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Extracting Radiant Cooling From Building Exhaust Air Using the Maisotsenko Cycle Principle

Théo Keeley-LeClaire[1], Eric Teitelbaum[2], Suin Shim[3], Michael Bozlar[2,4], Howard A. Stone[3], Forrest Meggers[2,4]

[1]Princeton University, Department of Chemical and Biological Engineering, USA
[2]Princeton University, School of Architecture, USA
[3]Princeton University, Department of Mechanical and Aerospace Engineering, USA
[4]Princeton University, Andlinger Center for Energy and the Environment, USA

[*]*Corresponding email: eteitelb@princeton.edu*

ABSTRACT
Indirect evaporative cooling has exciting implications for air based thermal comfort. With recent advances in the research and commercialization of Maisotsenko Cycle (M-Cycle), or dew-point, evaporative cooling, thermodynamics can be fully leveraged to provide effectively free air cooling. However, few studies seek to generate cool surfaces by evaporation for radiant cooling. As a method to reduce building energy consumption, such an evapo-radiative system would maintain occupant thermal comfort at higher ventilation air temperatures and provide cooling at low cost. This study explores an analytical model for an M-Cycle evapo-radiative cooling system that derives a 1-D temperature profile throughout an experimental module and compares the outputs to experimental data to begin the model validation process.

KEYWORDS
Radiant Cooling, Thermal Comfort, M-Cycle, Evaporative Cooling

INTRODUCTION
Maintaining comfortable conditions in buildings accounts for a significant portion of energy use and carbon dioxide emissions, both in the United States and around the world. According to the US Energy Information Administration, nearly 40% of total energy consumption in the US is accounted for by commercial and residential buildings (EIA, 2018); of total building energy use, 19% is used for ventilating and cooling building spaces (EIA, 2012). At a global scale, the International Energy Agency reports that the residential sector consumes approximately 20% of world energy demand (IEA, 2015).

Current air conditioning technologies rely on cooling air by exchanging heat with a refrigerant vapor compression cycle. Air conditioning refrigeration cycles typically have coefficients of performance, COP, between 2 and 4, and the compressor in the refrigeration cycle and the fan which circulates the building air account for the majority of the system energy demand (Duan et al. 2012). Using water for evaporative cooling is a common strategy for heat rejection and cooling.

Conventionally, the wet-bulb temperature is the maximum temperature depression achievable by conventional evaporative coolers, which are viable options only in very dry climates. But advancements in heat and mass exchanger integration and geometry have led to dew point evaporative cooling, known as the M-Cycle. The M-cycle was initially conceptualized by Valeriy Maisotsenko as a performance-enhancing modification of traditional IEC heat exchangers (Duan et al. 2012). Both processes cool product air via heat exchange with a wetted working air channel, usually through a metallic plate separating the two channels. Whereas in the traditional process the working air is completely separate from the product air and is at the same temperature at the inlet, the M-cycle diverts some of the product air to the working air channel. This pre-cools the working air and the higher temperature difference between the two streams drives more effective heat exchange. This modification allows the M-cycle to approach much lower temperatures than the traditional process, which is desirable for a radiant panel because the temperature of the radiant surface determines its ability to cool effectively.

Hydronic radiant cooling systems have been gaining traction in the United States and abroad for providing comfort during cooling conditions. Traditionally, chilled water is circulated to cool ceiling panels, which condition spaces through both radiative and convective heat transfer, and are often effective at reducing energy consumption compared to forced air systems. However, radiant cooling still faces significant challenges, namely the need for precise temperature control and humidity sensing to prevent water from condensing on the panel and falling on building inhabitants below. The hydronics for a radiant system are rarely integrated with an evaporative system.

This project seeks to simultaneously address challenges to both hydronic radiant cooling and indirect evaporative cooling systems by developing a dew-point evaporatively cooled radiant panel. Using evaporation to cool the radiant surface promises to be more efficient than circulating chilled water and inherently avoids the possibility of condensation, as the lowest temperature that can be attained by evaporative cooling is above the dew point of the inlet air. Such a technology links surface temperatures in the room to air temperature and relative humidity, further simplifying controls and leveraging other low exergy technologies, such as desiccant dehumidification, to control for comfort using air dehumidification and subsequently lowered radiant temperatures for occupant control rather than air conditioning.

METHODS

Few studies exist on indirect evaporative cooling conducted in an integrated manner for radiative cooling. Younis et al. (2015) model such a similar system for small office environments, however the primary mode for occupant comfort is through chiller-based air conditioning. In this study, we propose a design methodology maintaining occupant thermal comfort in the absence of a chiller using a truly integrated approach to radiative and evaporative cooling requiring only control of the relative humidity of the inlet air.

In our integrated evaporative-radiant cooling system, it is crucial to have a highly conductive, but also easily tunable, material for thin film formation of water in the wet section. We suggest a boehmite nanostructure on an aluminum surface as a key material for the cooling system. Aluminum, when exposed to high temperature water, forms a nano-scale porous layer of aluminum oxide. A stable boehmite phase is achieved for a sufficiently high and also wide range of temperature of water (Bernard and Randall, 1960; Vedder and Vermilyea, 1968; Abdollahifar et al. 2015; Rani and Sahare, 2015); we choose the hydrothermal synthesis at ~80 ºC for 24 hours as a standard treatment (Shim et al. 2018). The presence of the boehmite layer provides changes not only to the chemical composition of the substrate, but also to its physical structure, in terms of crystallography and surface roughness, which naturally turn the aluminum surface from hydrophilic to superhydrophilic (Shim et al. 2018). Such a superhydrophilic surface improves evaporative cooling performance by keeping the thermal mass of coolant (water) low compared to the regular hydrophilic aluminum surface.

Generating a thin film of water allows a large ratio of evaporating area to volume, which can be one of the most effective ways to enhance evaporative heat transfer.

Analytical Model

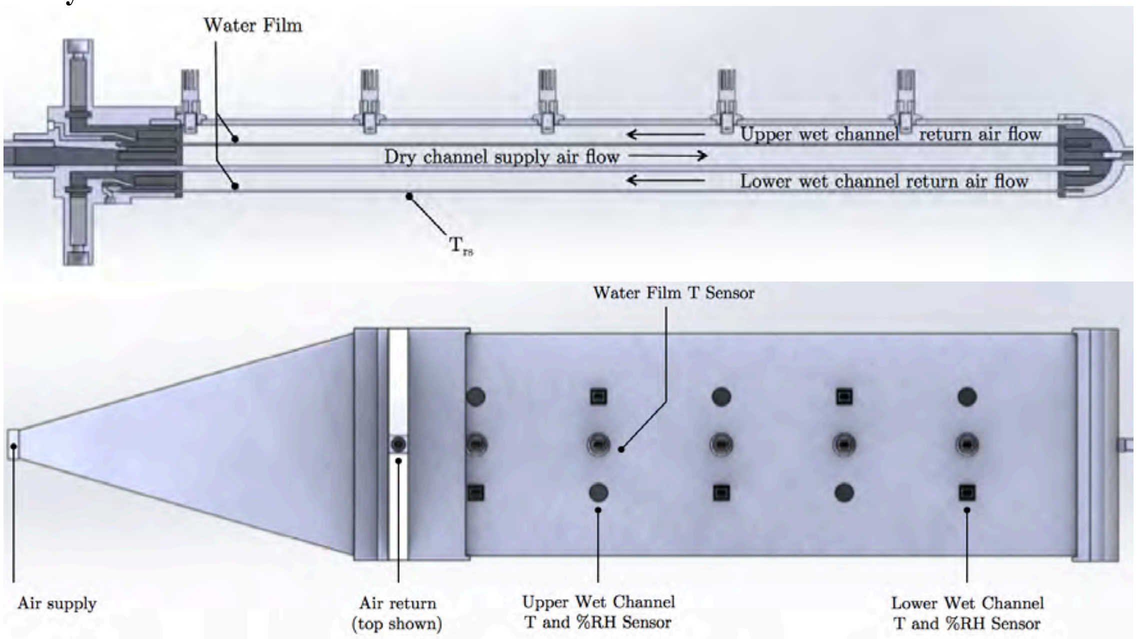

Figure 1: Section (top) and plan (bottom) drawings of the M-Cycle test apparatus, showing sensor locations and air flow directions. Starting from the top, the first and third horizontal plates are acrylic, chosen as an insulator; the second and fourth horizontal plates are wetted boehmitized aluminum. Air flow splits in the rounded baffle to the right, and flow can be modulated between 0 and 100% of supply air to either top or bottom stream.

A one-dimensional model tracking bulk average values was chosen to assess the performance of the prototype evapo-radiant device shown in figure 1. Because of its simplicity and relative accuracy (less than 10% error in the temperature of the product air), a previous simulation used by Hasan (2010) to model regenerative (M-cycle) evaporative coolers was used as a conceptual basis for the model developed here.

The conceptual basis is straightforward: air enters the panel (via the dry channel inlet) at ambient temperature and humidity and is cooled by heat exchange with the wet channel through the aluminum plate. At the end of the dry channel, the air flow is split into two streams that recirculate along the wet and radiant channels, respectively, at the end of which the air is collected and exhausted from the building. Thus, the bulk heat transfer equations account for (1) heat exchange between air in the dry channel and air in the wet channel, and (2) exchange between the room (via radiation and natural convection) and air in the radiant channel. The equations model the variation of the temperature, enthalpy, and humidity of the air streams with position in the panel (which is expressed synonymously by area $dA = Wdx$). Beginning with exchange between the dry and wet channels, the overall energy balance is given by:

$$M_w \frac{dh_w}{dA} = M_d C_p \frac{dT_d}{dA} \tag{1}$$

The heat (enthalpy) lost by air in the dry channel is given by Fourier's law as:

$$M_d C_p \frac{dT_d}{dA} = U(T_d - T_f) \tag{2}$$

where U, the overall heat transfer resistance from the dry air to the water film-wet air interface, is given by adding the heat transfer resistances of the air, aluminum, and water film, respectively:

$$U = (1/\alpha_d + \delta_{Al}/k_{Al} + \delta_f/k_f)^{-1} \tag{3}$$

The enthalpy gained by the air in the wet channel is given by both latent (water vapor mass transfer) and sensible (heat transfer) components:

$$M_w \frac{dh_w}{dA} = \alpha_w(T_f - T_w) + \beta(W_s - W_w) \tag{4}$$

However, assuming that the Lewis relation (5), which compares energy and mass propagation rates, can be taken as unity under these conditions, this equation can be simplified as an enthalpy driving force (6) between the water film and the bulk air as shown in Hasan (2010):

$$Le = \frac{\alpha_w}{\beta C_H} \approx 1 \tag{5}$$

$$\dot{M}_w \frac{dh_w}{dA} = \beta(h_s - h_w) \tag{6}$$

Solving equations (1), (2), and (6), and (7) will require relating the enthalpy and humidity of the saturated boundary layer at the air/water interface with the temperature of the water film. These relationships are tabulated in Chapter 6 of the 2001 ASHRAE Fundamentals Handbook and can be approximated as linear over the temperature range of interest with little loss of accuracy, the novel contribution to the field in this model. Thus, we can write the following linear relationships:

$$h_s = a_{s,h} T_f + b_{s,h} \tag{7}$$

Because the dry channel is insulated from the radiant channel by an acrylic plate, these equations fully define the temperature, humidity, and enthalpy profiles in the wet and dry channels. Before they can be solved, the heat and mass transfer coefficients need to be defined; for laminar flow between parallel plates, Burmeister (1983) gives the Nusselt number:

$$Nu = \frac{\alpha D_h}{k} = 4.861 \tag{8}$$

Thus, the heat transfer coefficient can be calculated, from which the mass transfer coefficient is known by the Lewis relation. Equation (1) can be integrated, which allows us to solve equations (2), (6), and (7) for the dry channel temperature, water film temperature, and wet channel enthalpy. The only difficulty here is in initial conditions; though the ambient conditions are known, none of the outlet conditions are given. Thus, we cannot set $x = 0$ to be the inlet of the panel because initial conditions would not be known for both the wet and dry air streams. Instead, we must guess the temperature at the recirculation point (the outlet of the dry channel where the air flow splits and recirculates to the wet and radiant channels). We can then calculate the outlet temperature and compare with known ambient conditions in order to adjust the guess and iterate to a solution. Heat transfer between the room and the radiant channel were then addressed to calculate the steady state temperature profile of the evapo-radiative M-Cycle apparatus for a given plate area and convective and radiative exchange with a fixed room condition. First, we assume that the heat transferred from the room to the water film is equivalent to the enthalpy lost by the water film to the air flow in the radiant channel:

$$\frac{dq_{room}}{dA} = \dot{M}_r \frac{dh_r}{dA} \tag{9}$$

The enthalpy gained by the air flow is analogous to that in the wet air channel, so we can write:

$$\dot{M}_r \frac{dh_r}{dA} = \beta(h_s - h_r) \qquad (10)$$

Finally, the heat gained from the room is given by summing radiant and convective contributions. As shown by Lienhard and Lienhard,[3] radiant heat transfer can be expressed with a heat transfer coefficient analogous to that of natural convections for small temperature differences, so we can write an overall heat transfer coefficient:

$$U_{room} = (1/(\alpha_{radiant} + \alpha_{convective}) + \delta_{Al}/k_{Al} + \delta_f/k_f)^{-1} \qquad (11)$$

$$\frac{dq_{room}}{dA} = U_{room}(T_{ambient} - T_f) \qquad (12)$$

Equations (9), (10), and (12) can be solved analogously to the wet channel/dry channel case by first solving for T_f and then integrating.

RESULTS AND DISCUSSION

Initial experiments of the M-Cycle model produce intuitive results with reasonable outputs. Certain initial criteria were chosen to facilitate model validation in future experimentation. Figure 2a shows 9 data points where the ratio between air flowing to top and bottom wet channels is varied between 0 and 1. For example, a ratio of 0.25 indicates that 25% of the air supplied in the dry channel is diverted to the top wet channel, and the remaining 75% is diverted to the lower wet channel with the radiant panel facing the room. The y-axis is the average temperature across the lower radiant plate exposed to the room. Points are placed at the lowest average temperature value, with the entire dataset extending to the right y-axis to show the initial inlet air temperature. For all runs, the initial inlet air absolute humidity was 9 g/kg. Additionally, three runs of the model were conducted with plate spacings of 3.5, 6, and 10 mm.

Figure 2b shows experimental data compared to model outputs for the wet and dry channel temperatures as well as surface temperature data for the exposed panel. This data was taken at an air flow ratio of 0.23 and an inlet temperature of 20.7 °C and absolute humidity of 6 g/kg. Airflow was at a Reynolds number of *Re* = 90 inside the apparatus.

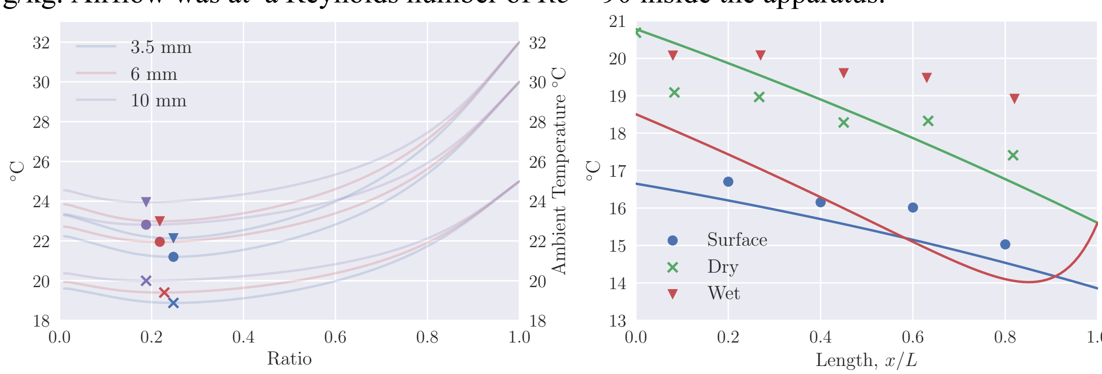

Figure 2: (a-left) The optimal ratio for air flow between top and bottom wet channels for producing a cold lower plate for three inlet air conditions and plate spacings. (b-right) Experimental data points compared to model outputs.

The wet bulb temperature for the experiment's inlet air was 13 °C and the dew point was 7 °C. The average measured radiant surface temperatures from figure 2b was 16.0 °C for a wet bulb efficiency of 61% and a dew point efficiency of 34%. These low efficiencies of the test apparatus could be explained by the apparent ineffectiveness of the wet channel at remaining wetted, which was visibly confirmed and shown in the data with the higher than expected temperatures for the wet channel compared to the model. This indicates that not enough precooling of the air occurs for the air entering the wet channels, and the lower channel is effectively a wet bulb radiant cooler rather than a dew point radiant cooler. Optimizing the setup was outside the scope of the experiment, as the ultimate goal is to begin validation on the analytical M-Cycle evapo-radiative cooling model. However, we will re-examine the surface wetting to make sure the physical data represents the modeled evaporative phenomena.

CONCLUSION

A major goal of the study was to develop an analytically solvable model for M-Cycle enhanced evapo-radiative cooling apparatus. The model is an exciting advance for describing this type of system. Linking air temperature, humidity, and surface temperatures is an innovative concept for control purposes, as fewer degrees of freedom simplify control and system response. Future work will be conducted to fully validate the model, drawing conclusions for the system design parameters, such as optimal airflow ratios between the channels, controls for comfort conditions based on climate data, and eventually an energy budget for the entire system.

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