Validation of radiant and convective heat transfer models of photonic membrane using non-invasive imaging of condensation pattern

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Abstract. Cooling a sample of a material until condensation is observed is a standard technique for accurately measuring the dewpoint and associated relative humidity in a volume. When conducting an experiment with a membrane-assisted radiant cooling panel, we found that membrane surface temperatures were difficult to measure directly. Instead, the onset of condensation was used to infer the membrane’s surface temperature. However, the radiant cooling panels displayed variations of membrane surface temperature at steady state, and thus a resulting condensation contour was observed, forming a curve on which the membrane surface temperature was accurately known and constant - the dewpoint. The curve was in equilibrium between the internal panel temperature driven by internal free convection in the air gap and the view factor to surrounding surfaces, which can be evaluated at each point along the curve. In this paper, we assess the convective and radiative heat transfer balances using simulations. Our methods expand the “sensing” of condensation to provide information about view factor and thermal stratification, both of which are quantities that are difficult to measure adequately in the field.
1. **Introduction**

Obtaining high accuracy field measurements under uncontrolled conditions is a challenge to many researchers, particularly in the thermal comfort domain where many variables influence thermal comfort and often large assumptions are associated with results. In our outdoor radiant pavilion known as the 'Cold Tube' constructed in Singapore in early 2019 [1], the goal was to assess occupant thermal comfort provided by the radiant system outdoors in the humid Singaporean climate, while also learning how the novel membrane-based radiant cooling system could be controlled to avoid condensation [2] (see Figure 1a). As shown in figure 1c, there is a thermally transparent membrane that allows 83% of infrared radiation to pass through. This membrane acts as a convection barrier, isolating the chilled tubes (blue) from the warm, humid air. Without this membrane, condensation forms quickly and plentifully. The membrane-assisted radiant cooling method was first proposed by Morse in 1963 [3].

The membrane itself is not in thermal isolation from the hot humid environment, and the resulting membrane temperature is therefore at an equilibrium between the ambient air temperature and the chilled water temperature. To develop an empirical relationship between the driving factors, namely the ambient air temperature convectively heating the membrane, the circulated chilled water in the blue tubes radiatively cooling the membrane, and the internal chilled air convectly cooling the membrane, and the thermal conductivity through the membrane, an accurate membrane temperature measurement is critical.

However, due to the thermally transparent nature of the membrane, locally monitoring the temperature proved difficult. Thermocouples placed on the membrane either inside or outside locally blocked the transparency, and even highly reflective sensors locally distorted equilibrium temperatures. Using non-contact surface temperature sensors such as thermal imaging was also not a viable option due to the transparency of the membrane - i.e., thermal images provide a good estimate of the temperature of the water tubes surface behind the membrane rather than of the membrane itself (Fig. 1b). In previous work by Raman et al. [4], in which a thin membrane was also used for its photonic properties, an adhesive resistance temperature detector sensor was placed at the back of the radiative cooler layer, but that layer is not transparent and the polyethylene membrane used to prevent convective losses above the cooler was not measured.

In order to obtain the infrared transparent polyethylene membrane temperature, we used a technique common in measurement science, observing the onset of condensation to obtain an accurate measurement of the membrane temperature using condensation as a temperature sensor. Modern techniques have been proposed to be implemented to predict and monitor condensation events [7]. We extended this concept one step further, understanding that view factor changes from point to point. The view factor changes the heat transfer creating an equilibrium line differentiating the portion of the membrane with and without condensation. Thus we can directly observe heat transfer equilibrium between the cool air stratified within the panel and view factor to the warm radiant external environment. While the demonstrated use-case may be highly specific, this method and corresponding simulation-based validation can be expanded beyond this specific application and demonstrates a useful technique for field-validating complex heat transfer in as-built conditions.

2. **Methods**

2.1. **Physical Experiment**

To generate equilibrium conditions with a significant portion of the membrane in a condensing regime, chilled water was circulated at 17°C. The ambient dewpoint temperature was 23.5°C, and the ambient air temperature was 32°C. We waited for the membrane to begin condensing moisture, and then photographed the observed pattern. Internal air temperature stratification measurements in 50 cm height intervals. This was repeated at different water temperature and ambient temperature conditions to produce a range of data. For the simulation, only one datapoint was used accompanying the photograph in Figure 1.
Assuming the panel was at steady state, we equate the sum of all radiative and convective heat transfer to 0. The radiative heat transfer can be broken into two components, the fraction of radiant heat transfer heating the membrane from sources warmer than the temperature of the membrane, and all sources cooling it. These two radiative heat transfer components have a combined view factor of 1. Additionally, a mean radiant temperature was measured at the location of the condensation interface of 23.8°C. This information allows for a system of equations from convective, radiative and mean radiative temperature relationships to determine the view factor of all portions of the environment heating the panel and the average temperature of everything heating the panel. We are also assuming that conduction through the membrane is negligible since it is thin (50 microns).

![Figure 1](image1.jpg)

**Figure 1.** (a) View of the Cold Tube pavilion’s interior with surrounding membrane-assisted cooling panels; (b) Infrared thermography of an active panel (c) Diagram Of membrane absorption, transmission, and reflection coefficients; (d) Diagram of the overall pavilion with the active radiant panels marked in light blue and the the test panel in purple.

### 2.2. View factor and radiant temperature simulation

We used a custom simulation constructed in Grasshopper algorithmic modeling software for the Rhino3D modeling environment. The 1.2x2.4 m panel was subdivided into a grid of 288 test points. The view factor at each point calculated using ray tracing [5]. The cold panel temperature is simulated.
as 17°C and the outdoor environment is simulated as varying between 30°C and 42°C based on thermography readings. While most of the pavilion’s metal and wood surfaces, painted matt orange and white, were assumed to be highly emissive in the longwave, the floor, made of reflective patterned metal, was assumed to be 50% reflective.

The radiant temperature $T_r$ (°C) at each grid point was calculated using Equation 1:

$T_r = \sqrt{\sum_{i=1}^{n} T_i^4 F_{p\rightarrow i} - 273.15}$ (1)

where $T_r$ (°C) is the radiant temperature at a point $p$, $F_{p\rightarrow i}$ are the view factors between the point $p$ and all the surrounding surfaces used to weigh the surface temperatures $T_i(°K)$. A color mesh is then constructed to the resulting surface radiant temperature gradient.

2.3. CFD simulation

In order to examine air temperature stratification within the air gap inside of the panel, between the cold surface of the capillary mat and the PE membrane, we conducted a CFD simulation in RhinoCFD plugin (version 2.1, powered by PHOENICS, CHAM, UK) into Rhino3D modeling software [6].

The dimension of the model is 1.2 m (length) × 0.15 m (width) × 2.4 m (height). The boundary conditions for the panel were defined as follows: the cold surface was specified as a smooth wall with the temperature of 17 °C. The membrane was set as a smooth wall and its temperature was simulated as 32°C, at steady-state with the ambient air temperature. The panel was considered a closed system with regular atmospheric pressure in the air gap between the membrane and the cold surface. The insulation layer is on the other side of the cold surface and was specified as an adiabatic non-slip wall. The top, side and bottom surfaces of the panel frame were simulated as non-slip walls with surface temperature of 30 °C, 27 °C and 25 °C separately.

The computational grids consist of cells in rectilinear shapes. The maximum size change to adjacent cells was no more than 20% for smoothness. For the grid sensitivity analysis, two grids containing different amounts of cells were tested, i.e., coarse grid with 144,000 cells and refined grid with 486,864 cells. The simulation results on a vertical line of the cross-section (see Figure 5) at a distance of 0.06 m from the membrane were used for comparison. As is shown in Figure 2, the temperature plots for the two grids are very close to each other, but the refined grid shows clearer temperature variations near the bottom surface. Therefore, the refined grid with ($27 \times 98 \times 184$) cells was selected for this study, with a grid size was 0.01 m.

![Figure 2](image_url)  
**Figure 2.** A comparison of temperature simulated with two different grids.

The Chen-Kim k-ε turbulent model was used in this study, which is a variant of the widely used standard k-ε model. It has been improved based on the two-equation k-ε model and has better prediction of separation and vortices than the classic k-ε model. This model has been validated with
experiments in simulating the flow recirculation [8]. A standard wall function was adopted to provide near-wall boundary conditions for turbulence-transport equations. In RhinoCFD, the global residuals were normalized for assessing convergence. Convergence was considered to be achieved when the normalized residuals fell below $10^{-5}$ and the error values percentage fell below 1% for all variables.

3. Results

3.1. Physical prototype results
Figure 2 shows the condensation pattern of the panel marked in purple in Figure 1d. The resulting line at the front between condensation and no condensation on the membrane is an isothermal line at $T_{\text{membrane}} = T_{\text{dewpoint}}$. Initially, this process was used to measure the membrane temperature to generate the diagram in Figure 2b to learn how to control the supply water temperature to the Cold Tube given ambient conditions. For this analysis, the extrapolated view factor of the hot environment at a point of this line occurring at the center of the middle radiant cooling panel is 0.472 and the average warm environmental temperature is 31.5 °C. This matches closely to the simulated results of a view factor of 0.455 and an average warm temperature of 32.0 °C. This is an exciting method to analytically calculate view factor from such a simple set of equations. However, the method is very sensitive to air temperature measurements, but the air temperature was easy to obtain during these experiments.

3.2. View factor and radiant temperature simulation results
The resultant mesh is shown in Figure 3a. The radiant temperature of the part of the membrane that is closer to the exit and to the floor is higher than the inward upperpart of the membrane. This pattern, created by the view factor of the cold surfaces versus that of the surrounding hot environment resembles the condensation pattern shown in Figure 2 insofar that the side of the membrane closer to the exit undergoes higher radiant temperature and therefore does not reach dewpoint temperature. In contrast, the side of the membrane far from the exit has a larger view factor of the interior cooling panels and smaller view factor towards the hot environment, and therefore its radiant temperature is lower and it condenses more quickly. However, the photographed condensation pattern does not match the difference between the membrane’s top and bottom radiant temperatures. This can be accounted for by the stratification of air inside the panel, as shown in the next subsection.

![Figure 3](image-url)
point, $T_{dp}$ (where $T_{dp} - T_w$ is the below dew point temperature difference) as constrained by the warmth of outdoor air temperature $T_{air}$, heating up the membrane above the dew point, (described by $T_{air} - T_{dp}$). Each point was produced by recording the air, water, and depoint temperatures at the moment that condensation was observed.

3.3. CFD simulation results
The CFD simulation results as displayed in Figure 5 clearly show the air stratification within the panel due to buoyancy. The bottom portion of the polyethylene membrane therefore will have less convective gains than its top portion, which will heat up faster. The region at the top surface of the membrane has a high temperature which is slightly lower than the ambient air temperature outside of
the panel. The temperature difference between the top and the bottom regions reaches over 10 °C. In addition, on the outside face of the cooler membrane surface, free convection would occur as well, by which air cooled by the membrane flows downward, adding to the difference in convective losses between the top and bottom regions of the membrane. Because of this stratification, the upper part of the membrane maintains higher temperatures than the lower part, which can explain the condensation pattern of the membrane showing variation from a colder bottom to a hotter top. Combining the results displayed in Figure 4 and in Figure 5 can supply an explanation for the specific condensation pattern observed in Figure 3a.

4. Conclusion

We demonstrate a sophisticated use of condensation on a membrane transparent to thermal radiation to determine temperature and validate heat transfer models. We present a novel set of methods for determining the view factor of warm and cold surfaces for radiant heat transfer calculations from observation using both a simulation and analytical model. Both methods have reasonable agreement. Extending the observation of condensation to a spatially resolved measurement is a novel process, built on methods of high precision humidity and temperature measurements at a point. Future work should replicate this method and determine sensitivity to the input parameters and assess further applications of these methods.

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