Simplified model and performance analysis for radiant cooling panel with serpentine tube arrangement and thin insulation layer for moisture control in tropical climate

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Abstract. Radiant cooling panel (RCP) is widely known to improve thermal comfort and has the potential for energy saving. However, when applied the system in countries with a tropical climate, a major issue arises which is condensation on the surface panel. In this paper, a simplified model of top insulated RCP panel with serpentine tube arrangement integrated with thin insulation layer is developed to predict its performance. The numerical model is based on the principle of superposition of heat resistance and validated by using simulation software. The result produced by both models shows a good match in which the relative error calculated were 1.3% and 2.7% for cooling capacity and surface mean temperature respectively. Thus, the correctness of the numerical model to determine the cooling capacity and surface mean temperature of radiant cooling panel is validated. The thickness of the thin insulation layer plays a significant role as moisture control while maintaining the cooling capacity at 767W for surface mean temperature of 18.2°C. When comparing various materials used as the thin insulation layer in RCP, it is also found that RCP with gypsum used as the thin insulation layer demonstrates better performance in maintaining the cooling capacity of RCP while controlling moisture condensation.

1. Introduction
The growing population has caused high demand in the building sector which mainly divided between the residential and commercial end-users. This causes higher energy demand for housing globally [1]. Although there is no immediate effect of the energy waste on the environment, higher demand in energy consumption will lead to a major issue concerning carbon dioxide and greenhouse gas emission in future [2]. Therefore, a high energy efficiency system is required to counter the high building energy consumption while creating a comfortable indoor environment.

In recent years, a radiant cooling system has been broadly used due to its potential in energy-efficient and improvement of thermal comfort [3,4]. In the system, more than 50% of the heat exchange occurs within the surrounding environment is mainly through thermal radiation [5]. By taking advantage of radiation heat transfer between radiant surfaces and the human body, a radiant cooling system can reach an equal level of thermal comfort at higher air temperature [6], thus, proven able to be energy efficient. The utilization of the system when used together with the ventilation system has been extended from mild climate to tropical climate for examples are Malaysia, Thailand and Singapore [3,7–9]. However,
the major limitation when using such a system in the tropical climate is that the condensation might occur on the radiant surface if the surface mean temperature surpasses the dew-point temperature of the entire room. The condensation issue occurred mainly at the area nearer to the cooling pipes due to uneven temperature distribution which in severe cases, can cause dripping [10].

Currently, there are a lot of studies focusing on the condensation problem for the radiant cooling panel. Standard design and guidebooks suggested using higher chilled water temperature to prevent condensation. However, the cooling capacity is sacrificed during the operation [11]. S. Wongkee et al. [8] reported the performance of RCP in tropical climate compared to the conventional air conditioning for energy consumption. It demonstrated 70% energy saving relative to the conventional air conditioning. However, higher chilled water temperature is recommended to avoid condensation and the cooling capacity of the system is limited to 30W/m² during hot weather in April. Y. L. Yin et al. [12] used a high-speed camera to study the condensation process on RCP and analysed the heat transfer performance on three-panel types; pure tube, gypsum panel and metal panel. The gypsum panel shows the best performance of heat transfer and capable of controlling water condensation under identical conditions. M. S. Shin et al [13] suggested an open-type RCP to enhance cooling capacity by utilizing the cooled plenum air moving through the opening. The cooling capacity can achieve 54-80% higher than conventional closed-type RCP. B. Ning et al. [10] used a thin air layer to achieve uniform surface temperature distribution. The design and integration of the air layer were modified to improve the cooling capacity to 43-46% compared to the original cooling RCP. X. Su et al. [14] also suggested an inside air layer to the improved RCP. However, it requires a lower temperature of chilled water which reduced the COP of chillers. G. Lv et al. [15] proposed a grooved RCP filled with heat transfer liquid surrounding the cooling pipes for application in the tropical climate. The finding showed the cooling capacity of RCP is 2.9°C smaller than conventional RCP while preventing condensation on the surface panel as uniform temperature distribution is achieved.

L. Zhang et al. [16] developed a simplified numerical model to evaluate cooling capacity, uniformity of temperature distribution and the radiant surface temperature. The heat transfer process from chilled water to the indoor environment is treated as heat resistance in series based on the principle of superposition of heat resistance. The error between the proposed model and experimental results is within 8%. X. Wu et al. [17] established a model to calculate radiant surface temperature and heat transfer of radiant floor cooling using conduction shape factor. For the average water temperature was between 10-20°C the maximum difference between the calculated and measured data were 0.3°C and 2.0W/m². H. Tang et al. [18] applied simplified Navier-Stokes equations with Boussinesq approximation to analyse the condensation rates. The condensation rate on the radiant ceiling is found to be 3.5 times greater than the radiant floor and 25% greater than the radiant wall with similar air state and surface temperature. Q. Kong et al. [19] used a coupling strategy between hourly computational fluid dynamics (CFD) and building energy simulation (BES) based on real-time

| Nomenclature | Greeks |
|--------------|--------|
| b_w          | λ      |
| D            | thickness (m) |
| h            | convection heat transfer coefficient (W/m²K) |
| k            | thermal conductivity (W/mK) |
| L            | length (m) |
| N            | number of section strip |
| q_c          | cooling capacity (W) |
| R            | thermal resistance (K/W) |
| T            | temperature (°C) |
| W            | pipe spacing (m) |

Subscripts:
- h indoor air
- k aluminium
- L silicon bond
- N copper pipe
- q_i inner
- R o outer
- T p panel
- W s solid layer
- w chilled water

| Nomenclature | Subscripts |
|--------------|------------|
| λ            | thickness (m) |
| b_w          | bond width (m) |
| D            | pipe diameter (m) |
| h            | convection heat transfer coefficient (W/m²K) |
| k            | thermal conductivity (W/mK) |
| L            | length (m) |
| N            | number of section strip |
| q_c          | cooling capacity (W) |
| R            | thermal resistance (K/W) |
| T            | temperature (°C) |
| W            | pipe spacing (m) |
information which were then solved using finite element method (FEM). The result shows dynamic heat transfer characteristic where the external wall and indoor air in the conditioned room change with the outside weather condition. M. G. Subawomo [20] studied the transient thermal behaviour of radiant cooling fin subjected to a magnetic field using the Galerkin finite element method. The model then verified using the exact solution developed by Laplace transform. The result revealed the rate of heat transfer and efficiency of the fin increased in increment of Biot number, convective, radioactive and magnetic parameters. G. Yu et al. [21] developed a simplified model for ceiling RCP with serpentine tube arrangement by considering the radiant panel as a straight strip fin. The tube spacing is proven to have a significant effect on the cooling capacity. B. Ning et al. [22] proposed a simplified numerical model using Matlab software to explore the cooling load dynamics for integrated operation of RCP and a dedicated outdoor air system (DOAS). The results show the peak cooling load is 16% larger than the all-air system while the operative temperature is 1.1°C lower.

This paper aims to study the performance of heat transfer and moisture condensation phenomenon on RCP with serpentine tube arrangement. In addition, a thin insulation layer is introduced between the cooling pipes and the radiant cooling surface to enable moisture control. A simplified model for the RC panel is established to calculate the cooling capacity and radiant surface mean temperature by using the principle of superposition of heat resistance. The mathematical model is validated by a simulation model using Fluent 18.1 software [23]. The effects of thin insulation layer on the RCP performance based on its thickness and types of material used are investigated by the simplified mathematical model.

2. Simplification of the improved RCP

2.1. System structure and working mechanism

In standard RCP structure as described in ASHRAE handbook [5], the cooling pipes are usually in direct contact to the metal panel which caused relatively large temperature difference on the radiant surface. This leads moisture condensation to be centralized on the radiant surface area which can easily be accumulated into droplets. In this paper, an improved RCP embedded with thin solid insulation layer between the cooling pipe and radiant cooling surface as shown in figure 1 was studied. The cooling pipes are in serpentine arrangement and thermal insulation layer covered the top part of the panel. Compared to the standard RCP structure, there is no direct contact of cooling pipes to the radiant surface panel, hence, encouraging uniform temperature distribution on the radiant surface area. The heat transfer between the cooling pipes to radiant surface panel is mainly through conduction.

2.2. Simplification and assumption of the model

M. Tye-Gingras et al. [24] through theoretical analysis and numerical simulation on the serpentine tube arrangement have concluded that the heat transfer efficiency of RCP is not significantly affected compared to results obtained by assuming thermal symmetry between tubes. Additionally, in the heat transfer analysis of the RC panel, the semi-circular tube extremities can be ignored, and straight tube modelling is appropriate to measure the heat transfer. Thus, following simplification and assumption can be made to the model:

1. The RCP can be divided into N-sections (figure 1(a)) and be considered as a N x L strip (figure 1(b)). The heat transfer between cooling pipe can be neglected and the semi-circular tube extremities are considered straight. The temperature at each tube inlet is similar as to the tube outlet of previous strip;
2. Heat transfer is calculated under steady-state condition;
3. Backward heat transfer is neglected as the panel has good thermal insulation;
4. The temperature gradient in the direction of flow can be neglected and heat transfer between the pipes is regarded as symmetrical.
3. Heat transfer process in RC panel
Heat transfer in a radiant cooling panel system can be divided as the following sections:

1) Inside section: heat transfer between chilled water and pipe outside surface;
2) Core section: heat transfer between pipe outside surface and panel surface;
3) Outside section: heat transfer between panel surface and indoor environment.

3.1. Heat transfer of RC panel inside section
The local Nusselt number of turbulent flows in circular pipes can be obtained from equation (1).

\[ N_u = 0.0023 R_e^{4/5} P_r^{0.4} \]  \hspace{1cm} (1)

where \( N_u \) is the local Nusselt number of the internal flow; \( R_e \) and \( P_r \) are the Reynolds number and Prandtl number at the mean temperature of inlet and outlet chilled water.
The heat transfer resistance from water to the external surface of the pipe can be calculated using equation (2) until equation (4).

\[
h_w = \frac{k_w Nu_f}{D_i} \tag{2}
\]

\[
R_w = \frac{1}{\pi D_i h_w L} \tag{3}
\]

\[
R_{cu} = \frac{\ln(D_o/D_i)}{2\pi k_{cu} L} \tag{4}
\]

where \( R_w \) and \( R_{cu} \) are the thermal resistance in chilled water and pipes, \( K/W \); \( h_w \) is the convection heat transfer coefficient of the chilled water, \( W/m^2K \); \( k_w \) and \( k_{cu} \) are the thermal conductivity of water and pipes, \( W/mK \); \( D_i \) and \( D_o \) are the internal and external diameter of copper pipes, \( m \); \( L \) is the strip length, \( m \).

3.2. Heat transfer in the core section

The dimension of silicon bond contacted along with the pipe can be considered as the same width as pipe outer diameter. The thermal resistance of the silicon bond can be calculated using equation (5). The thermal resistance through the core structure of radiant cooling panel is similar as the flat-plate type. In the proposed radiant cooling panel, there are a few surface layers that can be considered as slab with uniform thermal conductivity. The heat resistance of uniform slab and can be obtained by equation (6) to equation (8).

\[
R_b = \frac{\lambda_b}{k_b b_w L} \tag{5}
\]

\[
R_{al} = \frac{\lambda_{al}}{k_{al} L W} \tag{6}
\]

\[
R_s = \frac{\lambda_s}{k_s L W} \tag{7}
\]

\[
R_p = \frac{\lambda_p}{k_p L W} \tag{8}
\]

where \( R_b \), \( R_{al} \), \( R_s \) and \( R_p \) are the thermal resistance in silicon bond, aluminium layer, solid insulation layer and metal panel layer, \( K/W \); \( k_b \), \( k_{al} \), \( k_s \) and \( k_p \) are the thermal conductivity of silicon bond, aluminium layer, solid insulation layer and metal panel layer, \( W/mK \); \( \lambda_b \), \( \lambda_{al} \), \( \lambda_s \) and \( \lambda_p \) are the thickness of silicon bond, aluminium layer, solid insulation layer and metal panel layer, \( m \); \( b_w \) is the width of silicon bond, \( m \); \( W \) is the pipe spacing, \( m \).

According to the principle of superposition of heat resistance, as shown in figure 2 the total heat resistance of the radiant cooling panel from chilled water to surface panel, \( R_1 \) is as shown in equation (9):

\[
R_1 = R_w + R_{cu} + R_b + R_{al} + R_s + R_p \tag{9}
\]

![Figure 2. Heat resistance from chilled water to indoor environment.](image-url)
3.3. Heat transfer of RC panel outside section

The heat exchange between panel surfaces with the room thermal environment includes convective heat transfer between panel surface and air, and radiation heat transfer between panel surface and room surfaces including building envelope, furniture, etc. To simplify the calculation, a combined heat transfer coefficient is adopted. From Table 10 in Chapter 26 of the 2017 ASHRAE Handbook – Fundamental [25], the total heat transfer coefficient for vertical surface (wall) is 8.29W/m²K. Therefore, the total resistance heat transfer between chilled water and indoor air environment, $R_2$ can be obtained from equation (10) and equation (11).

$$R_a = \frac{1}{h_a W_p} \quad \text{(10)}$$

$$R_2 = R_w + R_{cu} + R_p + R_g + R_p + R_a \quad \text{(11)}$$

where $R_a$ is the thermal resistance in indoor air, K/W; $h_a$ total heat transfer coefficient of indoor air, W/m²K; $W_p$ is the width of radiant panel, m.

3.4. Cooling capacity and surface panel mean temperature

From the model proposed, the most important limitation of radiant cooling panel is the heat transfer between radiant panel and indoor air environment. In cooling condition, the influence of $h_a$ increases due to large convective heat transfer coefficient between radiant panel and indoor environment. The thickness and heat conductivity of each layer also plays important roles on performance of radiant floor. Copper pipe for example, although it is very thin, still has influence on total heat resistance which should not be ignored.

Based on the results of heat resistance, the panel surface mean temperature and can be expressed as equation (12).

$$T_p = \frac{T_w + R_a T_a}{R_a + 1} = \frac{T_w + (R_2 R_a - 1) T_a}{R_a} \quad \text{(12)}$$

where $T_p$, $T_w$ and $T_a$ are the temperature of surface panel, chilled water and indoor environment respectively, °C;

It should be noted that the cooling capacity calculated above is for single straight strip (N=1) thus, the total cooling panel for the radiant cooling panel can be calculated as shown in equation (13) until equation (15).

$$q_c = \Delta T_{lm} N (R_a)^{-1} \quad \text{(13)}$$

$$\Delta T_{lm} = \frac{(T_a - T_{w,o}) - (T_a - T_w)}{\ln(T_a - T_{w,o} / T_a - T_w)} \quad \text{(14)}$$

$$N = \frac{W_p}{d_a + W} \quad \text{(15)}$$

Where $q_c$ is the cooling capacity, W; $N$ is number of sections; $T_{w,o}$ is the temperature of chilled water outlet, °C.

4. Model validation

A simplified numerical model of RC panel embedded with thin insulation layer was established and validated with the simulation model. The heat transfer process is simplified as a steady-state two-dimensional problem as shown in figure 1(c). A CFD software Fluent 18.0 [23] is used as it provides fundamental heat transfer module and capable to solve the problem. The simulations are conducted with various type of weak insulation materials used as a thin insulation layer embedded in the RC panel. The materials tested are gypsum, air and water. For brevity, only the result of RC panel with thin gypsum layer is shown in this paper. The geometry parameters and thermal properties of the RC
panel are listed in table 1. Assumed inputs of the proposed RC model are shown in table 2 were used in the simulation and the result of validation is shown in table 3.

Figure 3 shows the temperature gradient of the cooling panel integrated with a thin insulation layer using gypsum material at 4mm thickness. From the figure, the distribution of temperature is very uniform which can be achieved by applying the thin insulation area to the cooling panel. From table 3, the numerical model predicted the cooling capacity of the panel is at 765W while the simulation model shows the reading of 775W. The surface temperature of the panel predicted from both models shows small temperature difference which 18.7°C from numerical model and 18.2°C from the simulation model. The error of the mean surface temperature and cooling capacity of the numerical model relatively to the simulation model is 2.7% and 1.3% respectively. This indicates the numerical and simulation models are in good agreement with each other. Thus, the correctness of the numerical model of the proposed RC panel is validated.

![Figure 3. Temperature gradient of cross-section RCP.](image)

| Table 1. Geometry parameters and thermal properties of RCP. |
|------------------------------------------------------------|
| Components | Thickness and geometry parameters (mm) | Thermal conductivity, k (W/m²K) |
|-------------|-----------------------------------------|-------------------------------|
| Aluminium plate | 2.0 | 237 |
| Aluminium sheet | 0.5 | 237 |
| Silicon bond | 1.0 | 191 |
| Copper pipes | D₀= 9.0, Dᵢ=8.0, W=10.1 | 390 |

| Table 2. Assumed inputs for simulation. |
|-----------------------------------------|
| Inlet chilled water temperature, Tᵢ (°C) | Indoor dry-bulb temperature, Tᵢₑ (°C) | Indoor relative humidity, φ (%) | Average uncooled temperature, AUST, (°C) | Water speed, v (m/s) |
|-----------------|-----------------|-----------------|-----------------|-----------------|
| 12              | 24              | 65              | 24              | 0.78            |

| Table 3. Cooling capacity and radiant surface meant temperature for numerical and simulation models. |
|------------------------------------------------------------------------------------------------------|
| Methods comparison | Numerical model | Simulation model | Relative error |
|---------------------|-----------------|-----------------|----------------|
| Cooling capacity (W) | 765             | 775             | 1.3%           |
| Average surface temperature, Ts (°C)              | 18.7            | 18.2            | 2.7%           |
5. Result and analysis
In this section, the effects of the material properties, geometry parameters and operating conditions on the surface mean temperature and cooling capacity of RCP are studied. All the geometry parameters and initial inputs used are listed in table 1 and table 2 except for the specific parameter in the following subsection. All results are calculated using the numerical model presented in this paper.

5.1. Effect of thin insulation thickness on cooling capacity
The significant feature integrated into the cooling panel of this study is the development of a thin insulation layer to avoid direct contact of metal panel and the cooling pipe. The thin insulation layer is proven to act as moisture control to avoid condensation from occurring on the panel while simultaneously preventing the cooling capacity to further dropped from its original design. In this study, the temperature of the panel surface must exceed the dew-point temperature of the indoor environment which tested at 19°C. As shown in figure 4, the thickness of the thin insulation layer has a significant effect on cooling capacity is significantly affected by the thickness of thin insulation layer within the range from 1mm to 10mm. Although the cooling capacity dropped abruptly at the thickness of 1mm to 3mm, the surface temperature started approaching the dew-point temperature of the indoor environment. The surface temperature finally exceeding the dew-point temperature at 18.7°C for 4mm thickness of thin insulation layer while maintaining the cooling capacity at an acceptable value of 765W.

![Figure 4](image_url)

**Figure 4.** Effect of thin insulation thickness on cooling capacity: (a) Radiant surface mean temperature, $T_p$; (b) cooling capacity, $q_c$. 
5.2. Effect of integration of thin insulation layer on the cooling capacity

The effect of the thin insulation layer is further studied by comparing the cooling capacity with the conventional radiant panel. In the conventional radiant panel, the thickness of the metal plate is adjusted so that the thickness is similar to the radiant panel with thin insulation layer to observe its function as the moisture control. Figure 5 shows that the difference of cooling capacity in both panels is extremely large that the cooling capacity of the radiant cooling panel without thin insulation layer can archive more than 14kW compared to when the thin insulation is applied, the cooling capacity can achieve the highest reading at 2.6kW only. However, when considering the condensation issues, the surface temperature of the radiant panel without a thin insulation layer is way below the dew-point temperature of the indoor environment marked about 12.7°C at any thickness applied. Thus, defy the performance of the radiant panel as condensation can occur on the surface which causes discomfort to the users.

![Diagram of surface mean temperature and cooling capacity](image)

**Figure 5.** Effect of integration of thin insulation layer on cooling capacity: (a) Radiant surface mean temperature, $T_p$; (b) cooling capacity, $q_c$. 
5.3. Effect of thermal media used as thin insulation layer

In this study, the thin insulation layer used solid gypsum as the medium to distribute the temperature evenly to the surface panel. It is generally known that solid is the better thermal conductor compared to liquid and gas. Thus, the effect of gypsum material as thin insulation layer is investigated by comparing its cooling capacity to water and air as the thermal medium. The thermal conductivity of air, gypsum and water are 0.024 W/m·K, 0.18 W/m·K and 0.606 W/m·K respectively. Figure 6 shows that the cooling capacity of all medium used dropped as the thickness increased and almost similar towards 10mm thickness. Water noted the highest cooling capacity at all thickness followed by gypsum and air. However, when comparing the surface mean temperature of the panel, air recorded the temperature higher than the dew-point temperature of indoor environment at all thickness while water stays below the dew-point temperature. Thus, it is proven that gypsum is the better material to use as thin insulation layer to maintain higher cooling capacity while avoiding condensation problem on the cooling panel.

Figure 6. Effect of thermal media used as thin insulation layer on cooling capacity: (a) Radiant surface mean temperature, $T_{p}$; (b) cooling capacity, $q_c$. 

![Figure 6](image-url)
6. Conclusions
The improved radiant panel with thin insulation layer is proven to be advantageous for moisture control while maintaining higher cooling capacity. The numerical model established in this paper for radiant cooling with serpentine tube arrangement and thin insulation layer is validated by simulation model to predict the cooling capacity and mean temperature of surface panel. The main conclusions are as follows:

1. The simplified model neglects the semi-circular tube extremities by considering the radiant panel as a straight strip. The model can be calculated using the thermal resistance principle approach which validated by showing a good agreement with simulation model with relative error of 1.3% and 2.7% for cooling capacity and surface mean temperature respectively.

2. A trend is established as the thicker the thin insulation layer, the lower the cooling capacity and surface mean temperature of cooling panel thus prove its significant role in determining the performance of cooling panel. Insulation layer with thickness of 0.4 mm is endorsed to achieve the high cooling capacity at 765 W while maintaining the surface mean temperature at 18.7 °C to avoid condensation on the panel.

3. The thermal resistance in insulation layer helps to distribute the temperature evenly to the surface panel thus serve its purpose as the moisture control. The absence of thin insulation layer in the radiant panel will cause condensation to occur when similar thickness of insulation layer is applied to the metal panel. The surface mean temperature without the thin insulation layer stays below the dew-point temperature although the thickness of metal panel is increased to the maximum of 10 mm.

4. Thin solid insulation layer gives the best performance when works as the thermal medium between the pipes and metal panel. It able to maintain the surface mean temperature at higher than dew-point temperature without abandon the cooling capacity compared to air and water that will sacrifice the performance of cooling panel.

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