Parametric analysis of the free-suspended ceiling radiant cooling panel using CFD simulation

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Abstract. In this study, the parametric analysis was conducted on different designs of the free-suspended ceiling radiant cooling panel (CRCP) and panel area to improve its cooling performance. A three-dimensional CFD simulation model was developed to investigate the cooling capacity and heat transfer coefficient of the CRCPs. Four parameters were used to be the design parameters: the panel length \(L\), the panel curvature radius \(r\), the void distance between each panel or panel segment \(d\), and the panel coverage area \(A_c\). The \(L/r\) ratio was also considered, representing the curvature of panel. Twenty-three designs were compared under seven different cooling load conditions. The cooling capacity curve of each design was given as the regression curve of the cooling capacity and temperature difference between air and panel surface. The results show that the CRCPs with curved shape and void have better cooling performance compared to the reference results. The cooling capacity and heat transfer coefficient increase with increasing curvature radius and coverage area and decreasing the panel length. Additionally, the nominal cooling capacity is the largest when \(L/r\) is 0.5. The distance between adjacent panels or panel segments and the panel to the wall is most significant in enhancing cooling capacity.

1 Introduction

The radiant heating and cooling (RHC) systems mostly depend on the radiation heat transfer for more than half of the total heat exchange. Basically, it is classified into three types, which are embedded surface system, thermally activated building system, and radiant panel system [1].

The radiant panel system commonly circulates the fluid thermal medium (such as water) in the pipe loop, directly touching the panel. Then, the panel heats or cools the space according to the temperature difference between the panel surface and surrounding areas. Unlike other two systems, the radiant panel system is more flexible for installation and easier to maintain as it is separated from the envelope. It is also defined as low-temperature heating or high-temperature cooling system, as the supplied water temperature in the radiant panel system is closer to the desired air temperature. Additionally, this system allows to adopt renewable energy as the heat source. Therefore, the system has recently become an energy-efficient alternative to customary air-conditioning systems and has been widely applied to achieve low-energy or zero-energy buildings.

However, the radiant cooling panel system still has a risk of dew condensation on the cooling surface. High manufacturing and installation costs also limit its extensive applications. Thus, it is highly required to address these issues by improving system performance through advanced strategies[2], such as integrating with air circulating devices or improving the panel structure.

In this study, a free-suspended CRCP with segmented and curved structure was proposed. The objective of this study is to evaluate the effects of the panel design parameters of the novel panel structure and explore the ideal design for maximizing the cooling capacity of CRCP. A parametric analysis was conducted on the free-suspended ceiling radiant cooling panel (CRCP) with curved shapes and voids between the adjacent panels. A three-dimensional CFD simulation model was developed using ANSYS Fluent to study the cooling capacity and heat transfer coefficient of the panel. Twenty-three panel designs were proposed, and the cooling capacity curves were given and compared with the reference panels. The influence of the panel structure on the cooling performance was investigated according to the comparison. The optimal panel structure was discussed based on the cooling performance.

2 Description of CRCP

The ceiling radiant cooling panel (CRCP) system is one of the popular radiant cooling systems installed beneath the ceiling. Some CRCPs are well insulated on the upper surface between the panel and ceiling to maximize the heat absorption at the bottom surface.

On the other hand, some researchers investigated the open-type CRCP, freely suspended under the ceiling without insulation, as illustrated in Table 1. The void between each panel allows the warmed air to rise and be
cooled by the panel’s upper surface and promotes the processed air moving down to the occupied conditioned space. The cooling capacity increased significantly by enhancing the air movement around the panel and increasing the convection heat transfer. Therefore, it performs better than the closed type [4]. In the previous study [5], an open-type free-suspended panel was investigated experimentally and numerically. The results showed that only changing the panel shape to the curved and adding the void can increase the radiation heat transfer coefficient by 31% and convection heat transfer by 174% and decrease the indoor air temperature by 1 K. The heat transfer of the panel is possibly being further improved. The relationship between the panel structures and cooling performance is still unclear. Therefore, the design methodology was proposed in this study for the consideration of system optimization and efficiency enhancement.

Table 1. Open-type CRCPs in reference

<table>
<thead>
<tr>
<th>Reference</th>
<th>structure</th>
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<tr>
<td>Zhang et al. [3]</td>
<td>Concrete ceiling</td>
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<td>Shin et al. [4]</td>
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<td></td>
<td></td>
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<td>Curved type in this</td>
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<td>study</td>
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In this study, four parameters are used to be the design parameters of the panel shape, as shown in Fig. 1, which are the panel length \( L \), the curvature radius \( r \), the void distance between each panel or panel segment \( d \), and the panel cover area \( A_c \). The \( L/r \) ratio is also discussed, which is parameter depending on the length and curvature radius. Especially the \( L/r \) ratio is a parameter representing the curvature of the panel, and the curvature increases with a larger value.

![Fig. 1. Panel design parameter](image)

Overall, twenty-three different designs were generated and compared to show the effect of each parameter in this study. The values of each parameter in each case are summarized in Table 2. The panel length \( L \) is changed from 0.03 to 0.12 m, the panel curvature radius \( r \) is changed from 0.03 m to 0.3 m, the \( L/r \) ratio is changed from 0.3 to 2, the void distance \( d \) is changed from 0.01 m to 0.14 m, the panel cover area \( A_c \) is changed from 7.58 m\(^2\) to 12.96 m\(^2\). Accordingly, the panel surface area \( A_s \) varies from 9.02 m\(^2\) to 33.82 m\(^2\).

Table 2. Panel parameter design.

<table>
<thead>
<tr>
<th>No.</th>
<th>( L ) [m]</th>
<th>( r ) [m]</th>
<th>( d ) [m]</th>
<th>( A_c ) [m(^2)]</th>
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3 Model development
3.1 Geometry
The room model is 4 m (L) × 4 m (W) × 2.9 m (H), which has the same dimension and arrangement with a room model for a suspended flat panel validated by Shin et al. [4]. Twelve cylindrical occupant dummies with the dimension of 0.3 m (D) × 1.1 m (H) are deployed symmetric in the room to mimic human bodies, which generate energy dissipated to the space as cooling load. The panels are suspended beneath the ceiling at 0.3 m and arranged along the central line.

Moreover, the symmetry boundary condition is applied to the middle plane to simplify the modelling and accelerate the simulation speed due to the completely symmetrical characteristic of the room. In this study, all the simulations were conducted in half of the space, as shown in Fig. 2.

3.2 Mesh
The mesh was generated by ANSYS Meshing tool using tetrahedron grid and hexahedral grid near the cylinder surface (Fig. 3). Mesh independent analysis was carried out for one curved panel design to minimize the impact of element size on the simulation accuracy. Finally, the preferred number of elements is 1,432,275 in the present study, as the results changed only 0.17% of heat flux and 0.81% of average air temperature compared with the finer mesh (Fig. 4).

3.3 Model description
3.3.1 Assumptions
The assumptions applied in this study are listed as follows:
1-The heat transfer is calculated under steady state condition
2-The air density difference is ignored, and only the gravitational force effect is considered
3-The heat transfer between the water pipe and the panel surface is ignored, and the panel surface temperature is uniform.
4-The emissivity is constant and a property of the surface, which is independent of wavelength.
5-The surface is opaque, and only the transferred radiation between two surfaces is considered

3.3.2 Calculation models
In this study, the standard \( k-\varepsilon \) model proposed by Launder and Spalding (1972) was used to describe the effect of turbulence.

\[
\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_i} (\rho k u_i) = \frac{\partial}{\partial x_j} \left[ \mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho \varepsilon - Y_M + S_k
\]

\[
\frac{\partial}{\partial t} (\rho \varepsilon) + \frac{\partial}{\partial x_i} (\rho \varepsilon u_i) = \frac{\partial}{\partial x_j} \left[ \mu + \frac{\mu_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1\varepsilon} \frac{k}{\varepsilon} (G_k + C_{3\varepsilon} G_b) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} + S_\varepsilon
\]  

Here \( \mu_t \) is the turbulent viscosity, \( u_i \) is the velocity, \( G_k \) is the turbulence kinetic energy generated by the mean velocity gradients, \( G_b \) is the turbulence kinetic energy generated by buoyancy. \( Y_M \) is the dilatation.
constant. 

The radiative heat transfer occupies a significant part of total heat transfer in CRCP systems. Therefore, the Surface to Surface (52S) radiation model was used in the simulations. In this model, the radiation heat flux into a surface $k$ was represented as a summation of the heat flux out of the surrounding surface $j$.

$$A_k q_{k,\text{in}} = \sum_{j=1}^{N} A_j F_{jk} q_{j,\text{out}}$$ (3)

Here $A_k$, $A_j$ is the area of surface $k$ and surface $j$, $F_{jk}$ is the view factor between surface $j$ and surface $k$, $q_{k,\text{in}}$ is the radiative heat flux incident on the surface $k$ from the surroundings, $q_{j,\text{out}}$ is the radiative heat flux leaving the surface $j$.

Then, the radiation heat flux of a given surface $k$, which is composed of both directly emitted and reflected energy, can be written as:

$$q_{k,\text{out}} = \varepsilon_k \sigma T_k^4 + \rho_k \sum_{j=1}^{N} F_{jk} q_{j,\text{out}}$$ (4)

Here $\varepsilon_k$ is the emissivity and $\sigma$ is the Boltzmann’s constant.

Furthermore, the Boussinesq model was used to model the natural convection in the closed space driven by buoyancy force.

$$(\rho - \rho_0)g \approx -\rho_0 \beta (T - T_0)g$$ (5)

Here $\rho_0$ is the specified constant density of the flow, $T_0$ is the operating temperature, and $\beta$ is the thermal expansion coefficient.

### 3.3.3 Numerical schemes

The SIMPLE algorithm was chosen for coupling pressure and velocity, the first-order upwind discretization scheme was applied for turbulent kinetic energy and turbulent dissipation rate, and the second-order upwind discretization scheme was used for the rest variables.

### 3.4 Boundary conditions

Table 3 lists the boundary conditions and emissivity. The room is assumed to be well insulated without heat transfer so that the envelope is assigned to the adiabatic condition, the panel surface temperature is set constant at 15.83 °C, which is the experimentally measured value given by Shin et al. Each panel design is investigated under seven different cooling load conditions, which are 621.69 W, 746.03 W, 870.37 W, 994.71 W, 1119.04 W, 1243.38 W, 1405.02 W owing to different heat flux emitted from cylindrical dummies. Finally, 189 cases were carried out for the analysis.

### 3.5 Heat transfer analysis

The cooling capacity $q$ (W/m²) is defined as the total heat transfer rate through all the panel surfaces divided by the panel surface area $A_s$.

$$q_{\text{tot}} = \frac{Q_{\text{tot}}}{A_s}$$ (6)

The radiation and convection heat transfer coefficient are then calculated based on Eqs. (7)-(8).

$$h_r = \frac{q_r}{A_s(T_{\text{AUST}} - T_p)}$$ (7)

$$h_c = \frac{q_c}{A_s(T_{\text{AUST}} - T_p)}$$ (8)

Where $A_{\text{AUST}}$ is the area-weighted uncooled temperature of the surfaces excluding the panel surface, $T_a$ is the air temperature, and $T_p$ is the panel surface temperature. Then, the operative temperature can be then roughly determined as follows.

$$T_{\text{ap}} = \frac{h_c T_a + h_r A_{\text{AUST}}}{h_c + h_r}$$ (9)

### Table 3. Boundary conditions

<table>
<thead>
<tr>
<th>Name</th>
<th>$T$ [°C]</th>
<th>$q$ [W/m²]</th>
<th>Emissivity</th>
</tr>
</thead>
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<tr>
<td>Floor</td>
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### 4 Results

#### 4.1 CFD validation

The $CFD$ model was validated with the experimental results for flat panel firstly presented by Shin et al. In the validation case, the panel is one flat solid panel with a dimension of 3.6 m (L) × 3.6 m (W) × 0.03 m (H) with the cooling load of 1409.76 W. The simulated temperature shows marginal differences, with average error of 1.01% in Case 1, 0.89% in Case 2, and 0.86% in Case 3. It indicates that the $CFD$ model agrees well with the experimental temperature measurements conducted for the free-suspended panel under different cooling load conditions.
### 4.2 Panel surface temperature

Different ceiling radiant cooling panels can be compared and evaluated using the cooling capacity curve present in the standard [6], which is given by the cooling capacity and the difference between the operative and panel surface temperature. The previous studies gave the curve under different cooling load conditions while adjusting the panel surface temperature to ensure the indoor air temperature is within a comfort range [4]. Nevertheless, in this study, the panel surface temperature is maintained at 15.83 °C in each case, with the cooling load increasing from 621.69 W to 1405.02 W.

Fig. 6 compares the results of heat flux, heat transfer coefficient, average indoor air temperature, and the difference between operative and panel surface temperature, which are obtained under the condition of different panel surface temperatures. The panel surface temperature is set from 14.83 °C to 19.83 °C with an interval of 1 °C. Except for indoor air temperature increasing with the panel surface temperature increase, the heat flux and temperature difference are almost the same, keeping the difference within 5%. It can be concluded that different panel surface temperature settings only affect indoor thermal condition instead of panel thermal performance. Therefore, the setting of the panel surface temperature can be simplified in this study.

### 4.3 Parametric study

As stated above, the cooling capacity curve is used to show the characteristics of CRCPs. The curve is approximated by a regression relationship between the cooling capacity and the difference between operative and panel surface temperature.

\[ q = k(T_{op} - T_s)^n \]  

"Standard" curve (Fig.7-Fig.11) is for a closed-type CRCP proposed in the European standard [7]. 'A-d' curve is calculated for open-type panel with distributed layout, which has the largest cooling capacity in previous study [4]. The nominal cooling capacity is given under the condition of \( \Delta T = 8 \) K.

In addition, the radiation heat flux, convection heat flux as well as radiation and convection heat transfer coefficient are compared under the condition of the largest cooling load (\( q_{cylinder} = 113 \) W/m²)
4.3.1 Panel length

The effect of panel length is shown by comparing the results of Design No.1-No.4. The panel length of Design 1, Design 2, Design 3, and Design 4 is 0.03 m, 0.06 m, 0.09 m, and 0.12 m, respectively. While the panel curvature radius is kept at 0.06 m, and the void distance is maintained at 0.03 m. As shown in Fig. 7, the cooling capacity and both radiation and convection heat transfer coefficient decrease with increasing the panel length.

![Fig. 7. Comparison of different panel length](image)

4.3.2 Panel curvature radius

Fig. 8. illustrates the comparison with the different curvature radii using the results from Design 5, Design 2 and Design No.6-No.9. The curvature radius is 0.03 m, 0.06 m, 0.09 m, 0.15 m, 0.2 m and 0.3 m in each Design. The results show that the cooling capacity and heat transfer coefficient increase with increasing the panel curvature radius. Moreover, the nominal cooling capacity increases by 9.1% when $r$ increases from 0.03 to 0.06, while it only rises by 4.2% when $r$ rises from 0.06 to 0.3.

![Fig. 8. Comparison of different curvature radius](image)

4.3.3 Void distance

Five different void distances are compared, as shown in Fig. 9. The void distance is 0.01 m, 0.03 m, 0.06 m, 0.1 m, and 0.14 m, respectively, in Design No.10-No.14. The nominal cooling capacity increases significantly by 21.9 % from 120.35 W/m² to 146.67 W/m² when increasing the distance from 0.01 m to 0.06 m. While the nominal cooling capacity and heat transfer coefficient change moderately when the distance is larger than 0.06 m, which increase only 3.2 % and 7.4 % with increasing the distance from 0.06 m to 0.14 m. In addition, the nominal cooling capacity become the largest about 150 W/m² and remained essentially unchanged when the void distance is larger than 0.1 m.

![Fig. 9. Comparison of different void distance](image)

4.3.4 Panel coverage area

Fig. 10. shows the comparison between different panel coverage area varying from 7.58 m² (Design 19) to 12.96 m² (Design 2). When the coverage area is 11.43 m² (Design 15), the nominal cooling capacity is 12.8 % higher than 7.58 m², and 5.8 % higher than 12.96 m².
The total heat transfer coefficient increases 4.3% from 8.4 to 8.76 with reducing the coverage area by 5.38 m². In other word, expanding the distance between the side of the panel and the wall within a proper range can also enhance the cooling performance as same as increasing the void distance between adjacent panels.

4.3.5 Panel curvature and L/r ratio

The L/r ratio is discussed by comparing Design No.20-No.23, which have the same void of 0.03 m and surface area of 11 m² ± 0.4 m². It should be aware that L and r is different in each design as L/r ratio is a dependent parameter. As a result, when the L/r ratio is 0.5, the nominal cooling capacity is the largest, which is 129.65 W/m², and is about 1.5% larger than the minimum design. On the other hand, the convection heat transfer coefficient decrease by 5% and the radiation heat transfer coefficient increase by 4.3% when decreasing the L/r from 1.1 (Design 20) to 0.3 (Design 23). It illustrates that the curved shape can effectively promote air movement over the top surface and enhance convective heat transfer because the curved structure has a streamlined shape.

5 Conclusion

1- The free-suspended CRCP with curved shape and void in this study has better cooling performance compared to the reference results.
2- The nominal cooling capacity and heat transfer coefficient increase with increasing curvature radius and decreasing the panel length. The nominal cooling capacity is largest when L/r ratio is 0.5.
3- The distances between adjacent panels and the panel to the wall plays a significant role in improving the cooling performance of the panel.
4- It is possible to achieve the same or even better indoor thermal conditions by applying fewer panels.

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Reference