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Numerical Analysis of a Hygrothermal Environment during Hot and Humid Seasons Considering Room Hygroscopicity and Air Conditioner Driving Mode

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Abstract

This study aims to investigate the hygrothermal environment by estimating the room air conditioner (RAC) heat load in a room with hygroscopic materials. The sensible and latent heat capacity and the water retention quantity in the RAC indoor unit are tested in cooling and dehumidification operation modes. Then, a model is developed to predict their values and investigate room temperature and humidity. The RAC model numerical analysis estimates the heat load by taking into account indoor hygroscopicity. In particular, the water is retained from several hundred grams to approximately 1 kg in the indoor unit. The RAC model is implemented considering the heat exchanger temperature distribution. In the weak cooling dehumidification mode, the room relative humidity is maintained between larger than 5% and 8% of the target value with and without hygroscopicity, respectively. Hygroscopicity does not affect the heat load in the bedroom during night time.

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Keywords: Hygroscopic materials, Dehumidification, Cooling, Hot and humid season, Coupled heat and mositure transfer

1. Introduction

Houses in Japan commonly use room air conditioners (RACs) to control the indoor temperature and humidity during the hot and/or humid season. The RAC operation modes—cooling and dehumidification—are adjusted

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depending on the room conditions without prior knowledge on how feasible the target temperature and humidity are by using RAC. In particular, the RAC's heat load characteristics and electricity consumption are unclear. The several models of the heat exchange in the cooling and the dehumidification of the coil of heat exchanger were developed [1]—[4], but the retention and re-evaporation of the condensation water at the evaporator's fin surface were not taken into account. Hygroscopic materials are known to reduce air relative humidity changes in the room. Therefore, it is necessary for the RAC and hygroscopicity (hereafter referred to as hygry) to work in conjunction to realize the target temperature and humidity. For example, an increase in room air humidity may be suppressed by hygroscopic materials, and the room air environment comfortably maintained. On the contrary, cooling load may be enhanced by hygroscopic materials which absorb moisture.

In this study, the hygrothermal environment characteristics are investigated by examining the RAC-adjusted heat load in a room with hygroscopic materials. First, the control method is clarified by determining the sensible and latent heat capacities and the water retention quantity in the indoor RAC unit in cooling and dehumidification operation modes. Next, a model is developed to predict their value. Finally, the room temperature, humidity and heat load are estimated through numerical analysis using the RAC model and by taking into account the room's hygry.

2. RAC climatic chamber experiment

2.1. Outline of the climatic chamber experiment

The RAC experiment was conducted in climatic chambers, which adopted a room and outdoor climate. The RAC's indoor and outdoor units were installed in the indoor and outdoor chambers, respectively (Table 1). The test RAC included two dehumidification modes: weak cooling mode and re-heat dehumidification mode. The former mode maintains the heat exchanger temperature below the dew point temperature of the inlet air and reduces the fan speed and/or cooling area so that sensible heat exchange will decrease. The latter mode uses part of the condenser as a heating source. This study only investigates the weak cooling mode.

Table 1: Outline of the experimental apparatus

Rated capability	Power consumption	Dehumidification method
2.2 kW	0.45 kW	Weak cooling dehumidification

Table 2: Experimental conditions

Operation mode of the RAC	Cooling Dehumidification	Operation of Air flow of the RAC	Automatic
Target value of temperature	27 ºC	Target value of relative humidity	Dehumidification:50%
Outdoor temperature	Cooling mode: 35ºC Dehumidification mode : 27ºC	Outdoor relative humidity	Cooling mode: 40% Dehumidification mode : 60%
Indoor temperature	13–29 ºC	Indoor relative humidity	35–90%

2.2. Indoor experimental conditions

Fixed temperature and humidity conditions were maintained in the indoor climatic chamber for about 1–2 h to obtain a steady state of driving by the RAC and clarify the RAC's heat exchange and dehumidification characteristics. The purpose of the experiment was to measure these characteristics and develop a model of the RAC dehumidification. The experimental conditions are presented in Table 2.

2.3. Measuring the retained condensation water in the indoor unit heat exchanger

During cooling or dehumidification, the condensation water, which is attached to the evaporator's fin surface, is not completely exhausted through the drain hose and is retained on the fin. Water partly evaporates and returns to the room. Hence, experiments were performed to quantify the retained dew condensation water during cooling. The retained water is obtained by two methods: the air enthalpy method, which measures the inlet and outlet temperature and humidity and the air flow rate, and the weight measurement method, which obtains values before and after drying the indoor unit. Before drying the retained water, the RAC is operated on cooling mode with 17 or 18 °C as the target, and the room temperature and humidity in the

Table 3: Measured retained water left in the indoor

	Air enthalpy method	Weight measuring method
Amount of retain water	468g	1014g

indoor chamber are maintained at 27 °C and 40%, respectively. Table 3 presents the measured retained water left in the indoor unit. The retained water ranges from several hundred grams to approximately 1 kg.

3. Experimental results of RAC climatic chamber and modeling of RAC

3.1. Experimental observation on the RAC's weak cooling dehumidification mode

To investigate the evaporator's dehumidification characteristics, the heat exchanger surface temperature was measured at 18 points of the indoor unit (Fig. 1(a)). Fig. 1(b) presents the temperature profile of the heat exchanger in the dehumidification mode when the indoor temperature and humidity are fixed at 27° C and 60%, respectively. The measured temperature at points 1–6 is less than that at the dew point temperature of the inlet air. However, the temperature at points 11–18 is nearly the same as the input air temperature. It appears that the refrigerant's flow rate is suppressed to decrease the sensible and latent heat loads. Fig. 1(c) provides the temperature profile of the heat exchanger in the cooling mode. The evaporator is dehumidified when the temperature at all points is less than that at a dew point. However, the sensible heat load is greater than that in the dehumidification mode because the whole evaporator heat exchanger absorbs the sensible heat.

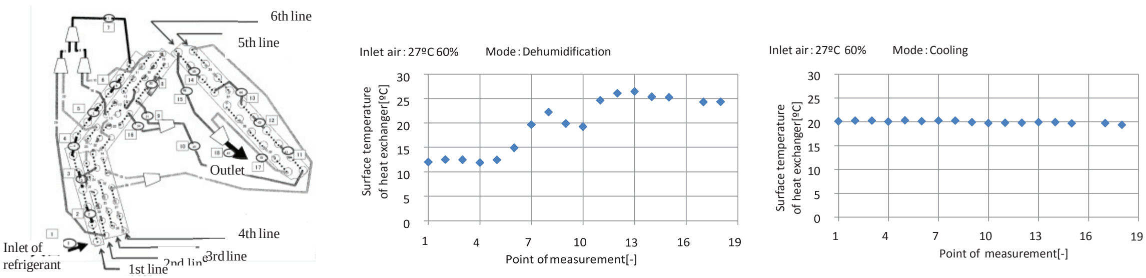

Fig. 1. (a) Measurement points of the heat exchanger; (b) Temperature profile of the heat exchanger in the dehumidification mode; (c) Temperature profile of the heat exchanger in the cooling mode apparatus

3.2. Predictive model of the RAC's weak cooling dehumidification mode

A model was developed for predicting the sensible and latent heat loads and the retained water in the RAC's indoor unit, taking into account the temperature distribution of the heat exchanger. This model is based on the following assumptions: (1) The heat exchange between the air flow and heat exchanger can be assimilated using the bypass factor (BF). (2) The surface temperature of the heat exchanger is assumed to be the same as the refrigerant temperature. (3) The heat exchanger is divided into two zones, and the air flow through each zone is separated using the factor ϕ without mixing the air of the two zones. (4) The air from each lane of the heat exchanger is represented by one mass point. (6) Until the dew condensation water in the heat exchanger surface exceeds an upper limit, water is retained, and the excess is run off. (7) The heat that a refrigerant receives is used only for phase conversion in the liquid and vapour mixture region, and the temperature of refrigerant depends only on the pressure. The sensible and latent heat exchange flow between the evaporator surface and the air is calculated by the following equations.

$$Q_{shin}(i) = \emptyset * M_{rin}(1-BF)\{C_a(T_{evin} - T(i)) + C_v(X_{evin} * T_{evin} - X_{sat}(T(i))T(i))\}/n \quad (1)$$

$$Q_{lhin}(i) = \emptyset * M_{rin} * (1-BF) * L_o * (X_{evin} - X_{sat}(i))/n, \quad (2)$$

$$Q_{thin}(i) = Q_{shin}(i) + Q_{lhin}(i), \quad (3)$$

where Q_{shin}, Q_{lhin} and Q_{thin} [kW] are the sensible, latent and total heat loads, respectively; M_{rin} [kg'/s] is the inlet air flow rate, BF [-] is the bypass factor, Ca and Cv [kJ/kgK] are the specific heat of air and vapour, respectively; T_{evin} and T(x)[K] are the temperature of the inlet air and heat exchanger surface, respectively; X_{evin} and X(x) [kg/kg of dry air] are the absolute humidity of the inlet air and the heat exchanger surface when the surface temperature is lower than the dew point, respectively and Lo [kJ/kg]is the latent heat of water; n is a division number of the heat exchanger[−].

The conservation equation of the retained water is calculated from equation (4).

$$\partial W_{cap}(i)/\partial t = \emptyset * G_{a} * (1 - BF) * (X_{in} - X_{sat}(T(i)))/n, \tag{4}$$

where W_{cap} [kg] is the heat exchanger retained water. The heat conservation of the refrigerant is expressed by equation (5).

$$(c\rho)_{ref} V_{ref}\, \partial T(i)/\partial t = -(q_{ref}(i+1) - q_{ref}(i)) + Q_{hi}(i), \tag{5}$$

where $(c\rho)_{ref}$ [kJ/m^{3}K] is the refrigerant's heat capacity, V_{ref} [m^3] is its volume and q_{ref} [kWm] is its enthalpy flow rate. A heat exchanger was divided into A and B (Fig. 2). Fig. 3 presents the comparison between the calculated and measured values for temperature and humidity from the fixed experiment. The measured temperature and enthalpy of the refrigerant, its flow rate, the inlet air temperature and humidity and the air flow rate are the input conditions. The calculated sensible and latent heat loads are consistent with all the measured results if BF and ϕ are 0.15 and 0.55, respectively.

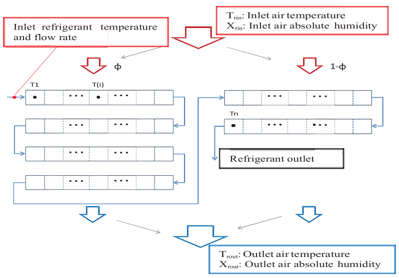

Fig. 2 Schematic of the heat exchanger model

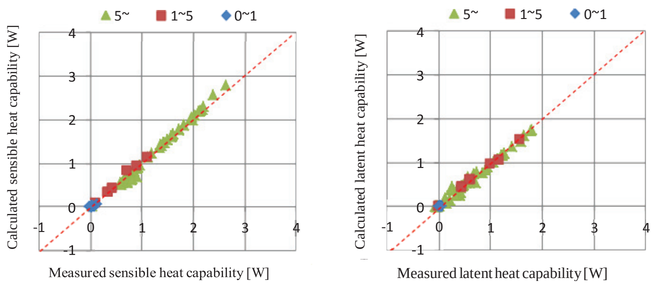

Fig. 3 Comparison between calculated and measured results in sensible and latent heat capacity

4. The RAC control model

The RAC's refrigerating cycle depends on the indoor and outdoor temperature, the target temperature and humidity. Thus, it is necessary to preset the temperature and the flow rate of the refrigerant to control the RAC's function. On the basis of the experimental results, a RAC control model was developed by considering the driving mode. In the cooling mode, the sensible heat load was estimated using the indoor temperature and target temperature and by choosing the appropriate refrigerant temperature to satisfy the load. The refrigerant finishes evaporation at the evaporator's terminal, and the refrigerant temperature is maximized. In the weak cooling dehumidification mode, an evaporator entrance temperature was chosen to be lower than the room air's dew point. The refrigerant flow rate was selected to satisfy the sensible heat load. The lower refrigerant temperature was chosen so that the indoor humidity would be higher than the targeted value.

5. Influence of hygry on indoor climate and RAC heat load under the weak cooling dehumidification mode

5.1. Numerical analysis outline

The developed RAC model was used to investigate the influence of the different driving modes, taking into account the room's hygry under indoor temperature and humidity; the sensible and latent heat loads as well as the amount of retained water to the indoor unit were estimated. Hygroscopic range coupled heat and moisture equations were used in the present analysis. The conservation equations of heat and moisture are given as follows [5]:

$$(c\rho + r\nu)\, \partial T/\partial t = \partial/\partial x\, (\lambda\, \partial T/\partial x) + r\kappa\, \partial X/\partial t \tag{6}$$

$$(\Phi_{0}\gamma' + \kappa)\, \partial X/\partial t = \partial/\partial x\, (\lambda'_{x}\, \partial X/\partial x) + \nu\, \partial T/\partial t \tag{7}$$

$$\kappa = \rho_{w}(\partial\varphi/\partial X)_{T},\ \nu = -\rho_{w}(\partial\varphi/\partial T)_{X}, \tag{8}$$

where Φ_o [m^3/m^3] is the absolute dry porosity of the material, γ' [kg/m^3] is the dry air density, X [kg/kg of dry air] is the absolute humidity, t [s] is the time, T [K] is the temperature, $\lambda'x$ [kg/ms (kg/kg of dry air)] is the moisture conductivity, x [m] is the horizontal displacement, L [J/kg] is the latent heat of vaporization, cρ [J/m^{3}K] is the apparent volumetric heat capacity of the material and ψ [m^3/m^3] is the moisture content.

The hypothetical analysis object is a one-room apartment within a complex with windows of 6 m^2 double-grazed glass (Fig. 4). The adjacent rooms are considered to have the same temperature and humidity conditions as the object room. Table 4 presents the material's hygrothermal properties. The hygroscopic material is diatomaceous earth. If the wall does not have hygry, the moisture transfer coefficient of the indoor surface is assumed to be zero. Moisture flow of the floor surface is assumed to be zero. Table 5 presents the outdoor climate conditions, air change rate and heat and moisture generation rates used in numerical analysis.

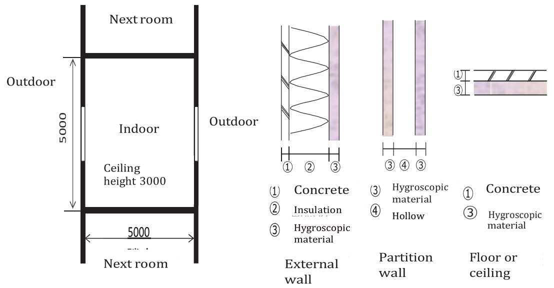

Fig. 4 Plan and wall composition of the analysis object

Table 4: Hygrothermal properties of materials

	Thickness [mm]	Density [kg/m^3]	Thermal conductivity [W/mK]	Heat capacity [J/kgK]	Water vapour permeability [ng/msPa]
Concrete	10.0	2200	1.89	840	1.0
Insulation	50.0	25	0.03	1470	1.0
Hygroscopic material	12.0	700	0.12	725	8.7

Table 5: Conditions for numerical analysis

Air change rate	Outdoor temperature	Outdoor absolute humidity	Target temperature	Target humidity	Heat generation	Moisture generation
0.5 1/h	29 – 35 ºC (1day periodic cycle)	16.32 g/kg	27 ºC	50 % Dehumidification Mode	100 W	60 g/h
		Operation time	0 - 6	0 – 6	0 - 24	0 - 6

5.2. Results and discussions

Figs. 5 and 6(a) present the calculated temperature, humidity and heat load of the room. The temperature during the RAC running with hygry is almost the same as that without hygry.

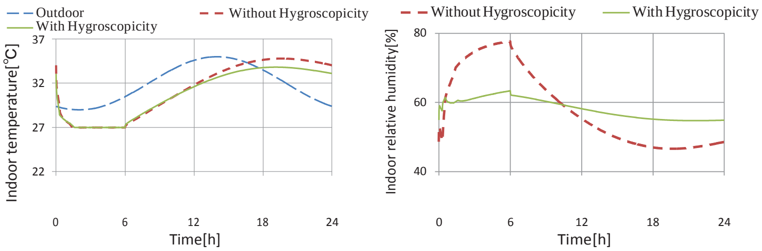

Fig. 5 Calculation results of temperature and humidity of the cooling mode

There are a few differences in room temperature between the two cases. In particular, during the RAC's operation in a room with hygry, the temperature is lower. This is because the latent heat is absorbed by the wall surface during evaporation. For the same reason, the room with hygry is lower in relative humidity than that without it. Latent heat load is large just after the RAC starts, then they become small. The room with hygry is larger in latent heat load than that without it just after the RAC starts,

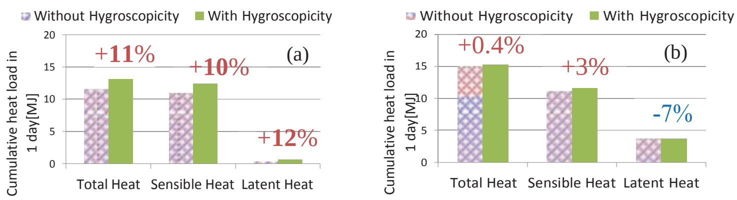

Fig. 6 Cumulative heat load in 1 day
(a) the cooling mode, (b) dehumidification mode

and then the latent heat load becomes zero both with and without hygry. This is because the room with hygry is smaller in temperature than it would be without it. In addition, the wall with hygry absorbs the moisture in the room, thus increasing the temperature because of the condensation heat by absorbing moisture at the wall with hygry. Thus, the room with hygry has 11% larger cooling load than that without hygry (Fig. 6(a)).

Fig. 7 presents the temperature and humidity estimates. Temperature during the RAC running with hygry of the room is almost the same as that without hygry. The room's relative humidity is maintained larger than 5% and 8% of the target value with and without hygry, respectively. When the room temperature approaches the target value, the room with hygry has a stable relative humidity, whereas the relative humidity in the room without hygry increases. This is because the latent heat load decreases with decreasing the sensible heat load, and the wall with hygry absorbs moisture from the room air. The sensible, latent and total heat loads are nearly the same in the room with and without hygry, as shown in Fig 6(b). The retained water in the indoor unit increases just after the RAC starts but decreases with decreasing sensible heat load, and re-evaporation occurs at that time. When the RAC stops, the retained water is about 20 g, which is not so large as the experimental result.

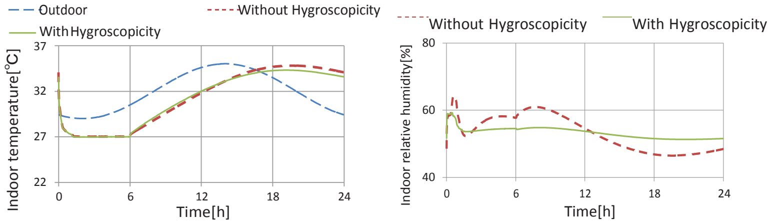

Fig. 7 Calculated results of temperature and humidity of the dehumidification mode

6. Conclusions

This study offers experimental and modeled data on the hygrothermal environment by focusing on a room with RAC and hygroscopic materials. The main conclusions of this study are 1) The indoor unit retains several hundred grams to approximately 1 kg water. 2) In the weak cooling dehumidification mode, dehumidification is conducted near the entrance of the evaporator and at a temperature lower than the dew point temperature. Temperature rises from the middle to the terminal part of the evaporator after the refrigerant evaporates. 3) We developed a model for predicting the latent and sensible heat loads and the RAC indoor unit retained water by taking into account the heat exchanger temperature distribution. When BF and φ are 0.15 and 0.55, respectively, the modeled values for the sensible and latent heat loads are consistent with the measured results. 4) In the cooling mode, the room with hygry exhibits lower relative humidity than that without it. The room with hygry has humidity greater by 11% than that without hygry when the RAC is supposed to be controlled in the bedroom during the night time. 5) In the weak cooling dehumidification mode, the room's relative humidity is maintained larger than 5% and 8% of the target value with and without hygry, respectively. The total heat load is nearly the same in both rooms.

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