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# Simplified cooling capacity estimation model for top insulated metal ceiling radiant cooling panels

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## Abstract

The main thrust of this research was to develop a simplified cooling capacity estimating correlation for a top insulated metal ceiling radiant cooling panel (CRCP). By statistically analyzing the impact of various panel design parameters on the panel cooling capacity, a linear regression equation was derived. A validated analytical CRCP model was used to collect panel performance data for the various combinations of design parameters. In this analysis, it was found that eight single design parameters and eleven two-factor interactions significantly affect the panel cooling capacity. Consequently, a first order linear regression equation, or the simplified CRCP model, was derived as a function of the major single parameters and two-factor interactions. The proposed model returns the cooling capacity of a top insulated CRCP not only for the natural convection condition but also for the mixed convection condition present in mechanically ventilated spaces. The predicted panel cooling capacities for both aluminum and steel panels corresponded well with manufacturer's data and experimental results reported in the literature.

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*Keywords:* Radiant cooling; Ceiling radiant panel; Mixed convection; Natural convection; Radiant panel model

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## 1. Introduction

The ceiling radiant cooling panel (CRCP) system was first seriously investigated in European countries [1], and has become one of the popular design alternatives for space cooling in north-west Europe where the climate is relatively mild and dry. However to date, the CRCP system has

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## Nomenclature

$A$	area (m <sup>2</sup> )
AUST	area-weighted average temperature (°C)
$b_w$	bond width (m)
$C_p$	specific heat of the fluid (kJ/kg K)
$D_e$	characteristic diameter of room surface ( $= 4A_c/P$ ) (m)
$D$	tube diameter (m)
$d$	room position index
$F$	fin effectiveness
$F'$	panel efficiency factor
$F_R$	panel heat removal factor
$h$	convection (radiation) heat transfer coefficient (W/m <sup>2</sup> 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 <sup>2</sup> )
$q'_{fin}$	transferred energy to the fin (W/m)
$q'_{tube}$	heat gain from above the tube region (W/m)
$q'$	total sensible heat gain of the panel (W/m)
$T$	temperature (°C)
$T_{OA}$	outside air temperature (°C)
$\Delta T$	temperature difference between the space and the panel mean surface temperature (°C)
$U$	heat transfer coefficient (W/m <sup>2</sup> K)
$V$	inlet air velocity (m/s)
$w$	distance between the tubes (m)
$W$	width of nozzle diffuser (m)
$y$	flow direction
$z$	adjustment factor for AUST (°C)

### Greeks

$\delta$	panel thickness (m)
$\gamma$	bond thickness (m)

### Subscripts

a	air, space
b	bond material, fin base
c	ceiling, convection
e	equivalent
f	forced, fluid
fi	fluid inlet

fo	fluid outlet
i	inside
m	mean
n	natural
o	outside, total, overall
p	panel
r	radiation

not been able to penetrate US market because of traditional negative perceptions of condensation, leakage, first-cost, and cooling capacity.

In early 1990s, interest in the CRCP system began to increase once the condensation issues had been addressed by independent ventilation systems designed to meet the entire space latent load and the required ventilation rate. First of all, the energy conservation and indoor air quality (IAQ) benefits of the decoupled (or hybrid) system began to attract attention [2–4]. In addition, it was also indicated that the traditional negative perceptions on the CRCP system need no longer be valid [5].

The various types of hydronic radiant cooling systems include: metal ceiling panels, chilled beams, and tube imbedded ceilings-walls-floors. Metal CRCPs are widely used and are frequently installed on T-bar grids designed to support the dropped acoustical ceiling. The panels in the dropped ceiling are top loaded with insulation to prevent heat gain from the plenum space. This top insulated CRCP system is also called a closed type CRCP system by the manufacture.

The CRCP system is able to accommodate varying space sensible loads by surface temperature control. Heat is transferred to the radiant panel by the heat transfer mechanisms of convection and radiation. ASHRAE [6] panel cooling capacity equations are based on radiation and natural convection (NC). ASHRAE chose to neglect the contribution of forced convection (FC) in mechanically ventilated systems, reasoning conservatively that not all designs could reliably increase the total cooling capacity of CRCPs. Consequently, the existing or draft CRCP test standards in US and Europe, such as DIN 4715 [8], ASHRAE/ANSI SPC 138P [25], and prEN 14240 [26], are based on NC and adiabatic wall conditions.

However Kochendörfer [7] indicated that cooling output of CRCPs are significantly higher (over 25%) in a real building than laboratory measured panel capacities under standard testing condition expressed in DIN 4715 [8]. The reasons for this higher capacity are non-standard surrounding conditions, such as warm windows and outside walls, and mechanical ventilation systems. Actually, the DIN 4715 test is required to be performed in a test room with adiabatic walls and no mechanically induced air motion. The underestimated panel cooling capacity results in higher investment and operating costs.

Mumma [9] also indicated that if air is supplied to the space via high induction diffusers the convective heat transfer to the panels increases by about 15% more than panels operating in still air. In another analytical work [10], it was found that the panel cooling capacity can be enhanced

by the FC effect in a mechanically ventilated room from 5% to 35%. The convective heat transfer on a panel surface can be estimated more closely to the real one by considering not only the buoyancy effect but also the FC effect on a CRCP.

For the last couple of decades, several numerical and analytical models for a CRCP have been proposed [11–14]. However, most models were developed under the NC condition without considering the FC effect on a panel. No models currently exist to estimate the panel cooling capacity in a mechanically ventilated space appropriately.

In this research, a simplified yet reliable top insulated metal CRCP model was developed that is capable of estimating the panel cooling capacity for either the NC or the mixed convection (MC) (i.e. combined NC and FC) condition typical of mechanically ventilated spaces.

## 2. Overview of existing CRCP models

In principle, CRCPs respond transiently to a change in room loads. However, the response time constant in a metal CRCP is very short (<5 min). This justifies developing a steady-state or quasi-steady-state radiant panel model which may be sufficient for engineering calculations and hourly thermal analysis procedures.

ASHRAE [6] uses the correlations developed by Min et al. [15] to estimate the NC heat flux received by CRCP (Eq. (1)), and to use the mean radiant temperature (MRT) method proposed by Walton [16] to estimate the radiation heat flux received by CRCP (Eq. (2)). Finally, given the effective panel surface temperature ( $T_p$ ) and the area-weighted average temperature (AUST) of uncontrolled surfaces in a room, the convective and the radiation heat transfer to a panel can be calculated:

$$q_c = 2.13 \cdot (T_a - T_p)^{0.31} (T_a - T_p) \quad (1)$$

$$q_r = 5 \times 10^{-8} \cdot [(AUST + 273)^4 - (T_p + 273)^4] \quad (2)$$

Chen and Kooi [11] treated the cooled ceiling as an individual surface that exchanges heat convectively with room air and radiantly with other building surfaces in their energy simulation program. Heat conduction within the ceiling panels was treated as one-dimensional, and the chilled water temperature variation along the panels was approximated using an empirical linear correlations.

A typical in-slab type panel model was proposed by Kilkis et al. [12]. They noted that the heat transfer in a panel-cooled room and the cooling panel itself might be represented by a quasi-steady-state natural convection model. Initially, they assumed uniform panel surface temperatures; however this assumption is only true for an infinite number of tubes. To predict the non-uniform temperature profile between the tubes, they proposed a composite fin model, and ignored the temperature variation in the direction of fluid flow in the tube. Using this steady state panel model, Kilkis [17] developed a computer based design tool for a CRCP system, and also proposed a monograph for manual panel capacity estimates.

Stetiu and Feustel [13] proposed an in-slab type, 2-D radiant panel model. They simplified Fourier's heat conduction equation and the heat diffusion equation by introducing lumped

thermal resistance and thermal capacitance for finite-differenced panel layers, and then determined temperature distribution in the panel.

Bohle et al. [14] developed a finite element model (FEM) for plaster ceilings containing embedded plastic tubing. They assumed steady state heat transfer, and that the AUST was equal to the space temperature. Temperature distributions, especially at the panel surface, were calculated as a function of the fluid temperature. After averaging the resultant local panel surface temperature profiles, the heat flux between the panel surface and the space could be calculated using an experimentally determined overall heat transfer coefficient. They also derived a single power function (Eq. (3)) from systematic calculation and variation of parameters with their FEM model:

$$q_o = B \cdot \prod_{i=1}^j a_i^{m_i} \cdot (T_a - T_f) \quad (3)$$

Conroy and Mumma [18] derived an analytical model for a top insulated metal CRCP with parallel tubes, which was based on the basic derivation by Hottel and Whillier [19] and Whillier [20]. They assumed the natural convection and steady state condition, and the topside of a panel is perfectly insulated. In this model, the panel cooling capacity ( $q_o$ ) is determined by finding unknown mean panel surface temperature ( $T_{pm}$ ) in an iterative process.

On the other hand, ASHRAE [6] indicated that an analytical panel model proposed by the Turkish Standard Institute (TSI) [24] may also be used to design and analyze panels. This model looks very similar to Conroy and Mumma's analytical panel model. However, there are some differences between those two models: First, the fluid temperature variation in the direction of flow in the tube is ignored in the TSI model. Second, the panel capacity ( $q_o$ ) required to meet the sensible cooling load allocated to the panel should be known before the calculation. In other words, this model returns the mean panel surface temperature ( $T_{pm}$ ) and mean fluid temperature ( $T_{fm}$ ) for the known panel capacity, while Conroy and Mumma's model estimates the unknown panel capacity by determining the unknown mean panel surface temperature in an iterative process for given boundary conditions.

### 3. Analytical model for top insulated metal CRCP

In this research, Conroy and Mumma's [18] analytical panel model was modified to enable consideration of both NC and MC in a mechanically ventilated space. The typical cross-sectional geometry of a panel (Fig. 1) and details of the modified analytical model are presented below.

The sensible heat absorbed by the CRCP must be conducted along the panel (fin) to the region of the tubes. The temperature distribution between the tubes (Eq. (4)) is derived from the energy balance on a fin element by temporarily assuming that the temperature gradient in the flow direction is negligible. By evaluating Fourier's heat conduction equation at the fin base, the energy transferred to the fin base per unit length in the flow direction ( $q'_{fin}$ ) is given by Eq. (5). 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. (6)):

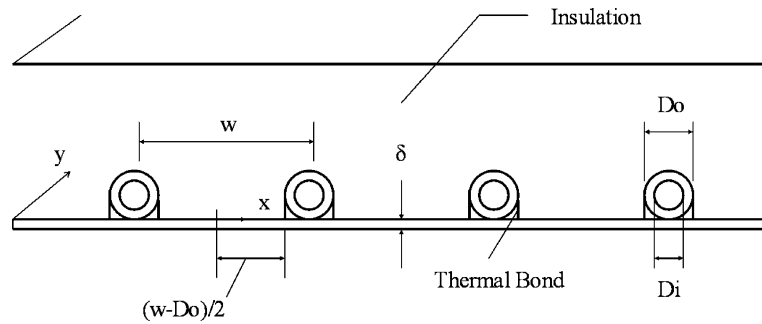


Fig. 1. Cross-sectional geometry of top insulated metal CRCP.

$$\frac{T_p(x) - T_a}{T_b - T_a} = \frac{\cosh(mx)}{\cosh(m(w - D_o)/2)} \quad (4)$$

$$q'_{\text{fin}} = -FU_o(w - D_o)(T_b - T_a) \quad (5)$$

$$F = \frac{\tanh(m(w - D_o)/2)}{m(w - D_o)/2} \quad \text{where } m = \sqrt{U_o/k \cdot \delta} \quad (6)$$

The sensible heat gain by the panel also includes the heat gain from the area immediately below the tube (Eq. (7)). The applicable boundary conditions are, the heat flux in the fin is zero midway between the tubes and the fin temperature immediately below the tubes is the fin base temperature ( $T_b$ ). By adding Eqs. (5) and (7), the total sensible heat gain by the panel per unit length ( $q'$ ) can be expressed as Eq. (8):

$$q'_{\text{tube}} = -D_o U_o (T_b - T_a) \quad (7)$$

$$q' = -[(w - D_o)F + D_o] \cdot U_o \cdot (T_b - T_a) \quad (8)$$

Ultimately,  $q'$  must be transferred to the fluid. The resistance path from the fin to the fluid consists of the thermal bond resistance between the tube and fin and the tube-to-fluid film coefficient resistance (Eq. (9)). By eliminating  $T_b$  from Eqs. (8) and (9),  $q'$  can be expressed in terms of known physical dimensions, thermal/physical parameters, and the local fluid temperature ( $T_f$ ) (Eq. (10)). The panel efficiency factor ( $F'$ ) (Eq. (11)) is a ratio of overall heat transfer coefficient between fluid and room to overall heat transfer coefficient between fin and room:

$$q' = \frac{T_b - T_f}{\frac{1}{h_i \pi D_i} + \frac{\gamma}{k_b b_w}} \quad (9)$$

$$q' = -wF'U_o(T_f - T_a) \quad (10)$$

$$F' = \frac{1/U_o}{w \left[ \frac{1}{U_o[D_o + (w - D_o)F]} + \frac{1}{h_i \pi D_i} + \frac{\gamma}{k_b b_w} \right]} \quad (11)$$

The forced convection heat transfer coefficient for turbulent flow inside the tube ( $h_i$ ) was determined by Eq. (12) [21]. It was assumed that the thermal resistance between the tube wall and the fin (i.e.  $\gamma/k_b b_w$  term in Eq. (11)) is negligible based upon current common CRCP construction:

$$h_i = \frac{Nu_D \cdot k_f}{D_h}, \quad \text{where } Nu_D = 0.023 \cdot Re_D^{4/5} \cdot Pr^{0.4} \tag{12}$$

As heat is absorbed by the panel the fluid temperature increases in the flow direction (i.e.  $y$ -direction). The temperature distribution in the flow direction can be derived by applying mass and energy balances and is given by Eq. (13):

$$\frac{T_f(y) - T_a}{T_{fi} - T_a} = \exp\left(-\frac{nU_o w F'}{MC_p} y\right) \tag{13}$$

The mean fluid temperature ( $T_{fm}$ ) can be determined by integrating Eq. (13) from  $y = 0$  to  $y = L$  (at the outlet). Performing this integration, and after some algebraic manipulation, the mean fluid temperature can be expressed as Eq. (14):

$$T_{fm} = T_{fi} + \frac{q_o}{F_R U_o} \left(1 - \frac{F_R}{F'}\right) \tag{14}$$

On the other hand, it is convenient to define the panel heat removal factor ( $F_R$ ) that 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. (15)), then the total sensible heat absorbed by the panel ( $q_o$ ) can be expressed as Eq. (16). The total sensible heat gain can also be expressed in terms of the mean panel temperature ( $T_{pm}$ ) (Eq. (17)). By equating these two equations and solving for  $T_{pm}$ , the expression for the mean panel temperature is given by Eq. (18):

$$F_R = \frac{MC_p(T_{fo} - T_{fi})}{A_p U_o (T_a - T_{fi})} \tag{15}$$

$$q_o = F_R U_o (T_a - T_{fi}) \tag{16}$$

$$q_o = U_o (T_a - T_{pm}) \tag{17}$$

$$T_{pm} = T_{fi} + \frac{MC_p(T_{fo} - T_{fi})}{A_p F_R U_o} \cdot (1 - F_R) \tag{18}$$

This analytical model for top insulated metal CRCP requires knowledge of the overall heat transfer coefficient ( $U_o$ ). However determination of  $U_o$  is not a simple problem because of the fact that space temperature ( $T_a$ ) is generally not at the AUST of the space enclosure during radiant cooling. In principal, the total heat flux ( $q_o$ ) is the summation of the convective heat flux ( $q_c$ ) and the radiation heat flux ( $q_r$ ) as given by Eq. (19a), and each component of the total heat flux is expressed by Eqs. (19b) and (19c), respectively:

$$q_o = q_c + q_r \tag{19a}$$

$$q_c = h_c \cdot (T_a - T_{pm}) \tag{19b}$$

$$q_r = h_r \cdot (AUST - T_{pm}) \tag{19c}$$

In Eq. (19),  $U_o$  can be readily determined by summing the convection heat transfer coefficient ( $h_c$ ) and the radiation heat transfer coefficient ( $h_r$ ) if  $T_a = \text{AUST}$ ; however, they are generally not equal. Therefore, in this research the equivalent overall heat transfer coefficient ( $U_e$ ) has been obtained by substituting Eq. (19) into Eq. (20). Final form of  $U_e$  is given by Eq. (21). The radiation heat transfer coefficient  $h_r$  (Eq. (22)) was found in the literature [6], and the simplified MC heat transfer coefficient  $h_c$  (Eq. (23)) developed by Jeong and Mumma [22] was used to consider MC effect on the CRCP capacity:

$$U_e = \frac{q_o}{(T_a - T_{pm})} \quad (20)$$

$$U_e = \frac{q_c + q_r}{(T_a - T_{pm})} = h_c + h_r \cdot \frac{(\text{AUST} - T_{pm})}{(T_a - T_{pm})} \quad (21)$$

$$h_r = 5 \times 10^{-8} \cdot [(\text{AUST} + 273)^2 + (T_{pm} + 273)^2] \cdot [(\text{AUST} + 273) + (T_{pm} + 273)] \quad (22)$$

$$h_c = F_c + 2.13 \cdot \Delta T^{0.31} \quad (23a)$$

$$F_c = \alpha_0 + \alpha_1(\Delta T) + \alpha_2(V) + \alpha_3(W) + \alpha_4(V \cdot W) \quad (23b)$$

where,

$\alpha_0$	$\alpha_1$	$\alpha_2$	$\alpha_3$	$\alpha_4$
0.28021	-0.13931	0.11416	1.25013	1.22058

The AUST in Eqs. (21) and (22) can be expressed as Eq. (24) given by Kilkis et al. [12]. This equation gives good approximation of the impact of outdoor conditions and solar radiation on the AUST:

$$\text{AUST} \approx T_a - d \cdot z \quad (24a)$$

$$z \cong \frac{7}{(T_{OA} - 45)} \quad \text{where } 26 \text{ }^\circ\text{C} \leq T_{OA} \leq 36 \text{ }^\circ\text{C} \quad (24b)$$

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, 2.0 for a room with fenestration greater than 5%, and 3.0 for a room with two or more outdoor exposed sides. The AUST adjustment factor ( $z$ ) is a function of outdoor air temperature ( $T_{OA}$ ).

The equivalent overall heat transfer coefficient ( $U_e$ ), defined in Eq. (21), can replace the overall heat transfer coefficient ( $U_o$ ). However  $U_e$  cannot be determined explicitly because the mean panel surface temperature ( $T_{pm}$ ) is still unknown.  $T_{pm}$  can be determined by solving the panel model equations (Eqs. (4)–(19) and Eq. (21) for any specific boundary conditions in an iterative process from an initial trial value of  $T_{pm}$ . Once  $U_e$  and  $T_{pm}$  have converged to an acceptable tolerance, the panel cooling capacities ( $q_o$ ,  $q_c$ , and  $q_r$ ) and heat transfer coefficients ( $h_c$ , and  $h_r$ ) are easily determined.



#### 4. Simplified top insulated metal CRCP model

From the analytical panel model presented in Section 3, the panel cooling performance data generated for various combinations of panel design parameters were analyzed statistically. The goal was to derive a simple correlation which returns the panel cooling capacity as a function of major design parameters and their interactions, which significantly impacted the panel cooling capacity.

##### 4.1. $2^k$ factorial experiment design approach

The  $2^k$  factorial experiment design method [23] was used to determine which design parameters and their interactions would significantly impact the panel cooling capacity and to derive a linear regression equation as a function of those parameters with a small number of experiments (or simulation). The superscript  $k$  means the total number of parameters considered in an experiment, and for each parameter only two values (i.e. maximum and minimum) are generally considered.

This method is frequently applied to design physical experiments. In the case of physical experiments, it is typical to repeat each experiment several times (so called replication), and to compute effects of each considering variables on the response variable using the average response. And then estimating the error associated with the measurement of each response variable and the probability that differences in average responses would have occurred due to experimental error.

On the other hand, the  $2^k$  factorial experimental design method is also applicable to the analysis of simulation results. In the case of simulations, the replication is not required, and no estimate for experimental error can be made. Since computer simulations are not subject to randomized experimental error, such an analysis is not necessary.

The single parameters and their combinations which have significant influence on the response variable can be identified visually from a normal probability plot of effects. Effects are defined as the average change in response that occurs as a result of changing each parameter from a low value to high value. If some parameters have negligible effects, a linear relationship that estimates the response as a function of significant parameters can be derived.

##### 4.2. Derivation of simplified top insulated CRCP model

In this research, 13 parameters (Table 1) were selected as major design variables which should be generally considered in a CRCP design. The high and low values for each design parameter required for the  $2^k$  factorial experiment design were defined using the manufactures' data and typical room design conditions.

In this particular case,  $2^{13}$  (= 8192) experiments are required to perform the full factorial experiment, since it has 13 parameters to consider. However, if it is assumed that certain high order interactions (i.e. ABC, BCD, ..., ABCDEFGHJKLMN) have negligible effects on the response variable or the panel cooling capacity compared with single and low order interactions (i.e. A, B, C, ..., AB, CD, ..., MN), information on the main effects and low order interactions may be obtained by running only a fraction of the complete factorial experiment. It is called the fractional factorial design method [23].

Table 1  
Design parameters for a top insulated metal CRCP

Label	Parameter	Low	High
A	Tube spacing ( $w$ ), m	0.15	0.3
B	Panel thickness ( $\delta$ ), m	0.0007	0.002
C	Panel thermal conductivity ( $k$ ), W/m K	60	237
D	Tube outside diameter ( $D_o$ ), m	0.01	0.015
E	Number of tubes ( $n$ )	2	4
F	Panel length ( $L$ ), m	0.6	1.2
G	Inlet fluid temperature ( $T_f$ ), °C	10	18
H	Fluid velocity ( $V_f$ ), m/s	0.25	0.7
J	Diffuser discharge velocity ( $V$ ), m/s	2	6
K	Diffuser width ( $W$ ), m	0.2	0.8
L	Room air temperature ( $T_a$ ), °C	26	28
M	Outside air temperature ( $T_{OA}$ ), °C	26	36
N	Room position index ( $d$ )	0.5	3

This method is generally categorized by three types (Resolution III, IV, and V) based on the grouping (or alias) structure of the single factor or low order interactions with high order interactions. In general, the higher the resolution, the less restrictive the assumptions that are required regarding which interactions are negligible to obtain a unique interpretation of the data. Therefore, in this research, Resolution V design was used.

In the Resolution V fractional factorial design, the single factors and the two factor interactions do not have other single factors and two factor interactions as their aliases. This is very powerful design, allowing that all the three factor and higher order interactions are negligible. Consequently, the full factorial ( $2^{13}$ ) experiments could be reduced to  $2^{-5}$  fractional factorial or  $2^{13-5}$  ( $=2^8$  or 256) experiments by using the Resolution V fractional factorial design method.

By analyzing the panel performance data obtained from  $2^8$  simulations designed by Resolution V fractional factorial experiment design method, the single factors and two factor interactions which have significant effects on the cooling performance were identified visually from a normal probability plot of effects.

Fig. 2 shows that the normal probability plot of effects of individual parameters and their interactions on the total (convection plus radiation) cooling capacity ( $q_o$ ) of a CRCP. In this analysis, eight single parameters and eleven two-factor interactions (Table 2) showed significant effects on the response variable compared with other parameters and interactions. In Table 2, the percent contribution of each selected parameters to the response variable or the total cooling capacity of a panel is also presented. The percent contribution comes from adding up the total sum of squares and then taking each term's sum of squares and dividing by the total to get a percentage [23]. The selected terms showed 0.30% to 49.51% contribution, while the other terms showed negligible contributions ( $<0.30\%$ ).

Based on this result, a first order linear regression equation which returns the panel cooling capacity in  $W/m^2$ , was derived as a function of above selected parameters and interactions (Eq. (25)). The  $R^2$  value for this proposed correlation is 0.98. The required coefficients for the panel capacity equation are presented in Table 3:

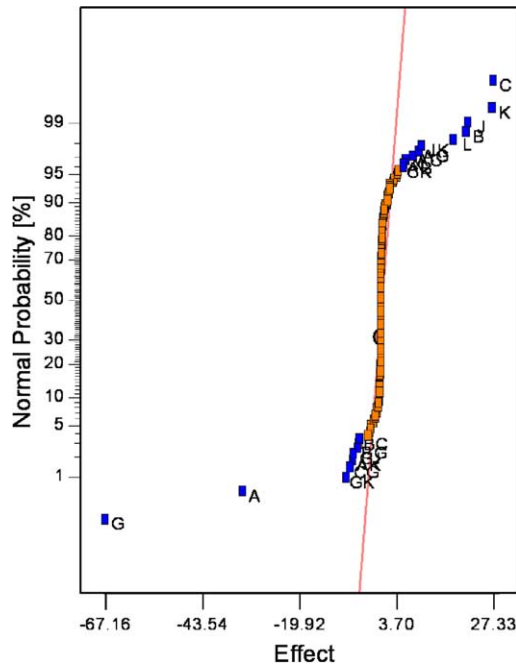


Fig. 2. Normal probability plot of the effects on the total cooling capacity ( $q_o$ ).

Table 2  
Selected parameters and percent contribution

8 Major parameters								
Label	A	B	C	G	J	K	L	N
Parameter	$w$	$\delta$	$k$	$T_{fi}$	$V$	$W$	$T_a$	$d$
Contribution (%)	14.43	4.68	8.20	49.51	4.88	8.02	3.40	0.39
11 Two-factor interactions								
Label	AB	AC	AG	AK	BC	BG	CG	CK
Parameter	$w \cdot \delta$	$w \cdot k$	$w \cdot T_{fi}$	$w \cdot W$	$\delta \cdot k$	$\delta \cdot T_{fi}$	$k \cdot T_{fi}$	$k \cdot W$
Contribution (%)	0.33	0.67	0.93	0.52	0.30	0.36	0.62	0.32
Label	GJ	GK	JK					
Parameter	$T_{fi} \cdot V$	$T_{fi} \cdot W$	$V \cdot W$					
Contribution (%)	0.49	0.81	1.05					

$$\begin{aligned}
 q_o = & \alpha_0 + \alpha_1(w) + \alpha_2(\delta) + \alpha_3(k) + \alpha_4(T_{fi}) + \alpha_5(V) + \alpha_6(W) + \alpha_7(T_a) + \alpha_8(d) + \alpha_9(w \cdot \delta) \\
 & + \alpha_{10}(w \cdot k) + \alpha_{11}(w \cdot T_{fi}) + \alpha_{12}(w \cdot W) + \alpha_{13}(\delta \cdot k) + \alpha_{14}(\delta \cdot T_{fi}) + \alpha_{15}(k \cdot T_{fi}) \\
 & + \alpha_{16}(k \cdot W) + \alpha_{17}(T_{fi} \cdot V) + \alpha_{18}(T_{fi} \cdot W) + \alpha_{19}(V \cdot W)
 \end{aligned} \tag{25}$$

The reliability of the above regression equation can be confirmed by inspection of the normal probability plot of residuals. The residual is defined as the difference between actual data and predicted data. If the points on this plot lie reasonably close to the straight line representing

Table 3  
Coefficients for simplified top insulated metal CRCP model

$\alpha_0$	$\alpha_1$	$\alpha_2$	$\alpha_3$	$\alpha_4$
-55.0617	-526.899	25332.08	0.182118	-5.32906
$\alpha_5$	$\alpha_6$	$\alpha_7$	$\alpha_8$	$\alpha_9$
7.030547	81.64985	8.805391	2.398	56608.97
$\alpha_{10}$	$\alpha_{11}$	$\alpha_{12}$	$\alpha_{13}$	$\alpha_{14}$
0.58883	15.35339	-152.417	-45.6568	-1099.94
$\alpha_{15}$	$\alpha_{16}$	$\alpha_{17}$	$\alpha_{18}$	$\alpha_{19}$
-0.01065	0.101018	-0.41732	-3.56953	8.166797

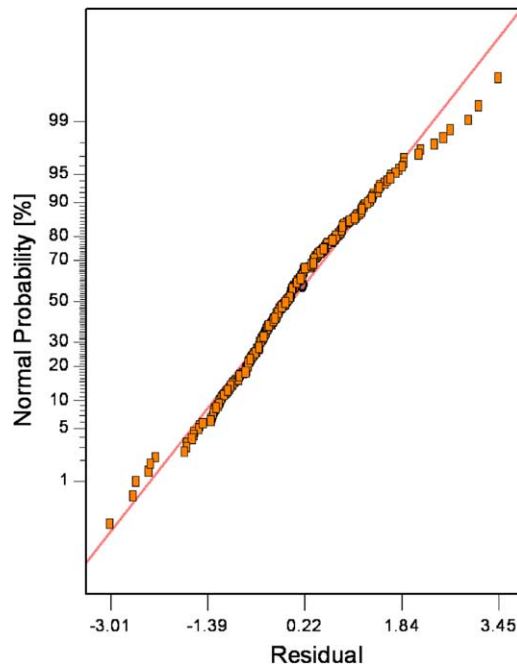


Fig. 3. Normal probability plot of residuals of the simplified CRCP model.

normal probability distribution, one can conclude that significant effects have been taken into account. Fig. 3 shows that the normal probability plot of residuals of the proposed panel model (Eq. (25)) is very close to the straight line. In addition, the panel cooling capacities predicted by the proposed correlation are compared with the analytical model results in Fig. 4. It also shows that the simplified CRCP capacity model predictions are close to the analytical model.

Significantly, the proposed panel model can be used to estimate the panel cooling capacity not only for the NC condition but also for the MC condition. For example, if a space has no

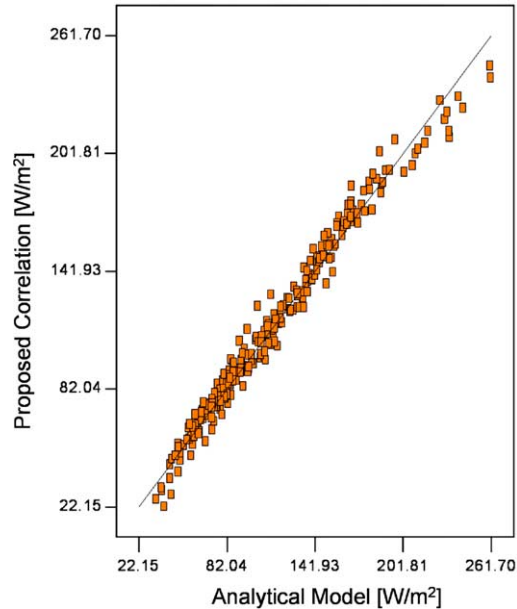


Fig. 4. Analytical model versus proposed correlation.

mechanical ventilation (i.e. set  $V$  to zero) the proposed panel model returns the panel cooling capacity for the NC condition.

## 5. Validity of simplified CRCP model

In order to confirm the validity of the proposed regression model, panel cooling capacities for the NC condition were computed by ASHRAE equations (i.e. Eqs. (1) and (2)), TSI model recommended by ASHARE, and Conroy and Mumma's model. Three different performance data from British, Canadian, and Italian manufacturers were also considered in this research, although only one of them is presented in this piece of work. Those manufacturers' data were not significantly different from other one.

In general, manufacturers offer actual panel performance data measured in accordance with the standard test conditions defined by DIN 4715 (i.e. the NC condition with adiabatic walls). Correction factors for other design conditions, such as radiation asymmetry, hot window surfaces, and room height, which are different from the DIN 4715 test condition, are also presented. The total uncertainty of each manufacturer's data is  $\pm 3\%$  as limited by the test standard.

On the other hand, confirming the validity of the panel performance for MC conditions is more difficult because manufactures' data are based on the NC condition only and the actual measured panel cooling capacity data under the MC condition, or in a mechanically ventilated room, are vary rare. Therefore the MC condition case was compared with existing preliminary research

results found in the literature [7,10]. Extensive experimental panel capacity enhancement research in a mechanically ventilated space is sorely needed.

### 5.1. Model space conditions

Two kinds of panel materials, aluminum ( $\delta = 1.0$  mm,  $k = 206$  W/m K) and steel ( $\delta = 0.7$  mm,  $k = 60$  W/m K), which are widely used in the CRCP industry were considered in this research. The cooling capacity of each panel was estimated for the following assumed room conditions (Fig. 5).

Five  $0.6 \text{ m} \times 3 \text{ m}$  radiant cooling panels are installed on the ceiling of the model space ( $3 \text{ m} \times 3 \text{ m}$ ). Four rows of parallel copper tubes are attached on the topside of each panel, and insulation materials are applied to the top side to prevent heat gains from the plenum space. The tube inside fluid velocity is  $0.3$  m/s, the spacing between the tubes is  $0.15$  m, and the outside ( $D_o$ ) and the inside diameter ( $D_i$ ) of the tube are  $0.01$  and  $0.0085$  m, respectively. The model space has one exterior wall, and less than 5% of the total surface area is fenestration area (i.e. room position index  $d$  is 1.0). The outdoor air temperature ( $T_{OA}$ ) is  $30$  °C, and the space temperature ( $T_a$ ) is maintained at  $26$  °C. The air is supplied to the space at neutral temperature through the  $0.5 \text{ m} \times 0.05 \text{ m}$  nozzle diffuser located on a wall near the ceiling. The diffuser discharge air velocity varies from  $0$  to  $6$  m/s.

### 5.2. Validity of proposed model for the natural convection condition

The proposed CRCP model (Eq. (25)) can be used to estimate the panel cooling capacity in  $\text{W/m}^2$  for the NC condition (i.e. no mechanical ventilation) by setting the diffuser discharge air velocity ( $V$ ) to zero. In this research, total cooling capacities for both steel and aluminum panels, in the model space with no mechanical ventilation, were estimated using the proposed correlation, and then compared with the manufacturer's data and the results from other analytical CRCP models: ASHRAE equations, TSI model, Conroy and Mumma's model.

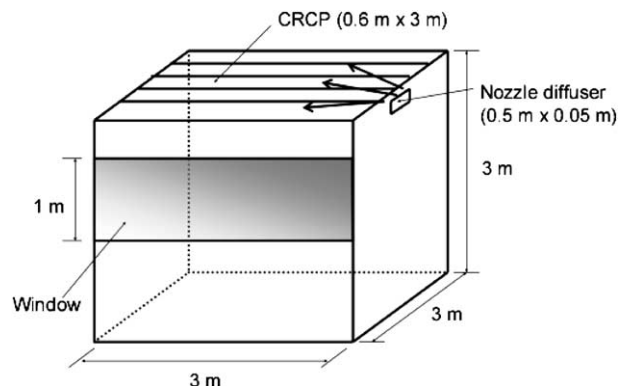


Fig. 5. Schematic of model room conditions.

Table 4  
Comparison of predicted cooling capacity for the NC condition

Inlet fluid temperature ( $T_{fi}$ ), °C	Proposed correlation Eq. (25)	Conroy and Mumma's model	ASHARE equation Eq. (1) + (2)	TSI analytical model	Manufacturer's data
<i>Aluminum panel, W/m<sup>2</sup></i>					
14	103.7	97.8	104.0	104.5	98.8
15	95.6	90.1	94.5	96.2	88.7
16	87.5	82.2	85.3	87.9	79.5
17	79.4	74.3	76.0	79.7	70.0
18	71.3	66.3	66.9	71.4	60.9
19	63.2	58.2	58.0	62.1	52.4
20	55.1	50.1	49.2	53.8	43.9
<i>Steel panel, W/m<sup>2</sup></i>					
14	80.6	81.4	84.2	83.5	94.6
15	74.4	74.8	76.6	76.7	85.3
16	68.1	68.2	69.0	70.0	76.6
17	61.9	61.5	61.6	62.2	67.7
18	55.7	54.8	54.3	56.4	59.0
19	49.5	48.1	47.1	49.6	50.9
20	43.3	41.4	40.1	42.5	42.5

The results for various inlet chilled water temperatures ( $T_{fi}$ ) (e.g. 14–20 °C) are presented in Table 4. It shows that the simplified CRCP model panel cooling capacity estimates for the NC condition case are close to both manufacturer's and analytical models' data.

5.3. Validity of proposed model for the mixed convection condition

In principle, the MC effect caused by the mechanical ventilation increases the convective portion of the panel cooling capacity, although the industry has and continues to overlook this increased capacity. There are a few research works investigating the MC effect on the panel cooling capacity.

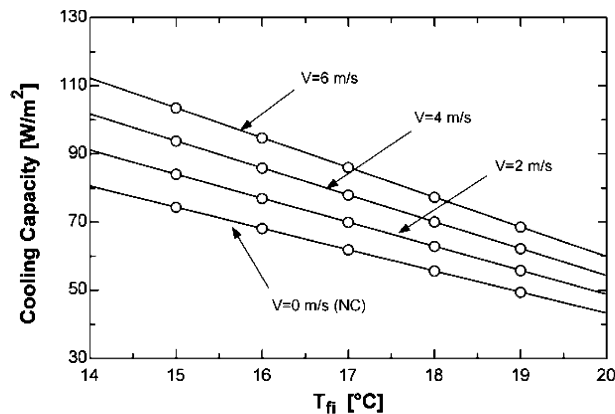


Fig. 6. Total cooling capacity for the steel panel.

Kochendörfer's [7] model room test with ceiling air diffusers showed that cooling panel outputs increase 10–15% with additional ventilation systems. In addition, his field measurements for real buildings showed that the cooling output increased over 25% compared with DIN 4715 test results. Recently, Jeong and Mumma [10] also indicated that the panel cooling capacity can be enhanced by the MC effect from 5% to 35%.

In this research, total cooling capacities predicted by the proposed model for the MC condition were preliminarily compared with existing research results [7,10]. Figs. 6 and 7 clearly show that cooling panel outputs are enhanced in both steel and aluminum panels at higher diffuser discharge air velocity. In other word, the higher the discharge air velocity, the more enhanced cooling capacity can be obtained.

On the other hand, the cooling capacity enhancement rates or percent ratios of capacity increment to the cooling capacity for the NC condition in both panels are presented in Table 5. It shows that panel cooling outputs for the steel panel and the aluminum panel are enhanced from 13% to 39% and from 10% to 30%, respectively. These results correspond very well to Kochendörfer's [7] and Jeong and Mumma's [10] research results.

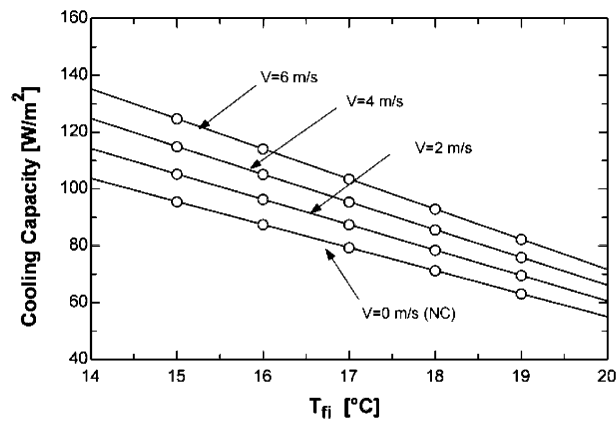


Fig. 7. Total cooling capacity for the aluminum panel.

Table 5  
Predicted cooling capacity and enhancement rate

$T_{fi}$ (°C)	Steel panel, W/m <sup>2</sup>			Aluminum panel, W/m <sup>2</sup>		
	$V = 2$ m/s	$V = 4$ m/s	$V = 6$ m/s	$V = 2$ m/s	$V = 4$ m/s	$V = 6$ m/s
14	91.1 (13%) <sup>a</sup>	101.7 (26%)	112.2 (39%)	114.2 (10%)	124.8 (20%)	135.3 (30%)
15	84.1 (13%)	93.8 (26%)	103.5 (39%)	105.3 (10%)	115.0 (20%)	124.7 (30%)
16	77.0 (13%)	85.9 (26%)	94.8 (39%)	96.4 (10%)	105.2 (20%)	114.1 (30%)
17	70.0 (13%)	78.0 (26%)	86.0 (39%)	87.4 (10%)	95.5 (20%)	103.5 (30%)
18	62.0 (13%)	70.1 (26%)	77.3 (39%)	78.5 (10%)	85.7 (20%)	92.9 (30%)
19	55.9 (13%)	62.2 (26%)	68.6 (39%)	69.6 (10%)	75.9 (20%)	82.3 (30%)
20	48.8 (13%)	54.3 (26%)	59.9 (38%)	60.6 (10%)	66.2 (20%)	71.7 (30%)

<sup>a</sup> Panel cooling capacity enhancement rate.



## 6. Conclusions

The objective of this piece of research was to propose a simplified cooling capacity estimation model for a top insulated metal CRCP. The proposed model was derived by statistically analyzing panel performance data collected from verified analytical panel model. Thirteen parameters were selected as major factors which should be considered in panel design. And then the effects of each design parameter on the panel performance were estimated by the factorial  $2^k$  fractional experiment design method of Resolution V.

In this analysis, eight single factors and eleven two-factor interactions showed significant effects on the panel cooling capacity. Consequently, a first order linear regression equation or the simplified CRCP model was derived as a function of those major single parameters and two-factor interactions.

The proposed CRCP model estimates the cooling capacity not only for the NC condition but also for the MC condition. The predicted panel cooling capacities for both aluminum and steel panels corresponded well to the manufacturer's data and other preliminary experimental results found in the existing literature. More extensive experimental panel capacity enhancement research in a mechanically ventilated space is sorely needed.

The simplified model also clearly showed that the panel cooling capacity is enhanced by the mechanical ventilation system installed in a space within the range of 10–39% in both steel and aluminum panels when the diffuser discharge air velocity and the inlet fluid temperature vary within their typical ranges.

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