Cooling and Heating performance of Ceiling Radiant Textile Air Conditioning System with PAC

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ABSTRACT

There are two types of air conditioning systems, convective and radiant air conditioning system. Radiant air conditioning systems have attracted many people's interest due to the high capabilities of energy saving and maintaining a comfortable indoor environment at the same time. However, it is difficult to install this system and maintain system performance. There is also a problem about the drought from convective air conditioning system. In this study, the authors suggest a new radiant air conditioning system, ceiling radiant textile air conditioning system with package air conditioner (Package Air Conditioner). In a room with PAC installed, we introduced the textile layer covering the ceiling below the PAC. By using existing PAC, the complexity of installation and the chance of having drought will become lower compared to the typical air conditioning system. The aim of this study is to investigate the thermal environment when operating the new radiant system and to develop an optimal method of using this system. This system controls the indoor environment utilizing the radiant effect of textile and the airflow through textile. In order to clarify the actual phenomenon, we conducted several experiments under 6 conditions, changing the supply airflow angle, pre-set temperature of PAC, airflow rate from PAC or heat generation rate. We also set up a "guide" below the inlet of PAC which prevents supply air from being drawn into the inlet of PAC and is made of non-flammable corrugated carton. Moreover, in order to increase the airflow rate through textile, we cut the part of textile under the guide and made an opening. Experiments in a total of 18 cases were carried out and compared under three conditions, "no guide", "guide on textile" and "guide with opening", combined with previous 6 conditions, thus we conducted the experiment in 18 cases.
In this study, the temperature, concentration of carbon dioxide as a tracer gas, and heat transfer rate by radiation were measured. Airflow rate through textile is calculated by tracer gas method. The heat transfer was also modelled considering exchange air through membrane. A new coefficient to quantify the heat transfer was introduced and performed by experiment. In addition, the representative temperatures of experimental room were calculated using the new coefficient and compared with the values obtained from the full-scale experiment which was reported in our previous study.

KEYWORDS

Textile, Radiant Air Conditioning, Distribution of Temperature, Exchange Air, Tracer Gas Method, Modelling of Heat Transfer

1 INTRODUCTION

The air conditioning system is generally classified into two types; one is convective and the other is radiant air conditioning system. In Japan, the convection air-conditioning system such as package air conditioner(PAC) has been the most commonly applied to office buildings. However, there is still problem about the drought from diffusers, and additional flaps are often attached to avoid it. Radiant air conditioning systems have attracted many academic interests over the last several decades due to its high capabilities of energy savings, thermal comfort achieved by radiation, quiet operation, and saving space *[1]*. However, it is difficult to install this system and maintain system performance, especially for a water cooling system.
Therefore, it should be an important matter to develop the technology to improve productivity. To solve these problems, the authors suggest a new radiant air conditioning system, ceiling

Nomenclatures	
C: concentration [-]	cp: specific heat capacity [W・h/kg・K]
Q: airflow rate [m^3/h]	q: heat quantity [W]
M: CO_2 emission rate [m^3/h]	S: area[m^2]
V: Volume [m^3]	v: face velocity [mm/s]
α_{cm}: heat transfer coefficient of textile [W/m^2K]	Δt: Measurement Interval [h]
θ: temperature [°C]	ρ: density of air [kg/m^3]
-Subscript-	
a: attic	*a*': through textile from attic
af: airflow	*back*: back side
c: convection	*ce*: ceiling
cu: ceiling upside space	*cm*: textile and air through textile
co: conduction	*e*: additional experiment
f: floor	*front*: front side
h: heat source	*i*: indoor space
i': through textile from indoor space	*j*: number of data collecting
l: lighting	*n:* number of walls
o: outer space	*op*: optimal
PAC: Package Air Conditioner	*r*: radiation
T: textile	*T1*: the part of textile (downward airflow area)
T2: the part of textile (upward airflow area)	*tr*: heat transfer
v: face velocity	*W*:wall (Indoor space)
W': wall (Attic)	

radiant textile air conditioning system with PAC installed in a room (see Figure 1). Compared to the typical air conditioning system, the drought from new radiant air conditioning system will be smaller. It can provide thermal comfort by radiant effect from cooling/heating textile, and airflow through the porous textile.

In order to clarify the actual phenomenon under the new radiant air conditioning system with textile, we conducted several full-scale experiments to measure the temperature, concentration of carbon dioxide and heat transfer rate by radiation. Moreover, in order to understand the indoor thermal environment generated by this system, a simplified calculation model is proposed where heat transfer through textile is considered. A new coefficient of textile to quantify the heat transfer was introduced, and determined by additional experiment in this study.

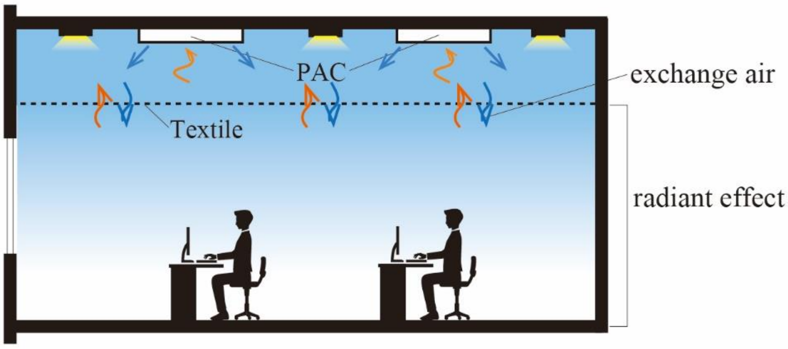


Figure 1: Outline of Ceiling Radiant Textile Air Conditioning System with PAC

2 FULL SCALE EXPERIMENT

2.1 Experiment Room

The full-scale experiments were performed from November 23, 2017 to February 5, 2018 in a laboratory located at Osaka University in Japan (see Figure 2), which assumed cooling experiment in summer. The dimensions of the laboratory are 7.00m (d) × 6.13 m (w) × 3.44 m (h). This laboratory was divided it into 2 spaces, the test room (Indoor space & Attic) and outer space. The partition wall consisted of glass wool (50mm-thick) and plaster board (12.5mm-thick), and its surface was covered by aluminium sheets to avoid radiation. The dimensions of the test room are 4.01m (d) × 4.41 m (w) × 3.44 m (h). The textile was suspended at FL+3.14m. The porosity of the textile was 2.6%. In addition, two ceiling cassette type air conditioners which had 4 diffusers were installed.
All the heat sources were located in the test room, consisting of 4 black lamps (53W×4) at FL+0.6m assuming sitting human, 4 light bulbs (100W×4) as apparatus heat load and 4 carpets as lighting heat load (50W×4). The total heat generation rate was 812W, except for Case2 (low heat load) and Case6 (high heat load). In Case2, the apparatus heat load wasn't located and total heat generation was 412W. In Case6, eight black lamps (53W×8) were added and the total heat generation rate was increased to 1,236W.

Vertical air temperature profiles were measured at five points of P1-P5s at six heights of 0.1, 1.1, 1.7, 2.4, 3.0 and 3.29m above the floor, and wall surface temperature profiles were measured at 12 points at three heights on back and front side. Vertical CO_2 concentrations profiles were also measured at the points of P1-P5s at the heights of 0.1, 1.1, 1.7 and 2.4 m. Additionally, CO_2 recorders were located at north, east and south in outer space. These measurement points were shown in Figure 2.

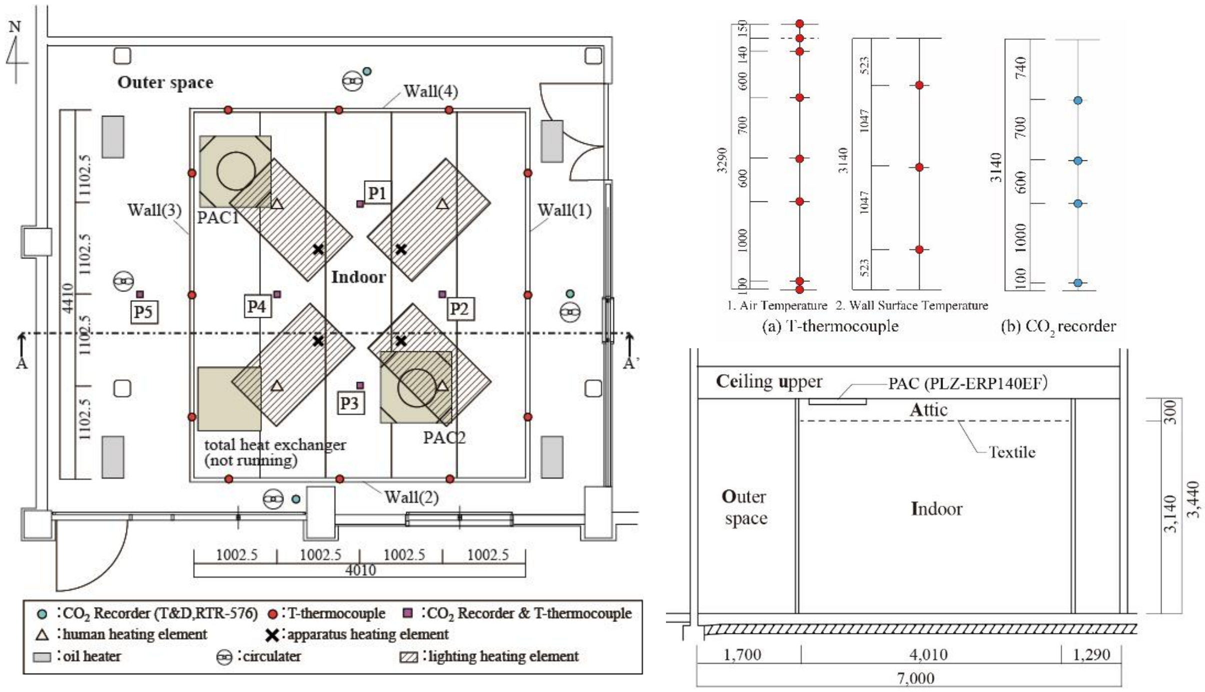


Figure 2: Plan of Experiment Room and A-A' Elevation and Measurement Points

2.2 Experimental Conditions and Method

The application of this new air conditioning system has not used yet in Japan. Thus, in order to clarify the actual phenomenon under this system being used, several experiments were conducted under 6 conditions, by changing the supply airflow angle (0° means the normal direction to ceiling), pre-set temperature of PACs, airflow rate from PACs and heat generation rate as shown in Table 1.

A case with "guide" provided for the inlet of PACs was also studied, which prevents the short-circuiting of supply air (Figure 3(b)). The guide was made of non-flammable corrugated carton (Figure 3(d)) and it is easy to process and use in construction sites. In addition, another case with the guide was investigated where the part of textile was cut under the guide to make an opening to increase the airflow rate through textile (Figure 3(c)). Thus, three conditions of guide, "no guide", "guide on textile" and "guide with opening" were studied, and combined with above-mentioned 6 conditions. Consequently, the experiments were conducted under 18 conditions in total.

Because the experiment was conducted in winter, oil heaters were used to increase the temperature in the laboratory and to simulate the outdoor condition for cooling (Figure 2) in “guide on textile” and “guide with opening” conditions.

Figure 3: Conditions of Textile and Photo of Guide

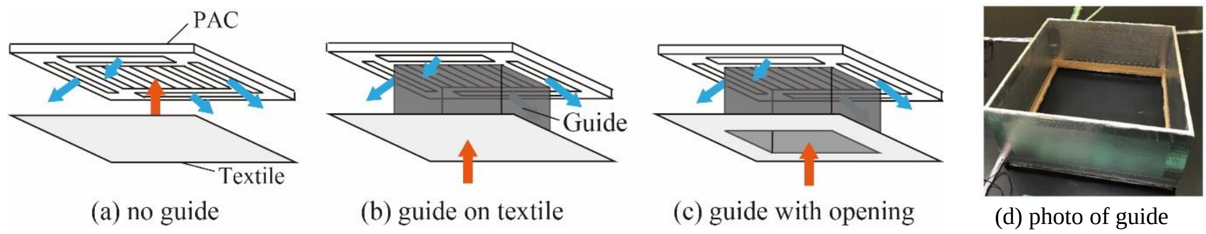


Table 1: Experimental Conditions

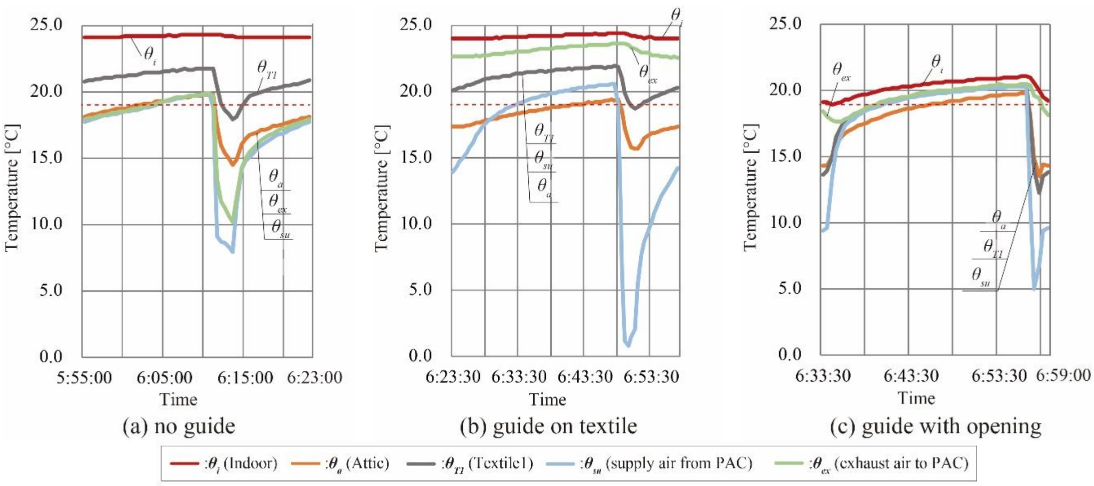
<table>
<tr><th rowspan="2">Condition</th><th rowspan="2" colspan="2">Pre-set indoor Temperature [°C]</th><th rowspan="2">Airflow Rate of PAC [m^3/h]</th><th rowspan="2">Angle of Supply [°]</th><th rowspan="2">Total Heat Generation [W]</th><th colspan="3">Oil Heater Pre-set Temperature [°C]</th></tr>
<tr><th>(a)</th><th>(b)</th><th>(c)</th></tr>
<tr><td>Case1 : Normal</td><td rowspan="3" colspan="2">19</td><td>1,016×2</td><td rowspan="4">60</td><td>812</td><td rowspan="6">-</td><td rowspan="3">14</td><td rowspan="6">18</td></tr>
<tr><td>Case2 : Low heat load</td><td>717×2</td><td>412</td></tr>
<tr><td>Case3 :Small flow rate of PAC</td><td rowspan="4">1,016×2</td><td rowspan="3">812</td></tr>
<tr><td>Case4 :High temperature</td><td>(a)25</td><td>(b)(c)22</td><td>18</td></tr>
<tr><td>Case5 :30°supply</td><td rowspan="2" colspan="2">19</td><td>30</td><td rowspan="2">14</td></tr>
<tr><td>Case6 :High heat load</td><td>60</td><td>1,236</td></tr>
</table>

The first step in the experimental procedure was to run PACs and start temperature measurement. After indoor thermal environment became a steady state (after 19 hours), CO_2 gas emitted at the inlet of PACs. The emission rate of CO_2 was set at 1.0L/min by using a mass flow controller. The CO_2 concentration was measured for 3 hours to investigate the airflow rate through textile.

2.3 Evaluation Index and Results

The PACs installed in the test room were ON-OFF control apparatus, and their sensors were located above the inlet of PACs (body thermostat). Under cooling condition, when the temperature at the sensor became 0.5°C higher than pre-set temperature, it would be turned on and when it became 1.5°C lower than pre-set temperature, it would be turned off. Since the repeated on/off operation was observed throughout the experiment, the average temperature during 1 cycle (Figure 4) was evaluated to assess the thermal indoor environment of this system. In the case of "no guide" condition, the indoor space temperature (θ_i) and its variation were totally different from those of the PAC inlet (θ_{ex}). On the contrary, the inlet temperature (θ_{ex}) is almost the same as supply temperature (θ_{su}), which indicates the short-circuit of the supplied airflow within the attic. On the other hand, in the cases of "guide on textile" and "guide with opening" conditions, variation of θ_i and θ_{ex} were similar, thus the guides prevented supply air from being drawn directly into the inlet of PACs in the attic.

Figure 4: Change of Temperature in a Cycle (Case1)



Vertical temperature distributions are shown for each case in Figure 5. In all cases, the temperature difference between FL+100mm and FL+1700mm is less than 3°C, which meets the recommendation in ASHRAE 55 *[2]*. Additionally, the result shows that cooling air from attic could reach to the floor.

By comparing studied cases, in "no guide" and "guide on textile" conditions, it was shown that the heat generation rate in indoor space affected air temperature. On the other hand, in the cases of "guide with opening" condition, heat generation rate didn't affect air temperature, but the pre-set temperature of PAC affected air temperature, and temperature was close to pre-set temperature of PAC. Thus, the cooling effect was biggest in the cases of "guide with opening" condition.

As for radiant effect, it is shown that surface temperature on textile is lowest in "guide with opening" condition, and the average of temperature difference between the textile surface and room air is 3.3°C in "no guide" condition, 3.4°C in "guide on textile" condition, and 1.6°C in "guide with opening" condition. Thus, radiant effect is larger in the cases of "no guide" and "guide on textile" conditions.

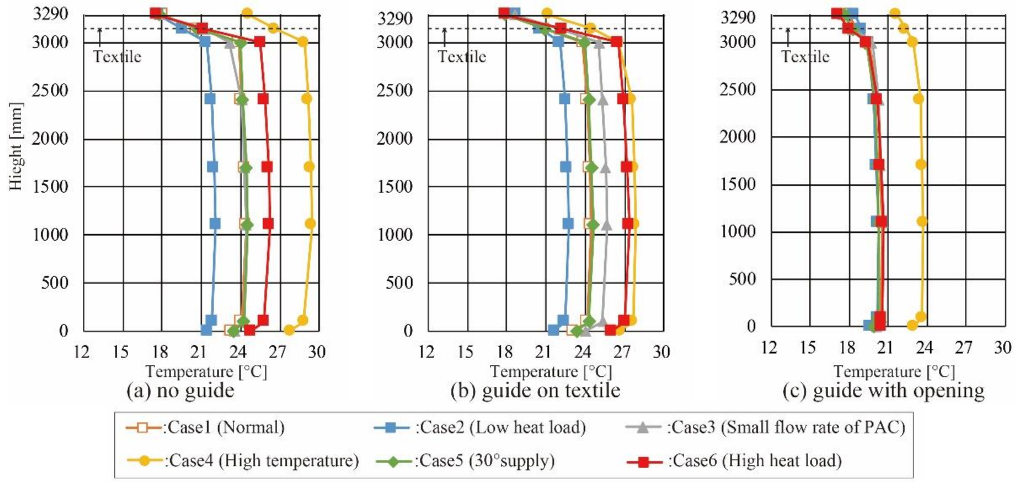


Figure 5: Vertical Temperature Distribution

Considering the airflow of tracer gas, the exchange airflow rate through textile can be estimated based on the following equations:

$$C_a^{n+1} = C_a^n + (C_i^n Q_2 + M - C_a^n Q_1 + C_o^n Q_5 - C_a^n Q_6)\frac{\Delta t}{V_a} \quad (1)$$

$$C_i^{n+1} = C_i^n + (C_a^n Q_1 + C_o^n Q_4 - C_i^n Q_3 - C_i^n Q_2)\frac{\Delta t}{V_i} \quad (2)$$

$$Q_1 - Q_2 - Q_3 + Q_4 = 0 \quad (3)$$

$$-Q_1 + Q_2 + Q_5 - Q_6 = 0 \quad (4)$$

Eq. (1) and Eq. (2) indicates the balance of CO_2 concentration in attic and indoor space respectively. Eq. (3) and Eq. (4) indicates the balance of exchange airflow rate. The schematic diagram was shown in Figure 6.

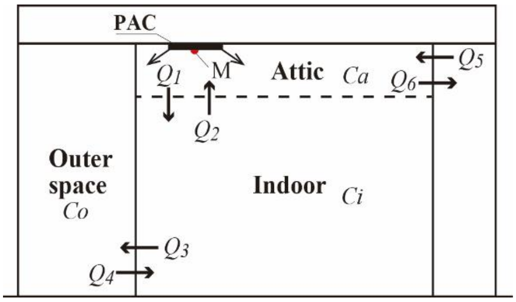


Figure 6: Schematic Diagram of Exchange Airflow Rate

Table 2 shows the results of estimated exchange airflow rate (Q_1 (downward) and Q_2 (upward) [m^3/h]). It can be seen that they are doubled when using guide, and increased by five times when the opening was provided below the inlet. The effect of opening at the bottom of the guides was larger than that of only using guide because the pressure loss across the textile was significant and its supply airflow rate decreases due to the resistance of textile. In cases of "guide with opening" condition, the convectional cooling effect is larger than other conditions. In order to understand how the air density difference caused by the temperature difference affects exchange airflow rate, the correlation between airflow rate and temperature difference (between air in attic and indoor space) was analysed, except for Case3 (because airflow rate was depended on PAC airflow rate in Case3). The temperature difference between attic and indoor space ($\Delta\theta$) [°C] was also shown in Table 2.
In "no guide" condition, the correlation coefficient between airflow rate and temperature difference was 0.91 (Q_1) and 0.88 (Q_2), which shows that the correlation between exchange airflow rate and temperature difference is stronger for "no guide" condition, compared to the other conditions with that resulted in smaller correlation coefficient.

Table 2: Temperature Difference ($\Delta\theta$) [°C] and Airflow Rate through Textile [m^3/h]

	(a)no guide			(b) guide on textile			(c) guide with opening		
	Δt	Q_1	Q_2	Δt	Q_1	Q_2	Δt	Q_1	Q_2
Case1	6.3	90.3	116.0	6.0	156.1	182.9	2.4	986.9	1016.1
Case2	3.8	79.9	94.7	4.0	154.8	182.3	1.5	990.6	1019.0
Case3	5.0	78.6	101.0	6.9	111.4	132.8	2.9	811.5	850.4
Case4	4.7	92.2	106.3	6.3	147.8	178.5	1.9	958.0	1008.8
Case5	6.5	85.9	106.4	6.3	144.2	175.8	2.3	975.3	1043.5
Case6	8.2	91.8	120.1	9.2	155.7	187.6	3.0	972.0	1018.7

3 MODELLING OF HEAT TRANSFER

In order to grasp the indoor thermal environment controlled by the air conditioning system with textile more easily, heat transfer was modelled considering exchange air through textile, based on the result of full-scale experiment. The model is developed so that it is used for the parametric study in the practical design phase when this system is applied to an office building.

3.1 Numerical Method

The heat balance equations regarding air temperature in the attic and indoor space, and surface temperature on textile1 (downward airflow was generated) and textile2 (upward airflow was generated) at steady-state are:

$$0 = q_{c(l->a)} + q_{c(T->a)} + q_{PAC} + q_{af(i->a)} + q_{tr(o->a)} + q_{tr(cu->a)} \tag{5}$$

$$0 = q_{c(h->i)} + q_{c(f->i)} + \sum_{n=1}^{4} q_{c(W_n->T2)} + q_{c(T1->i)} + q_{c(T2->i)} + q_{af(a->i)} + q_{af(o->i)} + q_{co(o->i)} \tag{6}$$

$$0 = q_{r(l->T1)} + q_{r(W'->T1)} + q_{r(ce->T1)} + q_{r(h->T1)} + q_{r(f->T1)} + \sum_{n=1}^{4} q_{r(W_n->T1)} + q_{T1} \tag{7}$$

$$0 = q_{r(h->T2)} + q_{r(f->T2)} + \sum_{n=1}^{4} q_{r(W_n->T2)} + q_{T2} \tag{8}$$

To express the heat transfer from air to textile, a new coefficient α_{cm} *[3]* was introduced and the heat transfer rate is calculated by these equations:

$$q_{T1} = \alpha_{cm} \cdot S_{T1}(\theta_a - \theta_{T1}) \tag{9}$$

$$q_{T2} = \alpha_{cm} \cdot S_{T2}(\theta_i - \theta_{T2}) \tag{10}$$

To calculate the temperature of air immediately after passing through textile, the following equations were used:

$$\theta_{i'} = \theta_i + \frac{S_{T2} \cdot \alpha_{cm}(\theta_{T2} - \theta_i)}{c_p \rho \cdot Q_2} \tag{11}$$

$$\theta_{a'} = \theta_a + \frac{S_{T1} \cdot \alpha_{cm}(\theta_{T1} - \theta_a)}{c_p \rho \cdot Q_1} \tag{12}$$

However, in full-scale experiment, the surface temperature on ceiling and wall in the attic was assumed to be the same as air temperature in the attic in this study, due to the difficulty of measuring the temperatures at these positions. By using heat transfer coefficients shown in Table 3, convective and radiative heat transfer rates were calculated.

Table 3: Heat transfer Coefficient used in the Calculation Model

Radiation		α_r=5[W/m^{2}K]
Convection	vertical plane	α_{c1} =4.5[W/m^{2}K]
	horizontal plane	(larger) α_{c2} =6.7[W/m^{2}K] (smaller) α_{c3} =1.7[W/m^{2}K]
Conduction (wall)		(indoor) $\alpha_{co\text{-}W}$ =0.655[W/mK] (attic) $\alpha_{co\text{-}W'}$ =0.5[W/mK]
Heat Transfer		(ceiling-upper) $\alpha_{tr\text{-}cu}$ = 4.12[W/m^{2}K]

For textile2 (upward airflow area), it was assumed that there was no convective heat transfer between textile2 and air in the attic in “guide on textile” condition, and there’s no textile2 portion in “guide with opening” condition. Schematic diagram of calculation model is shown in Figure 7. Details of heat transfer at the textile opening were illustrated in Figure. 8. In each case, α_{cm} was obtained using the least square method based on measured temperature, which is here expressed as $\alpha_{cm\text{-}op}$. However, in the case of “no guide” condition, the error between calculated temperature and measurement was quite large. Therefore, this paper only shows the results of other 2 conditions.

The results (Table 4) shows that $\alpha_{cm\text{-}op}$ was different between two conditions, “guide on textile” and “guide with opening” condition. Thus, the additional experiment was conducted to explore the heat transfer characteristics of the textile.

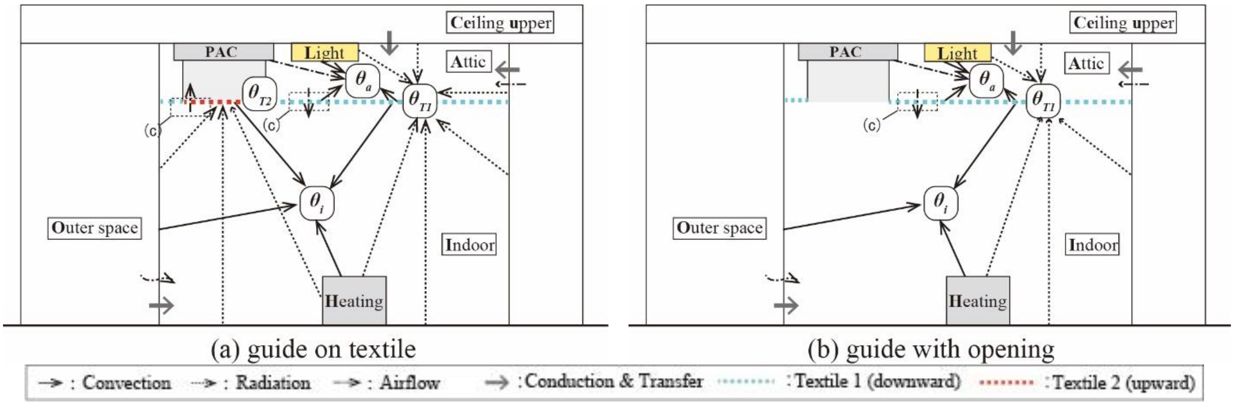


Figure 7: Modelling of Heat Transfer of this System

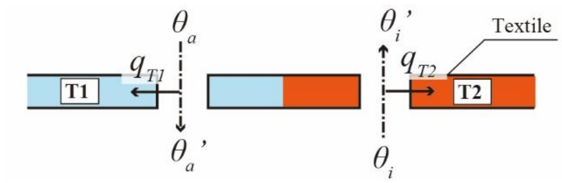


Figure 8: Modelling of Heat transfer (details of textile)

3.2 Additional Experiment

An additional experiment was conducted in May, 2018 in an experimental room which is also located at Osaka University in Japan. The airflow regulated by air mini pump flew into a cooling box, and after cooled, it flew into the measuring object. The textile was attached to the top of the measurement box, shown in Figure 9. 4 conditions of cooling box, changing the air temperature difference before and after passing the textile, and 10 conditions of airflow rate (2, 3, 4, 5, 15, 16, 17, 18, 19, 20 L/min) were set for this experiment. The α_{cm} was obtained by following equations:

$$\alpha_{cm-e} = \frac{c_p \rho \cdot Q_{1-e} \cdot (\theta'_a - \theta_a)}{S_{T1-e}(\theta_{T1-e} - \theta_a)} \tag{13}$$

where, θ_{T1-e} is calculated based on the following two equations:

$$\theta_{T1-e} = \frac{\theta_{T1-e-front} + \theta_{T1-e-back}}{2} \tag{14a}$$

$$\theta_{T1-e} = \theta_{T1-e-front} \tag{14b}$$

The correlations between face velocity and α_{cm} for all cases are shown in Figure 10. Airflow rate divides by the area of textile makes face velocity. It shows that two cases (using Eq. (14a) and (14b)) seem to fall on a straight line and it may be possible to predict α_{cm} by a linear function of face velocity (Eq. (14a) and (14b)), which is here expressed as $\alpha_{cm\text{-}v}$. This result was also confirmed in the previous study. *[3]*
To study the accuracy of the proposed calculation model, measured and calculated temperatures are compared, where two different values of α_{cm} are used, i.e., $\alpha_{cm\text{-}op}$ and $\alpha_{cm\text{-}v}$.
In this model calculation, linear correlation obtained from Eq. (14b) was used to give $\alpha_{cm\text{-}v}$, because only the temperature on the front surface of textile (high temperature) was measured

in the full-scale experiment, and Eq. (14a) cannot be applied to simulate the experiment by this model. Thus, $\alpha_{cm\text{-}v}$ is based on Eq. (14b) and given as;

$$\alpha_{cm-v} = 1.01 \times v + 0.13 \tag{15}$$

The α_{cm-v} obtained for each case is summarized in Table 4.

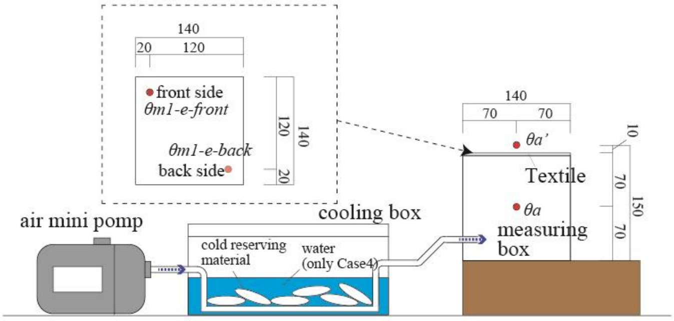


Figure 9: Outline of Additional Experiment

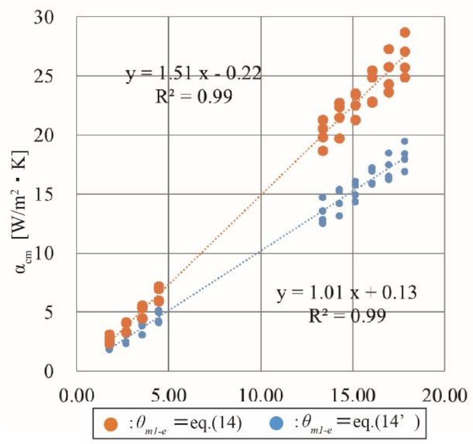


Figure 10 Correlation between Face Velocity and α_{cm}

Table 4: The Value of $\alpha_{cm\text{-}op}$ and $\alpha_{cm\text{-}v}$ in each case [W/m^2・K]

Condition	(a) guide on textile		(b) guide with opening	
	$\alpha_{cm\text{-}op}$	$\alpha_{cm\text{-}v}$	$\alpha_{cm\text{-}op}$	$\alpha_{cm\text{-}v}$
Case1 : Normal	7	2.71	16	16.16
Case2 : Low heat load	7	2.69	12	16.51
Case3 :Small flow rate of PAC	6	1.97	13	13.55
Case4 :High temperature	5	2.57	9	15.97
Case5 :30°supply	6	2.52	10	16.26
Case6 :High heat load	6	2.71	16	16.21
Average	**6.17**	**2.53**	**12.67**	**15.78**

3.3 Comparison between Measured and Calculated Temperature

Using α_{cm-v} and α_{cm-v} (Table 4), θ_i (air temperature in indoor space), θ_{T1} (surface temperature on textile1) and θ_a(air temperature in attic) were calculated and compared with the measured temperatures to examine the accuracy of the model.

The measured temperature vs calculated temperature was plotted and shown in Figure 11. The error between these two temperatures was less than 1.0 °C for all cases. However, the error in "guide on textile" condition using $\alpha_{cm\text{-}v}$ was larger than other cases. It seems that the difference between $\alpha_{cm\text{-}op}$ and $\alpha_{cm\text{-}v}$ was larger, and α_{cm} had an impact on temperatures in this condition.

4 CONCLUSIONS

To solve the problems of convective and radiative air conditioning systems, the authors proposed a new air conditioning system, "Ceiling Radiant Textile Air Conditioning System with PAC". The full-scale experiment was conducted and a heat transfer model was proposed which enables the investigation of indoor thermal environment for this system. The main findings are summarized as follows:

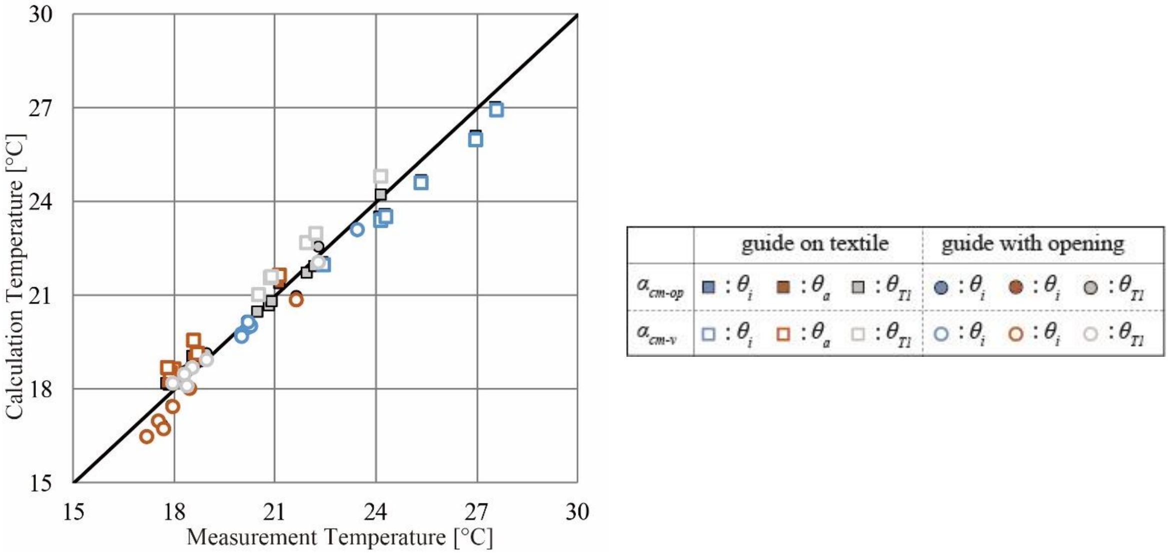


Figure 11: Accuracy of the Model

- Vertical distribution of temperature is uniform and appropriate indoor thermal environment is generated by the new radiant air conditioning system.
- In the cases of “guide with opening” condition, the room temperature well agreed with pre-set temperature of PAC.
- In the case of “no guide” condition, the exchange airflow rate through textile significantly decreased, and airflow rate was doubled when using guide, and was increased by five times when an opening was provided below the inlet of PAC.
- There was positive correlation between exchange airflow rate and air temperature difference between attic and indoor space for “no guide” condition.
- Heat transfer coefficient of textile can be expressed by a linear equation of face velocity.
- The error between measured and calculated temperatures was less than 1°C for all cases.

There are problems for “no guide” condition, such as small exchange airflow rate, unsatisfied accuracy of the model due to the complex temperature distribution on textile. Thus, as future prospects, these problems are to be solved, e.g., using fan and increasing exchange airflow rate.

5 ACKNOWLEDGEMENT

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