

Heat and mass transfer study in a direct evaporative cooler

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This paper presents a simple and efficient methodology to design a household desert cooler, predict performance of evaporative medium and determine pad thickness and height for achieving maximum cooling. Inputs for formulation include inlet air DBT and humidity ratio, air velocity, water temperature and flow rate, and geometrical properties of evaporative medium. A test rig was designed and fabricated to collect experimental data. Predictions of air condition at cooler outlet for given input conditions agree satisfactorily for air exit temperature ($\pm 15\%$) and humidity ratio ($\pm 10\%$). Results are useful for size selection of medium geometry required for an evaporative cooler design.

Keywords: Cross flow, Direct evaporative cooling, Heat and mass transfer studies, Rigid media

Introduction

With onset of energy crisis and environmental problems caused by conventional air-conditioners, importance of evaporative cooling has grown. Mathews *et al*¹ provided a detailed description of evaporative cooling. Baker & Shryock² analyzed cross flow-cooling tower using finite difference method. Kettleborough & Hsieh³ analyzed performance of counter flow indirect evaporative cooler. In a new type of direct evaporative cooler⁴, a thin plastic plate was utilized as surface for evaporative cooling. A theory⁵ was proposed to correlate heat and mass transfer coefficients of wet and dry surfaces. Characteristics and performance of a cross flow evaporative cooler using honeycomb paper as packing material has been studied⁶. Exhaustive studies have been carried out on different configurations of spray air interaction⁷⁻¹¹. Due to maintenance problems and a lower adiabatic efficiency of conventional pads ('khus' pads or wood wool pads), rigid media cellulose (RMC) pads¹² are preferred today.

This study presents development of a simple procedure to design and predict performance of cooler given by Dowdy & Karabash¹³ and validated with experimental results.

Experimental Section

Mathematical Formulations

In psychrometric process, dry air is considered as a single gas (av mole mass, 28.9645). In this work, humid air is considered as a mixture of two gases (dry air and water vapour). Heat transfer will occur between RMC wet surface (temp., T_s) and humid air flow (Temp., T) close to a wet surface (Fig. 1a). If absolute humidity (conc.) ω_s of air close to surface area (dA_s) is different from humidity of draft (ω), a mass transfer will occur. Elementary sensible heat (δQ_c) is given as

$$\delta Q_c = h_c dA_s (T_s - T) \quad \dots(1)$$

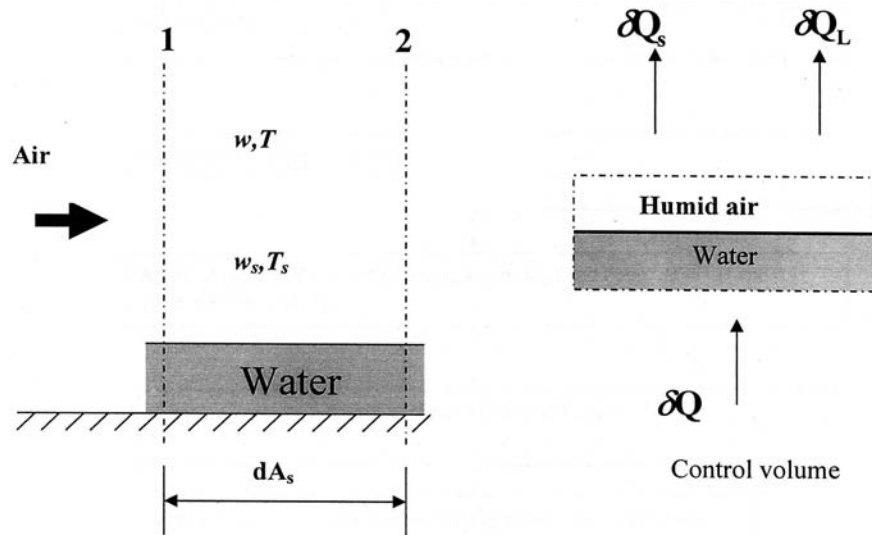
Latent heat (δQ_L) is determined by energy conservation law as

$$\delta Q_L = \delta Q - \delta Q_c = h_{vs} (dm_v) = h_{vs} h_m dA_s (\omega_s - \omega) \quad \dots(2)$$

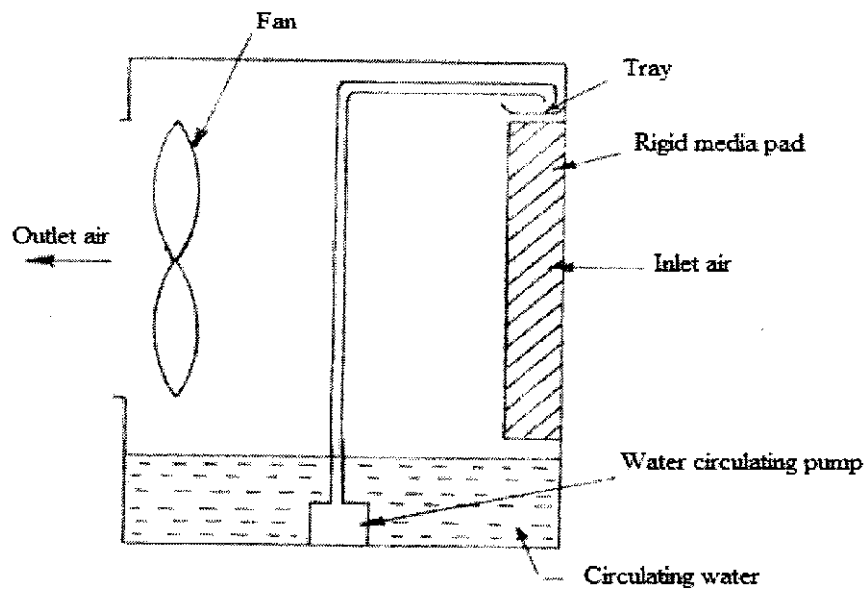
where, δQ = total energy ($\delta Q_c + \delta Q_L$); dm_v = mass flow rate of water vapour transfer; h_c = convective heat transfer coefficient; h_m = convective mass transfer coefficient; h_{vs} = specific enthalpy of vapour at surface temperature

In Eqs (1) and (2), water temperature varies throughout the medium and hence problem arises in

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(a)



(b)

Fig. 1—(a) Heat fluxes and control volume; (b) Schematic of a cross flow direct evaporative cooler [Inputs - Inlet air conditions (DBT, RH, air velocity), Water (temperature, flow rate) and Medium characteristics (medium thickness, height, volume, surface area); and Outputs - Outlet air conditions (DBT, RH)]

evaluation of T_s and ω_s . Dowdy & Karabash¹³ presented a set of equations for heat and mass transfer coefficients associated with evaporative process in a rigid impregnated cellulose medium. This study is based on elevated inlet water temperatures entering medium above that of ambient wet bulb temperature (WBT). Correlation presented for heat transfer is given as

$$Nu = \gamma \times (Re^{0.8}) \times (Pr^{0.33}) \quad \dots(3)$$

where $\gamma = 0.1 \times (D_s^{0.12})$; Reynold number, $Re = \frac{\rho_a \times u \times D_s}{\mu_a}$; Prandtl number, $Pr = \frac{c_{pa} \times \mu_a}{k_a}$; D_s , characteristic length (m); air density (kg/m^3), $\rho_a = \frac{P_{atm}}{287.08 \times T_{wb1}}$; μ_a , dynamic viscosity; k_a , thermal conductivity; u , air velocity; c_{pa} , specific heat of air; P_{atm} , atmospheric pressure; T_{wb1} , WBT of air at inlet.

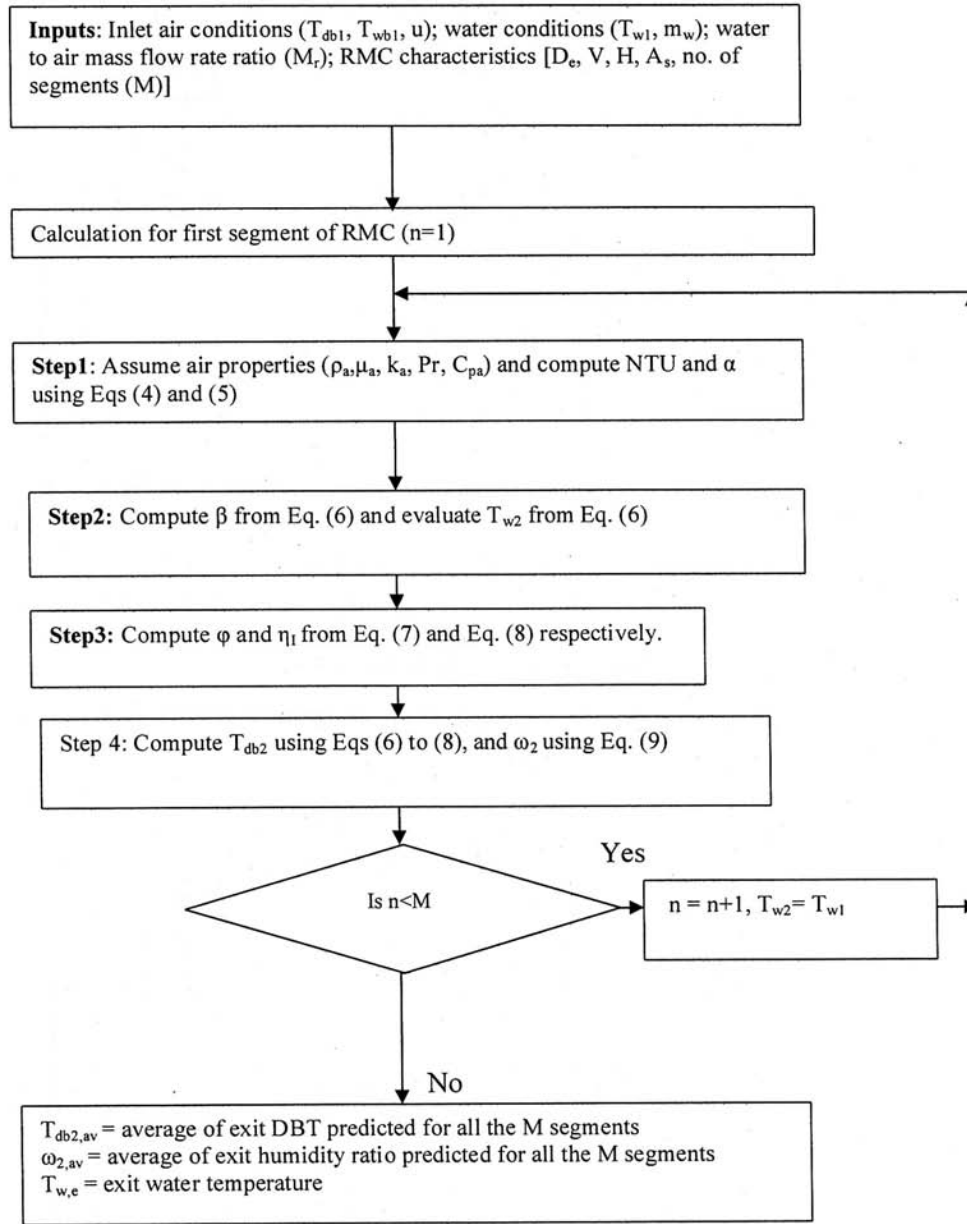


Fig. 2—Flow chart for calculation of average exit air and water conditions

Number of transfer units (NTU) is given by

$$NTU = \frac{h_c \times A_s}{m_a \times C_{pa}} \quad \dots(4)$$

where, h_c , heat transfer coefficient; A_s , surface area of RMC pad; m_a , mass flow rate of dry air.

In Eqs (3) and (4), properties of dry air (μ_a , k_a , and C_{pa}) were evaluated based on average dry air temperature¹⁴. Also, $D_e = V/A_s$, where, V and A_s

are volume and surface area of cellulose medium, respectively. Following dependent dimensionless parameters (α , β , ϕ) describe performance of medium with entering water temperature above ambient wet bulb

$$\alpha = \frac{T_{w1} - T_{wb1}}{T_{db1} - T_{wb1}} \quad \dots(5)$$

$$\begin{aligned} \beta &= 0.44 \times (\alpha^{0.932}) \times (NTU^{0.089}) \times (M_r^{-0.116}) \\ &= \frac{T_{w1} - T_{w2}}{T_{db1} - T_{wb1}} \quad \dots (6) \end{aligned}$$

$$\phi = 1 - e^{(-1.07(NTU)^{0.295})} \times \alpha^{-0.556} \times M_r^{-0.051} \quad \dots(7)$$

where, M_r , water to air mass flow rate ratio; T_{w1} , inlet water temperature; T_{w2} , outlet water temperature; T_{db1} , inlet air DBT.

Finally, actual exit DBT (T_{db2}) is calculated as¹³

$$\eta = \phi \eta_1 \quad \dots(8)$$

where, $\eta = (T_{db1} - T_{db2}) / (T_{db1} - T_{wb1})$, and $\eta_1 = 1 - \exp[-1.037NTU]$.

Humidity ratio¹⁶ is calculated as

$$\omega_2 = \frac{0.62198 P_{wv,2}}{p_{atm} - p_{wv,2}} \quad \dots(9)$$

where $p_{wv,2}$ = partial pressure of water vapour in exit air.

Simulation Procedures

Above formulations have been applied by keeping height of medium as a variable and limited to 30.5 cm only as above equations have been developed¹⁵ by keeping height of medium fixed as 30.5 cm. Therefore, height of medium is divided into multiple segments of 30.5 cm each. Exit water temperature from first segment becomes inlet water temperature for next segment. Outlet air temperature is average temperature of predicted outlet air temperature of each segment. Similarly, outlet air humidity ratio of each segment is predicted and average value will represent outlet air humidity ratio. In present case, a 61 cm height (equivalent to two segments) of medium was taken (Fig. 1). A detailed algorithm to solve mathematical formulation is shown in Fig. 2.

Experimental Setup

For verification of above model, an evaporative cooler (height, 91.4 cm) was fabricated using a galvanized iron sheet (Fig. 1b). Cross section of cooler sump and height are 71.1 cm x 71.1 cm and 16.5 cm respectively. A RMC pad (61 cm x 61 cm x 10.2 cm) is attached in upstream side of fan. Corresponding calculated value of equivalent diameter, d_e , for selected pad is 0.0089 m. Water circulating pump supplies water to water-distribution tray, which evenly distributes water to pad. Fan and water circulating pump wattage are 106 W and 18 W respectively. An electronic regulator was fitted to control flow rate of air.

Experimental Procedure

Variable parameters during experiments were air velocity, inlet water temperature and inlet air conditions. No attempts were made to control inlet air and water conditions respectively, which are totally governed by ambient air. At the start of experiments, water temperature is normally between the range of DBT and WBT of inlet air. During observations, water temperature approaches towards WBT of air. Sump water temperature was measured to characterize inlet water temperature at pad entry. Therefore, real controllable parameter was air velocity, which was measured at five locations of cooler exit (one at centre and remaining four between centre and periphery with a 90° phase difference) cross-section and consequently, average velocity was calculated. Based on minimum and maximum face velocity over evaporative pad, range for variation in mass flow ratio (M_r) was 0.15 to 0.4. Observations were collected for $M_r = 0.13, 0.2, 0.3$ and 0.4. At a particular instant of time, inlet and outlet air conditions (DBT and RH) were measured simultaneously by digital hydrometers. Total time for measurement of various readings for one observation was < 2 min, during which change in ambient condition were found to be insignificant. After an interval of 15 min, measurements were repeated for next observation. In each set, 5-7 observations were taken.

Experimental Uncertainties

For comparison of experimental data with model predictions, it is necessary to quantify uncertainties in measurements of ambient DBT and RH, outlet DBT and RH, and air velocity. Vane type anemometer accuracy was $\pm (2\% + \text{one digit})$, which results in uncertainty values between $\pm 15\%$ to $\pm 5\%$ for air velocities (0.8-7.0 m/s) used in experiments. Response time of anemometer was 12 s, required for each measurement and 1 min for a set of 5 readings. Resolution of DBT and RH measurements was limited by instrument readout to $\pm 0.1^\circ\text{C}$ and $\pm 1\%$, respectively. Maximum uncertainties in exit air DBT and RH measurements are $\pm 0.1^\circ\text{C}$ and $\pm 1\%$. Uncertainty in water temperature and flow rate measurement were $\pm 0.1^\circ\text{C}$ and ± 1 ml, respectively. Overall experimental uncertainty variation is found to be 6.2-15.4%.

Results and Discussion

Experimental results

During experimentation, inlet air conditions were as follows: DBT, 24.8-28.4°C; and air humidity ratio,

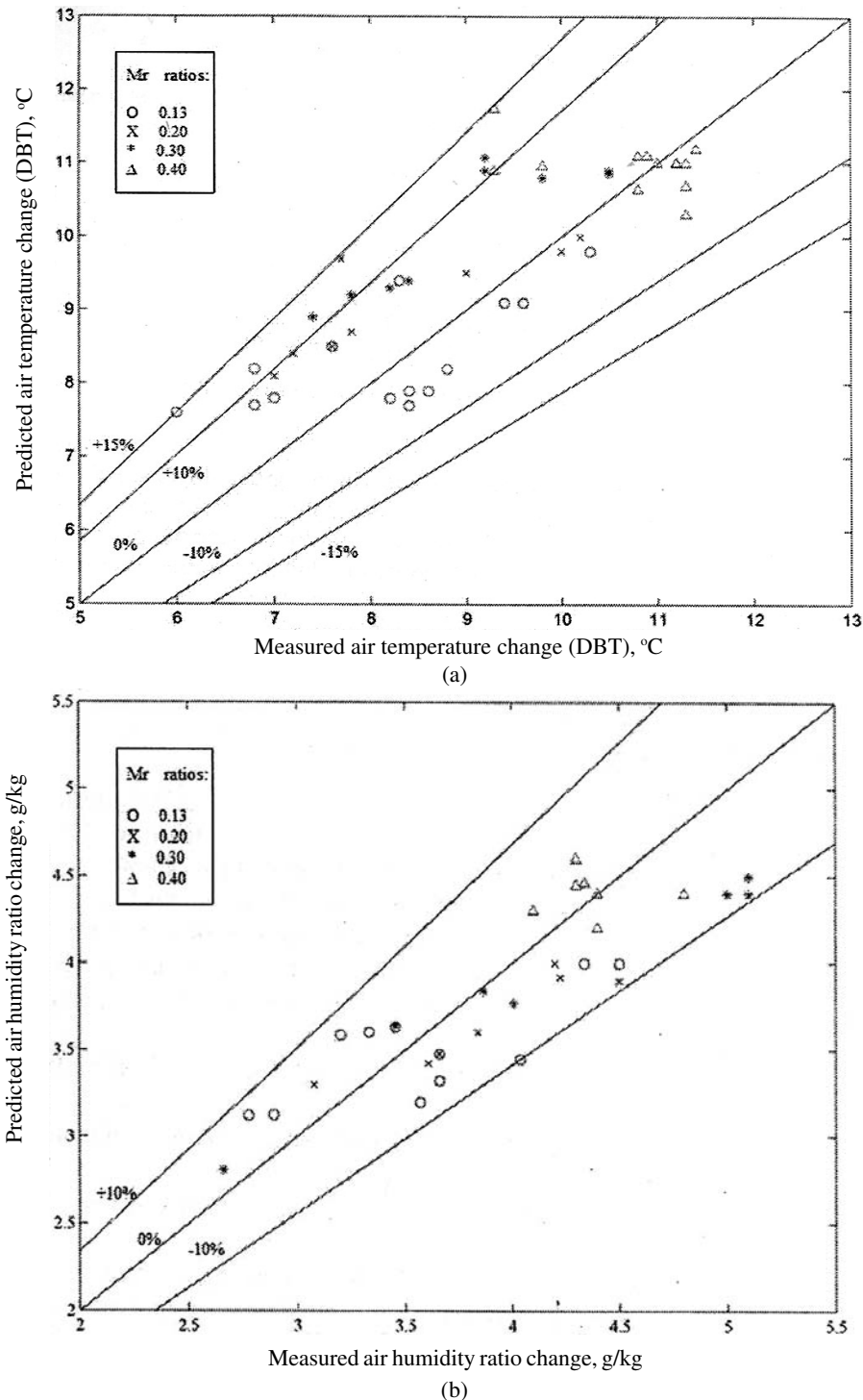


Fig. 3—Comparison of experimental data and model predicted readings of: a) Air temperature change (DBT); b) Humidity ratio change (w)

2.3–5.8 g/kg. For outlet air conditions, values were as follows: DBT, 16–21°C; and humidity ratio, 7–9 g/kg. Average air velocity range during experiments was 2–7 m/s. Average flow rate of pump was 7 l/min. Observations were collected for mass flow ratios 0.13, 0.2, 0.3 and 0.4. Water temperature range was 12–15°C.

Simulation Results and Comparison with Experiments

Simulation results are plotted (Fig. 3) for different values of mass flow rate ratios. Line (Fig. 3a) at 45° indicates perfect prediction of experimental values and other set of lines represent $\pm 10\%$ and $\pm 15\%$ matches. Predicted air temperature data (> 90%) lies within

$\pm 10\%$ deviation. Air temperature data ($<10\%$) lies between $\pm 10\%$ to $\pm 15\%$ deviation. Under predicted air humidity ratio change as compared with measured values, all predictions are well within accuracy of $\pm 10\%$ (Fig. 3b). Thus, overall predictions for temperature and humidity are within $\pm 15\%$ and $\pm 10\%$ respectively. In present formulation, average percentage uncertainty¹³ for various correlations to determine dimensionless parameters (α , ϕ , β etc.) vary between $\pm 7.1\%$ to $+12.2\%$. Further, average uncertainty (%) for present set of experimental data varies 6.2% to 15.4%. Keeping these facts into consideration, obtained prediction limits (exit air temperature, $\pm 15\%$; and humidity ratio, $\pm 10\%$) for may be quite reasonable. For design problem, size of RMC can be calculated in an iterative manner by assuming outlet air conditions (based on adiabatic evaporative cooling).

Conclusions

Experiments indicate that DBT decrease (up to 10°C) is attainable by employing evaporative cooling during dry months. Using present methodology, variations in temperature and humidity predictions are within the range of $\pm 15\%$ and $\pm 10\%$ respectively. Therefore, present methodology could be used for design and analysis of direct type evaporative coolers.

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