Radiant Ceiling Panel (RCP) for heating/cooling application
Part 1. Experiment study on the cooling performance of RCP with a novel curved surface under different operating condition

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This paper experimentally studies the heat transfer performance and assesses the thermal comfort conditions inside a laboratory scale room equipped with a novel suspended radiant ceiling panel with curve shape surface under different operating conditions. The result shows that the radiation heat transfer cover over 50 % of the total heat transfer and the convection heat transfer flux increases with the increment of the fluid inlet temperature. In addition, a more uniform air temperature distribution and comfortable indoor environment can be achieved at a higher inlet temperature and a lower flow rate.

Introduction

The radiant heating and cooling system(RHC system) are defined as the air-conditioning system in which radiant heating transfer covers more than 50 % of total heat transfer **[1]**. From the last several years, the application of the RHC system is becoming more and more popular in the world because of its characteristics of low energy consumption, high thermal comfort level, and high air quality **[2-3]**.

Although the RHC system has been approved to be more efficient and energy-saving than the traditional air-conditioning system, the cooling radiant ceiling panel (CRCP) has the disadvantage of dew condensation caused by low panel surface temperature.

The purpose of this paper is to study the cooling performance of a novel suspended metal RCP with a curved shape by operating experiments under different conditions. In the experiments, air temperature, mean radiant temperature, humidity, and air velocity in the test room are recorded to evaluate thermal comfort. Also, fluid flow rate, inlet and outlet temperature for each panel are recorded to calculate the cooling capacity.

1. Experimental set-up

1.1 Experimental Arrangement

(1) Panel arrangement

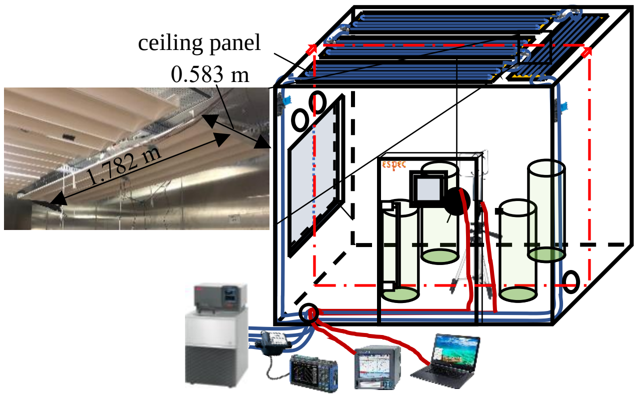

Figure 1. Schematic diagram of overall experiment set-up

The experiment set-up consists of twin identical environmental chambers with autonomous control. Each chamber has a dimension of length, width, and height of 2.7 m × 2.7 m × 2 m. Also, the two chambers are sharing a wall with a double-glazing window of dimensions 1.3 m × 0.9 m

Each chamber is well isolated, and the room temperature can be controlled independently. The overall arrangement of novel panels, internal load, measurement devices, and closed cooling water circuit system are shown in Fig.1. In the right chamber, four panels with a length of 1.782 m and width 0.583 m are hanged at 19 cm beneath the ceiling. The pump circulated the chilled water in a closed loop where the panel is connected in series mode.

(2) Internal load

In this experiment, the room is heated internally by four cylindrical heat-generating dummies. The cylinder is made from an aluminum sheet with a diameter of 20 cm and a length of 1.2 m, and it is painted with black color. The heat is generated by an incandescent bulb with a power of 42 W.

1.2 Case set-up

Table 1. Case set-up

Name	Toutside[°C]	Tinlet[°C]	Flow[L/min]
case1-1			3
case1-2	40	15	2.6
case1-3			2.1
case2-1			3
case2-2	40	18	2.6
case2-3			2.1
case3-1			3
case3-2	40	20	2.6
case3-3			2.1

Nine experiments are conducted with operating conditions as shown in Table 1.

1.3 Measurement set-up

(1) measurements for thermal comfort

The indoor air temperature is measured by 33 type-k thermocouples as shown in Fig.2. Each line carries 11 thermocouples with a 20 cm distance apart from top to bottom. Furthermore, each wall, floor, and ceiling have one measurement point to record wall temperature. Besides, each panel has 6 measurement points for recording the temperature of both the up and bottom surface. The indoor air relative humidity is measured by three relative humidity sensors. The air velocity at a local point is measured by a three-dimensional ultrasonic anemometer. The mean

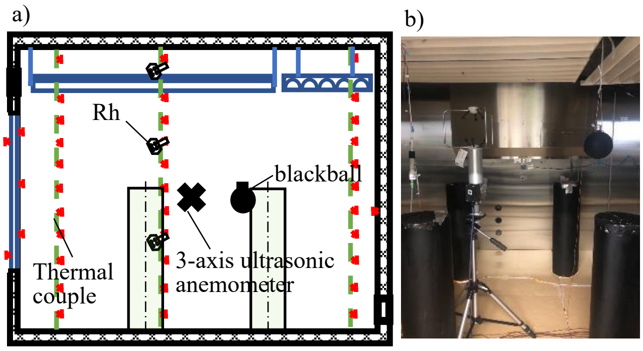

Figure 2. Measurement set-up a) Schematic diagram b) Interior view

radiant temperature (MRT) is measured by a blackball thermometer located at the mid-point.

(2) measurements for cooling capacity

Three thermocouples are measuring the temperature of the window inner surface lined in the window centerline. Also, two thermocouples are used to measure the window outer surface temperature. The fluid flow rate is measured by a compact magnetic flow sensor

2. Calculation methods

2.1 calculation method of PMV

Predicted Mean Vote (PMV) predicts the mean comfort response of a larger group of people, which is calculated from operative temperature, humidity, and the other four parameters. In addition, operative temperature (T_o) can be simply expressed as:

$$T_o = \frac{T_{mr}+T_{db}}{2} \tag{1}$$

Where, T_{mr} is the mean radiant temperature, T_{db} is the air temperature

2.2 calculation method of heat flux

The total heat flux (q_{tot}) consistent with convective heat flux (q_c) and radiant heat flux (q_r).

Radiant heat flux q_r is calculated by using the view factor of each surface F_{s-j} and radiation interchange factors $F_{\varepsilon_{s-j}}$ by Eq. (2)

$$q_r = \sigma \sum_{j=1}^{n} F_{\varepsilon_{s-j}}((T_j + 273.15)^4 - (T_s + 273.15)^4) \tag{2}$$

Radiation interchange factors $F_{\varepsilon_{s-j}}$ is calculated by Eq. (3). The view factor is obtained by using ANSYS Fluent 19.2

$$F_{\varepsilon_{s-j}} = \frac{1}{[\frac{1-\varepsilon_s}{\varepsilon_s}] + (\frac{1}{F_{s-j}}) + \frac{A_s}{A_j}[\frac{1-\varepsilon_j}{\varepsilon_j}]} \tag{3}$$

2.3 calculation method of cooling load

In the experiment, there are two main cooling loads in the room-the heat generated from the human body model and the heat escaped from the window to the left chamber. The heat transfer of the window is calculated by Eq. (4)

$$Q = \frac{A(T_{\infty 1} - T_{\infty 2})}{R} \tag{4}$$

Where R is the resistance to the movement of heat through the separating material, A is the area of window-

1.17 m^2

3. Result and discussion

3.1 Cooling capacity

The total heat transfer from the panel surface to/from the indoor air environment is the summation of the heat transferred by convection and radiation. Fig.3 presents the convection (q_c in W/m^2) and radiation heat (q_r in W/m^2) fluxes under different operating conditions. In all cases, radiation heat flux accounts for more than 50 % of the total heat flux (q in W/m^2), of 47 W/m^2. But, the ratio between the convection and total heat fluxes (q_c/q) increased significantly by 18 % from 22 % in case 1 to 40 % for case3.

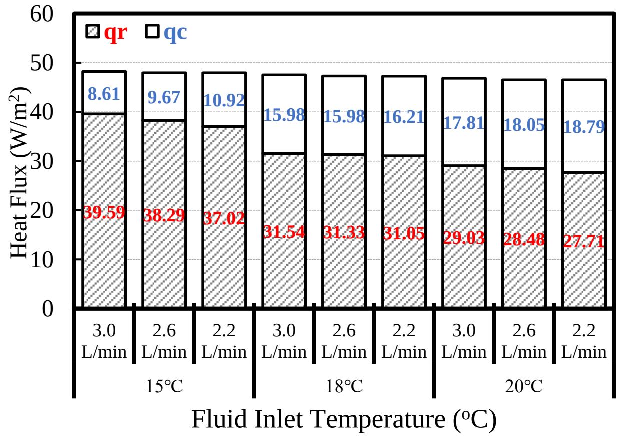

Figure 3. Convection and radiation heat fluxes

1) For the same fluid flow rate; as the inlet temperature increases, the panel surface temperature rises accordingly, hence the temperature difference decreases as well so the radiation heat transfer flux q_r decreases.

2) For the same fluid inlet temperature, increasing the flow rate also increases the heat absorbed by the flowing fluid which increases the panel surface temperature, as well as it deceases also the temperature difference between the panel surface and the An Uncooled Surfaces temperatures (AUST)

3.2 Heat transfer coefficient

The radiation and convection heat transfer coefficients (h_r and h_c) are calculated and the results are listed in Table 2. Increasing the inlet temperature has a significant impact on increasing the h_c, as the inlet temperature increases from 15 °C to 20 °C at a constant flow rate of 3 L/min, the h_c increases by 148 % from 1.532 W/m^2·K to 3.813 W/m^2·K. And the flow rate has a less impact, decreasing the fluid flow rate increases the convection heat transfer because of decreasing the hr.

Table 2. Convection and radiation heat transfer coefficients

	Inlet temperature	flow rate	h_r[-]	h_c[-]
Case1-1		3.0 L/min	6.07	1.532
Case1-2	15℃	2.6 L/min	5.81	1.686
Case1-3		2.2 L/min	5.42	1.745
Case2-1		3.0 L/min	4.90	3.199
Case2-2	18℃	2.6 L/min	4.62	3.026
Case2-3		2.2 L/min	5.01	3.134
Case3-1		3.0 L/min	5.35	3.813
Case3-2	20℃	2.6 L/min	4.79	3.811
Case3-3		2.2 L/min	5.12	3.839

3.3 Indoor air temperature profile

The air temperature distribution in the vertical direction is investigated by three lines at different locations, where 11 temperature sensors are attached to each line. Fig.4 illustrates the indoor air temperature profiles in each line. To quantify the air temperature distribution uniformity, the Standard Deviation (SD) in the measured air temperature along each line is calculated. Also, the temperature difference between uncle (Y=0.1 m) and head (Y=1.3 m) levels △T is calculated. The figure shows that the indoor temperature difference is quite small since △T is always less than 0.3°C. In case 2 with the inlet temperature of 18°C, △T is far smaller compared with other cases. Moreover, the SD value decreases as inlet temperature increases, but all of them fluctuate around 0.5. In general, a uniform indoor environment can be achieved by using this radiant pane

3.4 PMV

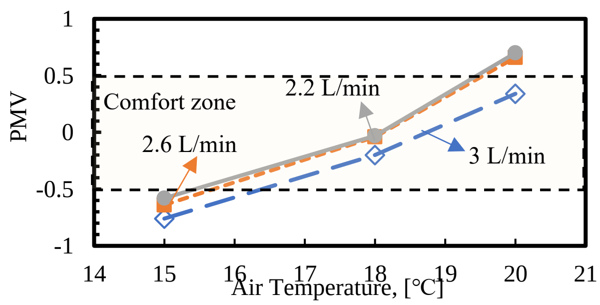

Figure 5. PMV at different condition

Predicted Mean Vote (PMV) equal to zero is representing thermal neutrality, and the PMV between -0.5 and 0.5 can be considered as a comfort zone. The calculated value of PMV is showed in Fig.5, it can be noticed that PMV remain very close to 0 value in the case with inlet temperature around 18ºC. In addition, the values have no obvious difference between the cases under the condition of approximate 2.2 L/min and 2.6 L/min inlet flow rate, but far from the result of the case using 3 L/min.

PMV value is mainly affected by operative temperature in this experiment because the difference of air velocity and humidity in each case is very small according to using the radiant cooling panel system without combining other air supply systems.

4. Conclusion

A comfortable indoor environment can be achieved at a higher inlet temperature and a lower flow rate. As a result, the application of the novel panel shows the potential of energy-saving.

Reference

1)Rhee KN, Olesen BW, Kim KW. Ten questions about radiant heating and cooling systems. Build Environ 2017;112:367–81.

2)Imanari T, Omori T, Bogaki K. Thermal comfort and energy consumption of the radiant ceiling panel system. Energy Build 1999;30:167–75.

3)Sastry G, Rumsey P. VAV vs. Radiant. ASHRAE J 2014;56:17–24.

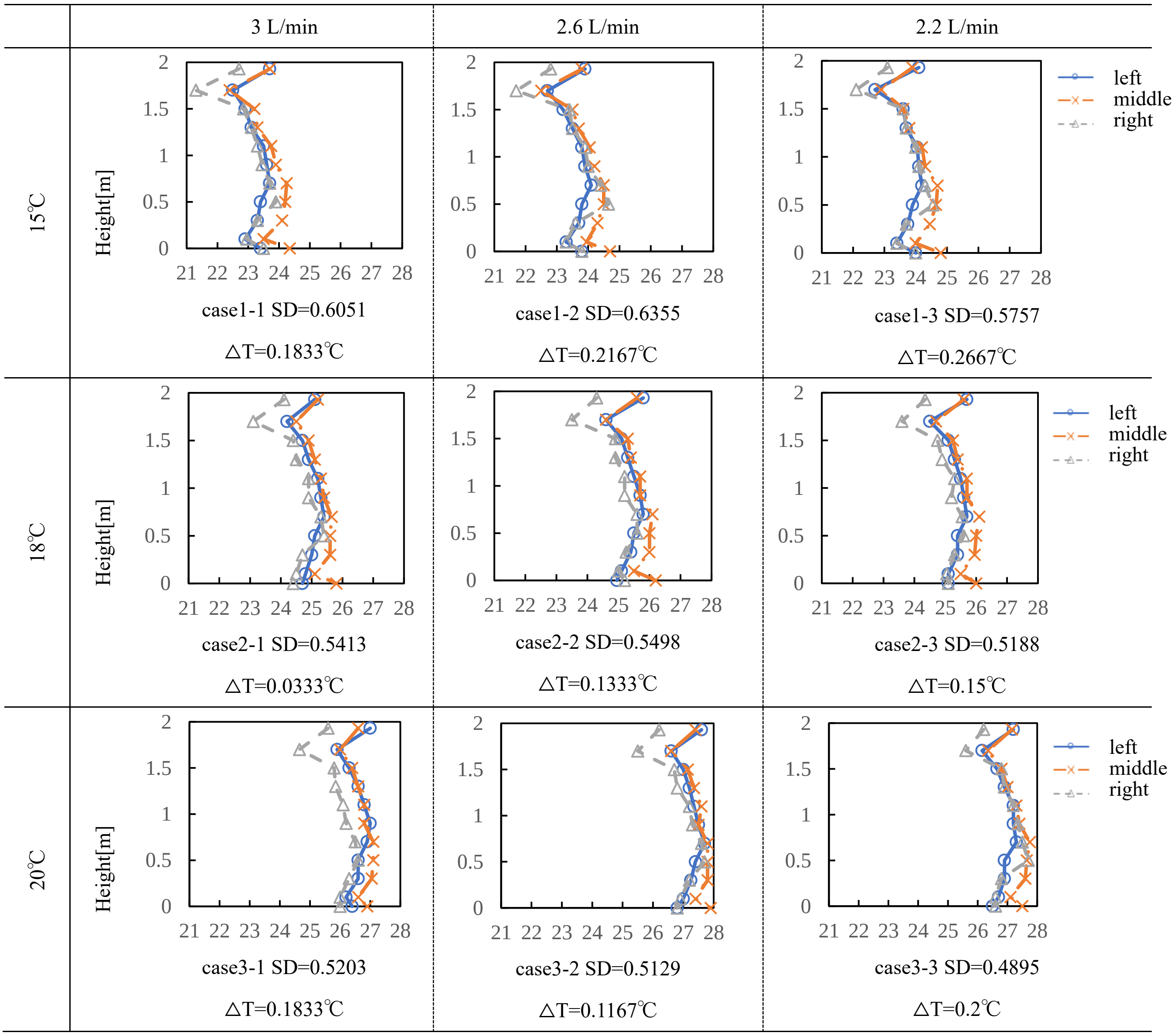

Figure 4. Temperature distribution profile