

## A STUDY ON THE APPLICATION OF THE RADIANT FLOOR COOLING SYSTEM INTEGRATED WITH A DEHUMIDIFICATION SYSTEM

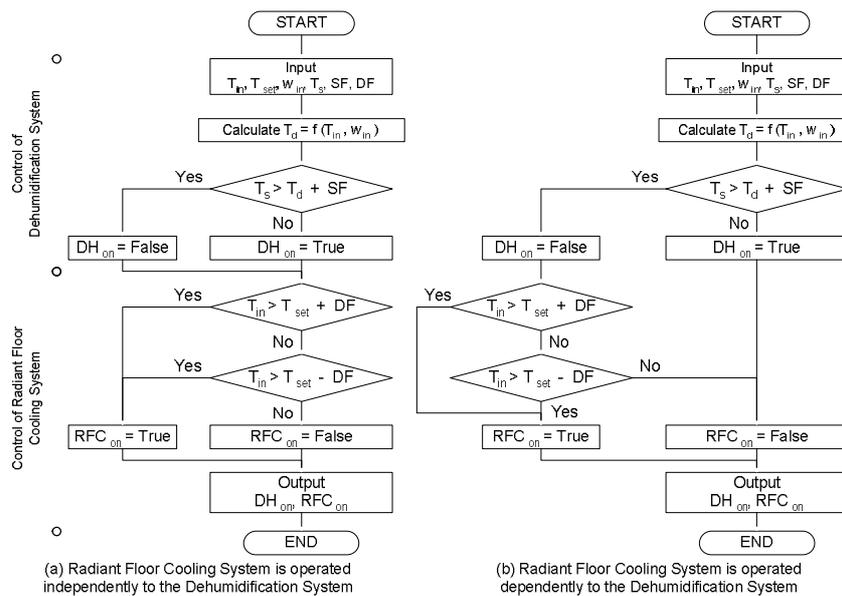
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### ABSTRACT

When applying the radiant floor cooling system, it is important to prevent condensation on the floor surface. To solve this problem, a radiant floor cooling system integrated with the dehumidification system has been proposed and evaluated. In doing so, the relationship between the control variables in preventing floor surface condensation is first analyzed, and the control methods are evaluated through simulations. After analyzing the control stabilities of room air temperature, condensation control of the floor surface, and the operation time of PAC with the simulation results, the control method is determined and tested through model experiments. Through these experiments, the applicability of the radiant floor cooling system integrated with dehumidification system is confirmed after analysis of stability in room air temperature and changes in the condensation control variables for varying internal loads.

### INTRODUCTION

In Korea, the radiant floor heating system has been widely used as a residential heating method, which has been modernized to use hot water running into the tubes embedded in the floor structure (Myoung-Souk, Y. et al, 1997, Jin-Young, L. et al, 1999, Yong-Yee, K. et al, 1999). Recently, there are many researches about radiant floor heating system. Also, with the requirement for cooling and the recent popularization of packaged air conditioners (PAC) in domestic apartment buildings, the peak demand for electricity in the summertime is greatly increasing (Jun-Kun, Y., 2000). In response to this phenomenon, the radiant floor cooling system has been proposed as an alternative, which utilizes the existing floor heating system. In doing so, it is important for the radiant floor cooling system to prevent condensation on the floor surface. The existing packaged air conditioner can be used effectively as a dehumidification system, which allows the electrical peak demand to be clipped and



**Fig. 1 Control algorithms of the dehumidification-integrated radiant floor cooling system.**

shifted if the cooling sources for the radiant floor cooling is diversified into the gas-fired absorption chiller or district cooling. However to apply the dehumidification system to the radiant floor cooling system for the condensation problem, research is necessary on the analysis of condensation control method, and stability in room air temperature with varying internal loads according to the condensation control alternatives.

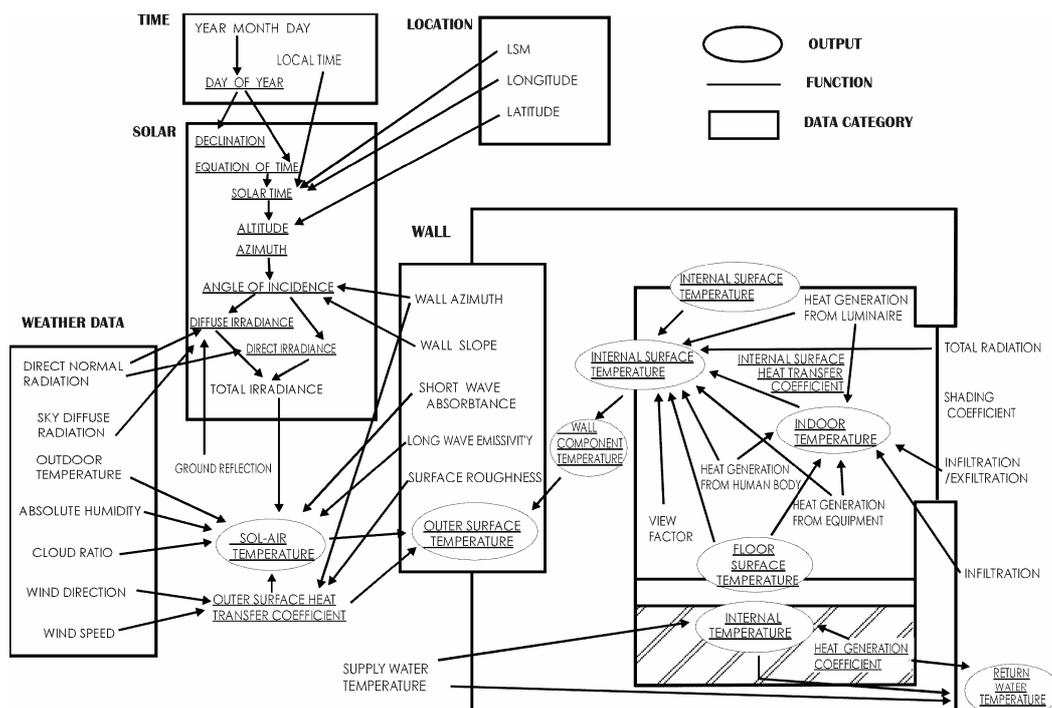
Therefore, this study applies the dehumidification system to the radiant floor cooling, and finds the method to yield the better control performance. Model experiments are performed for this control method during the cooling period, and the applicability of the radiant floor cooling system with dehumidification system is confirmed through the analysis of stability in room air temperature and changes in the condensation control variables for varying internal loads.

## **CONTROL FACTORS AND CONTROL METHODS**

In radiant floor cooling, condensation occurs with changes in floor surface temperature, room air temperature, and room air humidity. Changes in the room air temperature are caused by external and internal heat loads, such as solar radiation, occupants, lighting, and equipment. In general, the factors affecting room air temperature are as follows: convection and radiation at each surrounding surface, internal heat generations and infiltration. Considering the control of room air temperature, allowance for the time lag should be considered. Changes in the floor surface temperature occur

according to the flow rate and temperature of the cooling water, and also vary according to time lag characteristics of the floor structure. For this reason, it is appropriate to control the dew point temperature through dehumidification to prevent surface condensation.

In radiant floor cooling, the amount of heat absorbed depends on the flow rate and temperature of the supply water. Thus, the control methods may be classified into flow rate control, such as on/off control and variable flow control, and supply temperature control, such as outdoor reset control and outdoor reset with indoor temperature feedback control, which have been studied in the area of the floor heating system. When the radiant floor cooling with the dehumidification system controls both room air temperature and condensation occurrence, the control factors for room air temperature, floor surface temperature, and room air humidity interact with one another. In other words, dehumidification system normally removes both moisture and sensible heat from entering air, so overcooling of room air temperature can be occurred. Therefore, the control methods may be classified into the cases where the operation of radiant floor cooling is unaffected by the operation of dehumidification (the two are independent, as in Fig. 1 (a)), and the case where the operation of radiant floor cooling is affected by the operation of dehumidification (the radiant floor cooling is dependent on dehumidification, as in Fig. 1 (b)). In the current study, on/off bang-bang control (case I, case II in Table 1), which is commonly used in the heating systems of existing apartment buildings, and outdoor reset with indoor temperature



**Fig. 2 Diagram of the room thermal model for simulation program.**

feedback control (case III, case IV in Table 1), which was found to show good control performance in a previous study (Jae-Han, L. et al, 2001), are applied as the control methods. So, when the two are dependent, supplied water of the radiant floor cooling is stopped (on/off bang-bang control) or recirculated (outdoor reset with indoor temperature feedback control) if dehumidification system is operating. Packaged air conditioners with direct expansion coil (DX coil) are used for the dehumidification system.

## **SIMULATIONS**

### **Simulation algorithms**

The room thermal model used in simulation program is based on the implicit type of finite difference method for the calculation of unsteady state heat conduction of building envelopes, as shown in Fig. 2. The heat transfer in the panel is analyzed as a one dimensional unsteady-state heat transfer in the unit section perpendicular to the pipe. Sol-air temperature and conductive heat transfer into the interior of the structure are considered for the analysis of the exterior wall surfaces. For the interior wall surfaces, radiative heat transfer through the modified thermal balance model (Zmeureanu, R. and Paul F., 1988), convective heat transfer with room air, and solar heat gains are considered. The room air temperature is determined based on the convective heat transfer at each surface, convective heat loads from occupants, lighting, and equipment, and the amount of infiltration and ventilation. For the analysis of latent loads, which affect the occurrence of condensation, outdoor air and internal latent heat gains are considered. For the modeling of the direct expansion coil in packaged air conditioners (Michael, J.B., 1993), the dry bulb temperature and absolute humidity of the air inflow

into the coil, the amount of air inflow, and the dry-bulb temperature of the outdoor air are used as input to calculate the dry-bulb temperature and absolute humidity of the air outflow.

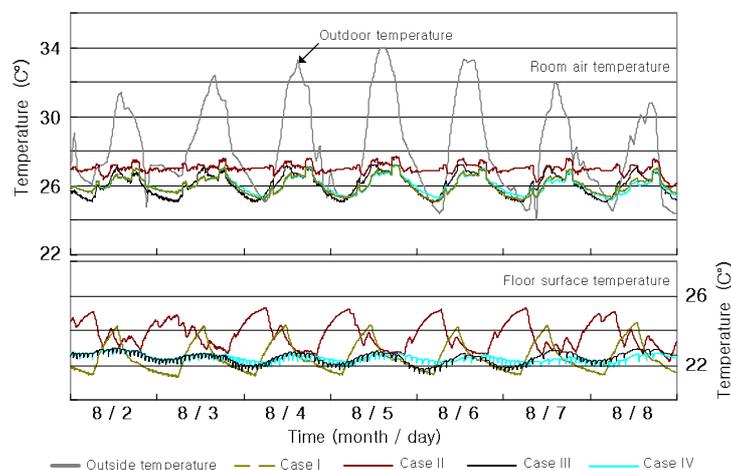
### **Validation of the simulation algorithms**

Validation of the algorithms has been performed by comparison between the experimental results and simulation calculations. Different control methods were applied to each experimental test room. In this experiment, water flow rate or water temperature was manipulated according to the control methods. Cooled water, which was manipulated by a solenoid valve or a 3-way valve, was supplied to the floor panel. Afterwards the temperatures of the supplied and returned water were recorded and were thus available for the simulation. Thermal behaviors such as the temperatures of indoor air, floor surface, also measured for comparison with the simulation results.

The simulation results for the air temperature and floor surface temperature show good agreement with the measured results within 0.5°C mean error for every case, and those for indoor relative humidity are within 3.2% mean error. These small discrepancies can be due to the difference between the input values for the simulation and actual constructed values of material properties or computational errors in rounding off to the nearest whole number. However it is thought that these will not greatly affect the overall tendency of the room's thermal environment.

### **Simulation method**

Outdoor reset control is firstly used during the week with the highest temperatures (Aug 2nd ~ Aug 8th) in the weather data to obtain the outdoor reset ratio, and this is used for the outdoor reset with indoor feedback control during the entire cooling period. Next, the minimum supply temperature is determined and applied to on/off control based on



**Fig. 3 Comparison of room air temperature, floor surface temperature during the week with the highest outdoor temperature (see Table 1).**

the relationship between outdoor temperature and supply water temperature. The control methods for the floor panels are applied separately for the case where the radiant floor cooling system and dehumidification system are independent (case I, case III in Table 1), and the case where the cooling system is dependent on the dehumidification system (case II, case IV in Table 1).

**Table 1 Comparison of room air temperature, floor surface temperature, and total operating time of packaged unit with DX coil.**

	Room air temp. (°C)		Floor surface temp. (°C)		Dew point temp. (°C)		Total operating time of packaged unit (hr)
	AVG	SD	AVG	SD	AVG	SD	
Case	26.0	0.6	24.2	1.1	21.4	1.7	6
Case	26.5	0.5	24.9	0.9	22.0	1.9	4
Case	26.0	0.4	24.3	0.6	20.9	2.2	3
Case	26.0	0.4	24.3	0.6	20.9	2.2	3
Case	26.1	0.3	25.3	0.6	22.8	2.4	299

- \* Case : Radiant floor cooling system (on/off control) is operated independently of the dehumidification system.
- \* Case : Radiant floor cooling system (on/off control) is operated dependently on the dehumidification system.
- \* Case : Radiant floor cooling system (outdoor reset with indoor feedback control) is operated independently of the dehumidification system.
- \* Case : Radiant floor cooling system (outdoor reset with indoor feedback control) is operated dependently on the dehumidification system.
- \* Case : Packaged unit is only used for room air temperature control.

## RESULTS AND DISCUSSIONS

### Control stability of room air temperature

During the entire cooling period, the control yielded an error of around 1°C from the set temperature range, while successfully preventing condensation. However, when the on/off control is integrated with the dehumidification system (case II) as in Table 1, the discontinuance of supply water causes the average room air temperature (AVG) to increase about 1.5°C when the valve is closed. In particular, during the week with the highest temperatures, the room air temperature and floor surface temperature are higher than those for the other control methods by around 0.9°C (see Fig. 3). It may thus be inferred that on/off control may cause the room to become

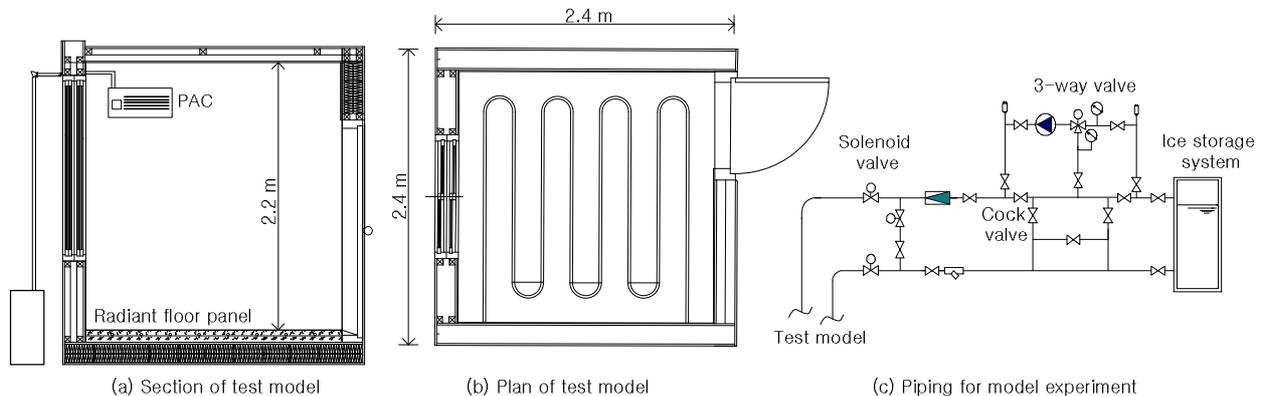
overheated for weather conditions of high temperature and high humidity. When the average room air temperatures are similar (case I, case III), the standard deviation for outdoor reset with indoor feedback control (case III) is lower by about 0.2°C compared to on/off control. (see Table 1)

### Condensation control of floor surface

In the result of applying the dehumidification system using direct expansion coils in order to control the floor surface condensation, condensation occurrence can be prevented by always keeping the room dew point below the floor surface temperature (see Fig. 3). The dehumidification system is mainly operated for a week when the highest air temperature was recorded. During this period, the dew point temperature of the room has an average of 22°C and 20°C in each case of on/off control (case I) and outdoor reset with indoor feedback control (case III). Outdoor reset with indoor feedback control shows more stability in the condensation control. In the cases with similar capability of control (case III, case IV), for the purpose of convenience in application, the case where the two systems are controlled independently (case III) can be considered to be more effective than the case where the two systems are influenced by each other.

### Comparison of the operation time of PAC

In order to evaluate the electric energy consumption in the case of applying a dehumidification system to the radiant floor cooling, the operation time is compared between the cases where packaged air conditioner is used to control the condensation (case I ~ IV) and to control the room air temperature (case V). In case V in Table 1, the compressor is controlled by the on/off method for the purpose of controlling the room air temperature, as in common in existing packaged air conditioners. When the air conditioner is applied to both radiant floor cooling and condensation control, the operation time is about 6hrs in on/off control (case I) and 3hrs in outdoor reset with indoor feedback control (case III), thus showing that the operation time of the air



**Fig. 4 Schematic diagram of test model.**

conditioner can decrease largely in the summer. However, in order to totally appraise the energy consumption in the radiant floor cooling integrated with the dehumidification system, the energy consumption of the cooling sources for both floor panel and dehumidification should be totally considered during the whole cooling period.

## **MODEL EXPERIMENT**

The condensation partly occurring on the floor surface can be influenced by the vertical distribution of temperature, the humidity, and the horizontal distribution of the floor surface temperature in an actual space. Consequently, it is necessary that we analyze the factors related to condensation on the floor surface through experiments in which the control methods are applied. Thus for case III, where the control is most effective in the simulation, experiments are performed, where room air temperature and floor surface condensation are controlled, with the condition of sensible and latent heat loads changed. In the results, the stability of the control of room air temperature is examined, and the changes of control variables related to floor surface condensation is analyzed.

### **Test model and equipment**

Test room is built with the same K-value and the same floor structure as of an existing apartment building, which has the dimensions of 2.4m x 2.4m x 2.2m (see Fig. 4). In supplying the cold water of constant temperature for radiant floor cooling, an ice storage tank is used as cooling source, and the temperature of the cold water provided for the floor panel is readjusted as about 16.5°C by using plate-type heat exchanger. The water is supplied for the floor panel by outdoor reset with indoor feedback control according to outdoor temperature and the room air temperature of the model space.

In order to measure the vertical distribution of room temperature, 5 T-type thermocouples are installed at intervals of 50cm, at the height of 10cm above the floor. The height where the room air temperature is controlled is 1.1m, the height at which room temperature controllers are generally installed. In order to analyze the distribution of the floor surface temperature according to cold water supply, T-type thermocouples are installed at 12 points. To control the floor surface condensation, the temperature of the upper surface of the floor where cold water is supplied is also measured.

### **Experimental conditions and method**

Room air temperature is controlled by adjusting the three-way valve through outdoor reset with indoor feedback control. In order to prevent the condensation of the floor surface, the compressor and fan of the packaged air conditioner are made to

be controlled independently of the operation of the floor panel with a safety factor of 1°C. Also, to study the reaction to the load change, sensible and latent heat are generated as described in Fig. 5.

**Table 2 Results of experiment according to the internal load conditions**

	Sensible heat generation 30W			
	Room air temperature (°C)	Floor surface temperature (°C)	Room air Humidity (g/kg(DA))	Outdoor air temperature (°C)
AVG	25.9	23.4	13.6	25.5
SD	0.6	0.5	0.4	2.9
Max	27.6	24.4	14.2	29.5
Min	24.7	22.5	13.0	19.1
Sensible heat generation 60W				
AVG	26.5	23.6	13.7	26.2
SD	0.9	0.5	0.8	3.1
Max	28.2	24.4	15.9	30.5
Min	25.6	22.5	11.2	19.7
Sensible heat generation 60W & Latent heat generation 30W				
AVG	26.8	23.1	14.1	26.3
SD	0.8	0.6	0.4	3.2
Max	28.2	24.1	14.8	30.2
Min	25.1	22.2	12.9	19.9

## **EXPERIMENTAL ANALYSIS**

### **Stability of room air temperature control**

As a result of analyzing the stability of the room air temperature control according to the load changes in the radiant floor cooling system integrated with the humidification, the averages of the room air temperatures(AVG) are 25.9°C, 26.5°C, 26.8°C as in Table 2. When the sensible heat of 30W is generated in the space, relatively small standard deviation (SD) is yielded. On the other hand, as the load increases, the control deviation of the room air temperature increases by some 50%, whose reason is as follows: In the case where the internal heat gain increases during the period when air temperature rises above 30°C as in Fig.5, as a result of fixing the lower limit of the supply water temperature, the floor panel cannot respond to cooling load immediately, and the space is overheated exceeding the set-point temperature (26°C) by a maximum of 2°C. Unless the lower limit of supply water temperature is fixed, the function of controlling the room air temperature may be more stable for cooling. However, because discomfort may occur owing to the drop of the floor surface temperature, auxiliary cooling facilities should be installed according to the amount of internal load. Particularly in cases where a packaged air conditioner is utilized for the purpose of cooling and dehumidification, a packaged air conditioner will be applicable easily by modifying the control algorithm.

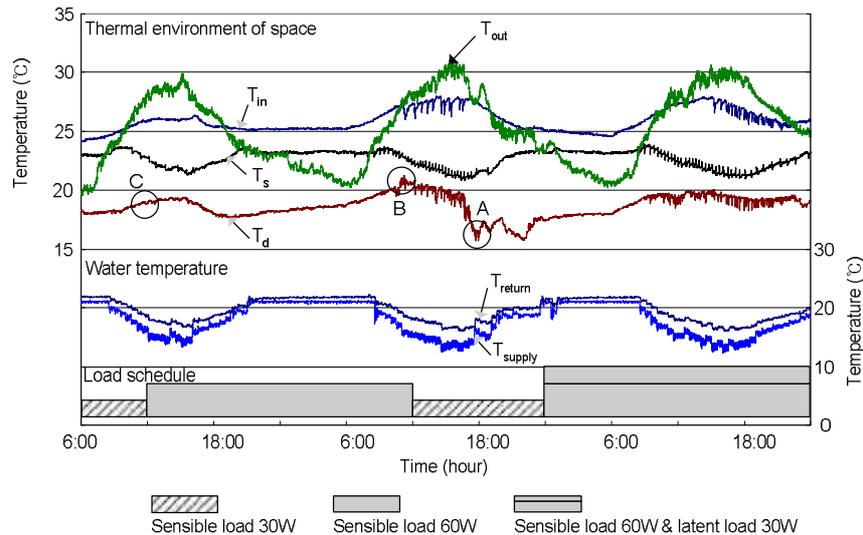


Fig. 5 Experimental results in room air temperature, floor surface temperature, dew point temperature, and supplied and returned water temperatures according to the load schedule.

In the case where internal heat gain during the nighttime is equal to the heat gain during daytime, despite the increasing internal heat gain of the room, overheating of the room does not occur, because the floor panel has been cooled down to about 22°C (see Fig. 5). So, it can be concluded that radiant floor cooling can be used properly in downtown apartments where the tropical night phenomenon frequently occurs, because of its characteristics of cooling storage.

#### Analysis of condensation control variables

In order to control the condensation occurrence on the floor surface, the dehumidification system is controlled by measuring the temperature of the upper floor where cold water is supplied and applying a safety factor (1°C) to room dew point. In the experiments on control methods, the floor surface temperature is kept at an average 23.5°C and at least 22.2°C in order to maintain the fixed room air temperature. The results show that the condensation on the floor surface does not occur, because the room dew point is controlled to an average 18.7°C and at most 21.2°C. These values are higher than those of simulation, because simulation results are obtained and analyzed throughout the whole cooling period. So, the average of the floor surface temperature is about 0.8°C less than the results of the simulation.

The vertical difference of the room air temperature is an average 2.1°C and maximum 2.9°C, as described in Table 3. Therefore local discomfort due to the vertical difference of room air temperature is not thought to be significant, because the vertical distribution is kept lower than the value proposed by the study on comfort (ASHRAE, 1992, Olesen, B.W., 2002).

Table 3 Difference of room air temperature, floor surface temperature, supplied and returned water temperature, and indoor air humidity

	Room air temperature difference (°C)	Floor surface temperature difference (°C)	Difference between supply and return water temperature (°C)	Indoor air humidity difference (g/kg(DA))
AVG	2.1	2.0	1.6	0.15
SD	0.4	0.7	0.9	0.18
MAX	2.9	3.2	6.2	1.9
MIN	1.3	0.8	0	0

The difference of indoor air humidity is an average 0.15g/kg(DA), maximum 1.9g/kg(DA), which makes a slight difference from the actual value(average 13.7g/kg(DA)). In the results, the indoor humidistat is assumed to be installed at the same height as of the existing thermostat.

The difference of floor surface temperature is an average 2.0°C, maximum 3.2°C, and a plot of this result with regard to the difference between supplied and returned water temperature is as Fig. 6. Local discomfort due to the drop of floor surface temperature is not thought to occur, because the overall distribution of floor surface temperature is kept higher than the minimum 22°C, if the condensation occurrence is controlled by the lowest floor surface temperature and room dew point. As regards the floor panel design, the acceptable difference in supplied water is a maximum 3~4°C, taking account of permissible temperature differences in the refrigerator and heat loss in pipes. If the difference exceeds the maximum, the amount of supplied water, and the sizes of the pipes and pumps applied for heating should be reexamined.

In case A of Fig.7, the dew point temperature in the model space has the lowest value (15.8°C). In this

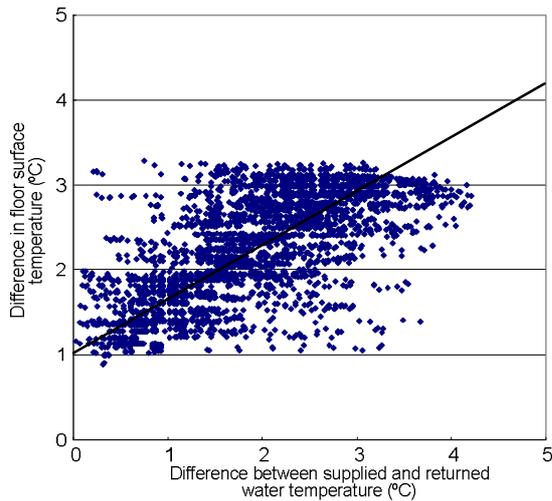


Fig. 6 Difference in floor surface temperatures according to the  $\Delta T(T_{\text{return}} - T_{\text{supply}})$ .

case, the absolute humidity in the model space decreases due to the operation of the dehumidification system in the previous time step. Thus floor surface temperature is maintained at 22.9°C owing to the time lag caused by thermal storage effect. In case B, the dew point temperature has the average value (18.2°C), and there is no operation of the dehumidification system. The room air temperature is kept stable at 26.3°C, with the floor surface temperature at 23.4°C. In case C, the dew point temperature has the highest value (21.1°C). Even though the room air temperature and floor surface temperature are kept high at 27.5°C and average 24.1°C respectively, the dehumidification system is operated because the floor surface temperature of the pipe inlet part goes below minimum 22.1°C. In other words, while radiant floor cooling controls room air temperature with about two hours of time lag, the dehumidification system has little influence on the control of room air temperature due to its rapid control over condensation according to the difference between dew point temperature and the temperature of the upper floor where water is supplied. The cooling and humidification system is thought not to bring about over-cooling, because the dehumidification system is operated when room air temperature is higher than the set-point temperature.

## CONCLUSIONS

In this study, to apply the dehumidification system to the radiant floor cooling system, control factors and methods are analyzed and suitable control methods are selected through simulations and the applicability of control methods is estimated through model test.

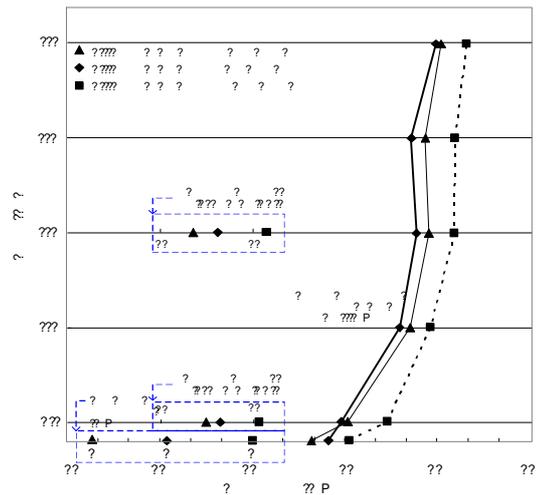


Fig. 7 Profiles in vertical room temperature, floor surface temperature, dew point temperature, and room air humidity.

Summarized results of this study are as follows.

- (1) Condensation occurrence on a floor surface is controlled by applying the dehumidification system to the radiant floor cooling, deciding whether to operate the system or not according to the floor surface temperature and the dew point temperature of room, and regulating the humidity of the air supplied to room.
- (2) In the case of combining the outdoor reset with indoor feedback control and the dehumidification system, the control deviation of indoor temperature is lower by 0.9°C and the control deviation of floor surface temperature is lower by 0.4°C compared with the on/off control. Moreover, the operating time of the dehumidification system for the control condensation decreases to one half.
- (3) As a result of model experiments that applied the dehumidification system to the radiant floor cooling system, the room temperature is kept at average 25.9~26.8°C. However, overheating of about 2°C occurs, by setting the minimum of the supply water temperature. Considering the local discomfort caused by a decrease in the floor surface temperature, it is needed to establish an auxiliary cooling device. When a packaged air conditioner (PAC) is used as a dehumidification system, correcting control algorithm is helpful for the easier application. And, by time lag characteristics according to the cooling of the floor panel, the radiant floor cooling is appropriate for the application to an urban apartment house in which tropical night phenomenon occurs frequently.

(4) The vertical difference in room air temperature is roughly 2.1°C average, 2.9°C maximum. Therefore, local discomfort is not significant according to existing comfort standards. And, it is possible to install the indoor humidity sensor at the height of the existing room temperature controller, because vertical differences in the absolute humidity are small (0.15g/kg(DA)). The deviation of floor surface temperature is 2.0°C average, 3.2°C maximum. Because the temperature is kept above at least 22°C, discomfort due to the decrease of floor surface temperature does not occur. Also, overcooling due to dehumidification system does not occur either, because a time delay is created in controlling room air temperature and condensation.

### **ACKNOWLEDGEMENTS**

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### **NOMENCLATURE**

AVG : average value  
 DA : dry air  
 DF : control differential  
 DH<sub>on</sub> : operation of dehumidification system  
 I : irradiation  
 Q<sub>lat</sub> : latent load  
 Q<sub>sen</sub> : sensible load  
 RFC<sub>on</sub> : operation of the radiant floor cooling  
 SD : standard deviation  
 SF : safety factor  
 T<sub>d</sub> : dew-point temperature  
 T<sub>in</sub> : indoor air temperature  
 T<sub>out</sub> : outdoor temperature  
 T<sub>return</sub> : returned water temperature in floor cooling  
 T<sub>s</sub> : floor surface temperature  
 T<sub>supply</sub> : supplied water temperature in floor cooling  
 W<sub>in</sub> : absolute humidity  
 W<sub>out</sub> : absolute humidity in outside air  
 T<sub>set</sub> : set point temperature in space  
 W<sub>supply</sub> : absolute humidity passed through the dehumidifying coil