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Ceiling radiant cooling panel capacity enhanced by mixed convection in mechanically ventilated spaces

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Abstract

The main thrust of this research is to estimate the impact of the mixed convection effect on the cooling capacity of a ceiling radiant panel in mechanically ventilated spaces. To estimate panel cooling capacity enhancement caused by mixed convection, a verified analytical panel model was used. The simplified correlation for mixed convection heat transfer coefficient which can be easily adopted in panel cooling capacity estimation was derived from established mixed convection and natural convection correlations. It was found that the total cooling capacity of radiant panels can be enhanced in mixed convection situations by 5–35% under normal operating panel surface temperatures.

Keywords: Ceiling radiant panel; Radiant cooling; Mixed convection; Natural convection; Mechanical ventilation

1. Introduction

Total heat transfer capacity of a radiant panel is determined by two heat transfer mechanisms: convection and radiation. ASHRAE [1] proposed to use the natural convection (NC) heat transfer coefficient developed by Min et al. [2] for estimating the convective heat flux on a ceiling panel, and to use the mean radiant temperature method proposed by Walton [3] for calculating the radiation heat flux.

When calculating the convective heat transfer rate of a ceiling radiant cooling panel (CRCP) in a naturally ventilated space, the NC heat transfer coefficient can be used, but in a mechanically ventilated room the air movement is obviously greater, and the convective heat transfer on a

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Nomenclature

A	area (m^2)
AUST	area-weighted average temperature of un-cooled surfaces (°C)
b_w	bond width, (m)
C_p	specific heat of the fluid (kJ/kg K)
D	tube diameter (m)
D_e	characteristic diameter of room surface ($= 4A_c/P$) (m)
F	fin effectiveness
F'	panel efficiency factor
F_c	correction function (W/m^2 K)
F_R	panel heat removal factor
h_i	fluid to tube heat transfer coefficient (W/m^2 K)
h	heat transfer coefficient (W/m^2 K)
k	heat conductivity of the panel (W/m K)
M	mass flow rate to the panel (kg/s)
n	number of tubes
P	parameter of the room (m)
q	heat flux to the panel (W/m^2)
q'_{fin}	transferred energy to the fin per unit length (W/m)
q'_{tube}	heat gain from above the tube region per unit length (W/m)
q'	total sensible heat gain of the panel per unit length (W/m)
T	temperature (°C)
ΔT	temperature difference between the space and the panel mean surface temperature (°C)
U_o	overall heat transfer coefficient (W/m^2 K)
U_e	equivalent overall heat transfer coefficient (W/m^2 K)
V	diffuser discharge air velocity (m/s)
w	distance between the tubes (m)
W	width of nozzle diffuser (m)

Greeks

δ	panel thickness (m)
γ	bond thickness (m)

Subscripts

a	air, space
b	bond material, fin base
c	ceiling, convection
f	forced, fluid
i	inside
m	mean
n	natural
o	outside, total, overall
p	panel
r	radiation

radiant panel would be expected to be enhanced; therefore, the use of the NC heat transfer coefficient is inappropriate for a mechanically ventilated room.

Kochendörfer [4] indicated that in real buildings cooling outputs of CRCPs are significantly higher (over 25%) than measured panel capacities in the laboratory under standard testing condition expressed in DIN 4715 [5]. The reasons for this higher capacity are non-standard surrounding conditions, such as warm windows and outside walls, and mechanical ventilation systems. However, those effects have not been considered in panel capacity estimation, and NC is still a general assumption in CRCP system design. If the higher performance of CRCPs is ignored in the design phase, unnecessary panel area is specified, and the cost of the panels is excessive.

Therefore, in this research the enhanced cooling capacity of a CRCP by mixed convection (MC) will be estimated with verified analytical steady-state panel model. The simplified yet reliable correlation for MC heat transfer coefficient which can be easily adopted in cooled ceiling capacity estimation will also be derived from the established MC and NC coefficients.

2. Literature review

2.1. Natural convection coefficient

For the last couple of decades, many researchers focused on buoyancy driven heat transfer coefficients for small free-edge heated plates. Widely used NC heat transfer coefficients are those in the ASHRAE Handbook Fundamentals [7] and Alamdari and Hammond [8]. However, the use of above coefficients to room surfaces is somewhat debatable, because the air movement over a surface in a room is different from that over a small free-edge heated plate.

Some researchers carried out their experiments using full-size enclosures [9–11]. The work of Min et al. [2] is the first to investigate the NC heat transfer in full-size enclosures. They used three differently sized testing chambers for their experiments to consider the room size effect. Their correlation for a cooled ceiling or heated floor is presented in Eq. (1).

$$h_c = 2.13 \cdot (T_a - T_{pm})^{0.31} \tag{1}$$

2.2. Mixed convection coefficient

The need for the MC heat transfer coefficient has been growing because the NC heat transfer coefficient cannot accurately estimate the convective heat flux from a heated or cooled surface in a mechanically ventilated room. However, there has been a few works into convective heat transfer from the heated or cooled surfaces in ventilated rooms. Chen et al. [12] performed experiments to determine MC heat transfer coefficients for the floor and ceiling of the room. The experiments were carried out in a 5.6 m $\times$ 3.0 m $\times$ 3.2 m chamber with two mechanically driven air inlets located at the floor level. They proposed the MC heat transfer coefficient for cooled ceiling (4.0 W/m^2 K) for mechanically driven ventilation rates ranging from 3 to 7 air changes per hour (ACH).

Spitler et al. [13] also conducted measurements for the convection heat transfer coefficient in a rectangular office-sized enclosure (5.48 m $\times$ 3.65 m $\times$ 3.35 m). Although their experiments were focused on the forced convection (FC) heat transfer coefficient with large ventilation rates (15–100 ACH), they found that the space could fall into the MC flow regime at the low ventilation rate.

Fisher and Pedersen [14] proposed MC heat transfer coefficients for the low ventilation rate (3–12 ACH) using Spitler et al.'s test chamber. Actually, their correlations for walls, floor, and ceiling were derived for the isothermal room where buoyancy caused by surface-to-air temperature differences is negligible. However, they found that the correlations can be applied to non-isothermal rooms with surface temperature differences of less than 20 °C. Eq. (2) is their MC correlation for a ceiling.

$$h_c = 0.49 \cdot ACH^{0.8} \tag{2}$$

Beausoleil-Morrison [15] built MC heat transfer correlations by blending well known NC and FC correlations using the Churchill and Usagi's approach [16] which has been used to correlate heat transfer problems which are governed by two or more driving forces. He assumed that, when FC effect is overwhelmed by buoyancy, surface convection can be adequately characterized with the Alamdari and Hammond [8] correlations. Whereas when FC effect is dominant, the Fisher's correlations [17] are sufficient to calculate surface convection.

The most noted study is by Awbi and Hatton [18]. They proposed MC heat transfer coefficients for heated room surfaces partially covered by air jet. Experiments were conducted in an office-sized, well-insulated environmental chamber (2.78 m × 2.78 m × 2.3 m) with heating plates fixed to the internal surfaces. A fan box with an adjustable nozzle was placed at one end of the surfaces within the chamber to create the convective flow. Their correlation for a heated floor or cooled ceiling is presented in Eqs. (3).

$$h_c = (h_{cn}^{3.2} + h_{cf}^{3.2})^{1/3.2} \tag{3a}$$

$$h_{cn} = \frac{2.175}{D_e^{0.076}} (T_a - T_{pm})^{0.308} \tag{3b}$$

$$h_{cf} = 4.25 \cdot W^{0.575} \cdot V^{0.557} \tag{3c}$$

3. Simplified mixed convection coefficient

3.1. Simplified correlation for mixed convection

In a mechanically ventilated space, the convection heat transfer on a heated (or cooled) surface is occurred by the combination effect of both NC and FC on the panel surface. Pamelee and Huebscher [19] tried to include the FC effect on the heat transfer from heated panels as an increment to be added to the NC coefficient. However, their work was not widely accepted by the industry because one could not be sure the validity of their predicted FC effects.

Recently, Awbi and Hatton [18] proposed noticeable MC correlations for heated room surfaces by correlating the combined effects of both NC and FC on each surface as a function of four parameters; characteristic diameter of a space (D_e), space-to-panel temperature difference (ΔT), diffuser width (W), and diffuser discharge air velocity (V).

In this research, Awbi and Hatton's MC correlation for a cooled ceiling Eq. (3) was extensively analyzed, and it was found that the characteristic diameter of a space (D_e) or the space size effect has negligible impact on the MC heat transfer coefficient on a CRCP in most spaces usually faced in the HVAC design projects. As a consequence of this observation, a simplified MC correlation which can be easily adopted in panel cooling capacity estimation without concerning the space size effect was derived as follows.

As shown in Eq. (4), the simplified MC correlation has the form of adding the FC effect to well known Min et al.'s NC correlation Eq. (1) widely used to estimate design cooling capacity of CRCPs. The correction function F_c in Eq. (4) represents additional FC effect on a panel in a mechanically ventilated room.

$$h_c = F_c + 2.13 \cdot \Delta T^{0.31} \tag{4}$$

In principle, the difference between the MC heat transfer coefficient and the NC heat transfer coefficient represents the impact of the FC on a panel. The FC effect data required to derive the correction function (F_c) were collected by calculating differences between Awbi and Hatton's MC coefficients and Min et al.'s NC coefficients calculated by Eqs. (3) and (1) respectively for the various values of four parameters: characteristic diameter of a space (D_e), space-to-panel temperature difference (ΔT), diffuser width (W), and diffuser discharged air velocity (V). These FC effect data were analyzed statistically, and it was found that the space size effect (D_e) has insignificant impact on the convection heat transfer on a CRCP surface. Finally, the correction function (F_c) for considering additional FC effect on a panel was derived as a function of three parameters (i.e. ΔT, W, and V). More details of F_c were presented in the following sections.

3.2. 2^k factorial experiment design approach

The 2^k factorial experiment design method [20] was used to determine which parameters and their interactions would show significant effects on the increment of the convection heat transfer, and to derive the correction function (F_c) with a small number of experiments (or calculations). The superscript k means the number of parameters considered in an experiment. In the 2^k factorial experiment design, it is typical to select only two values (i.e. minimum and maximum) for each parameter. Table 1 shows the ranges of four parameters considered in this research. Once the individual parameters and their interactions which have significant effects on a dependent parameter are identified, a linear correlation for the dependent parameter can be developed as a function of those parameters.

Table 1
Typical ranges of each parameter

Label	Parameter	Low	High
A	Temperature difference between the space and the panel mean surface temperature (ΔT), °C	1	14
B	Diffuser discharged air velocity (V), m/s	2	6
C	Characteristic diameter of a space (D_e), m	1	30
D	Diffuser width (W), m	0.2	0.8

In this particular case, the dependent parameter is the difference between the MC and the NC coefficient (Δh_c), and the full factorial (or 2^4) experiment can be performed without making assumptions to reduce the number of calculations because only four parameters (i.e. D_e, ΔT, W, and V) are considered in this problem.

Individual parameters or interactions which have significant effects on the dependent parameter can be identified by a simple way proposed by Daniel [21]. He suggested examining a normal probability plot of the effects of each parameter. Effects are defined as the average change in response that occurs as a result of changing each parameter from its low value to its high value. Formulas for computing effects can be found in statistics texts [20]. According to this method, the negligible effects are normally distributed, and will tend to fall along a straight line representing the normal probability distribution, whereas significant effects will not lie along the straight line.

3.3. Derivation of the correction function F_c

Fig. 1 clearly shows that three individual parameters (ΔT, V, W) and one interaction ($V \cdot W$) have significant effect on the dependent parameter (Δh_c). Based on this analysis result, a first order regression equation for the correction function (F_c) was derived as a function of those parameters Eq. (5). The coefficients for F_c were presented in Table 2.

$$F_c = f(\Delta T, V, W) = \alpha_0 + \alpha_1(\Delta T) + \alpha_2(V) + \alpha_3(W) + \alpha_4(V \cdot W) \quad (5)$$

On the other hand, the regression equation can be verified by the normal probability plot of residuals or the differences between the actual Δh_c (collected data) and the predicted Δh_c by Eq. (5). If the points on this plot lie reasonably close to the normal probability distribution line, one

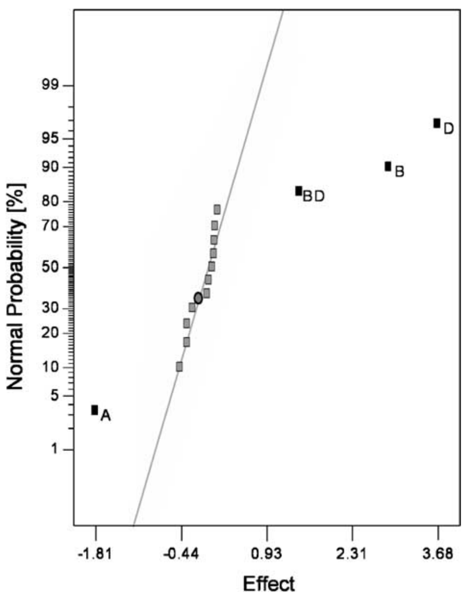

Fig. 1. Normal probability plot of effects on response variable Δh_c.

Table 2
Coefficients for correction function

α_0	α_1	α_2	α_3	α_4
0.28021	−0.13931	0.11416	1.25013	1.22058

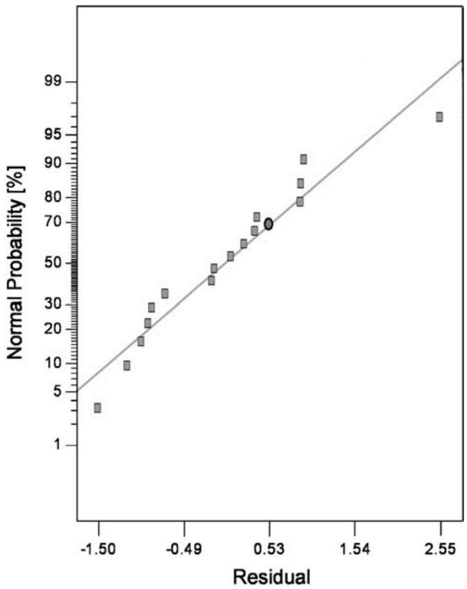

Fig. 2. Normal probability plot of residuals of the correction function F_c.

can conclude that the regression equation is satisfactory. Fig. 2 clearly shows that Eq. (5) is satisfactory.

By substituting Eq. (5) into Eq. (4), the simplified MC correlation is represented. The proposed correlation will return to the NC correlation if a space is not mechanically ventilated (i.e. V and W equal 0), although there is a small deviation (i.e. $\alpha_0 + \alpha_1(\Delta T)$) from original Min et al's NC coefficients. To estimate the MC impact on a cooling capacity of a CRCP, the proposed correlation was integrated into the verified analytical CRCP model presented in the following section.

4. Ceiling radiant cooling panel model

The analytical CRCP model developed by Conroy and Mumma [6] was modified to consider the MC effect on panel cooling capacity. Fig. 3 shows the cross section and the geometry of a CRCP.

The temperature distribution between the tubes Eq. (6) was derived from the energy balance on the panel (fin) element with temporarily assuming that the temperature gradient in the flow

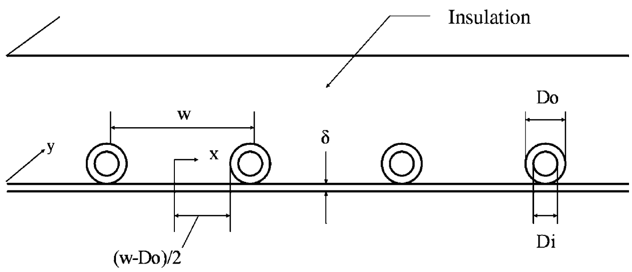

Fig. 3. CRCP cross section and the geometry.

direction (y-direction) is negligible. The energy transferred to the fin base per unit length in the flow direction (q'_{fin}) is calculated with Eq. (7). The fin effectiveness (F) is a ratio of the actual heat transfer to the ideal heat transfer when the entire fin is at its base temperature T_b Eq. (8).

$$\frac{T_p(x) - T_a}{T_b - T_a} = \frac{\cosh(mx)}{\cosh[m(w - D_o)/2]} \tag{6}$$

$$q'_{\text{fin}} = -FU_o(w - D_o)(T_b - T_a) \tag{7}$$

$$F = \frac{\tanh[m(w - D_o)/2]}{m(w - D_o)/2} \quad \text{where,} \ m = \sqrt{U_o/k \cdot \delta} \tag{8}$$

The sensible heat gain of the panel also includes the heat gain from above the tube region Eq. (9). The temperature above the tubes can be considered at uniform temperature (T_b). By adding Eqs. (7) and (9), the total sensible heat gain of the panel per unit length (q') is determined Eq. (10).

$$q'_{\text{tube}} = -D_o U_o (T_b - T_a) \tag{9}$$

$$q' = -[(w - D_o)F + D_o] \cdot U_o \cdot (T_b - T_a) \tag{10}$$

The resistance to heat flow to the fluid results from the bond and the tube-to-fluid resistance Eq. (11). By eliminating T_b from Eqs. (10) and (11), q' can be expressed in terms of known dimensions, physical parameters, and the local fluid temperature (T_f) as shown in Eq. (12). The panel efficiency factor (F') Eq. (13) is a ratio of overall heat transfer coefficient between fluid and room to overall heat transfer coefficient between fin and room. For more simplicity, the bond and the tube-to-fluid resistance are neglected, then F' can be reduced to Eq. (14).

$$q' = \frac{T_b - T_f}{\dfrac{1}{h_i \pi D_i} + \dfrac{\gamma}{k_b b_w}} \tag{11}$$

$$q' = -wF'U_o(T_f - T_a) \tag{12}$$

$$F' = \frac{1/U_o}{w\left[\dfrac{1}{U_o[D_o + (w - D_o)F]} + \dfrac{1}{h_i \pi D_i} + \dfrac{\gamma}{k_b b_w}\right]} \tag{13}$$

$$F' \cong \frac{D_o + (w - D_o) \cdot F}{w} \tag{14}$$

The temperature distribution in the flow direction was derived from the mass and energy balance. If the fluid enters the panel at temperature T_{fi} and increases in temperature until at the exit it is T_{fo}, the fluid temperature at any position in flow direction can be expressed as Eq. (15).

$$\frac{T_f(y) - T_a}{T_{fi} - T_a} = \exp\left(-\frac{nU_o wF'}{MC_p} y\right) \tag{15}$$

If the panel length is L, the mean fluid temperature (T_{fm}) is found by integrating Eq. (15) from $y =$ 0 to L. Performing this integration, and after some algebraic manipulation, the mean fluid temperature can be expressed as follows;

$$T_{fm} = T_{fi} + \frac{q_o}{F_R U_o}\left(1 - \frac{F_R}{F'}\right) \tag{16}$$

The panel heat removal factor (F_R) relates the actual sensible heat gain of a panel to the heat gain if the whole panel surface were at the fluid inlet temperature Eq. (17), then the total sensible heat flux of the panel (q_o) can be expressed as Eq. (18). The total sensible heat flux can also be expressed in terms of the mean panel temperature (T_{pm}) Eq. (19). By equating these two equations and solving for T_{pm}, the expression for the mean panel temperature is derived as shown in Eq. (20).

$$F_R = \frac{MC_p(T_{fo} - T_{fi})}{A_p U_o (T_a - T_{fi})} \tag{17}$$

$$q_o = F_R U_o (T_a - T_{fi}) \tag{18}$$

$$q_o = U_o (T_a - T_{pm}) \tag{19}$$

$$T_{pm} = T_{fi} + \frac{MC_p(T_{fo} - T_{fi})}{A_p F_R U_o} \cdot (1 - F_R) \tag{20}$$

In principal, the total heat flux (q_o) is the summation of the convection heat flux (q_c) and the radiation flux (q_r) as shown in Eq. (21a), and each heat flux can be expressed as Eqs. (21b) and (21c), respectively.

$$q_o = q_c + q_r \tag{21a}$$

$$q_c = h_c \cdot (T_a - T_{pm}) \tag{21b}$$

$$q_r = h_r \cdot (\mathrm{AUST} - T_{pm}) \tag{21c}$$

The overall heat transfer coefficient U_o can be easily determined by summing the convection heat transfer coefficient (h_c) and the radiation heat transfer coefficient (h_r) if $T_a = \mathrm{AUST}$; however, it is usually not true. Therefore, the equivalent overall heat transfer coefficient (U_e) was defined Eq. (22).

$$U_e = \frac{q_o}{(T_a - T_{pm})} \tag{22}$$

By substituting Eq. (21) into Eq. (22),

$$U_e = \frac{q_c + q_r}{(T_a - T_{pm})} = h_c + h_r \cdot \frac{(\mathrm{AUST} - T_{pm})}{(T_a - T_{pm})} \tag{23}$$

The simplified mixed convection heat transfer coefficient Eq. (4) was derived in previous section, and the radiation heat transfer coefficient Eq. (24) was found in the literature [1].

$$h_r = 5 \times 10^{-8} \cdot \left[(\mathrm{AUST} + 273)^2 + (T_{pm} + 273)^2\right] \cdot [(\mathrm{AUST} + 273) + (T_{pm} + 273)] \tag{24}$$

An approximated expression for AUST given by Kilkis [22] was used in this study.

$$\mathrm{AUST} \approx T_a - d \cdot z \tag{25a}$$

$$z \cong \frac{7}{(T_{oa} - 45)} \quad \text{where, } 26\ {}^\circ\mathrm{C} \leqslant T_{oa} \leqslant 36\ {}^\circ\mathrm{C} \tag{25b}$$

Where, d is the room position index; d is 0.5 for an interior space, 1.0 for a room with one outdoor exposed side with fenestration less than 5% of the total room surface area or 2.0 for a room with fenestration greater than 5%, and 3.0 is for a room with two or more outdoor exposed sides.

The equivalent overall heat transfer coefficient Eq. (23) can replace the overall heat transfer coefficient (U_o) of the panel model; however, U_e can not be determined explicitly because the mean panel surface temperature (T_{pm}) is unknown. This unknown is determined by solving panel model equations Eqs. (6)–(20) and Eq. (23) iteratively for given boundary conditions. Once U_e and T_{pm} are converged to certain values, other quantities such as the panel cooling capacities (q_o, q_c, and q_r), heat transfer coefficients (h_c, and h_r) are determined.

5. Panel cooling capacity estimation

5.1. Model space

It is assumed that five aluminum radiant cooling panels (0.6 m × 3 m) are installed on the ceiling of a 3 m × 3 m × 3 m model space (Fig. 4). The topside of the panel is perfectly insulated,

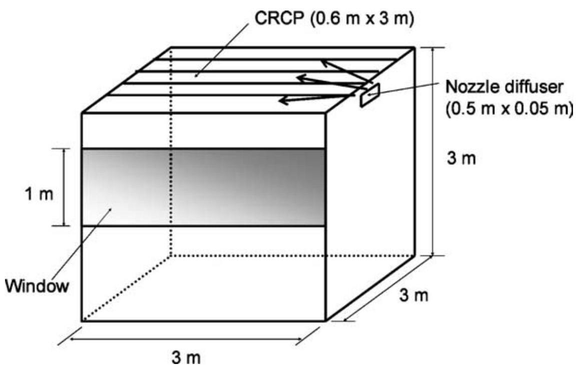

Fig. 4. Schematic of model space.

and four-row of parallel copper tubes is attached on the topside of each panel by high conductivity bond material. The tube outside diameter is 0.01 m, and the mass flow rate is 0.04 kg/s for each panel. The thermal resistance between the panel and the fluid is negligible. The model space has one exterior wall with small window area. The outdoor air (OA) temperature is 30 °C, and the space temperature is maintained at 26 °C. The air is supplied by the 0.5 m × 0.05 m nozzle diffuser located on a wall near the ceiling. The diffuser discharge air velocity varies from 2 to 6 m/s, and the panel inlet chilled water temperature changes from 12 to 25 °C.

5.2. Convection and radiation heat fluxes

Fig. 5(A)–(C) clearly shows that the convection heat flux (q_c) is enhanced by increasing discharge air velocity (V). The q_c values for simplified correlation, Awbi and Hatton's, and Fisher and Pedersen's correlation increase with discharge air velocity except for Chen et al.'s coefficient. Chen et al.'s coefficient is inherently insensitive to the diffuser discharge air velocity because they proposed a constant value (4W/m^2 K) as a MC coefficient for cooled ceiling.

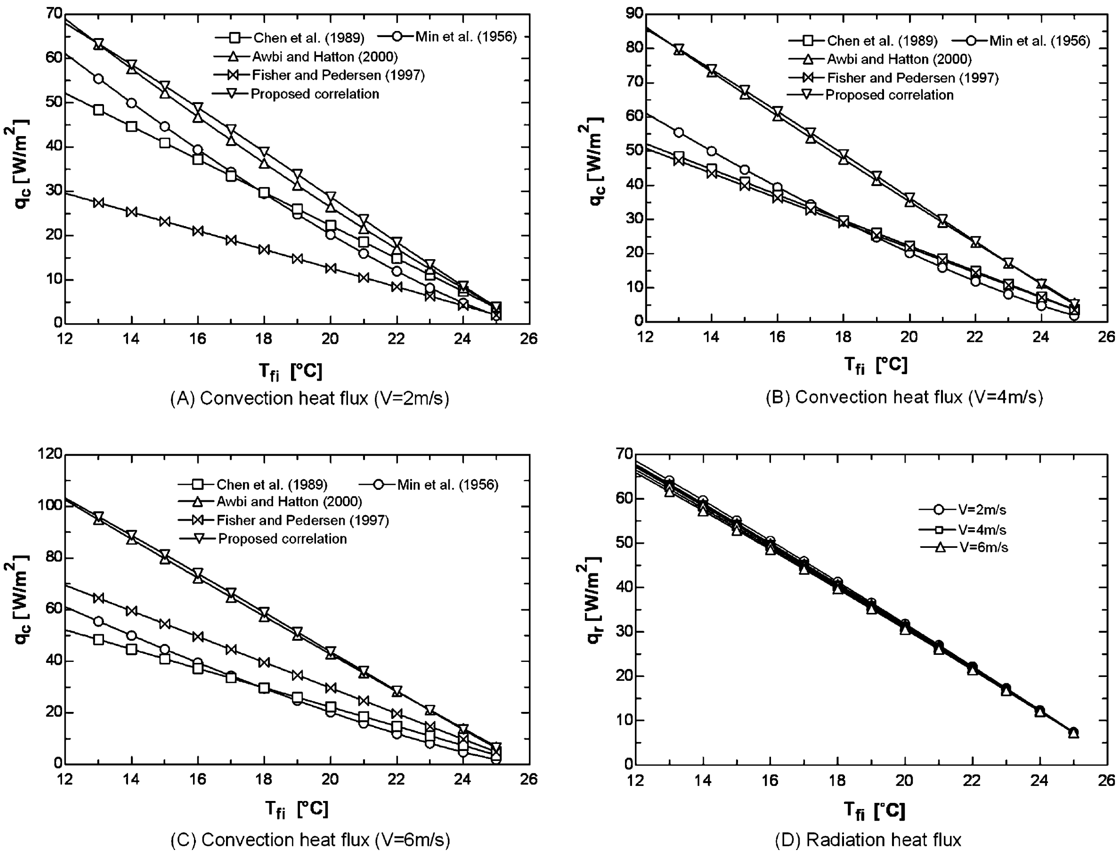

Fig. 5. Convection and radiation heat flux.

Note that the simplified correlation and Awbi and Hatton's correlation return higher convection heat fluxes than those determined by Min et al.'s NC correlation. Whereas convection heat fluxes for Fisher and Pedersen's and Chen et al.'s MC correlation are not much different from or even smaller than those for the NC correlation.

Fig. 5(D) shows that radiation heat fluxes (q_r) are not much affected by diffuser discharge air velocities and kind of convection coefficients since the panels operate with a very small water temperature rise or a nearly constant surface temperature.

6. Cooling capacity enhancement

6.1. Total cooling capacity

In Fig. 6, as the discharge air velocity increases from 2 to 6 m/s, the total cooling capacity (convection heat flux plus radiation heat flux) of the panel increases significantly due to the convection heat transfer enhancement, while the radiation part of the panel capacity is not much affected by the MC effect. On the other hand, when the diffuser discharge air velocity is less than 2 m/s, the total panel cooling capacity is not enhanced noticeably.

The percent of total cooling capacity enhancement caused by the MC effect is presented in Fig. 7. In practice, the inlet chilled water temperature of the cooling panel is around 15 °C (with 16 °C design surface temperature). In Fig. 7, the total cooling capacity is enhanced by the MC effect from 5% ($V = 2$ m/s) to 35% ($V = 6$ m/s) at 15 °C panel inlet chilled water temperature. This result

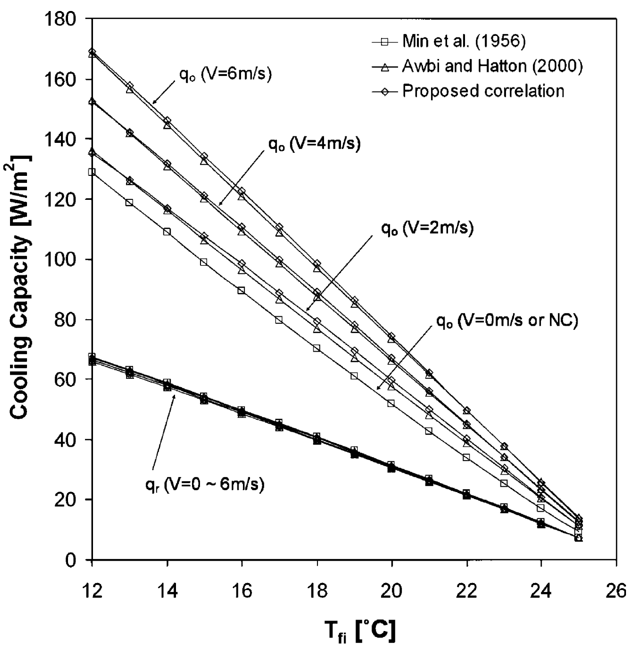

Fig. 6. Total cooling capacity of the panel.

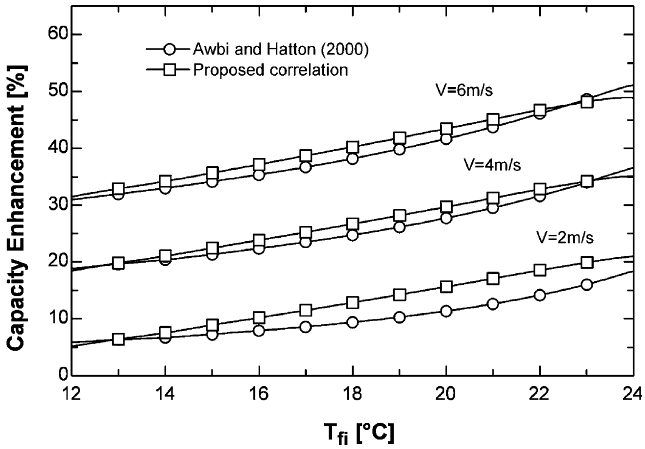

Fig. 7. Panel cooling capacity enhancement.

corresponds very well to the experiment results of Kochendörfer [4]. His field measurement for real buildings showed the cooling output of the panel increases over 25% compared with DIN 4715 test results. This increased capacity will finally reduce required panel area and initial cost.

7. Conclusions

The objective of this piece of work was to show the impact of the mixed convection on the cooling capacity of a ceiling radiant panel, which occurred in most of the mechanically ventilated space. The total cooling capacity of the panel was enhanced significantly with increasing diffuser discharge air velocity from 5% to 35% when the panel was at typical design temperature. It means that engineers can use 5–35% increased panel cooling capacity usually overlooked at the design stage. This increased capacity will finally reduce required panel area and initial cost. However, when the diffuser discharge air velocity is less than 2m/s, the impact of MC on the panel cooling capacity is small. Therefore the correlation for natural convection heat transfer coefficient can be used to estimate panel cooling capacity instead of the MC correlation for low velocities.

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