A study of the sensible and latent heat flux of a ceiling radiant cooling panel with superhydrophobic treatment

Ziwen Zhong [a], *Wei* Ma [b], *Jianlei* Niu [a], *Shuhuai* Yao [b].

[a] Department of Building Environment and Energy Engineering, The Hong Kong Polytechnic University, Hong Kong, China

[b] Department of Mechanical Engineering, The Hong Kong University of Science and Technology, Hong Kong, China

Abstract. Condensation is one of the major factors that limit the application of radiant cooling systems in hot and humid areas. The need for condensation control restricts the temperature difference between a panel surface and indoor spaces, limiting the cooling capacity of the panel. Previous studies indicated that condensation risks of a ceiling radiant cooling panel can be greatly mitigated by applying superhydrophobic surface materials, making a panel usable with a lower temperature even below air dew point. We performed a case study to show how the total heat flux of a ceiling radiant cooling panel with latent heat transfer can be enhanced compared with a panel with only sensible cooling. Based on empirical methods and heat and mass analogy, as indicated by a series of natural convection condensation heat transfer experiments for a ceiling positioning superhydrophobic aluminum surface showing the condensation heat transfer of a superhydrophobic surface can be well predicted by the method, we investigated both the sensible and latent heat flux of a panel placed in the air with a temperature of 25^{o}C and relative humidity between 40% and 70%. The case study shows an increment between 4% and 300% for the total heat flux of the panel compared with only sensible cooling under different humidity conditions.

Keywords. Radiant cooling system, heat flux, condensation, superhydrophobic surface materials.
DOI: https://doi.org/10.34641/clima.2022.202

1. Introduction

Radiant cooling technology has many benefits like energy saving [1], better or equal thermal comfort compared with all-air conditioning [2], and quiet operation [3], but its application is limited in hot and humid areas majorly due to condensation concerns. The principle of condensation control is to maintain the radiant surface temperature above the indoor air dew point, but the raised surface temperature also narrows the temperature difference between the radiant surface and indoor spaces, resulting in a limited cooling capacity of a radiant cooling system.

A vertically positioning metal radiant cooling panel can be operated with condensation on its surface if equipped with a drain unit [4-7]. For ceiling radiant cooling panels (CRCP), however, the cooling capacity is stilled limited as the temperature of the radiant surface facing the occupied space must be controlled [8] to prevent dripping of large condensate droplets. Since panels are usually made of hydrophilic metal alloy materials, droplets with sizes up to 7.3 mm [9] can be formed on a panel surface. Such a large dropping droplet will cause discomfort feelings to space occupants.

Different from hydrophilic surfaces, condensate droplets formed on superhydrophobic surfaces can be much smaller. Condensate droplets can leave a superhydrophobic surface with a tiny size even smaller than people's sensory threshold, hence condensation risk can be greatly mitigated if superhydrophobic surfaces are applied for CRCP [10]. Recent research [11] indicated that the size of condensate droplets formed on certain areas of a superhydrophobic aluminum surface placed on a practical CRCP can be constrained below 150 microns during an 8-hour period of condensation with surface temperature 8^{o}C lower than air dew point under typical indoor air conditions, showing the potential application of CRCP with much lower surface temperature by superhydrophobic surface treatment.

The cooling capacity of a CRCP can be substantially increased with the expanded temperature difference between the radiant surface and indoor spaces. If the surface temperature is further lowered below air dew point without dangerous condensation, as in the case of the condensation regime on a super-hydrophobic surface, additional latent heat flux can be achieved by the panel. However, there is no discussion revealing the potential cooling capacity of CRCP with latent heat transfer to the best of our knowledge.

On the other hand, the condensation heat transfer coefficient between a ceiling positioning super-hydrophobic surface and humid air under natural convection remains little known, hindering the prediction of the condensation heat flux of a panel with latent heat transfer.

In this paper, we performed a case study of a CRCP placed in indoor space with an air temperature of 25°C and relative humidity between 40% and 70%, attempting to indicate the potential enhancement of the total heat flux of CRCP with superhydrophobic treatment which can be operated with lower surface temperature below air dew point, namely, with latent heat transfer, based on our experimental investigation for the condensation heat and mass transfer of a ceiling placed super-hydrophobic aluminum surface.

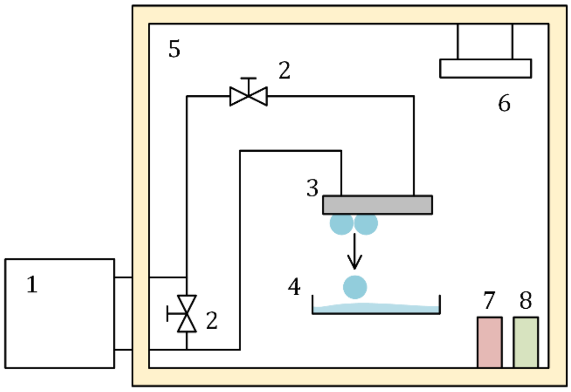
2. Methods

2.1 experimental methods for condensation heat transfer of hydrophilic and super-hydrophobic surfaces

Square aluminum plates with a length of 80 mm and a thickness of 5 mm were prepared to fabricate superhydrophobic surfaces by methods mentioned in [11-13]. Same plates were also prepared without any treatment and remained hydrophilic.

The schematic experimental setup is shown in Fig. 1. Condensation experiments were performed in a climate chamber with a size of 4m x 2.7m x 2.9m. To control indoor thermal conditions while avoiding forced airflow, a hydronic radiant panel, a thermal radiator, and a humidifier were placed inside the chamber.

The hydrophilic aluminum plate and super-hydrophobic aluminum plate were mounted onto a cooling stage by bolts and rubber gasket, with a ceiling positioning. The cooling stage was connected to a water chiller. Hence, the plates can be cooled directly by circulated chilled water.



1 - chiller 2 - valve
3 – testing plates 4 – water collection pan
5 - chamber 6 – hydronic panel
7 – thermal radiator 8 - humidifier

Fig. 1 – The schematic experimental setup.

The plate surface temperature was measured by calibrated ultrafine thermocouples attached to the surface via a small foil tape. The condensate rate was measured through the dew collection method. A plastic drain pan was used to collect condensate dripping from the plates. The mass of the condensate was measured using a calibrated weighing scale. The ambient air temperature and humidity were measured by a temperature and humidity transmitter placed near the plates. The accuracy of sensors and calibration devices is shown in Table 1.

Tab. 1 - Accuracy of sensors and calibration devices.

Sensors/Calibration devices	Accuracy
Thermocouples	±0.5°C
Reference thermometer	±0.03°C
Air temperature/humidity transmitter	Temp: ±0.35°C RH: ±2.5%
Weighing scale	0.01 g
Standard weight	0.0005 g

The ambient air temperature (T_a) and relative humidity (RH) in the chamber were maintained at 24.3°C ± 0.5°C and 65% ± 2.5%, and the air dew point was maintained at around 17.3°C. The plate surface temperature was controlled by adjusting the chilled water temperature setpoint (T_{set}) between 5°C and 13°C for different experimental sets to achieve various subcooling degrees. The conditions of each experimental set are shown in Table 2.

Tab. 2 - Experimental sets.

Set	T_{set} (°C)	T_a (°C)	RH (%)
S1	5	24.3 ± 0.5	65 ± 2.5
S2	8		
S3	10		
S4	13		

When performing experiments, the indoor air condition was firstly controlled to the expected level and the chilled water temperature was controlled to the setpoint, then the chilled water valve of the cooling stage was switched on, and the plate was cooled quickly and the plate surface temperature can reach stable in five minutes. After three-hour condensation, the total mass of dew collected in the water collection pan and that remained on the plate surface was measured.

The tiny droplets from the superhydrophobic aluminum plate surface can easily evaporate after falling into the pan, leading to underestimation of the condensate mass. To avoid the error due to the evaporation of tiny droplets, water was added to the water collection pan as a buffer during experiments. The evaporation loss can be firstly measured by implementing a condensation test on the hydrophilic aluminum plate since condensate water film kept remaining on the hydrophilic aluminum surface and won't drip down into the water collection pan during three-hour condensation.

The condensation mass rate $\dot{m}_{cond}$ was calculated by equation (1), considering the dew mass m_{dew}, condensation duration $\Delta\tau$, and the plate surface area A.

$$\dot{m}_{cond} = \frac{m_{dew}}{\Delta\tau A} \quad (1)$$

2.2 methods for the case study of the total heat flux of CRCP with latent heat transfer

The condensate mass rate is considered to be equal to the convection mass transfer rate between humid air and the plate surface, which can be calculated by equation (2).

$$\dot{m}_{cond} = \dot{m}_{conv} = h_m\left(\rho_{vapor,\infty} - \rho_{vapor,sat}\right) \quad (2)$$

Where h_m represents mass transfer coefficient, $\rho_{vapor,\infty}$ is the density of water vapor in ambient air, $\rho_{vapor,sat}$ is the saturated vapor density at the surface temperature.

Heat and mass transfer analogy [14] is applied to estimate mass transfer coefficient.

$$\frac{h}{h_m} = \rho c_p Le^{2/3} \quad (3)$$

Where h is the free convection heat transfer coefficient of a cold horizontal bottom surface, ρ and c_p the density and specific heat capacity of ambient air. Le is the Lewis number, for air, $Le \approx 1$.

Nusselt number (Nu) is defined as equation (4). As recommended in [14], the empirical correlation proposed by Lloyd and Moran [15] is applied to calculate Nu of a cold horizontal bottom surface over a wide range of Raleigh number (Ra).

$$Nu_L = \frac{hL}{k} \quad (4)$$

$$Nu_L = \begin{cases} 0.54Ra_L^{1/4}\ (10^4 \lesssim Ra_L \lesssim 10^7, Pr \gtrsim 0.7) \\ 0.15Ra_L^{\frac{1}{3}}\ (10^7 \lesssim Ra_L \lesssim 10^{11}, \text{All } Pr) \end{cases} \quad (5)$$

Where Ra is calculated based on equation (6).

$$Ra = GrPr \quad (6)$$

Grashof number (Gr) is calculated by equation (7).

$$Gr = \frac{g\beta(T_a - T_s)L^3}{\upsilon^2} \quad (7)$$

Where g is the gravity acceleration, β the coefficient of thermal expansion, T_a the ambient air temperature, T_s the surface temperature, L the length of the plates, υ the kinematic viscosity.

By comparing our experimental data of condensation mass rate and the empirical convection mass transfer rate, it can be determined whether the heat and mass analogy-based empirical methods can be applied to predict the condensation heat transfer of super-hydrophobic surfaces under natural convection in humid air.

To predict the total cooling capacity of CRCP with latent heat transfer, heat flux by thermal radiation, natural convection, and condensation should all be considered.

$$q_s = q_{rad} + q_{conv} + q_{cond} \quad (8)$$

The MRT method [16] was applied to estimate the heat flux by thermal radiation based on the equation (9).

$$q_{rad} = 5 \times 10^{-8}\left|(t_s + 273.15)^4 - (\text{AUST} + 273.15)^4\right| \quad (9)$$

Where t_s is the effective panel surface temperature, AUST is the area-weighted temperature of all indoor surfaces excluding active panel surfaces. For simplification, AUST is considered equal to indoor air temperature as in the case of little outdoor exposure of walls.

The natural convection heat transfer coefficient between a cooled ceiling surface and indoor air is determined by equation (10) [16].

$$h_{conv} = 2.13|t_s - t_a|^{0.31} \tag{10}$$

Then the convection heat flux can be calculated.

$$q_{conv} = h_{conv}|t_s - t_a| \tag{11}$$

The heat flux induced from condensation equals the latent heat released through the condensation process, which can be calculated from the condensation mass rate $\dot{m}_{cond}$ and the vaporization latent heat of water vapor h_{fg}.

$$q_{cond} = \dot{m}_{cond} h_{fg} \tag{12}$$

3. Results and discussions

3.1 condensation heat transfer of a ceiling positioning superhydrophobic surface

The experimental condensation mass rate of the hydrophilic aluminum plate and the super-hydrophobic aluminum plate are compared with the data predicted by the heat and mass analogy and the empirical convection heat transfer coefficient based on equation (4) – (10). The data are displayed in Fig. 2 with relation to the subcooling degrees (SCD) which is defined as the temperature difference between the plate surface temperature and the air dew point. The error between the experimental condensation mass rate of the two plates and the empirical data are both within 15%, proving the applicability of using heat and mass analogy to predict the condensation rate of a superhydrophobic surface. For the hydrophilic plate, all the condensate mass rates are very close to the empirical one, while the condensation mass rate of the superhydrophobic surface is a bit higher than the empirical data when SCD is lower than 8 K. With higher SCD, the condensate mass rate of the superhydrophobic surface becomes closer to empirical data.

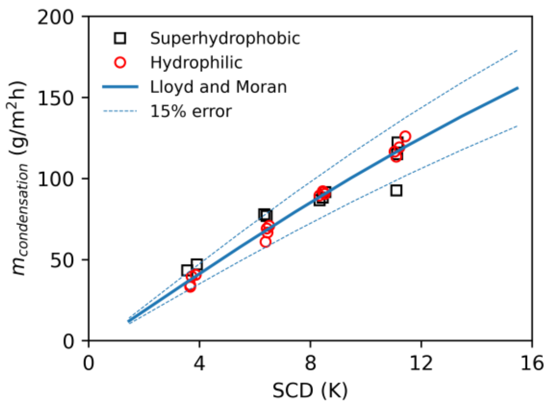


Fig. 2 - The condensation mass rate of the superhydrophobic and hydrophilic aluminum plate, with a comparison with empirical correlation.

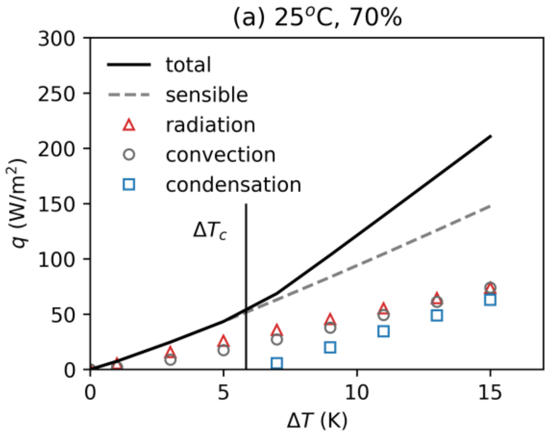
Our experimental data showed that the condensation mass rate of the ceiling positioning super-hydrophobic aluminum surface is in the similar range of the hydrophilic aluminum surface under natural convection in humid air, and the comparison with the empirical data shows both the condensate rates can be predicted based on the heat and mass analogy after knowing the convection heat transfer coefficient. Based on the finding, the condensation heat flux of a CRCP with superhydrophobic treatment can be determined using equations (2), (3), and (10).

3.2 the case study of the sensible and latent heat flux of CRCP with latent heat transfer

Studies [10, 11] confirmed that the jumping condensation regime of superhydrophobic surfaces can be applied to mitigate condensation risks of CRCP by constraining the size of dripping droplets, making CRCP usable with a lower surface temperature below air dew point. Under the circumstance, CRCP transfers heat from an indoor space and its enclosure surface by thermal radiation, natural convection, and condensation.

The heat flux through the CRCP is investigated by the method mentioned in section 2.2. In this case study, indoor air temperature is set as 25°C, relative humidity is set between 40% and 70%, the lowest panel surface temperature is set as 10°C to simulate the effective surface temperature of a CRCP with chilled water supplied at 7°C and returned at 12°C.

The heat flux of the CRCP with latent heat transfer in relation to the temperature difference between panel surface and indoor air, ΔT, is shown in Fig. 3. The sensible heat flux is the net heat flux by radiation and convection. The critical temperature difference ΔT_c is the temperature difference between the panel surface and air dew point, which represents the maximum temperature difference a sensible cooling panel can achieve.



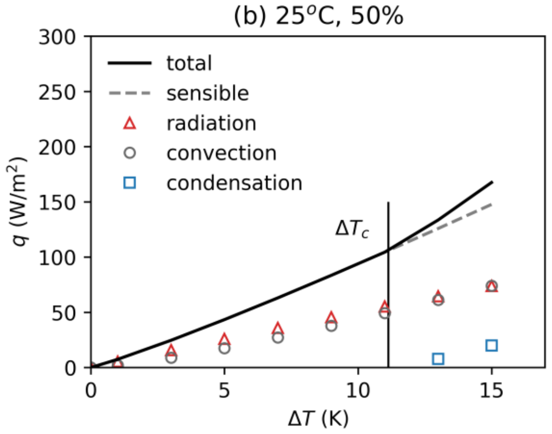


Fig. 3 - The heat flux of CRCP by radiation, convection, and condensation, $\Delta T = |T_a - T_s|$, $\Delta T_c = |T_a - T_{dew}|$. The sensible heat flux is the net heat flux by thermal radiation and convection.

For the highly humid condition (RH = 70%) shown in Fig. 3(a), the heat flux of a sensible cooling panel is limited to 52 W/m^2 at $\Delta T_c = 5.8$°C. With ΔT further increased, heat flux by condensation is included due to condensation on the panel surface and is increased with increasing ΔT, making the total heat flux higher than the sensible heat flux. With the highest temperature difference $\Delta T = 15$°C in this case, the total heat flux can reach 210 W/m^2, consisting of 147 W/m^2 of sensible heat flux and 63 W/m^2 of condensation heat flux. The ratio of the radiation heat flux is higher than 50% when there is no condensation on the panel surface, but it becomes close to the convection and condensation heat flux when ΔT is near 15°C.

The heat flux of a sensible cooling panel can reach a higher value of 105 W/m^2 in a relatively dry condition (RH = 50%), as indicated in Fig. 3(b), since ΔT_c can be as high as 11.1°C. If the panel surface is further cooled to the lowest temperature of 10°C, the total heat flux is 168 W/m^2 with a condensation heat flux of only 20 W/m^2. The heat flux by radiation contributes to near 50% for sensible heat flux, while the ratio of condensation heat flux can be much smaller when compared with the highly humid condition.

The total heat flux of CRCP with latent heat transfer in relation to the panel surface temperature is shown in Fig. 4. The sensible heat flux is the net heat flux by thermal radiation and convection heat transfer, which represents the heat flux of a CRCP without condensation. The maximum sensible heat flux can reach 150 W/m^2 at $T_s = 10$°C. When condensation control is considered, however, the sensible heat flux will be hugely limited. In the case of RH = 60%, the maximum sensible heat flux decreased from 148 W/m^2 to 76 W/m^2 since the panel surface temperature must be controlled higher than air dew point to prevent dew formation.

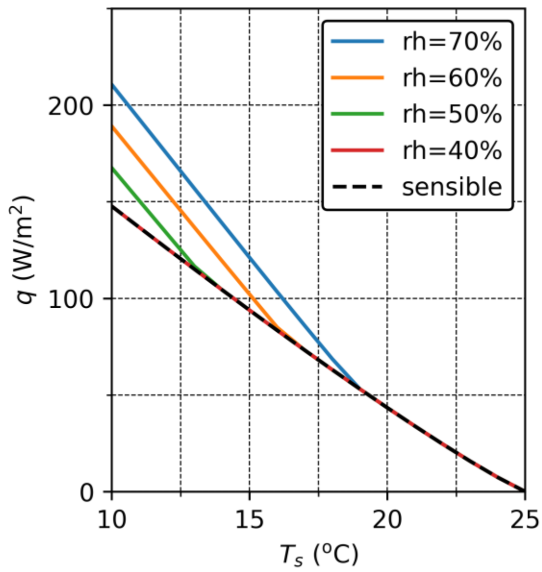


Fig. 4 - The total heat flux of CRCP with latent heat transfer in humid air, $T_a = 25$°C, RH = 40% − 70%, the panel surface temperature is set between 10°C and T_a. The sensible heat flux is the net heat flux by thermal radiation and convection.

An index E_{pl} is proposed to demonstrate the potential enhancement of the heat flux of a CRCP with latent heat transfer. E_{pl} is defined as the ratio of the total heat flux of a CRCP with latent heat transfer under a certain temperature difference between panel surface and indoor air, $q_{tot,\Delta T}$, to the maximum sensible heat flux of a CRCP with only sensible cooling, $q_{\Delta T_c}$.

$$E_{pl} = \frac{q_{tot,\Delta T}}{q_{\Delta T_c}} - 100\% \tag{13}$$

By equation (13), E_{pl} can show to which extent the total heat flux is increased when latent heat transfer is available compared with only sensible heat transfer.

Fig. 5(a) illustrates the evaluation of $q_{tot,\Delta T}$ and $q_{\Delta T_c}$. The E_{pl} of CRCP with a surface temperature of 10°C in the case is shown in Fig. 5(b). E_{pl} is generally increased with increased air temperature and relative humidity, showing that a more significant enhancement of heat flux of CRCP can be achieved especially in hot and humid weather. In relatively dry conditions (RH = 40%), E_{pl} is relatively small because a large temperature difference can be also achieved for a sensible cooling panel. But in highly humid conditions (RH =70%), E_{pl} shows three times of enhancement of total heat flux compared with only sensible cooling due to the extremely limited cooling capacity for sensible cooling CRCP, indicating the great potential of applying a panel with latent cooling to meet a high thermal load in indoor spaces.

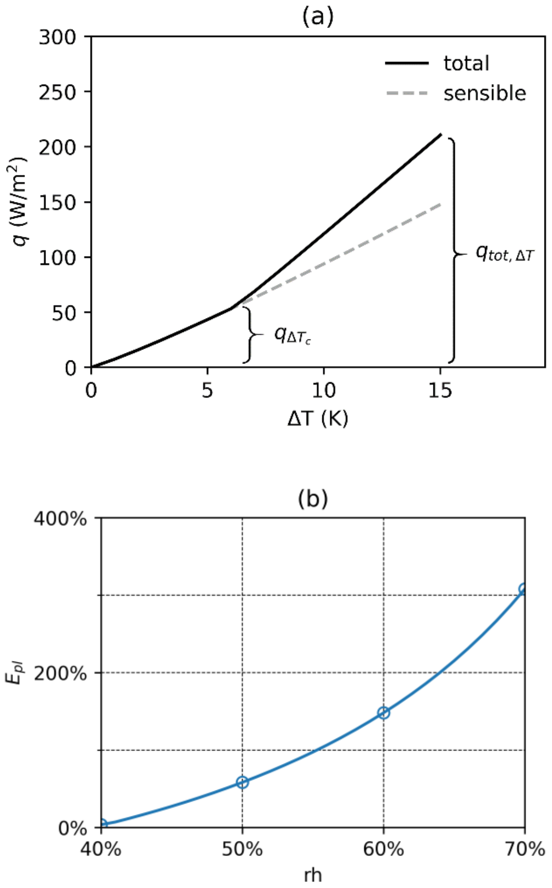


Fig. 5 – (a) The heat flux enhancement index $E_{pl} = \frac{q_{tot,\Delta T}}{q_{\Delta T_c}} - 100\%$, $q_{tot,\Delta T}$ is the total heat flux of CRCP with latent heat transfer at ΔT, $q_{\Delta T_c}$ is the maximum heat flux of a sensible cooling CRCP. (b) E_{pl} of CRCP with latent heat transfer, in this case, $T_s = 10$℃, $T_a = 25$℃, $\mathrm{RH} = 40\% - 70\%$.

3.3 cooling effects on air

The cooling effects of CRCP with latent heat transfer are dependent on the falling-escaping-evaporation behavior of tiny droplets jumping from the super-hydrophobic-treated panel surface. If jumping droplets are totally removed from the indoor space, the indoor air can be simultaneously cooled and dehumidified. If jumping droplets totally re-evaporate into the indoor air, the indoor air would be further cooled by an evaporative cooling-like process.

By creating a near-wall flow close to the panel surface through a ventilation system, some droplets can be exhausted and some latent cooling can be achieved. A primary estimation indicates that the ratio of exhausted droplets shall be diminutive. The complex falling-moving-evaporation behavior of tiny jumping droplets in airflow makes it difficult to predict the actual indoor humidity. The computational fluid dynamics technique can be applied to solve the issue in future research.

3.4 limitations of the research

There are also some limitations for this study. The prediction of heat flux by thermal radiation assumes that the panel surface emissivity is 0.9. However, the emissivity of a superhydrophobic surface with condensation has not been thoroughly investigated, though some studies [13] reported that high emissivity and super-hydrophobicity can be integrated on a surface by specific design.

On the other hand, the experimental study is performed with Ra between 5 x 10^5 and 10^6, lower than the Ra of a practically used CRCP, because the size of our superhydrophobic plate is relatively small compared with practical panels while Ra is highly dependent on the length scale of a surface. The fabrication of a large-scale superhydrophobic surface with mechanical and chemical robustness and good condensation mitigation performance is still a challenge, which is also the major limitation for constructing a practical superhydrophobic CRCP.

4. Conclusions

Condensation issue is the major limitation hindering the application of radiant cooling systems in hot and humid areas. Nonetheless, our case study revealed that the total heat flux of a CRCP can be remarkably enhanced by taking advantage of condensation heat transfer. The results of this study can be summarized as follows.

- The total heat flux of a panel with latent heat transfer can be enhanced by both the increased sensible heat flux due to expanded temperature difference between panel surfaces and indoor spaces, and latent heat flux by condensation of tiny droplets.

- Compared with a panel with only sensible cooling, a significant enhancement of up to 300% can be achieved for the total heat flux of a CRCP with latent heat transfer under a highly humid condition due to 1) the limited maximum sensible heat flux of the sensible cooling panel and 2) the extended total heat flux of the latent cooling panel.

This study draws a clear picture of how the cooling capacity of CRCP can be enhanced via a latent heat transfer process, showing a hidden potential of radiant cooling panels. Since the fabrication of large-scale superhydrophobic materials with mechanical and chemical robustness and good condensation mitigation performance is still a challenge, more efforts are needed to make the panels with latent heat transfer more practicable.

5. Acknowledgment

This study was fully supported by the General Research Funding of the Research Grant Council of the Hong Kong SAR, China (Project No. 15206620).

6. References

[1] J. Niu, L. Zhang, H. Zuo, Energy savings potential of chilled-ceiling combined with desiccant cooling in hot and humid climates, Energy and buildings, 34 (5) (2002) 487-495.
[2] C. Karmann, S. Schiavon, F. Bauman, Thermal comfort in buildings using radiant vs. all-air systems: A critical literature review, Building and Environment, 111 (2017) 123-131.
[3] K.-N. Rhee, B.W. Olesen, K.W. Kim, Ten questions about radiant heating and cooling systems, Building and Environment, 112 (2017) 367-381.
[4] Y. Wu, H. Sun, M. Duan, B. Lin, H. Zhao, Dehumidification-adjustable cooling of radiant cooling terminals based on a flat heat pipe, Building and Environment, 194 (2021) 107716.
[5] H. Teufl, M. Schuss, A. Mahdavi, Potential and challenges of a user-centric radiant cooling approach, Energy and Buildings, 246 (2021) 111104.
[6] H. Shu, X. Bie, H. Zhang, X. Xu, Y. Du, Y. Ma, L. Duanmu, G. Cao, Natural heat transfer air-conditioning terminal device and its system configuration for ultra-low energy buildings, Renewable Energy, 154 (2020) 1113-1121.
[7] D. Song, S. Kato, Radiational panel cooling system with continuous natural cross ventilation for hot and humid regions, Energy and Buildings, 36 (12) (2004) 1273-1280.
[8] W. Zhao, Y. Hu, Y. Wang, W. Qin, Thermal performance of a suspended ceiling fin heat transfer panel with drain pan, Building and Environment, 144 (2018) 622-630.
[9] H. Tang, X.-H. Liu, Experimental study of dew formation on metal radiant panels, Energy and Buildings, 85 (2014) 515-523.
[10] H. Tang, X.-H. Liu, H. Li, Y. Zhou, Y. Jiang, Study on the reduction of condensation risks on the radiant cooling ceiling with superhydrophobic treatment, Building and Environment, 100 (2016) 135-144.
[11] Z. Zhong, J. Niu, W. Ma, S. Yao, M. Yang, Z. Wang, An experimental study of condensation on an aluminum radiant ceiling panel surface with superhydrophobic treatment, Energy and Buildings, 252 (2021) 111393.
[12] Y. Hou, Y. Shang, M. Yu, C. Feng, H. Yu, S. Yao, Tunable Water Harvesting Surfaces Consisting of Biphilic Nanoscale Topography, ACS Nano, 12 (1112(12) (2018) 11022-11030.
[13] W. Ma, Y. Li, C.Y.H. Chao, C.Y. Tso, B. Huang, W. Li, S. Yao, Solar-assisted icephobicity down to −60°C with superhydrophobic selective surfaces, Cell Reports Physical Science, 2 (3) (2021) 100384.
[14] T.L. Bergman, F.P. Incropera, D.P. DeWitt, A.S. Lavine, Fundamentals of heat and mass transfer, John Wiley & Sons, 2011.
[15] J.R. Lloyd, W.R. Moran, Natural Convection Adjacent to Horizontal Surface of Various Planforms, Journal of Heat Transfer, 96 (4) (1974) 443-447.
[16] ASHRAE, ASHRAE Handbook 2016: HVAC Systems and Equipment, ASHRAE, 2016.

Data Statement

Data sharing not applicable to this article as no datasets were generated or analysed during the current study.