [MJ/day]
HER: Heat Extraction Rate [MJ/day]
MLR: Mixing energy Loss Ratio

time in the north perimeter ought to be the cause of most mixing energy loss.

In the case that the outer wall azimuth is for the east, heating and cooling simultaneity all day long, and mixing energy loss occurs. Specially the perimeter is heated from the air conditioner start to 10 a.m. and after 15 p.m. when solar radiation to the outer wall decreases. Therefore, a mixing energy loss occurs easily in these heating times.

In the case that the outer wall azimuth is for the west, mixing energy loss occurs between the heated perimeter zone and the cooled interior zone in the morning. But, mixing energy loss is often dissolved because a perimeter air conditioner becomes circulation operation in the afternoon by the solar radiation. And mixing energy loss is hard to occur even in when solar radiation is small, because the perimeter air conditioner becomes circulation operation if outdoor air temperature becomes high.

In the case that the outer wall azimuth is for the south, mixing energy loss occurs low in comparison with other azimuth, because the heating load of the perimeter is small by the solar radiation.

RESULTS BY SENSOR POSITION

The position of the air temperature sensor was examined about 9 cases with three sensor positions of combinations of the upper, middle and bottom with the perimeter and the interior. In this simulation the outer wall azimuth is for the north that most mixing losses occur. The amount of the mixing energy loss and MLR in February appear in the **Table 4**.

Mixing energy loss occurs lowest in the case of the combination of the both perimetr and interior sensor in the middle zone. And most mixing energy loss occurs in the case of the combination of the perimetr

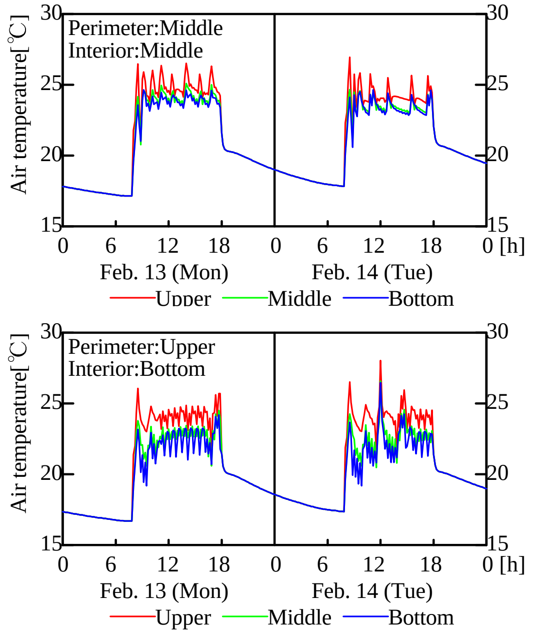

Fig. 8 Zone air temperature by sensor position

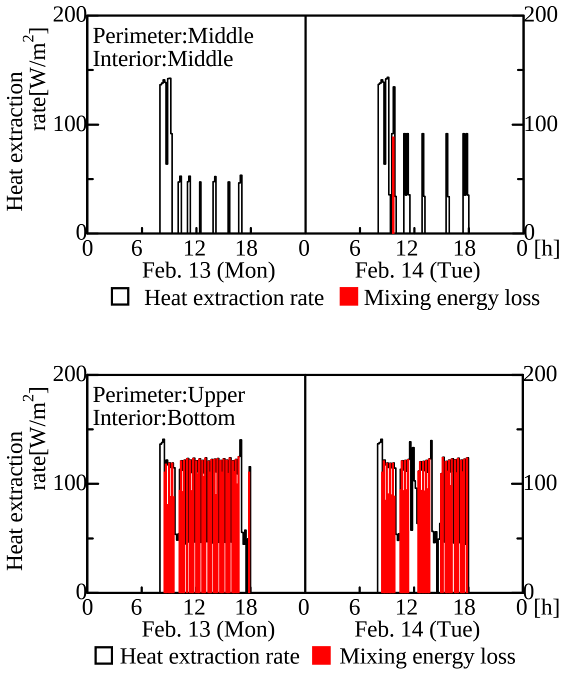

Fig. 9 Heat extraction rate and ML by sensor position

sensor in the upper zone and interior sensor in the middle or bottom zone.

Each zone air temperature on February 13 and 14 are plotted in the **Fig. 8**. And heat extraction rate and mixing energy loss on February 13 and 14 are shown in the **Fig. 9**. When both the perimeter and the interior sensor are middle zones, the temperature gradient is small. On the other hand, when the perimeter sensor in the upper zone and the interior sensor in the middle or bottom zone,the temperature gradient is big. And the room is cooling and heating simultaneously and mixing energy loss occurs all day long in these cases.

CONCLUSIONS

We examined macro model which considered airflow between zones using the calculation method of ventilation. Though it couldn't be inserted in detail in this paper, it learned to get the same results as the calculation results of the past study at a calculation speed of several times. A left subject is to calcluate the diffusion of air jets from air conditioners using the calculation method of ventilation.

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NOMENCLATURE

ρ Density of air [kg/ m^3]

V Airflow between blocks [m^3/h]

V_{peri} Air jet from perimeter airconditioner [m^3/h]

V_{in} Air jet from interior airconditioner [m^3/h]

Δp Pressure difference between zones [Pa]

A Boundary area between zones [m^2]

c Surface pressure coefficient $c = 1$ [−]

p_i Static pressure of i zone floor surface [Pa]

p_j Static pressure of j zone floor surface [Pa]

g Acceleration due to gravity 9.80665 [m/ s^2]

h Height of boundary area between zones [m]

T Temperature [K]

S Area of wall, floor, ceiling and window surface [m^2]

α_c Convective heat transfer coefficient [W/ m^2/K]

CH Internal heat generation [W]

N Heat distribution coefficient of perimeter air conditoner [−]

M Heat distribution coefficient of interior air conditoner [−]

Δt Calculation term [h]

B Coefficient of zone heat loss [W/K]

Q Thermal capacity of zone air [J/K]

H Heat loss from zone air [J]

C Specific heat of air [J/kg/K]

Subscripts

i Zone air

i' Zone air (no inflow of neighboring zone air)

j Neighboring zone air

k Surface of wall, floor, ceiling and window

l Neighboring room air

o Outdoor air

n Time

$peri$ Perimeter air conditioner

in Interior air conditioner