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Research Article

Variation of Water Temperature along the Direction of Flow: Effect on Performance of an Evaporative Cooler

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Abstract This paper presents a model for the evaluation of the variation of the water temperature along the direction of flow in an evaporating pad. The model has been used to evaluate the mean air exit temperature and the transient temperature of the water in the tank. The analytical results are in agreement with the observations in our experiments. The time variation of the temperature of water in the tank has been investigated and the new concept of using the tank water for cooling have been investigated theoretically and experimentally; the theory is in good agreement with experiment. It is seen that the penalty on the mean exit air temperature is negligible for thermal loads $\dot{Q}$ (for cooling) of the order of 1 kW; it is seen that it is 0.6 °C for $\dot{Q} = 2\,\text{kW}$. Further it is concluded that for typical coolers the steady state temperature of water in the tank is reached in a time of the order of one hour or less.

Keywords evaporation; cooling; energy balance; mass transfer

1 Introduction

The use of direct evaporative coolers (commonly known as desert coolers) is very widespread in the third world countries, during hot and dry weather. The cooling of air is accomplished by the flow of air in a direction, normal to a vertical porous pad, through which water flows from top to bottom; the water collects in a tray at the bottom of the pad and is pumped up to the top of the pad. The flow of air is induced by a fan, which draws in the outside air. It is suggested that in addition to cooling air, one can utilize the coolness stored in the bottom tray (or tank) for medium temperature cooling; as an example one can place a box in the tank to keep food cool or exchange heat with the help of a coil placed in the tank. The present paper explores this option and in the process makes further progress in modelling of such coolers, in particular by inclusion of the variation of the temperature of flowing water along the direction of flow; the expressions for the corresponding parameters also become different.

The classic books of Berman [1] and Watt [16] are early landmarks in the understanding of evaporative cooling. The two papers by Mathur and Jain [7,8] are typical of publications in the seventies and early eighties in this area. Sawhney et al. [9], Singh et al. [10] and Sodha et al. [11,12] analyzed the efficacy of evaporative cooling for thermal comfort in buildings. The present knowledge of evaporative cooling is based on theoretical and experimental work on heat and mass transfer in wet porous pads with cross flow of air and water and performance of devices based on evaporative cooling by Dowdy and Karabash [5], Zalewski and Gryglaszewski [19], Halasz [6], Stabat et al. [14], Camargo and Ebinuma [2], Dai and Sumanthy [4], Camargo et al. [3], and Wu et al. [17,18], amongst others. Mention may also be made of an interesting paper by Sodha et al. [13], which represents an attempt for getting rules of thumb, corresponding to design and applications of evaporative coolers. All these analyses correspond to the steady state and neglect the variation of the temperature of water along the direction of flow; the temperature of water has been assumed to be the wet bulb temperature, throughout the pad. In this paper the variation of the temperature of water along the direction of flow has been taken into account. The time variation of the temperature of water in the tank has also been investigated. The effect of addition of heat to the water in the bottom tank (for purpose of cooling) has also been studied. An expression for the average temperature of cool air, flowing out of the pad has also been derived. The theoretical results have been compared with the results of experiments conducted by the authors. Using the validated model, the steady state exit water temperature have been plotted as a function of parameter $\alpha = h_c F_p/\rho_a v_a c_{pa}$.

2 Analysis

2.1 Air & water temperature

Consider a vertical evaporating pad (Figure 1), with water flowing from the top to a bottom tray; the air flow is normal to the pad. The z and x axes may be taken along

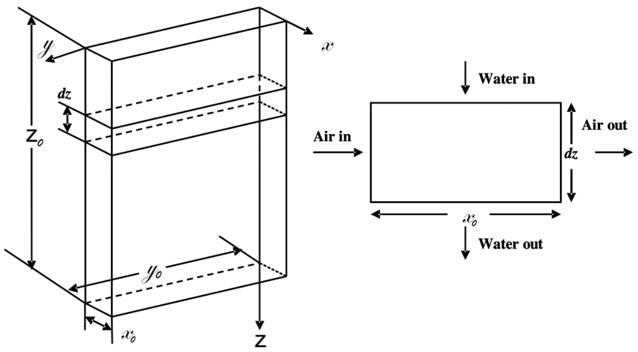

Figure 1: Pad profile and elemental volume.

the vertical and the direction of the air flow respectively. In common with the earlier analyses (mentioned in the introduction) the variation of the temperature of water along the x axis (direction of flow of air) has been neglected in this paper on account of the much larger heat capacity of water as compared to that of air. However, in contrast to earlier analyses the dependence of the temperature of water on z (the direction of flow of water viz the vertical) has been taken into account; the consequent dependence of the temperature of the exiting air on z and its mean value have also been evaluated. Dependence of the temperature of the tank water on time has also been investigated for transient conditions. A novel feature of the analysis is the incorporation of addition of heat, transferred to water in the tank by a body (to be cooled).

Following earlier work (e.g. Wu et al. [18]), the temperature of the air inside the pad is given by

$$\frac{T_a(x,z)-T_w(z)}{T_a(0,z)-T_w(z)}=\exp(-\alpha x), \tag{1}$$

where $\alpha=h_cF_p/\rho_a v_a c_{pa}$. The term $T_a(0,z)$ is just the temperature of the inlet atmospheric air, which may be denoted by T_{a0}, since it is independent of z. The parameter F_p is known as packing fraction or wettable surface area per unit volume.

From (1) the exit and the mean temperature of air is given by

$$T_a(x_0,z)=T_w(z)\{1-\exp(-\alpha x_0)\}+T_{a0}\exp(-\alpha x_0), \tag{2}$$

$$\overline{T}_a(x,z)=T_w(z)+\{T_{a0}-T_w(z)\}\{1-\exp(-\alpha x_0)\}/\alpha x_0, \tag{3}$$

where the bar indicates average over x.

Considering an element of pad of thickness dz. Figure 1, the energy balance of water may be expressed as

$$\dot{m}_w c_w\frac{dT_w}{dz}dz+\dot{Q}_L dz+\dot{Q}_S dz=0; \tag{4}$$

here $\dot{Q}_L\,dz$ and $\dot{Q}_S\,dz$ are the latent and the sensible heat transfer due to convection from water to air in the element of volume $x_0y_0\,dz$.

For unit Lewis number the heat transfer per unit area, associated with the mass transfer is given by (e.g. Tiwari [15])

$$\dot{q}_L=13\times10^{-3}h_c(P_w-\gamma P_a). \tag{5}$$

It is to note that the evaporating surface in the volume element $x_0y_0\,dz$ is $F_px_0y_0\,dz$

$$\dot{Q}_L=\dot{q}_LF_px_0y_0=13\times10^{-3}h_c(P_w-\gamma P_a)F_px_0y_0. \tag{6}$$

Further,

$$\dot{Q}_S=h_c(T_w-T_a)F_px_0y_0. \tag{7}$$

In the range of temperature of interest the saturation vapour pressure of water (tabulated by Tiwari [15]) can be represented to a very good approximation by

$$P=R_1T^2+R_2T+R_3, \tag{8}$$

where $R_1=6.42\,\mathrm{Nm}^2\,^\circ\mathrm{C}^{-2}$, $R_2=-135.4\,\mathrm{Nm}^2\,^\circ\mathrm{C}^{-1}$, and $R_3=2516\,\mathrm{Nm}^{-2}$.

From (3), (4), (5), (6), (7), and (8) one obtains,

$$\frac{dT_w}{d\zeta}=A{T_w}^2+BT_w+C, \tag{9}$$

where

$$\zeta=[(h_cF_px_0y_0)/\dot{m}_wc_w]z,$$
$$K_1=\{1-\exp(-\alpha x_0)\}/\alpha x_0,$$
$$A=0.083,$$
$$B=(1.76-K_1),$$
$$C=0.083\gamma {T_{a0}}^2+T_{a0}(K_1-\gamma1.76)+32.7(\gamma-1).$$

Integrating (9) one obtains

$$\frac{T_w(\zeta)-(B/2A)-C_1}{T_w(\zeta)-(B/2A)+C_1}=\beta\exp(-2AC_1\zeta), \tag{10}$$

where

$$\beta=\frac{[T_{w0}-(B/2A)-C_1]}{[T_{w0}-(B/2A)+C_1]},\quad C_1=\sqrt{(B/2A)^2+C/A},$$

and T_{w0} is the temperature at the top of the evaporative pad ($\zeta=0$), say T_0.

From (10) the exit and the mean (over z) temperature of water flowing through the cooler pad are

$$T_w(\zeta_0)=(B/2A)+C_1\left\{\frac{(1+\beta\exp(-2AC_1\zeta_0))}{(1-\beta\exp(-2AC_1\zeta_0))}\right\}, \tag{11}$$

$$\langle T_w(\zeta)\rangle=\frac{B}{2A}+\frac{1}{A\zeta_0}\ln\left\{\frac{\exp(2AC_1\zeta_0)-\beta}{1-\beta}\right\}-C_1. \tag{12}$$

From (2) and (12) the average exit temperature of air is given by

$$\langle T_a(\zeta)\rangle=\exp(-\alpha x_0)\{T_{a0}-\langle T_w(\zeta)\rangle\}+\langle T_w(\zeta)\rangle. \tag{13}$$

2.2 Temperature of the tank water

If the heat transfer during the flow of water from the tank to the top of the pad is negligible, the temperature of the water in the tank and at the top of the evaporating pad may be taken as the same, viz T_0.

If $\dot{Q}$ is the rate of heat rejected to the tank water on account of cooling an extraneous object (by circulation of water or otherwise), the energy balance equation for the water in the tank is

$$M_t c_w \frac{dT_0}{dt} = \dot{Q} - \dot{m}_w c_w \{T_0 - T_w(\zeta_0)\}. \tag{14}$$

From (11) and (14)

$$\frac{dT_0}{dt} = \frac{\dot{Q}}{M_t c_w} - \frac{\dot{m}_w}{M_t}\left[(T_0 - B/2A) - C_1\left\{\frac{1+\beta\exp(-2AC_1\zeta_0)}{1-\beta\exp(-2AC_1\zeta_0)}\right\}\right]. \tag{15}$$

Equation (15) can be solved using the boundary condition at $t = 0$, $T_0 = T_{00}$, where T_{00} is the initial ($t = 0$) temperature of the water in the tank.

2.3 Average air exit temperature coming out when heat is added to the tank water

The time variation of the average water temperature $\overline{T}_w(\zeta)$ can be evaluated from (12) by substituting $T_{w0} = T_0$ from (15). Similarly the time variation of average air temperature can be determined by using (15) and (13). It may be remembered that T_0 occurs in the expression for β.

3 Experiment

To have a controlled experiment, the cooler was fitted with an exit air duct going out of the room. The inlet air for the cooler was drawn from air in a $(6\times 6\times 3.6)\,\mathrm{m}^3$ room with an open window; Figure 2 is the schematic diagram of the arrangement. The temperature and humidity of the air in the room (i.e. the inlet air of the cooler) and that of washed air were determined by temperature and humidity sensors; the mean values were used for comparison of the data with theory. The velocity of the air incident on the pad was measured at four points by an anemometer and the average value used for computing h_c. Figure 3 is the schematic diagram of the cooler. The single evaporating pad with dimensions $(0.76\times 0.60\times 0.10)\,\mathrm{m}^3$ had the $(0.76\times 0.60)\,\mathrm{m}^2$ faces bare for inlet and exit air, the $(0.76\times 0.10)\,\mathrm{m}^2$ faces were covered by plywood; the top $(0.60\times 0.10)\,\mathrm{m}^2$ face was covered by the perforated bottom of a tray with an inlet for pumped water. The water from the bottom $(0.60\times 0.10)\,\mathrm{m}^2$ face is allowed to fall on a perforated tray (with sensors) placed over the tank. The pad was made of GLASdek (Munters Co., Switzerland) with evaporating surface of $440\,\mathrm{m}^2\mathrm{m}^{-3}$ (curtsey manufacturer) and convective heat transfer coefficient

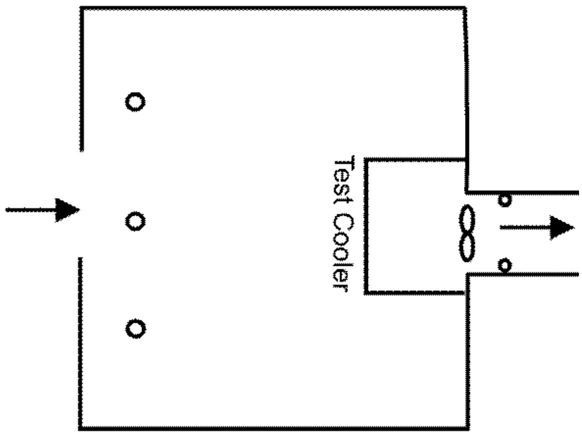

Figure 2: Climate room: (○) thermocouples and Hygrometers.

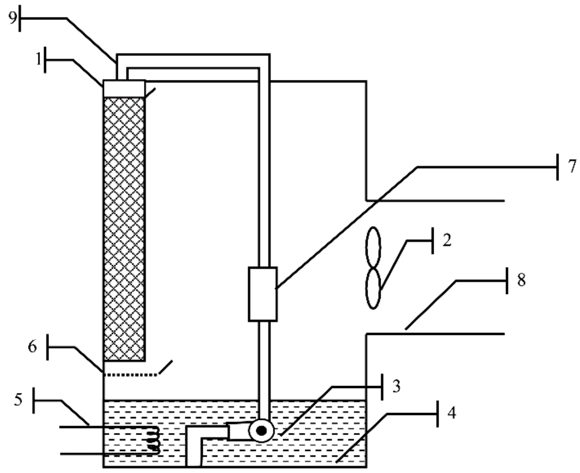

Figure 3: Test cooler: 1. cooler pad, 2. fan, 3. pump, 4. tank (with thermocouples and hygrometer), 5. heater element, 6. perforated tray (with thermocouples), 7. rotameter, 8. outside duct (with thermocouples and hygrometers), and 9. water tube (with thermocouple).

h_c for this material has been experimentally determined by Wu et al. [18], as

$$h_c = 25.2 v_a^{0.65}\,\mathrm{Wm}^{-2}\,^{\circ}\mathrm{C}^{-1}.$$

The water falling down the pad is collected in a tank and pumped back to the top of the pad through an insulated pipe and rotameter (to measure the flow rate) outside the cooler assembly. One could use the circulation of water in the tank for medium cooling of extraneous objects. To simulate the heat transfer, electric heating coils (500 W, 750 W, and 1000 W) were placed in the tank (one at a time). The water inlet and outlet temperature for the pad were also recorded. The temperature of water in the tank was also periodically monitored; it was seen that the temperature variation at

Table 1: Experimental and theoretical exit water and average air exit temperatures in steady state; pad is described in Section 3; *EXP.*, *Th.*, and *D.* represent experimental, theoretical, and percentage difference between the observed and theoretical values of the change in temperature of water and air (mean) after passage through the pad, respectively.

S.N.	v_a (ms^{-1})	$\dot{m}_w$ (kgs^{-1})	γ	Water temperature (°C)				Average air temperature (°C)			
				Inlet T_{w0}	*Exit Exp.*	*Exit Th.*	*% D.*	*Inlet* T_{a0}	*Exit Exp.*	*Exit Th.*	*% D.*
1.	1.66	0.116	0.335	33.0	24.9	25.0	1.3	39.3	30.8	31.6	10.4
2.	1.54	0.116	0.365	29.0	23.1	22.6	7.8	35.1	27.8	28.3	7.4
3.	1.26	0.116	0.252	30.0	24.1	23.4	10.6	40.9	29.8	30.9	11.0
4.	0.94	0.116	0.450	32.0	26.5	26.2	5.2	37.0	30.7	30.4	4.5

Table 2: Sets of conditions for which the time variation of tank water temperature was observed. ($\dot{m}_w = 0.116\,kgs^{-1}$); pad is described in Section 3.

S.No	$\dot{Q}$ (W)	v_a (ms^{-1})	γ	T_{a0} (°C)
1a	0	1.26	0.22	42.2
1b	0	1.55	0.24	43.6
1c	0	1.75	0.22	43.4
2	500	1.55	0.22	39.5
3	750	1.55	0.38	38.6
4	1000	1.55	0.22	39.5

different points in the tank was not more than 0.1 °C; this may be due to the shallow depth and turbulent mixing due to pumping and falling water.

4 Results and discussion

4.1 Comparison of theory with experiment (validation of model)

Table 1 presents a comparison of the steady state theoretical and experimental values of water outlet and average exit air temperatures for four sets of parameters viz velocity of air incident on the pad, mass flow rate of water, relative humidity and atmospheric air temperature. It is seen that the percentage difference between the theoretical ((11) and (13)) and experimental values of the drop in temperature of (i) water and (ii) mean air temperature, after flow through the pad is between 1.3% and 11.0%, respectively. The discrepancy may be ascribed to the assumption of unit Lewis number, the approximations inherent in the correlation for the convective heat transfer coefficient; further the dependence of latent and specific heat of air on humidity and temperature has been ignored in favor of a mean value (because the variation in range of interest is negligible).

Table 2 lists the sets of conditions under which the time variation of the tank water (mass 85 kg) temperature was observed. Table 3 presents comparison of the experimental and the theoretical (equation (15)) time variation of the tank water temperature for the corresponding conditions. The variation for set-2 condition. (Heating load of 500 W) is shown in Figure 4; the agreement between the theory and experiment is good. It is seen that for typical coolers the steady state is reached in a time of about one hour.

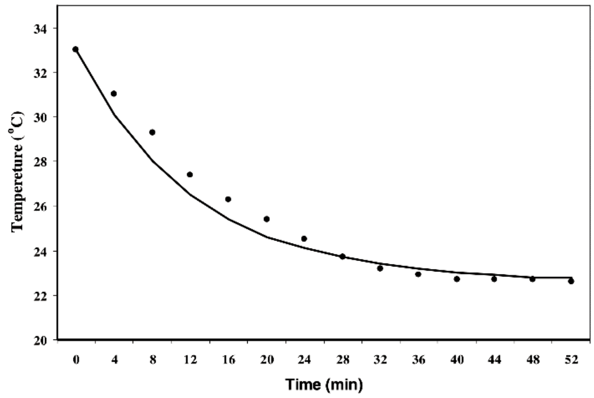

Figure 4: Time variation of tank water temperature with thermal load (set-2 conditions, Table 2): continuous curve is theoretical and dot (•) indicates experimental points.

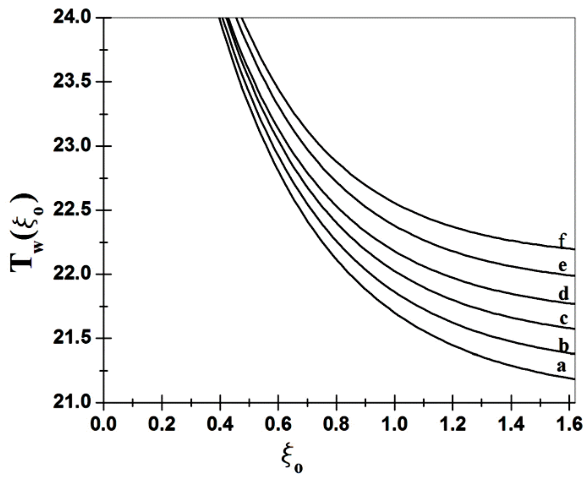

Figure 5: Computed dependence of exit water temperature $[T_w(\zeta_0)]$ on the height of the pad for $T_{a0} = 40\,°C$, $T_{w0} = 30\,°C$, $\gamma = 0.22$, and $\dot{m}_w = 0.116\,kgs^{-1}$; pad is described in Section 3. The letters a, b, c, d, e, f refers to $\alpha = 11, 10, 9, 8, 7, 6$, respectively.

4.2 Further computations

Figures 5 and 6 display the computed dependence of the exit water temperature and average air exit temperature on the

Table 3: Comparison of computed and observed time variation of tank water temperature (°C); subscripts "C" and "O" represents computed and observed values ($M_t = 85\,\text{kg}$).

Set → (Table 2)	1a		1b		1c		2		3		4	
Time (min) ↓	C	O	C	O	C	O	C	O	C	O	C	O
0	31.9	31.9	32.5	32.5	33.0	33.0	33.0	33.0	34.0	34.0	33.1	33.1
4	29.3	30.0	30.4	29.8	30.5	30.7	30.1	31.0	32.1	32.3	30.5	31.9
8	27.5	27.9	28.9	27.8	28.7	28.6	28.0	29.3	30.8	31.0	28.6	30.7
12	26.1	26.3	27.8	26.6	27.4	26.6	26.5	27.4	29.9	29.6	27.2	29.4
16	25.2	25.3	27.0	25.4	26.5	25.3	25.4	26.3	29.2	28.3	26.2	27.5
20	24.5	24.5	26.4	24.8	25.8	24.6	24.6	25.4	28.7	27.6	25.5	26.4
24	24.0	23.8	26.0	24.1	25.4	23.8	24.1	24.5	28.3	26.9	25.0	25.3
28	23.6	23.2	25.7	24.0	25.0	23.5	23.7	23.7	28.0	26.7	24.6	24.7
32	23.4	22.8	25.5	23.8	24.8	23.5	23.4	23.2	27.9	26.5	24.3	24.0
36	23.2	22.6	25.3	23.8	24.6	23.4	23.2	22.9	27.7	26.4	24.2	23.6
40	23.0	22.5	25.2	23.7	24.4	23.3	23.0	22.7	27.6	26.3	24.0	23.3
44	22.9	22.4	25.1	23.7	24.4	23.3	22.9	22.7	27.5	26.2	23.9	23.3
48	22.8	22.3	25.1	23.7	24.3	23.3	22.8	22.7	27.5	26.2	23.8	23.2
52	22.8	22.3	25.0	23.6	24.2	23.3	22.8	22.6	27.4	26.2	23.8	23.2

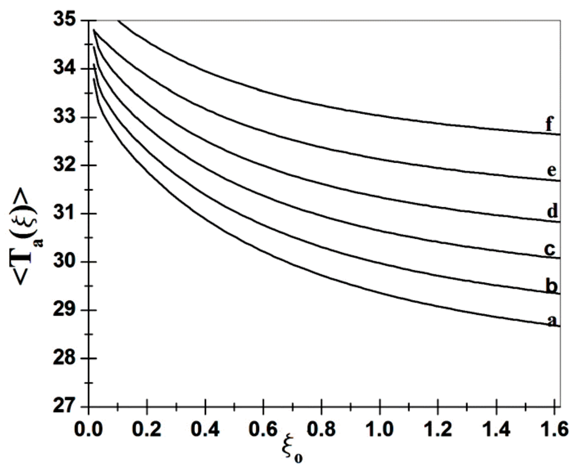

Figure 6: Computed dependence of exit average air (over z) temperature [$\langle T_a(\zeta)\rangle$] on the height of the pad for $T_{a0} = 40\,°\text{C}$, $T_{w0} = 30\,°\text{C}$, $\gamma = 0.22$, and $\dot{m}_w = 0.116\,\text{kgs}^{-1}$; pad is described in Section 3. The letters a, b, c, d, e, f refers to $\alpha = 11, 10, 9, 8, 7, 6$, respectively.

parameter $\alpha = (h_c F_p/\rho_a v_a c_{pa})$ for typical summer conditions at Raipur, India viz $T_{a0} = 40\,°\text{C}$, $T_{w0} = 30\,°\text{C}$, $\gamma = 0.22$, and $\dot{m}_w = 0.116\,\text{kgs}^{-1}$. It is seen that these temperatures are very near the saturating value for $\zeta \geq 1.6$.

Table 4 shows the steady state computed dependence of the temperature of tank water T_0 and average exit air temperature $\langle T_a(\zeta)\rangle$ on the refrigeration load $\dot{Q}$ for a typical set of parameters viz $T_{a0} = 40\,°\text{C}$, $T_{w0} = 30\,°\text{C}$, $\gamma = 0.22$, and $M_w = 85\,\text{kg}$; the pad is as described in Section 3. It is seen that the penalty on the mean exit air temperature is negligible (0.6 °C for $\dot{Q} = 2\,\text{kW}$) for loads of the order of 1 kW; this is due to the fact that regardless of inlet temperature, water acquires a steady state temperature, near the wetbulb temperature, after traversing a distance much smaller than

Table 4: Temperature of tank water and average air exit in steady state (after 1 hour) for, $T_{a0} = 40\,°\text{C}$, $v_a = 1.5\,\text{ms}^{-1}$, $\dot{m}_w = 0.116\,\text{kgs}^{-1}$, $\gamma = 0.22$, and $M_t = 85\,\text{kg}$.

$\dot{Q}$ (kW)	0	0.25	0.5	0.75	1.0	1.25	1.5	2.0
T_0 (°C)	22	22.5	23.0	23.5	24.0	24.5	25.0	26.0
$\langle T_a(\zeta)\rangle$ (°C)	30	30.1	30.2	30.3	30.3	30.4	30.5	30.6

the length of the pad. However the temperature of the tank increases rather fast with increasing load, limiting the useful of the concept.

5 Conclusions

A model for the evaluation of the performance of an evaporating pad, taking into account the variation of the temperature of water, along the direction of flow in an evaporating pad has been developed and validated experimentally. The model has also been extended to evaluate the temperature of the tank as a function of the rate of heat addition (to cool extraneous objects); it has been also validated experimentally. It is seen that for typical coolers the steady state is reached in a time of the order of one hour or less.

Nomenclature

- c_{pa} Specific heat of air at constant pressure (J.kg^{-1}.°C^{-1}).
- c_w Specific heat of water (J.kg^{-1}.°C^{-1}).
- F_p Packing fraction of the pad (m^{2}m^{-3}).
- h_c Convective heat transfer co-efficient between air and water (Wm^{-2} °C^{-1}).
- h_e Mass transfer co-efficient between air and water (kg.m^{-2}s^{-1}).
- $\dot{m}_w$ Mass flow rate of water (kgs^{-1}).
- M_t Mass of tank water (kg).
- P_a Saturated water vapour pressure in air (Nm^{-2}).
- P_w Saturated water vapour pressure in air at the temperature of water (Nm^{-2}).

$\dot{Q}$	Rate of heat added to tank water (W).
T_{a0}	Temperature of the inlet atmospheric air (°C).
$T_a(x_0, z)$	Temperature of the exit air (°C).
$\overline{T}_a(x, z)$	Mean (over x) temperature of air (°C).
$\langle T_a(\zeta)\rangle$	Mean (over z) exit air temperature (°C).
$T_{w0}(T_0)$	Temperature of the water at the top of evaporative pad (°C).
$T_w(\zeta_0)$	Temperature of water at the bottom (exit) of the evaporative pad (°C).
$\langle T_w(\zeta)\rangle$	Mean temperature (over z) of water flowing through pad (°C).
T_{00}	Initial temperature of water in tank (°C).
t	Time(s).
v_a	Velocity of air (ms^{-1}).
x_0	Thickness of pad (m).
y_0	Breadth of pad (m).
z_0	Height of pad (m).
γ	Relative humidity of air.
ρ_a	Density of air (kgm^{-3}).

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