EFFECT OF HEAT DISCHARGE BY NATURAL VENTILATION ON INDOOR ENVIRONMENT AND HEAT REMOVAL STRUCTURE

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ABSTRACT

At present, for energy conservation, conscious utilization of natural ventilation is encouraged. However, the relationship between heat discharge through natural ventilation and heat loads has not been clarified yet. Accordingly, in this study, the effect of heat discharge through natural ventilation on indoor environments and heat removal structure is examined using a simulation model that can estimate the occupant thermal control behaviour and the characteristics of air conditioners. The results suggest that the use of natural ventilation is effective not only for reducing of cooling capacity, but also for improving energy efficiency of air conditioners.

INTRODUCTION

In Japan, natural ventilation through large openings has traditionally been used for cooling in summer and medium seasons. At present, as global warming has become a serious problem, conscious utilization of natural ventilation is encouraged for energy conservation. In fact, the new Japanese standard enforced in 2009 first adopts a quantitative method for evaluating the energy conservation effectiveness of natural ventilation. This indicates that natural ventilation through large openings is expected to be an important technique for energy conservation.

To verify the effectiveness of natural ventilation for energy conservation, Habara et al. (2009) conducted experiments by simulating occupant behaviour in a multi-family residential building using reinforced concrete. Their experimental results indicate that although heat discharge through natural ventilation has positive effects on the reduction of cooling hours, thermal energy storage during the absence of occupants in the daytime could cause additional air conditioner usage in the evening and night. However, they did not clarify the relationship between the heat discharge through natural ventilation and the heat loads caused by inner heat emission, heat/radiation transfer and infiltration. This is because it was difficult to completely measure the heat loads in the experiment.

Accordingly, this study discusses the effect of heat discharge through natural ventilation on indoor environments and heat removal structure by using a simulation model that can estimate occupant thermal control behaviour and the characteristics of air conditioners.

SIMULATION MODEL

The simulation model (Habara et al., 2007) consists of a thermal model, a radiation model, a ventilation model, an air conditioner model and a thermal control behaviour model. The flowchart of the simulation model is shown in Figure 1 and the details of each component model are described below.

Thermal model

The room air temperature and the absolute humidity are calculated using a heat balance equation under the assumption of perfect mixing of air. The equation for sensible heat is as follows:

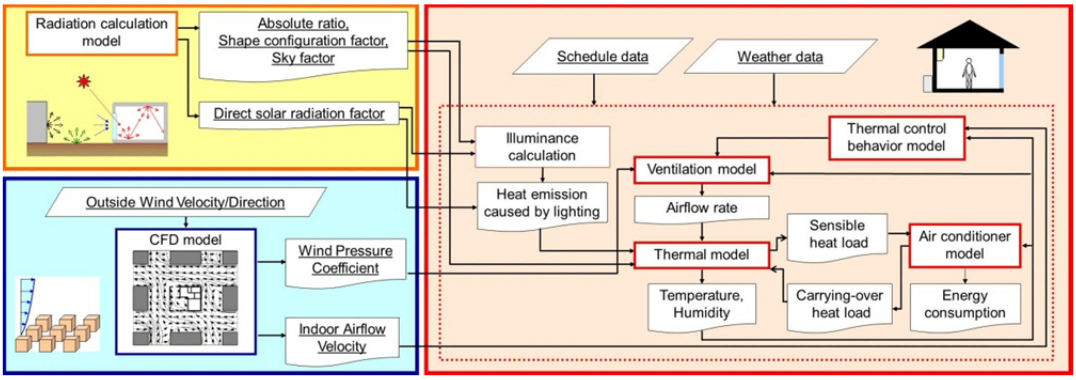


Figure 1 Flowchart of the simulation model

$$C_i V_i \frac{d\theta_i}{dt} = \sum_{j=1}^{M} \alpha_{c,j} \left(\theta_{w,j} - \theta_i\right) A_j + \rho C_p Q_{ai} \theta_a - \rho C_p Q_{ia} \theta_i$$

$$\sum_{k=1}^{n} \rho C_p Q_{ki} \theta_k - \sum_{k=1}^{n} \rho C_p Q_{ik} \theta_i + Q_{S,i} + Q_{s,i}, \quad (1)$$

where C_i is the thermal capacity of a room [kJ/(m^3•K)], V_i is the volume of room air [m^3], $d\theta_i / dt$ is the time derivative of room air temperature, M is the number of wall surfaces, $\alpha_{c,j}$ is the heat conductance of the wall surface j [W/(m^2•K)], $\theta_{w,j}$ is the surface temperature of the wall j [^{o}C], θ_i is the air temperature of the room i [^{o}C], θ_a is the outside air temperature i [^{o}C], A_j is the area of the wall surface j [m^2], ρ is the air density [kg/m^3], C_p is the specific heat at constant pressure [kJ/(kg•K)], Q_{ij} is the airflow from space i to space j [m^3/s], $Q_{S,i}$ is the internal heat emission from human bodies and electric appliances [W], and $Q_{s,i}$ is the sensible heat load [W]. The equation for latent heat is given by:

$$G_i V_i \frac{dX_i}{dt} = \rho\gamma Q_{ai} X_a - \rho\gamma Q_{ia} X_i$$

$$+ \sum_{k=1}^{n} \rho\gamma Q_{ki} X_k - \sum_{k=1}^{n} \rho\gamma Q_{ik} X_i + Q_{L,i} + Q_{l,i}, \quad (2)$$

where G_i is the moisture capacity [kJ/(g/kg(DA))], dX_i / dt is the time derivative of room absolute humidity, γ is the heat of vaporization [J/kg], X_i is the absolute humidity of the room i [g/kg(DA)], X_a is the outside absolute humidity [g/kg(DA)], $Q_{S,i}$ is the internal moisture emission from human bodies and electric appliances [W], and $Q_{s,i}$ is the latent heat load [W]. The thermal and the moisture capacities of room air are set to 12.6 kJ/(m^3•K) and 25.1 kJ/(g/kg(DA)) respectively, which include those of furniture. The airflow Q_{ij} is calculated using the ventilation model.

Radiation model

The radiate heat transfer is calculated, by considering external surroundings, shadings and other impediments. Direct solar radiation is assumed to diffuse perfectly on the building surface and the ground. The proportion of specular reflection to diffuse reflection is 1:9 on the inner surface of window glasses and 1:1 on that of walls. In addition, the exchange of long-wave radiation and diffusely reflected solar radiation between inner surfaces is simulated by the Gebhart absorption coefficient method (Gebhart et al., 1959).

Ventilation model

The airflow through large openings and cracks is simulated using an airflow network model with the pressure assuming calculation method. The characteristics of large openings and cracks are respectively described using the following equations:

$$Q = 3600 \alpha A \sqrt{\frac{2}{\rho} \Delta p}, \quad (3)$$

$$Q = a \Delta p^{\frac{1}{n}}, \quad (4)$$

Where Q is the airflow rate [m^3/h], α is the discharge coefficient, A is the opening area [m^2], ρ is the air density [m^2], Δp is the pressure difference [Pa], a is the air leakage coefficient [m/(s•Pa^n)], and n is the pressure exponent.

Air conditioner model

The energy consumption and the coefficient of performance (COP) are estimated with experimental formulas (Hosoi et al., 2010), some of which are incorporated into the new Japanese standard enforced in 2009. The normalized energy consumption (Pr_c) is represented by the energy consumption (P_c [W]) and the rated energy consumption ($P_{c,rtd}$ [W]) as follows:

$$Pr_c = \frac{P_c}{P_{c,rtd}} = f_{c,\theta}\left(qr_c'\right), \quad (5)$$

$f_{c,\theta}\left(qr_c'\right)$ is a function of the outside temperature (θ [^{o}C]) and the modified load rate (qr_c'):

$$f_{c,\theta}(qr_c') = a_3 qr_c'^3 + a_2 qr_c'^2 + a_1 qr_c' + a_0, \quad (6)$$

$$a_3 = 0.0148\theta + 0.0089, \quad (7)$$

$$a_2 = -0.0153\theta + 0.1429, \quad (8)$$

$$a_1 = 0.034\theta - 0.4963, \quad (9)$$

$$a_0 = -0.0012\theta + 0.288 + 0.0322, \quad (10)$$

where qr_c' is obtained by modifying the load rate (qr_c) to offset the differences of measurement conditions of the exhaust airflow and the intake air humidity between the latent load ($L_{c,lat}$ [W]) and the rated cooling capacity ($Q_{c,rtd}$ [W]).

$$qr_c' = qr_c \times \frac{1}{\frac{C_{af,c} + C_{hm}}{2}} = \frac{L_{c,sen}}{Q_{c,rtd}} \times \frac{1}{\frac{C_{af,c} + C_{hm}}{2}}, \quad (11)$$

where $C_{af,c}$ (=0.85) and C_{hm} (=1.15) are the correction coefficients for the exhaust airflow and the intake air humidity, respectively.

The latent load ($L_{c,lat}$ [W]) is calculated using the sensible load ($L_{c,sen}$ [W]) in the thermal model and the sensible heat factor (SHF) estimated from the following experimental formulas (Hosoi et al., 2010):

$$L_{c,lat} = L_{c,sen} \times \left(\frac{1}{SHF} - 1\right), \quad (12)$$

$$\begin{cases} SHF = 1.0 \quad \left(R_h < 0.385\right), & (13) \\ SHF = 1.1774 R_h^2 - 2.9042 R_h + 1.9427 & \\ \quad \left(0.385 \le R_h \le 0.9\right), & (14) \\ SHF = 0.28 \quad \left(0.9 < R_h\right), & (15) \end{cases}$$

The cooling capacity is determined from the balance between the total heat load ($L_{c,sen}+L_{c,lat}$ [W]) and the maximum cooling capacity of the air conditioner ($Q_{c,max}$ [W]).

$$Q_c = Q_{c,sen} + Q_{c,lat}\,, \tag{16}$$

$$Q_{c,sen} = \begin{cases} SHF \times Q_{c,max} & \left(L_{c,sen}+L_{c,lat} > Q_{c,max}\right), \quad (17) \\ L_{c,sen} & \left(L_{c,sen}+L_{c,lat} \leq Q_{c,max}\right), \quad (18) \end{cases}$$

$$Q_{c,lat} = \begin{cases} (1\text{-}SHF) \times Q_{c,max} & \left(L_{c,sen}+L_{c,lat} > Q_{c,max}\right), \quad (19) \\ L_{c,lat} & \left(L_{c,sen}+L_{c,lat} \leq Q_{c,max}\right), \quad (20) \end{cases}$$

If the total heat load is greater than the maximum cooling capacity of the air conditioner, room temperature does not reach the preset temperature of the air conditioner and the surplus heat load is carried over to the next time step. The maximum cooling capacity of the air conditioner is represented by the following experimental formula:

$$\begin{aligned} Q_{c,max} = & -1\times10^{-5}\times r_c \times(\theta\text{-}35)^3 \\ & +2\times10^{-4}\times(0.5+0.5\times r_c)\times(\theta\text{-}35)^2 \\ & -(0.0147+0.014\times(r_c\text{-}1))\times(\theta\text{-}35)+r_c\,, \end{aligned} \tag{21}$$

where r_c is the ratio of the rated maximum cooling capacity ($q_{c,max}$ [W]) to the rated cooling capacity ($q_{c,rtd}$ [W]), which are given in the air conditioner catalog:

$$r_c = \frac{q_{c,max}}{q_{c,rtd}} \tag{22}$$

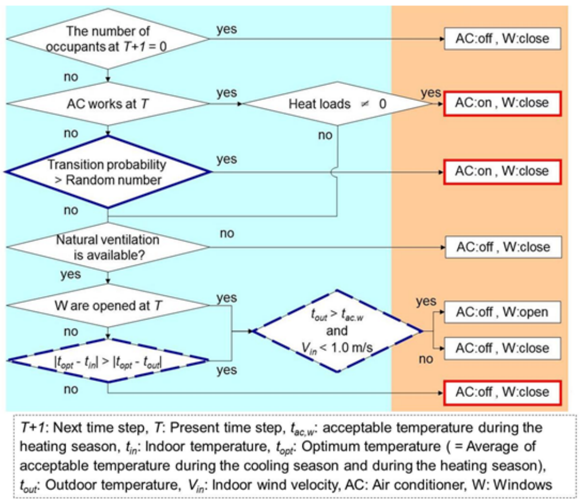


Figure 2 Flowchart of the thermal control behaviour model

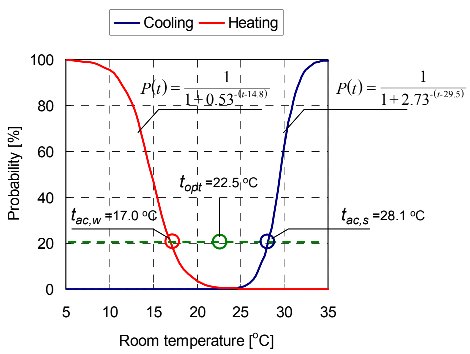


Figure 3 On-to-off transition probability for the air conditioner

Thermal control behaviour model

In a previous study, we have modeled the thermal control behaviour of occupants on the basis of a survey on the relationship between temperature and the usage of an air conditioner in residential houses (Habara et al., 2005). The flowchart of the thermal control behaviour model is shown in Figure 2, and the on-to-off transition probability for an air conditioner is presented as a sigmoidal response function in Figure 3. The thermal control behaviour is determined from the occupancy and room temperature every 15 min during waking hours and once before sleep. The types of thermal control behaviour are “air conditioner (AC)”, “natural ventilation through large openings (NV)” and “closed windows without air condition (CL)”.

SIMULATION SETUP

Outside conditions

Rectangular buildings (north-south width: 7.43 m, east-west width: 8.795 m, height: 5.9 m) are spaced at intervals of 6 m. The weather data measured from July 1 to September 1 2005 is obtained from the Automated Meteorological Data Acquisition System (AMeDAS). Figure 4 shows the daily average of the outside temperature and humidity during the measurement period.

Building and equipment

The house plan is based on the standard house model proposed by the Architectural Institute of Japan (AIJ), as shown in Figure 5. The house is wooden construction with 0.65 m deep overhangs above each window. The thermal insulation satisfies the Japanese 1999 standard, which means that the heat loss coefficient is about 2.7 W/(m^2•K).

The sizes of opening area and the ventilation parameters are given as Table 1. The airtightness of the air intake openings and inner doors are obtained from the experimental results (Shimizu et al., 1995). The airtightness of the exterior doors is set in proportion to the opening areas, and the airtightness of the whole house is assumed to be 5.0 cm^2/m^2, which satisfies the Japanese 1999 standard. The air

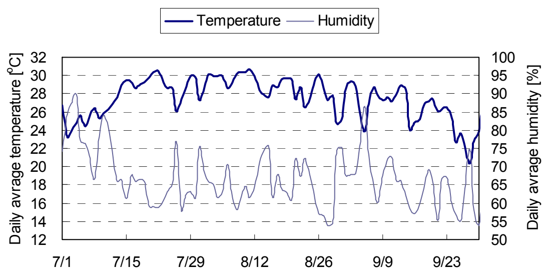


Figure 4 Daily average of outside temperature and humidity

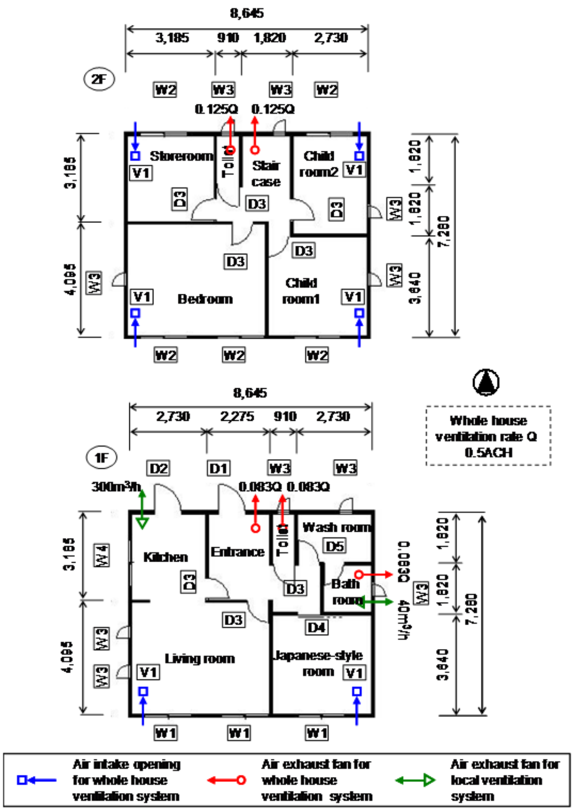


Figure 5 Ventilation system setup

Table 1 Sizes of the opening areas and ventilation parameters

Type		Size [m^2]	Air leakage coefficient [m/(s•Pa^n)]	Pressure exponen [-]	Discharge coefficient [-]
Exterior openings	W1	3.40	19.30	1.50	0.54
	W2	2.04	11.58		
	W3	0.60	3.41		
	W4	0.77	4.34		
	D1	2.00	11.35		
	D2	1.60	9.08		
Inxterior openings	D3	1.60	56.04	1.83	- (always closed)
	D4	1.60	110.71	2.13	
	D5	1.60	36.08	1.90	
Other Opening	V1	4.65	2.04	2.04	-
	Stair case	11.00	-	-	0.20

intake and exhaust openings for a mechanical ventilation system are placed as shown in Figure 5. The air exhaust rate for the whole house ventilation is set to be 0.5 ACH according to the Japanese law. The local ventilation system works at 300 m^3/h in the kitchen while cooking and at 40 m^3/h in the bathroom for 1 h after bathing. The wind pressure coefficient, which is required for calculating the natural ventilation rate, is simulated with a CFD model.

The specifications for lighting and air conditioners is set according to the floor area of the rooms, as shown in Table 2 and Table 3, in accordance with the guidelines.

Life-style

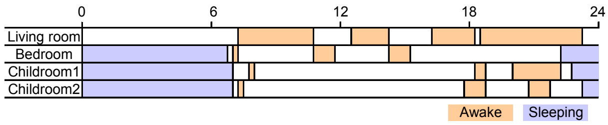
The family in this study consists of four members, namely, a couple (male office worker, housewife) and two children (female high school student, male junior high school student). The schedule of

Table 2 Specifications of air conditioners and lighting

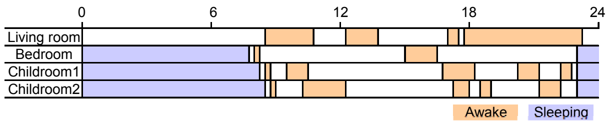
	Room name	Area [m^2]	Volume [m^3]	Spec of air conditioner	Lamp wattage [W]
1F	Kitchen	7.97	19.13	-	64
	Living room	19.37	46.5	c	155
	Japanese-style room	12.36	29.65	-	108
	Bathroom	2.92	7	-	45
	Wash room	4.42	10.6	-	45
	Toilet	1.37	3.3	-	45
	Entrance	6.66	15.97	-	45
	Corridor	2.96	7.1	-	45
2F	Storeroom	9.42	22.62	-	80
	Bedroom	19.37	46.5	c	155
	Child room 1	12.36	29.65	b	108
	Child room 2	9.16	21.98	a	80
	Toilet	1.34	3.21	-	45
	Corridor	3.55	8.5	-	45
	Stair case	3.09	7.41	-	45

Table 3 Details of air conditioner specification

Spec of air conditioner	Rated cooling capacity [kW]	Maximum cooling capacity [kW]	COP [-]
a	2.2	3.4	4.9
b	2.5	3.5	5.0
c	3.6	4.0	3.3

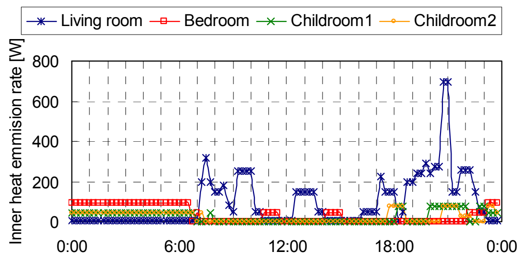


(a) Weekday

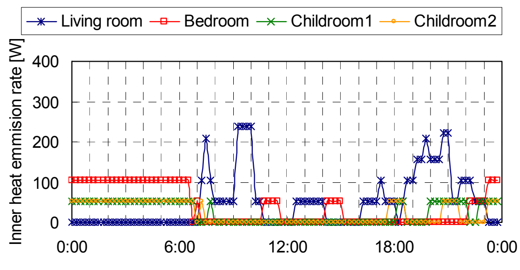


(b) Holiday

Figure 6 Occupation schedule



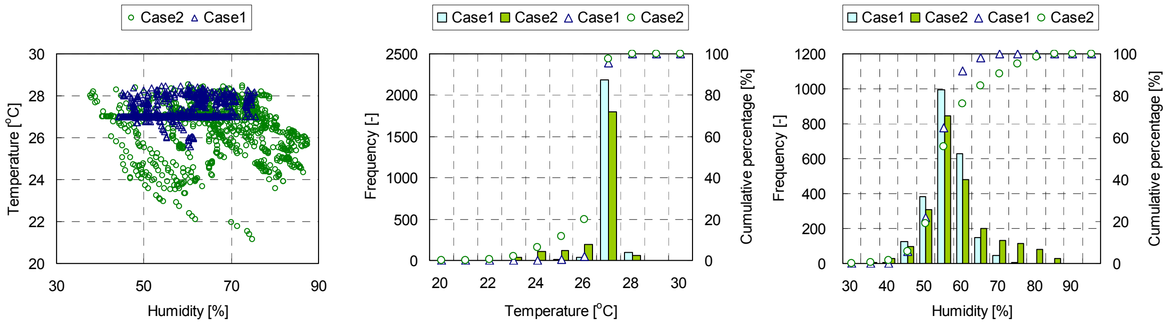
(a) Sensible heat emission rate



(b) Latent heat emission rate

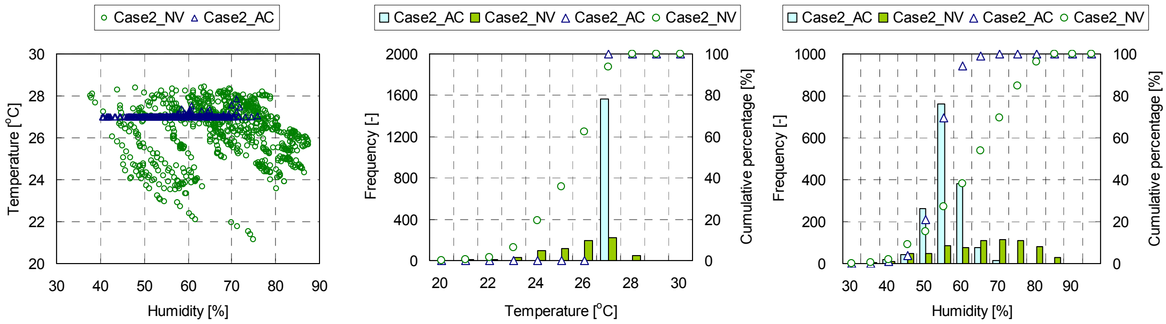
Figure 7 Inner heat emission pattern for a weekday

occupation and domestic appliances usage (excluding air conditioners and lightings) is determined from survey results (Broadcasting Culture Research Institute, 2001). Figure 6 shows the occupation



(a)Relationship between room temperature and humidity *(b)Histogram of room temperature* *(c)Histogram of room humidity*

Figure 8 Comparison of room temperature and humidity in the living room between Case 1 and Case 2



(a)Relationship between room temperature and humidity *(b)Histogram of room temperature* *(c)Histogram of room humidity*

Figure 9 Comparison of room temperature and humidity in the living room for "AC" hours and "NV" hours in Case 2

schedule for weekday and holiday. Figure 7 describes the heat emission pattern for a typical weekday.

The usage of air conditioner and natural ventilation in each room is determined using the thermal control behaviour model. During cooling hours, the preset temperature of air conditioner is 27 °C and all openings are closed. During natural ventilation hours, the windows and exterior doors facing the target room are opened, whereas all inner doors are closed.

Lighting is turned on when the room illuminance is below 75 lx without lighting. The lace curtain (rate of airflow decline: 0.35) is always closed and the shade curtain (rate of airflow decline: 0.58) is only closed during sleeping hours.

RESULTS AND DISCUSSION

Two cases are modeled; In Case 1, occupants can control their environment only with air conditioners, whereas in Case 2 occupants can control their environment with air conditioners and natural ventilation through large openings. The simulation results are discussed below.

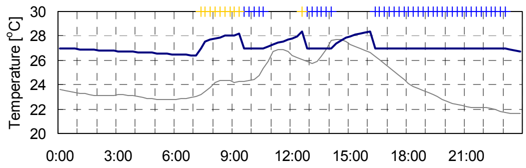
Room temperature and humidity

The relationship between room temperature and relative humidity and the histograms of room temperature and relative humidity during occupation of the living room are shown in Figure 8. The room temperature and humidity ranges are broader in Case 2. In both cases, room temperature is concentrated around 27 °C, which is the preset temperature of the air conditioners. The difference between room humidity in Case 1 and Case 2 is smaller than that between room temperature. The room humidity is mostly within 50-60 % in both cases, whereas it occasionally exceeds 80 % in Case 2.

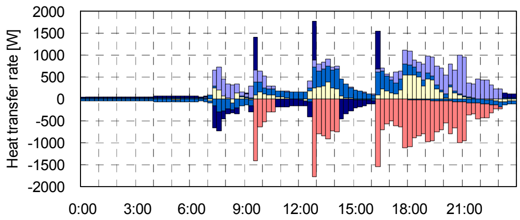
Figure 9 shows the relationship between room temperature and relative humidity and the histograms of room temperature and relative humidity in the living room during "AC" hours and "NV" hours in Case 2. As shown in Figure 9, the defined peaks of room temperature and humidity during "NV" hours does not appear. The average room temperature is 27.0 °C (S.D.: 0.0 °C) during "AC" hours and 26.3 °C (S.D.: 1.4 °C) during "NV" hours. The average room humidity is 58.0 % (S.D.: 4.0 %) during "AC" hours and 67.7 % (S.D.: 11.2 %) during "NV" hours.

Heat transfer structure

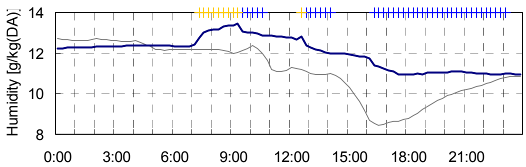
The time series data for outside/room temperature and sensible heat transfer rate in the living room are given in Figure 10 for Case 1 and in Figure 11 for Case 2. In addition, the time series data of outside/room absolute humidity and latent heat transfer rate in the living room are shown in Figure 10 for Case 1 and in Figure 11 for Case 2. "AC," "NV" and "CL" in Figure 10 and Figure 11 represent



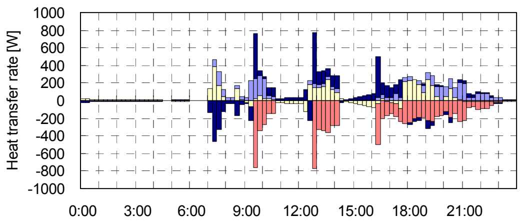
(a) Temperature



(b) Sensible heat transfer rate

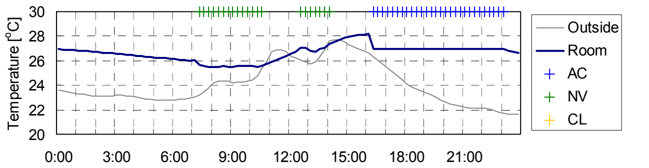


(c) Absolute humidity

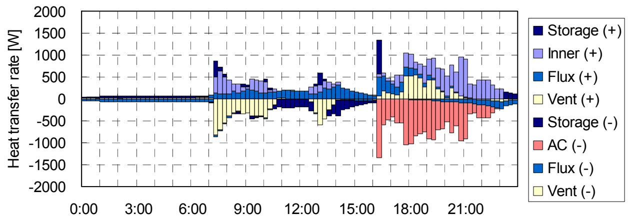


(d) Latent heat transfer rate

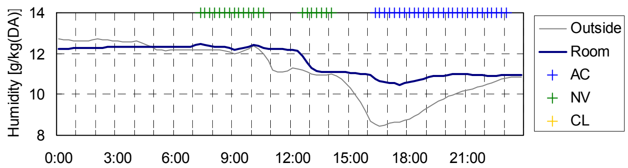
Figure 10 Time series data for outside/room temperature/humidity and heat transfer rate (Case 1)



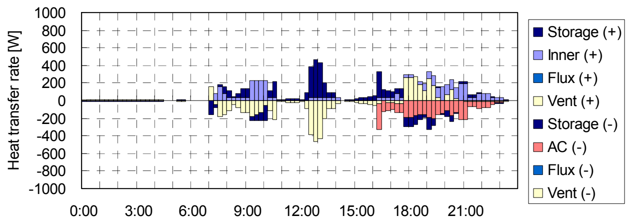
(a) Temperature



(b) Sensible heat transfer rate

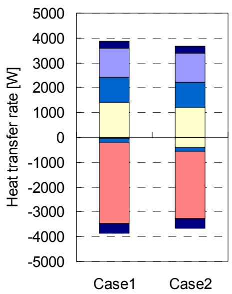


(c) Absolute humidity

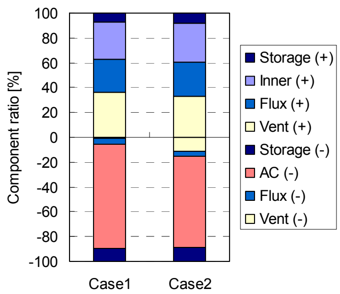


(d) Latent heat transfer rate

Figure 11 Time series data for outside/room temperature/humidity and heat transfer rate (Case 2)

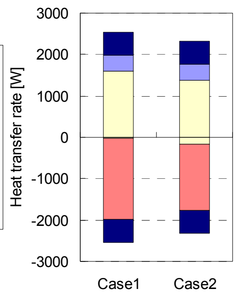


(a) Heat transfer rate

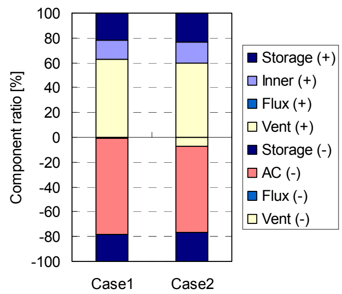


(b) Component ratio

Figure 12 Periodic sensible heat transfer rate in the living room



(a) Heat transfer rate

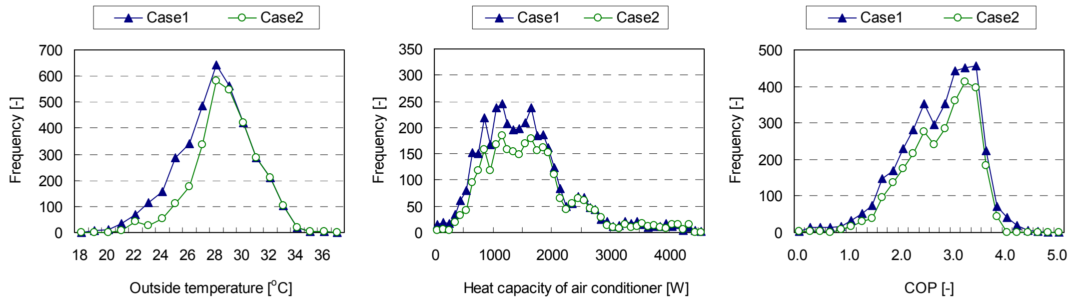


(b) Component ratio

Figure 13 Periodic latent heat transfer rate in the living room

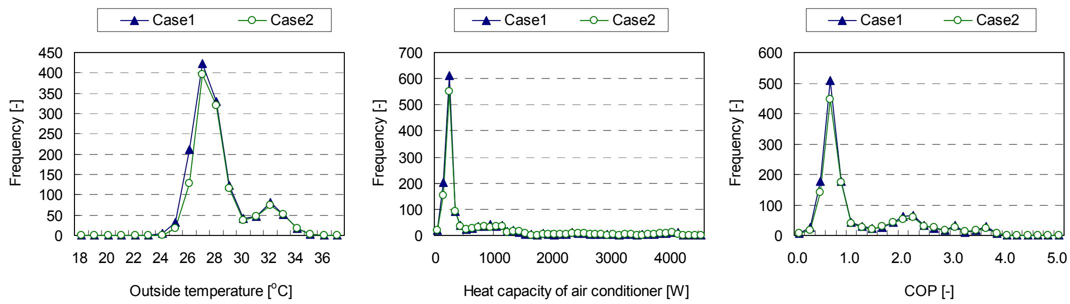
occupancy and occupant thermal control behaviour. "Storage," "Inner," "Flux," "Vent" and "AC" in Figure 10 and Figure 11 mean thermal storage in room air, inner heat emission, heat convection, heat transfer through ventilation and the heat removal by air conditioning respectively.

During the weekday shown in figures, occupants use an air conditioner in most of occupation hours in Case 1, whereas they use natural ventilation before evening in Case 2. The sensible heat removal rate of the air conditioner at 4 pm, when it has been turned on, is lower in Case 2 compared with Case 1. The reason for this is that the convective heat flux during the absence of occupants decreases and the room temperature at the start of air conditioning is lower because the heat discharge rate through natural



(a) Outside temperature *(b) Total cooling capacity* *(c) COP*

Figure 14 Outside temperature during cooling hours, total cooling capacity and COP of air conditioner (in the living room)



(a) Outside temperature *(b) Total cooling capacity* *(c) COP*

Figure 15 Outside temperature during cooling hours, total cooling capacity and COP of air conditioner (in the bedroom)

ventilation is equivalent to or greater than the inner heat emission rate and the convective heat flux. The latent heat removal rate of the air conditioner at 4 pm is also lower in Case 2. The room humidity increases during "CL" hours in Case 1, while it reaches to the outside humidity value during "CV" hours in Case 2. Therefore, the room humidity at the beginning of air conditioning is lower in Case 2.

The periodic heat transfer rate and the component ratio of the living room for the sensible heat and the latent heat are shown in Figure 12 and in Figure 13, respectively. The ratio of the heat transfer through ventilation to the heat gain is relatively large in Case 1, where natural ventilation is not available. This result suggests that although it is often said that natural ventilation through large openings could increase the heat load owing to induction of high temperature and humid outside air, mechanical ventilation has a much greater influence on the increase in heat gain resulting from ventilation. However, the heat discharge rate through ventilation is not sufficient for exceeding the inner emission rate and the convective heat flux. In particular, durign the absence of occupants, because the heat discharge through ventilation is minimal, room temperature increases owing to the convective heat flux. This indicates the effectiveness of heat discharge through ventilation for reducing air conditioner usage and cooling energy consumption when there are no occupants.

Air conditioner operation

Figure 14 and Figure 15 show the histograms of the outside temperature during cooling hours and total cooling capacity and COP of the air conditioner for the living room and the bedroom, respectively. For the living room, the cooling hours are reduced in the temperature range of below 28 ^{o}C owing to the use of natural ventilation. Consequently, the total cooling capacity of the air conditioner decreases to the range of below 1500 W, where the air conditioner operates at part load. It's generally known that the part-load operation degrades performance. Therefore, the use of natural ventilation has an additional effect for improving the energy performance of air conditioners by reducing the part-load operation. On the other hand, for the bedroom, the difference between Case 1 and Case 2 is small. The reason of this is that the air conditioner is used for the almost same number of hours because "CL" hours in Case 1 are a substitute for "CV" hours in Case 2.

Table 4 shows the periodic evaluation of air conditioner operation. The use of natural ventilation is most effective in conserving cooling energy in Child room 2. The influence of solar radiation is small in this room, because it is located in the northeast corner of the house. In addition, it is occupied during sleeping hours and the inner heat

Table 4 Periodic evaluation of air conditioner operation

Room name	Calculation case	Periodic sensible cooling capacity of air conditioner [MJ]	Periodic latent cooling capacity of air conditioner [MJ]	Periodic total cooling capacity of air conditioner [MJ]	Periodic electric power consumption [MJ]	Periodic cooling hours [h]	Average periodic COP [-]	Average periodic SHF [-]
Living room	Case 1	3261	1961	5222	1786	940.50	2.92	0.62
	Case 2	2683	1611	4294	1453	731.25	2.96	0.62
	Difference	-17.7%	-17.8%	-17.8%	-18.7%	-22.2%	+1.1%	-0.1%
Bedroom	Case 1	429	342	772	504	341.25	1.53	0.56
	Case 2	382	301	683	446	301.00	1.53	0.56
	Difference	-10.9%	-12%	-11.4%	-11.5%	-11.8%	-0.1%	+0.5%
Child room 1	Case 1	358	256	614	348	575.25	1.76	0.58
	Case 2	302	212	514	272	425.75	1.89	0.59
	Difference	-15.8%	-17%	-16.3%	-22%	-26%	+7.3%	+0.6%
Child room 2	Case 1	206	168	374	139	218.25	2.69	0.55
	Case 2	170	139	309	94	131.25	3.30	0.55
	Difference	-17.7%	-17.4%	-17.5%	-32.6%	-39.9%	+22.4%	-0.2%
Total	Case 1	4254	2727	6982	2777	2075.25	2.51	0.61
	Case 2	3537	2263	5800	2264	1589.25	2.56	0.61
	Difference	-16.9%	-17%	-16.9%	-18.5%	-23.4%	+1.9%	-0.1%

* Average periodic COP = Periodic total capacity of air conditioner / Periodic electric power consumption
** Average periodic SHF = Periodic sensible capacity of air conditioner / Periodic total capacity of air conditioner

emission rate is small. Hence, compared to the other rooms, part-load operation is more frequent in Child room 2 in Case 1. On the other hand, in Case 2, natural ventilation usage reduces part-load operation, which considerably improves the average periodic COP. Therefore, although the reduction ratio of cooling capacity is comparable to that of the other rooms, the reduction ratio of energy consumption is largest in Child room 2. The average periodic SHF is the same for Case 1 and Case 2, which means that the latent cooling capacity and the sensible cooling capacity decrease equally.

CONCLUSION

In this study, the effect of heat discharge by natural ventilation through large openings on indoor environments and heat removal structure was examined using a simulation model that can estimate the occupant thermal control behaviour and the characteristics of air conditioners. The results are as follows:

- The room temperature range was broader in Case 2, where occupants could use natural ventilation through large openings, compared to Case 1, where natural ventilation was not available.
- The room humidity was mostly in the rage of 50-60 % in both cases, while it sometimes exceeded 80 % in Case 2.
- Heat discharge through natural ventilation is effective for equally reducing the sensible heat load and the latent heat load.
- If the heat convective flux during the absence of occupants is exhausted by natural ventilation, air conditioner usage could be further restrained.
- The effectiveness of natural ventilation for cooling energy conservation is high, especially, for the rooms in which the heat load is small because of the improvement of energy efficiency of an air conditioner in addition to the reduction of cooling capacity.

In future works, to adequately evaluate the heat discharge rate through buoyancy ventilation, we will improve the thermal load calculation model so as to consider the spatial distribution of indoor air temperature.

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