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HEAT TRANSFER COEFFICIENT AND CONCEPT OF RELAXATION TIME IN FORCED AIR DIRECT EVAPORATIVE COOLING SYSTEM

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Abstract: Evaporative cooling system with palm fruit fiber as cooling pad was studied at different ambient air temperature and relative humidity. Experiment was conducted with a prototyped direct evaporative cooling system for preservation of fruits and vegetable of moderate respiratory rates at a low flow rate of 0.6 $m^3 \cdot s^{-1}$. Three different models were proposed and used to obtain the heat transfer coefficient at different evaporative effectiveness. For the three models, the heat transfer coefficient varied from 173 to 857 $W \cdot m^{-2} K^{-1}$. The relaxation time was predicted as a finite time process and the value calculated from the model can be regarded as a hypothetical value since other heat transfer methods like conduction or even radiation loss were neglected. Therefore the value calculated might be much higher than the real relaxation time. For ambient air temperature range of 26.1 to 34.8°C which was cooled to 23.2 to 25.8°C, the average relaxation time was calculated as 0.71 to 1.68 s.

Key words: *Ambient temperature, evaporative cooling, heat transfer, relative humidity*

INTRODUCTION

Evaporative cooling system is an environmentally friendly air-conditioning system that uses water as the working fluid [1] which can be adapted to cool residential houses,

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production process in metallurgical shops, preserve fruits and vegetables, cool automobile engine and tractor cabins. When water evaporates it draws a considerable energy from its environment which produces considerable cooling effects. The wet porous media provides the environment for this to take place.

Basic principle involves the movement of air that is not too humid across this wet porous media causing evaporation of water from the pad media. This air movement results in two favorable changes, a reduction in ambient air temperature and increase in relative humidity of incoming air. Evaporative cooling is an adiabatic heat transfer process involving the process fluids (air and water). The sensible heat of air is reduced proportionally to the amount of evaporation that takes place [2].

The parameter for the energy balance equation across the wet porous pad includes the heat transfer coefficients and the temperature difference [3] [4]. These parameters are influenced by the type of pad material and the air speed [5]. [6] Stated that equipment design depends greatly upon reliable equations for explaining the heat transfer process in a system.

Therefore theoretical analysis on evaporative cooling is important for revealing the heat and mass transfer laws in evaporative cooling process as well as for predicting the process outputs under various operating and environmental conditions. Some equations have been developed to consider the heat and mass transfer in evaporative cooling [5][7][8][9]. All these models include the heat transfer coefficient and temperature difference of the process fluids. Also literature on how long the cooling process of the ambient air to storage space temperature takes under the tested conditions, which will give room for any modification, is very scarce.

The objectives of this study is to present information on the heat transfer coefficient using some model equations and guess the relaxation time in an evaporative cooling system. This will provide important knowledge for evaporative cooler design especially in rural area in Africa where local waste is used as a substitute for expensive and unavailable pad media.

MATERIAL AND METHODS

Cooling pad

Palm fruit fiber was used as the cooling pad. It was obtained from local palm oil processing industries which generates this material as a waste in there factory. The material was thoroughly washed with detergent, rinsed several times with water and allowed to dry under the sun for several days.

Evaporative cooling system

The evaporative cooling system used for the experiment was developed at the research workshop of the Agricultural Engineering Department of Federal University of Technology Akure. The evaporative cooler (Fig. 1) was developed for the storage of fruits and vegetables of moderate respiratory rate and has been presented in [10].

Experimental measurements

Experiment was conducted with small prototype evaporative cooling system (Fig. 1) for preservation of fruits and vegetables. The porous cooling pad was rigidly parked inside the pad holder at a pad thickness of 30 mm and a parking density of 20 – 22 kg·m^{-3}. Dry bulb temperature was measured with thermocouple (Omega HHI147) for the draft or ambient air (T_1), the cold chamber of the cooler (T_2), the supply water temperature (T_3) and the sump water temperature (T_4). Also the wet bulb temperature of the cold chamber was also recorded. The air flow rate was measured at five locations 3cm from each point measured longitudinally between the pad media and the axial fan with vane microprocessor (AM-4826) digital anemometer (±0.1 m·s^{-1}).

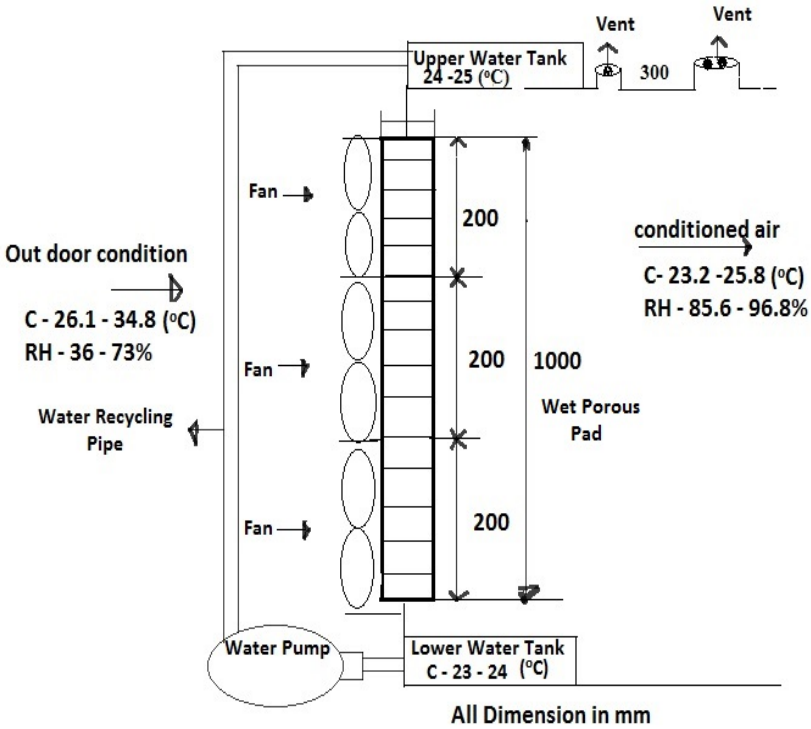

Figure 1. Schematic view of direct evaporative cooling system

The data was logged every two hours to record the temperatures. The cooling effectiveness (ε) was calculated from the equation presented by [11]. The relative humidity of draft air (Rh_1) was measured with humidity clock while that off the cold chamber (Rh_2) was calculated with psychometric calculator. The experiment was carried out from the month of January to June when the relative humidity is relatively low and evaporative cooling possible in southern part of Nigeria. Though by June, rain is becoming frequent and evaporative cooling less efficient. However for the sake of this paper and for easy assimilation, typical three consecutive day's data for January were presented and used in the model evaluation.

The evaluation parameters are as follows:

- Pad type: Oil palm fruit fiber
- Water holding capacity: 2.05 kg of water per kg of solid
- Bulk density: 47.8 kg·m^{-3}
- Air speed: 4 m·s^{-1}
- Air mass flow rate: 0.6 m^{3}·s^{-1}
- Ambient air temperature range: 26.1- 34.8°C
- Cold air temperature range: 23.2 - 25.8°C
- Ambient air relative humidity: 36.0- 73.0%
- Cold air relative humidity: 85.6 - 96.8%
- Evaporative effectiveness: 70.0 - 98.8%

Heat transfer coefficient

A total of three heat transfer equation was used to model the heat transfer coefficient of the system. Statistical parameter such as R^2 was used to assess the goodness of fitting.

Process modeling (A)

The energy balance applied to a small section of the pad media as a heat transfer surface can be written as:

$$dq = hA(T_w - T)dt = C_p \rho V dT \tag{1}$$

Where:

q	[W·m^{-2}]	- heat flux,
h	[W·m^{-2}K^{-1}]	- convective heat transfer coefficient of the outside air,
T_w	[°C]	- wet bulb temperature,
T	[°C]	- dry bulb temperature,
A	[m^2]	- surface area of the pad,
T	[s]	- time,
C_p	[J·kg^{-1}K^{-1}]	- specific heat of air,
V	[m^3]	- volume of the pad,
ρ	[kg·m^{-3}]	- density of air.

Integrating between the limits $T = T_0$ at $t = 0$ and $T = T$ when $t = t$ [6]:

$$q = \int_{T_0}^{T} \frac{dT}{(T_w - T)} = \frac{hA}{C_p \rho V} \int_0^t dt \tag{2}$$

Let the time taken for the air to flow past the pad media volume $t = \frac{V}{U}$, where V is the volume of the pad, u is the air flow rate, $A = H \times b, V = H \times b \times L$ and $u = Av$. L is the thickness of the pad media, b is the breath.

$$In\ \frac{(T_2 - T_w)}{T_1 - T_w} = -\frac{hHb}{C_p \rho HbLu}\int_0^V dV \tag{3}$$

Where:
T_2 [°C] - the temperature of cold air in the cooling space,
T_1 [°C] - the ambient air temperature.

$$In\ \frac{(T_2 - T_w)}{T_1 - T_w} = -\frac{hA}{C_p \rho u} \tag{4}$$

But $\frac{(T_2 - T_w)}{T_1 - T_w}$ is the evaporative effectiveness or efficiency (ε). Therefore:

$$In\varepsilon = -\frac{hA}{C_p \rho u} \tag{5}$$

and:

$$h = \frac{-C_p \rho u\ In\varepsilon}{A} \tag{6}$$

The average heat transfer coefficient can be calculated from the Eq. 6 considering the specific heat, and thermal conductivity of air is constant.

***Process modeling* (B)**

Also the heat transfer coefficient was fitted into the equation presented by Dowdy and Karabash (1987) [5]. Due to close packing of the pads, the pad was assumed to be rigid, therefore the heat transfer coefficient was calculated as:

$$N_u = \frac{h}{l} \tag{7}$$

Where:
Nu [-] - Nusselt number,
l [m^3] - characteristic length.

$$l = \frac{e}{A} \tag{8}$$

Where:
e [m^3] - the volume occupied by the pad.

$$N_u = 0.1\left(\frac{l}{x}\right)^{0.12} R_e^{0.8} Pr^{\frac{1}{3}} \tag{9}$$

Where:
X [m] - the thickness of the pad,
R_e [-] - Reynolds number.

$$R_e = \frac{v\,l}{\nu} \tag{10}$$

Where:
v [m·s^{-1}] - air speed,
ν [m^2·s^{-1}] - kinematic viscosity.

$$Pr = \frac{\nu}{\alpha} \tag{11}$$

Where:
Pr [-] - Prandtl number,
α [m^2·s^{-1}] - the thermal diffusivity which is given by:

$$\alpha = \frac{k}{\rho c_p} \tag{12}$$

The air mass flow rate was generated from the continuity equation as follows:

$$m_a = \rho A_1 v \tag{13}$$

Where A_1 [m] is the area of the pad covered by each of the three fans since the pad is divided into three compartments, each mounted with an axial fan of the same capacity controlled through a single gang switch of the same rheostat. The calculated value of some of the parameters used in the models is presented in Tab. 1 – 4.

Process modeling **(C)**

The heat transfer coefficient can also be calculated from the net change in the heat content of the air inside the cooler. The movement of the draft air through the wet pad results in the loss of heat. But the air inside the cooler usually picks up heat and moisture due to respiratory activities, there is a heat change also due to the incoming air at temperature T_2, replacing the air already inside the cooler at temperature T_b,. [12] stated that in this case change in enthalpy are proportional to temperature changes and considering the difference in temperature of the order involved, changes in their density and vapor pressure of the incoming and outgoing air are small and therefore negligible.

Therefore the energy equation can be written as:

$$H_i = uT_2\rho C_p \tag{14}$$

$$H_l = uT_b\rho C_p \tag{15}$$

Where u [m/s] is the air flow rate, H_i [J] is the heat input by the air coming into the cooler and H_l [J] is the heat loss by the air already inside the cooler. The air density (ρ) is taken to be constant.

Assuming no evaporation inside the cooling system and considering the low air flow rate of 0.6 m^3·s^{-1}, $T_b > T_2$ therefore the net heat gain by the incoming air is given by:

$$H_l - H_i = uT_b\rho C_p - uT_2\rho C_p \tag{16}$$

Because the other part of the evaporative cooling system is exposed to natural air motion, heat transfer through the walls and roof (h_{wr}) with area A_c, excluding the wet porous pad can be calculated from the general heat transfer equation as a function of wind speed [12] with an exchange coefficient h [W·m^{-2}K^{-1}] as follows:

$$h_{wr} = h(T_1 - T_b)A_c \tag{17}$$

The total energy balance equation for the evaporative cooling system can be written as:

$$dq = \rho c_p V\frac{dT}{dt} = h(T_1 - T_b)A_c + \mathrm{u}\rho C_p(T_b - T_2) \tag{18}$$

Table 1. Values of constants used in the models

Symbol	Value	Unit	Equation
C_p	1005.0000	$J{\cdot}kg^{-1}K^{-1}$	6,12 and 20
L	0.0300	M	3
H	1.0000	M	3
	0.6000	$m^3{\cdot}s^{-1}$	3
p_r	0.7130	-	11
A	0.3000	m^2	6
A_c	1.7400	m^2	20
k	0.0260	$W{\cdot}m^{-1}K^{-1}$	12

Table 2. Calculated properties of the air at day one

Time (h)	Re	Nu
9	3115	557
11	3096	557
13	3096	554
15	3090	554
17	3089	553

Table 3. Calculated properties of cold air at day two

Time (h)	Re	Nu
10	3103	555
11	3109	556
12	3115	557
13	3105	556
14	3078	551
16	3085	552
18	3095	554

Table 4. Calculated properties of cold air at day three

Time (h)	Re	Nu
10	3105	555
12	3109	556
14	3099	554
16	3067	550
18	3097	554

The time interval for the series of temperature measurement is assumed to be very short, the condition can be assumed to be steady state. Under steady state conditions $dT/dt = 0$ [12] and assuming $T_b \sim T_w$, substituting into Eq. 18, h can be calculated as:

$$h = \frac{-u\rho C_p(T_w - T_2)}{A_c(T_1 - T_w)} \tag{19}$$

The parameter $\frac{T_2 - T_w}{T_1 - T_w}$ is the evaporative effectiveness or efficiency, therefore Eq. 19 can be rearranged to give:

$$h = \frac{u\rho C_p(T_2 - T_w)}{A_c(T_1 - T_w)} \tag{20}$$

Relaxation time

In Heat Transfer, it is interesting to know the finite-time process, and a basic question is to know the thermal inertia of the system, i.e. how long the heating or cooling process takes under the tested conditions, usually with the intention to modify it, either to make the system more permeable to heat, more insulating, or more 'capacitive', to retard a periodic cooling/heating wave. For the case where the heat flux is not imposed but a temperature gradient is imposed, an order-of-magnitude analysis of the energy balance, shows that depending on the dominant heat-transfer mode the relaxation time is of the order [13]:

$$\frac{dH}{dt} = q \tag{21}$$

$$\Delta H = mc\,\Delta T \tag{22}$$

For a process dominated by heat transfer by convection like evaporative cooling process where the change in temperature of the ambient air and cold storage space temperature is given as ΔT:

$$q = \mathrm{hA}\,\Delta\mathrm{T} \tag{23}$$

Considering a wet porous pad of characteristic length l which is at the same temperature with the cold air inside the cooler at density ρ and assuming that the relaxation time is proportional to l.

$$\Delta t = \frac{\Delta H}{q} = \frac{mc_p\Delta T}{hA\Delta T} = \frac{\rho c_p\, l^3}{h\, l^2} = \frac{\rho c_p\, l}{h} \tag{24}$$

Note that m is the mass [kg], l is the characteristic length given as $\frac{V}{A}$ and $m = \rho V$.

The above equation can be used to hypothetically guess the relaxation time of the system although this might not be a realistic value but gives the maximum time required since the value is usually predicted at a little above the mid temperature of the hot and cold temperature [13].

Statistical analyses

Minitab 1513 was used to analyze the results of the fitted models. The value of h was fitted using the method of least square. R^2 value and standard deviation was used to assess the goodness of fit.

RESULTS AND DISCUSSION

Direct measurement from the experiment conducted yielded the ambient air temperature, the process air temperature, wet bulb temperature and the air flow rate used to obtain the heat transfer coefficients, temperature and relative humidity profile of the wet porous pad at each experimental condition (Fig. 2 - 7). The parameters used in the equation and the constant values were presented in Tab. 1 - 4. The dimensions of the wet porous pad used for the analysis were $A = 0.30\ m^2$, $A_c = 1.74\ m^2$ and $V = 0.009\ m^3$. The value of air flow rate tested is $0.6\ m^3 \cdot s^{-1}$ and for the wet porous pad of the dimension specified this is equivalent to ventilation rate of $0.0056\ m^3 \cdot m^{-2} s^{-1}$ and to approximately 8 air changes per hour. According to [12], higher ventilation rate improves the efficacy of cooling (lower temperature) but the improvement is non linear but has to be balanced with high cost of power and fan associated with increased fan capacity and also the high relative humidity needed for the storage space. Therefore the research evaporative cooling system was designed for low ventilation rate because, the targeted users of the system are rural dwellers, who are very poor and cannot afford the high cost of increased fan capacity. Also the temperature reduction and relative humidity achieved can maintain the quality of their produce in a short period at which they can sale their produce with minimum loss.

Simulation commenced at 7.00 hrs local time and proceeded in time steps of 2 hrs to 18.00 hrs local time corresponding to a day length of 11 hrs. In modeling with Eq. 6 and 7 the heat transfer is considered as depending partly on the characteristics of the wet porous pad and the flow of air into the system is only through the wet porous pad. However in Eq. 20 the whole enclosure is treated as porous material surrounded by air driven by the wind since the system is also exposed to the wind. The heat transfer is both through the wet porous pad and the walls of the enclosure. Representative curves showing the variation of heat transfer coefficient, derived from simulation runs for the three equations are presented in Fig. 2, 3 and 4. The variation of temperatures and relative humidity through the day is also presented in Fig. 5, 6 and 7. For day one the maximum ambient air temperature of 34°C occurred at 15.00 hrs local time, for day two, it was 34.8°C at 16.00 hrs and 34°C for day three at also 16.00 hrs. The results presented here showed that the single stage evaporative cooler can reduce the ambient air temperature by 13°C under high ambient temperature which usually results in high radiation intensity. The average heat transfer coefficients were calculated by fitting the evaluation parameters into Eq. 6, 7 and 20. The results obtained are higher for higher evaporative effectiveness for Eq. 20 and 7 but decreased for Eq. 6 at higher evaporative effectiveness for the three days period. It is observed from the equations that if the evaporative effectiveness is constant, the heat transfer coefficient will be constant since it directly or indirectly controls the parameters in the equations. Large standard deviations (130.94 – 282.85) obtained for the three days period for Eq. 6 compared to Eq. 20 (17.53- 26.3) and Eq. 7 (1.87 -1.99) might be because of small area for heat transfer and steep temperature drop in the afternoon compared to the morning period. Eq. 7 has been widely used by some authors [14, 4 and 15] to predict heat transfer coefficient in evaporative coolers and the results has been compared with experimental values with good results. Therefore Eq. 7 was used to test the level of accuracy of Eq. 6 and 20. The values obtained in Eq. 7 are very close to Eq. 20 with a maximum value of

483 and 396 respectively; therefore Eq. 20 can provide an alternative route for quick calculation of heat transfer coefficient in evaporative cooling because it contains less number of parameters to calculate than Eq. 20.

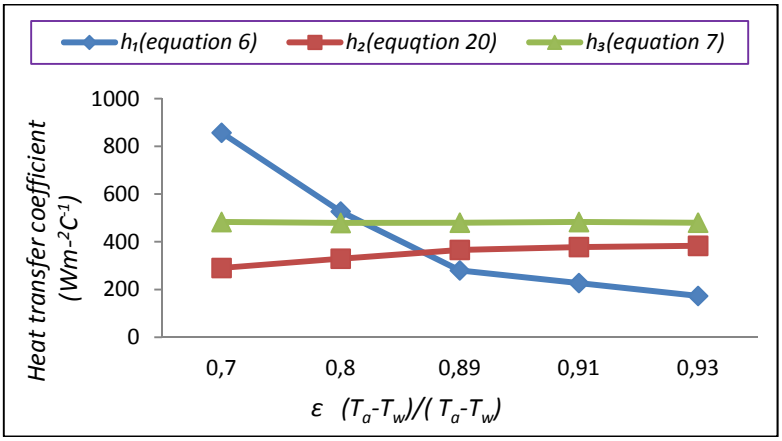

Figure 2. Heat transfer coefficient curve for day one

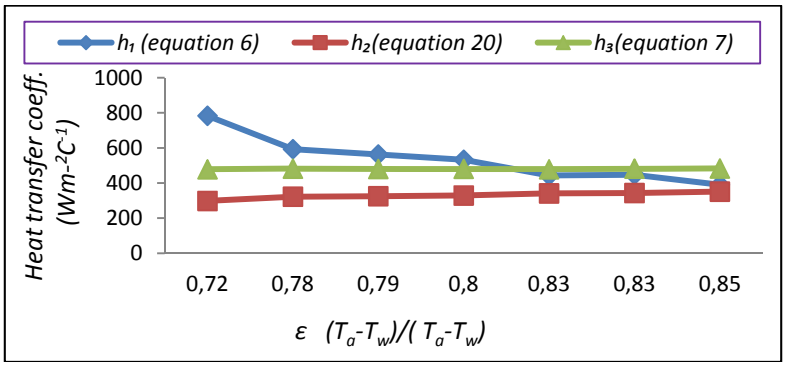

Figure 3. Heat transfer coefficient curve for day two

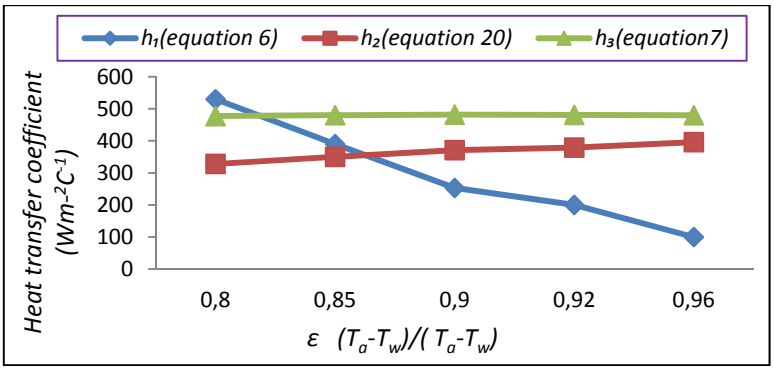

Figure 4. Heat transfer coefficient curve for day three

Result of relationship of h with ambient air temperature (T_1) and R^2 value derived by the method of least square is presented in Tab. 5. The various relationship for the three days showed that, h_1 has an exponential relationship with ε; h_2 has a linear relationship with ε while h_3 has a polynomial relationship with ε. Generally the R^2 value for various equations is above 90%.

Table 5. Relationship between heat transfer coefficient and evaporative effectiveness [$\varepsilon = (T_c - T_w)/(T_1 - T_w)$]

Day	h	Model	R^2
1	h_1	$h_1 = 1163e^{-40\varepsilon}$	0.959
	h_2	$h_2 = 23.5\varepsilon + 278.7$	0.895
	h_3	$h_3 = -0.916\varepsilon^3 + 8.54\varepsilon^2 - 23.54\varepsilon + 499$	0.974
2	h_1	$h_1 = 791.5e^{-0.10\varepsilon}$	0.928
	h_2	$h_2 = 7.75\varepsilon + 298.8$	0.911
	h_3	$h_3 = -0.02\varepsilon^5 + 0.337\varepsilon^4 - 1.623\varepsilon^3 + 1.291\varepsilon^2 + 6.522\varepsilon + 4715$	0.756
3	h_1	$h_1 = 842.3e^{-0.4\varepsilon}$	0.970
	h_2	$h_2 = 16.5\varepsilon + 315.3$	0.976
	h_3	$h_3 = 0.785\varepsilon^2 + 5.44\varepsilon + 472.4$	0.967

The relaxation time based on the various heat transfer coefficients calculated from Eq. 6, 7 and 20 is presented in Tab. 6, 7 and 8 for the period of three days. The average value for the three equations is approximately one second which shows consistency although Eq. 7 gave a constant value of 0.75 s. The value predicted from the equations can be regarded as a hypothetical value since other heat transfer methods like conduction or even radiation loss can contribute to the cooling of the air. Therefore the value calculated might be much higher than the real relaxation time. This value can serve as a basis in future design or modification of the system and its kind.

Table 6. Relaxation time for the process ambient air for day one at various heat transfer coefficient

Time (h)	Eq. 6	Eq. 7	Eq. 20
	t_1 (s)	t_2 (s)	t_2 (s)
9	0.42	0.75	1.20
11	1.58	0.74	0.95
13	1.07	0.75	0.94
15	1.28	0.75	0.98
17	0.68	0.75	1.00
Average value	1.00	0.75	1.01

The temperature and relative humidity profile is presented in Fig. 5, 6 and 7 for the period of three days. Considering the flowing of humid air to the wet porous pad, the heat transfer will occur if the surface temperature of the pad is different from the draft ambient air temperature. Also mass transfer will occur if the absolute humidity of the air close to the pad is different from the humidity of the draft ambient air [15].

Table 7. Relaxation time for the process ambient air for day two at various heat transfer coefficient

Time (h)	Eq. 6	Eq. 7	Eq. 20
	t_1 (s)	t_2 (s)	t_2 (s)
10	0.67	0.75	1.09
12	0.92	0.75	1.02
14	0.46	0.75	1.20
16	0.81	0.75	1.05
18	0.64	0.75	1.10
Average value	0.71	0.75	1.09

Table 8. Relaxation time per hour for the process ambient air for day three at various heat transfer coefficient

Time (h)	Eq. 6	Eq. 7	Eq. 20
	t_1 (s)	t_2 (s)	t_2 (s)
10	1.79	0.75	0.95
12	1.42	0.75	0.97
14	3.58	0.75	0.91
16	0.67	0.75	1.09
18	0.92	0.75	1.02
Average value	1.68	0.75	0.99

The temperature and humidity profile of the storage space showed a decrease and increase of the ambient air temperature and relative humidity respectively. It is clear from Fig. 5 that at 13:00 hours, the ambient air of 32.8°C with 36 % relative humidity could be brought to 23.2°C and 90.4 % relative humidity at the first day. It shows that the system can drop the ambient air temperature very close to its wet bulb temperature of 21.96°C. The maximum temperature reduction observed was 13°C. The relative humidity of the cooler was observed around 85.6 - 96.8 % throughout the experiment, which shows the maximum possible level of saturation of air by humidification.

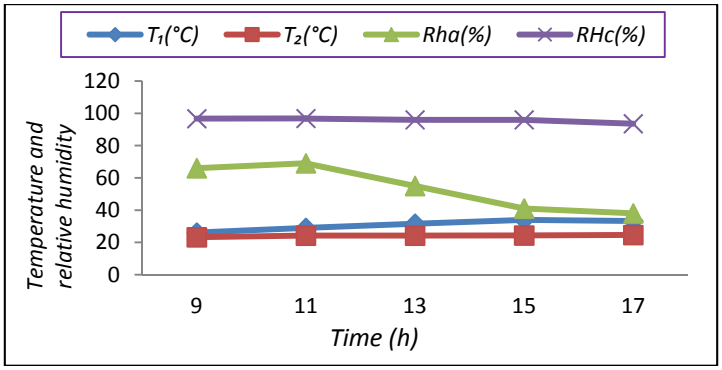

Figure 5. Temperature and relative humidity profile for day one

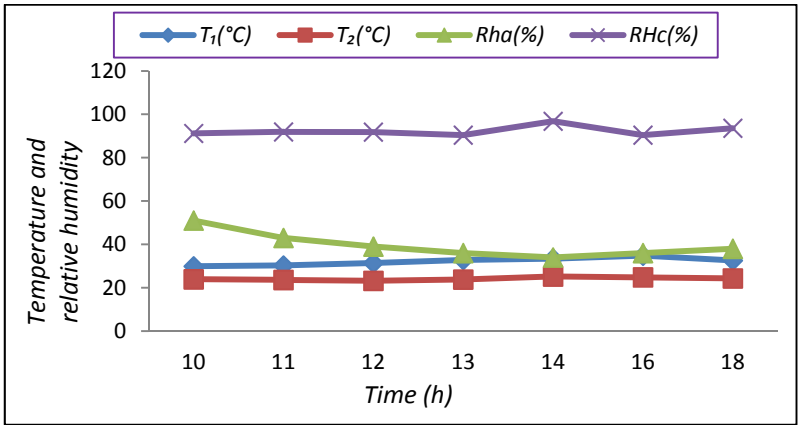

Figure 6. Temperature and relative humidity profile for day two

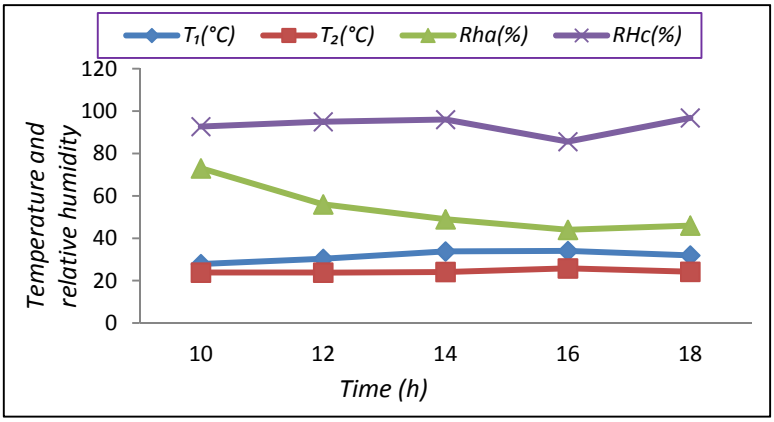

Figure 7. Temperature and relative humidity profile for day three

CONCLUSION

The equations for modeling heat and mass transfer can be used to predict process fluid behavior with temperature under different ambient conditions for direct evaporative cooling systems. This information is a valuable tool for designing and also modifying already existing evaporative cooling systems. Models were obtained for heat transfer and relaxation time and can be applied to different situation in heat and mass transfer process. Generally the temperature and relative humidity profile of the system showed that greater cooling is achieved in the afternoon. This is desirable since the afternoon presents higher heat load therefore requires more cooling to maintain the cooler condition at the desired temperature and humidity state. The maximum temperature reduction during the field evaluation was 13°C with the relative humidity of the cooler ranging from 85.6 – 96.8 % throughout the experiment, which shows the maximum possible level of saturation of air by humidification.

BIBLIOGRAPHY

[1] Xuan, Y.M.F., Xiao, X.F., Niu, X., Huang, S.W. 2012. Research and application of evaporative cooling in China: A review (I). *Renewable and Sustainable Energy Reviews*Volume 16, Issue 5, June , pp 3535 – 3546

[2] Gupta, C.P.A., Abbas, M.S. 1995. Thermal comfort inside a tractor cab by evaporative cooling system. *Transaction of ASAE* 38(6): 1667-1675.

[3] Liao, C., Chiu, K. 2002. Wind Tunnel Modeling the System Performance of Alternative Evaporative Cooling Pads in Taiwan Region. *Building and Environment*, p. 177-187

[4] Camargo, J.R., Ebinuma, C.D., Cardoso, S. 2003. A mathematical model for direct evaporative cooling air conditioning system. *Engenharia Térmica,* no.4,p. 30-34

[5] Dowdy, J.A., Karabash, N.S. 1987. Experimental Determination of Heat and Mass Transfer Coefficients in rigid Impregnated Cellulose Evaporative Media. *ASHRAE Transactions,* part 2, vol. 93. pp. 382-395.

[6] Ditchfield, C. C.C., Tadini, R.K., Singh, R.T. 2006. Velocity and temperature profile, heat transfer coefficient and residence time distribution of temperature dependent Herschel bukly fluid in tubular heat exchanger. *Journal of food engineering* (79), 632 -638

[7] Peterson, J.L. 1993. An effectiveness model for indirect evaporative coolers. *ASHRAE Transactions*; 99(2).

[8] Ren, C.Q., Yang, H.X. 2006. An analytical model for heat and mass transfer processes in indirect evaporative cooling with parallel/counter flow configurations. *Int. J. Heat and Mass Transfer*, 49 .pp. 617–627

[9] Wu, J.M., Huang, X., Zhang, H. 2009. Theoretical analysis on heat and mass transfer in a direct evaporative cooler. *Applied Thermal Engineering*, 29 (5–6), pp. 980–984

[10] Ndukwu M.C, Manuwa, S.I. Olukunle, O.J., Oluwalana, I.B. 2013. Development of an Active Evaporative Cooling System for short-term storage of tropical agricultural product. Agricultural engineering international. *The CIGR Journal* 15(4) 307-313.

[11] Wat, J.R., Brown, W.K. 1997. *Evaporative air conditioning handbook* (3rd ed.)The Fairmont Press, USA (1997)

[12] Landsberg, J.L., White, B., Thorpes, M.R. 1979. Computer analysis of the efficacy of evaporative cooling for glasshouse in high energy environment. *Journal of agricultural engineering research* 24, pp 29-39.

[13] Isidoro, M. 2013. *Heat and Mass Transfer*. Politecnica.

[14] http://webserver.dmt.upm.es/~isidoro/bk3/c12/Heat%20convection.%20Boundary%20layer%20flow.pdf. Downloaded 4th December 2013.

[15] Qureshi, B.A., Zubair, M. 2006. A comprehensive design and rating study of evaporative coolers and condensers. Part I. Performance evaluation: *International Journal of Refrigeration,* 29, pp 645–658

[16] Ghasem, H. 2008. Heat and mass transfer modeling of two stage indirect/direct evaporative air coolers: ASHRAE Thailand Chapter.*ASHARE Journal* 2007 – 2008 .Pp 2-8

KOEFICIJENT PRENOSA TOPLOTE I KONCEPT PERIODA RELAKSACIJE U SISTEMU EVAPORATIVNOG HLAĐENJA PRINUDNO USMERENIM VAZDUHOM

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Sažetak: Sistem evaporativnog hlađenja sa rashladnim uloškom od palminih vlakana je bio ispitivan pri različitim temperaturama i relativnim vlažnostima vazduha. Ogled je izveden sa prototipom direktnog sistema evaporativnog hlađenja za zaštitu voća i povrća od umerene respiracije sa niskim protokom od 0.6 $m^3 \cdot s^{-1}$. Ponuđena su tri različita modela za postizanje koeficijenta prenosa toplote pri različitim efikasnostima evaporacije. Za ova tri modela, koeficijent prenosa toplote varirao je od 173 do 857 $W \cdot m^{-2} K^{-1}$. Period relaksacije bilo je predviđeno kao konačan vremenski proces i vrednost izračunata iz modela može se prihvatiti kao hipotetička, obzirom da su drugi postupci prenosa toplote, kao provođenje ili čak i radijacioni gubici, bili zanemareni. Zato izračunata vrednost može da bude mnogo veća nego stvarni period relaksacije. Za opseg ambijentalnih temperature od 26.1 do 34.8°C, koje su hlađene na 23.2 do 25.8°C, za srednji period relaksacije bile su izračunate vrednosti od 0.71 do 1.68 s.

Ključne reči: *ambientalna temperature, evaporativno hlađenje, prenos toplote, relativna vlažnost*

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