Building and Environment 97 (2016) 1–10

Contents lists available at ScienceDirect

Building and Environment

journal homepage: www.elsevier.com/locate/buildenv

Theoretical and experimental study of condensation rates on radiant cooling surfaces in humid air

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ARTICLE INFO

Article history:
Received 23 September 2015
Received in revised form
13 November 2015
Accepted 1 December 2015
Available online 10 December 2015

Keywords:
Radiant cooling surface
Condensation rate
Different positions
Heat and mass transfer analogy

ABSTRACT

The condensation rate on a radiant cooling surface in humid air is a basic index used to evaluate the risk of condensation and is beneficial to the engineering design of radiant cooling systems. In this paper, condensation rates were studied theoretically using simplified Navier–Stokes equations with Boussinesq approximation, and the heat and mass transfer analogy was also derived. In the experiments, the condensation rates on radiant cooling panels of different lengths at various positions (the floor, the wall, and the ceiling) were measured. The experimental results of the mass transfer coefficients are represented in a correlation of average *Sh* number vs. *Ra* number. With a sub-cooled degree of 5 °C, the mass transfer coefficients of the radiant floor, wall and ceiling were 1.22 mm/s, 3.35 mm/s, and 4.15 mm/s, respectively. As a result, the condensation rate on the radiant ceiling was 3.5 times greater than that on the radiant floor and 25% greater than that on the radiant wall.

1. Introduction

Cooling and heating systems in buildings consume large amounts of energy in order to provide acceptable standards of thermal comfort. As a result of rapid economic development, building energy consumption accounts for about 23% of the total energy consumed in China [1]. Radiant cooling systems are now commonly used in new commercial buildings, apartments, and airports and their popularity is still increasing [2,3]. Currently, more and more attention is being directed to radiant cooling systems due to their high energy efficiency and adequate thermal comfort.

The studies on the evaluation of energy performance proved that the radiant cooling system can achieve energy savings by reducing fan energy, increasing chiller efficiency and so on. Hao et al. [4] studied energy consumption of the chilled ceilings with displacement system, combined with a desiccant dehumidification system in a hot and humid climate. The combined system yield energy saving of 68.5% for the chiller and 39.0% for the fans. Further, compared with the conventional air conditioning system, the radiant cooling system with evaporative cooling sources in hot and humid climate could save 80% energy consumption [5] and save 55% of cooling tower energy and 25% of pumping power consumption through model operative control [6]. The comparisons of energy consumption with cooling technologies and cooling load sharing rates were researched in the literature [7,8]. Researches on radiant cooling systems showed that occupants perceived better thermal comfort, compared with conventional air conditioning systems [9–11]. According to the field study conducted by Tian and Love [12], it was proved that the radiant cooling slab can also minimize draft rate (4%) with the mean air velocity of 0.06 m/s in summer. Additionally, the cooling capacity of the radiant cooling system significantly varies with the installation positions, i.e., floors, walls and ceilings [13–15]. As the radiant cooling systems only handle indoor sensible heat load, the indoor moisture load needs be handled by separate dehumidification systems [16,17]. Not surprisingly, indoor humidity control is crucial in the application of radiant cooling systems in order to avoid condensation on radiant surfaces.

Research on avoiding condensation on radiant cooling surfaces has mainly focused on control strategies and the characteristics of condensation on radiant panels. Zhang et al. [18] and Ge et al. [19] both indicated that the pre-dehumidification time is important for preventing condensation on radiant cooling surfaces where the air conditioning system operates intermittently. In these studies, neural network models were established to predict the risk of condensation and the optimal pre-dehumidification time in radiant cooling systems. Song et al. [20] proposed a radiant floor cooling system accompanied by dehumidifying ventilation to prevent the

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http://dx.doi.org/10.1016/j.buildenv.2015.12.003
0360-1323/

condensation of moisture on floor surfaces. The control system's responsiveness to variable indoor loads was also improved. Yin et al. [21] compared the condensation on three different types of radiant cooling ceiling panels: pure capillary panels, metal panels, and gypsum panels. Experimental results showed that the gypsum radiant cooling panel had the strongest ability to avoid condensation due to its moisture absorption capability. Tang and Liu [22] investigated experimentally the dew formation on metal radiant ceiling panels. The results indicated that the fall-off time of the dew droplets from the panel surface at a sub-cooled degree (air dew point minus cooling surface temperature) below 3 °C was over 10 h. It is clear from these studies that the condensation rate on the radiant cooling surfaces plays a central role in the responsiveness requirements of control systems.

It is well known that the heat transfer coefficients of thermal natural convection on radiant floors, walls, and ceilings vary considerably [23–26]. However, the extent to which condensation rates of radiant cooling systems at different positions (e.g. the floor, the wall, and the ceiling) vary remains unknown, which is a deficiency in the research on the condensation risk of radiant cooling systems. Gebhart and Pera [27] examined the vertical natural convection resulting from the combined buoyancy effects of thermal and mass diffusion, and indicated that the heat and mass transfer analogy is applicable to $Pr = Sc$. However, its applicability to condensation on horizontal plates requires further investigation.

This paper focuses on condensation rates on radiant walls, floors, and ceilings. The similarity between the condensation in humid air and thermal natural convection is indicated theoretically, and a unified analysis is presented. In the study, the parameters influencing the condensation rate were synthesized into Rayleigh number defined by the relative air density variation. Experiments examining the condensation rates on radiant cooling panels with various lengths at different positions were conducted in a climate room with constant air temperature and humidity. The mass transfer coefficients of condensation were obtained by the dew collection and energy balance methods independently and were regressed in a correlation of Sherwood number vs. Rayleigh number. This work gives empirical correlations for condensation on radiant floors, walls, and ceilings, and the results illustrate the tremendous differences in the condensation rates of radiant cooling systems at different positions. This study has important implications for the ability to determine quantitatively the responsiveness requirements of control systems, as well as for the optimal design of radiant cooling systems.

2. Theoretical analysis

2.1. Heat and mass transfer analogy

Fig. 1(a) shows a two-dimensional radiant cooling panel with adjustable orientation in ambient air. Consider a radiant panel of length L and a depth perpendicular to the plane of the paper. Condensation occurs on the radiant cooling surface as long as the temperature of the radiant cooling surface (T_i) is lower than the dew point temperature (T_d). The process of condensation in humid air involves natural convection flows resulting from the combined buoyancy effects of thermal and mass diffusion. As cold dry air is denser than hot humid air, cold dry air tends to sink. Thus, the condensation rate on a radiant cooling surface is significantly influenced by its orientation. The orientation of a 2D radiant panel is described by its inclination angle with the horizontal plane (α), which is 0° for the radiant floor shown in Fig. 1(b), 90° for the radiant wall shown in Fig. 1(c), and 180° for the radiant ceiling shown in Fig. 1(d). Therefore, condensation rates on radiant cooling surfaces at different positions can be analyzed in a unified model.

Here, the related temperature and humidity are defined as follows: the ambient air temperature is T_∞, the water vapor mass fraction is χ_∞, and χ_i represents the water vapor mass fraction corresponding to the saturated air of radiant surface temperature T_i.

The governing equations result from the conservation condition of mass, of force-momentum, of energy, and of the water vapor mass [28]. These equations are simplified by Boussinesq approximation which applies to the evaluation of the buoyancy force arising from small density differences [27–29]. The natural convection laminar flow of an incompressible fluid on a plate with negligible viscous dissipation and constant fluid properties is considered here. The secondary phenomena of species thermal diffusion and thermal energy diffusion effects are ignored. The governing equations are expressed as:

$$\nabla \cdot \mathbf{v} = 0 \tag{1}$$

$$\rho\left(\frac{\partial \mathbf{v}}{\partial t} + \mathbf{v}\nabla \cdot \mathbf{v}\right) = -\nabla p + \mu\nabla^2\mathbf{v} + \rho\mathbf{g} \tag{2}$$

$$\frac{\partial T}{\partial t} + \mathbf{v}\nabla \cdot T = \alpha\nabla^2 T \tag{3}$$

$$\frac{\partial \chi}{\partial t} + \mathbf{v}\nabla \cdot \chi = D_{AB}\nabla^2\chi \tag{4}$$

where the gravitation acceleration is represented by $\mathbf{g} = g[-\sin\alpha, -\cos\alpha]^T$ and g is the magnitude of the gravitational field.

The combined effect of the temperature and humidity difference between the ambient air and the radiant surface, which determines the buoyancy force of natural convection flows, can be synthesized into the air density variation ρ_i-ρ_∞. A series expression of ρ-ρ_∞ in terms of p, T, and χ at a given elevation indicates that the pressure effect may be neglected, and that only the linear terms of the temperature effect and the mass fraction effect need to be retained:

$$\rho_\infty - \rho = \rho\beta(T - T_\infty) + \rho\beta^*(\chi - \chi_\infty) \tag{5}$$

Denote

$$\beta = -(1/\rho)(\partial\rho/\partial T)_{p,\chi}, \quad \beta^* = -(1/\rho)(\partial\rho/\partial\chi)_{p,T} \tag{6}$$

where β is the volumetric coefficient of thermal expansion and β^* is the volumetric coefficient of expansion with mass fraction.

Because humid air is the only fluid of interest in this paper, the approximation $Le = 1$ [30] is applicable to the governing equations. The heat and mass transfer analogy was derived by the non-dimensional analysis detailed in Appendix A, which is expressed as

$$Nu = Sh \tag{7}$$

$$\text{where} \quad Nu = hL/k \text{ and } Sh = h_m L/D_{AB} \tag{8}$$

Furthermore, the correlation of the Sherwood number to condensation in humid air was verified to be the same as that of the Nusselt number to thermal natural convection through theoretical analysis in Appendix A. The theoretical and experimental results related to thermal natural convection in air on plates at different positions are adequate [23]. Among published heat transfer correlations, the correlation between the Nusselt number (Nu) and Rayleigh numbers (Ra) is generally given as a power function [23–27]:

$$Nu = CRa^n \tag{9}$$