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Studying the performance of evaporative cooler using pre-cooling water system

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Abstract

The big increase of the ambient temperature during the summer season leads to increase of energy consumption for air conditioning which in turn increases peak electrical study load. In this work, theoretical and experimental study are performed for hybrid system combined of evaporative air cooler and refrigeration unit. The experimental rig is build and tested during the summer months. The theoretical model is performed based on heat and mass transfer for the hybrid system. The effect of may parameters are studied such as, water temperature, water and air mass flowrates and wetted pad thickness. The results show that the pre-cooling water temperature improves effectiveness by about (5-10%). The results of the present work are compared with results of other works and show a good agreement.
Keywords: evaporation; cooling; energy balance; mass transfer; air conditioning.

1. Introduction

Evaporative cooling has been in use for many centuries in countries for cooling water and for providing thermal comfort in hot and dry regions. This system is based on the principle that when moist but unsaturated air comes in contact with a wetted surface whose temperature is higher than the dew point temperature of air, some water from the wetted surface evaporates into air. The latent heat of evaporation is taken from water, air or both of them. In this process, the air loses sensible heat but gains latent heat due to the transfer of water vapor. Thus the air gets cooled and humidified. The main characteristic of this process is that it is more efficient when the temperatures are higher, that means, when more cooling is necessary for thermal comfort. It has the additional attractiveness of low energy consumption and easy maintenance. Due to use total airflow renewal, it eliminates the recirculation flow and proliferation of fungi and bacteria, which represent a constant problem in conventional air conditioning systems. Several authors dedicated their researches to the development of direct evaporative cooling systems. **Heidarinejad, et al. [1]** investigated the behavior of the hybrid system of nocturnal radiative cooling, coil, and direct evaporative cooling has been during 8 h in Tehran. The chilled water for the cooling coil unit was provided by the nocturnal radiative cooling at previous night. The results showed that Tehran has the capability of providing cold water at night during summer. In addition, the effectiveness of the hybrid system was more than 100% considerably higher than stand-alone direct evaporative cooling. **Sodha and Somwanshi, [2]** presented a model for the evaluation of the variation of the water temperature along the direction of flow in an evaporating pad. They concluded that the penalty on the mean exit air temperature was negligible for thermal loads $\dot{Q}$ (for cooling) of the order of 1 kW; it was seen that it was 0.6 °C for $\dot{Q}$ = 2 kW. Further it was concluded that for typical coolers the steady state temperature of water in the tank was reached in a time of the order of one hour or less. **Jain and Hindoliya [3]** presented development and testing of a regenerative evaporative cooler. A conventional direct evaporative cooler has been modified by adding a water to air, heat exchanger in the path of outgoing air stream. The heat exchanger cools the air further by using cooled water available in the collecting tank Experiments have been conducted to study the performance of the regenerative evaporative cooler. They concluded that the efficiency and COP of regenerative system increases by 20-25%. The regenerative cooler having higher cooling capacity may be advantageous as it may attract more people for maximum utilization of this low energy consuming device leading to energy conservation in residential and commercial buildings. **Boukhanouf, et al. [4]** presented A computer model and experimental results of a sub-wet bulb temperature evaporative cooler using porous ceramic materials. The system uses porous/fired clay materials as wet media for water evaporation. The supply air and working air flows were staged in separate ducts and in counter flow direction. They concluded that the evaporative cooler can achieve high thermal performance in terms of low air supply temperatures and effectiveness. The structural stability and manufacturing controllability of ceramic materials lend them well to integration into buildings and performing the function of air conditioning in regions with hot and dry climatic conditions. This paper develops a mathematical model for direct evaporative cooling system and presents some experimental tests results in a direct evaporative cooler that take place in the Air Conditioning Laboratory at the University of Basrah Mechanical Engineering Department, located in the city of Basrah. All these analyses correspond to the steady state and neglect the variation of the temperature of water along the direction of air flow. The temperature of water has been assumed to be the wet bulb temperature, throughout the pad. Also, 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 and the effect of rejection of heat from the water (for purpose of cooling) has also been studied.

2. Mathematical Model:

The commonest direct evaporative coolers are essentially cubical metal or plastic boxes with large flat vertical air filters, called ‘‘pads”, in their walls. Consisting of very wet table porous material, they are kept moist by water dripped continuously onto their upper edges. Motorized centrifugal fans within the boxes draw in air through pads, which both cools and humidifies the air. The discharged air from coolers often named as ‘‘washed air” is used for cooling. Many coolers have two or three fan speeds; thus user can modulate outputs as needed. The following assumptions are used to simplify the solution of the proposed model:

1-The pad material is wetted uniformly and fully.
2-The convective heat transfer coefficient and mass transfer coefficient of moisture air on the surface of water film are constant.
3-The thermal properties of air and water are constant.
4-Water–air interface temperature is assumed to be uniform and constant.
5-Lewis number Le=1.
6-The heat transferred from the surroundings is neglected.
7-Air near the water–air interface is saturated and its temperature is assumed to be that of sprinkled water.
8-Air temperature changes only in flow direction denoted by x.
9-Kinetic and potential energy gain is negligible.
10-The process is assumed to be under steady state open system.
11-The sensible heat removed from air is equal to the latent heat gained from water evaporation.

2.1 Analysis of Air temperature:

Analysis of an evaporative cooling process requires the application of heat and mass transfer analogy. The following theoretical analysis will focus on the heat and mass transfer of a direct evaporative cooler. The simplified sketch of a direct evaporative cooler is shown in Fig.1 which shows the configuration of a pad module.

Thus the sensible heat removed from the air stream through the pad material within dx in air flow direction can be written as [9]:

$$dQ_s = -\dot{m}_a.cp_a.dT = h_c.(T_a - T_w).dA \quad (1)$$

$$dA = F_p.y_o.z_o.dx \quad (2)$$

The rate of evaporated water into air within dx may be decided by Fick’s law:

$$d\dot{m}_w = \dot{m}_a.dw = h_m.(w_s - w_a).dA \quad (3)$$

Therefore, the latent heat gained by air through water evaporation is decided as:

$$dQ_l = h_{fg}.d\dot{m}_w = h_{fg}.h_m.(w_s - w_a).dA \quad (4)$$

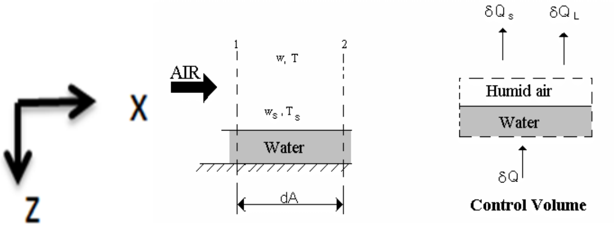


Fig. (1) Schematic direct evaporative cooler [5]

Considering assumption (11), we find that:

$$h_c.(T_a - T_w)dA = h_{fg}.h_m.(w_s - w_a).dA \quad (5)$$

$$Le = \frac{h_c}{h_m.cp_a} \quad (6)$$

Which is based on the assumption of Le = 1,
The inlet boundary conditions for Eq. (1) are:
$T_a=T_{a1}$, $w_a=w_{a1}$ at x=0
By solving Eqs. (3) - (6), the changes of temperature and humidity ratio of moisture air along with x are obtained as follows:

$$T_a = T_s + (T_{a1} - T_w)\exp(-\frac{h_c.F_p.y_o.z_o.x}{\dot{m}_a.cp_a}) \quad (7)$$

$$w_a = w_s - \frac{cp_a}{h_{fg}}(T_{a1} - T_s)\exp(-\frac{h_c.F_p.y_o.z_o.x}{\dot{m}_a.cp_a}) \quad (8)$$

Then effectiveness (saturating efficiency) is usually defined as follows [10]:

$$\varepsilon = \frac{T_{a1} - T_{a2}}{T_{a1} - T_s} * 100\% \quad (9)$$

where T_{a1} is the entering air dry-bulb temperature, T_{a2} is the leaving air dry-bulb temperature, T_s is the entering air wet-bulb temperature. As stated previously, the process of air being cooled and humidified approximates isenthalpic. The water film temperature T_w is approximately equal to T_s. Then the effectiveness can be derived from Eq. (6) based on Eq. (9) where the boundary condition $T_a = T_{a2}$ at $x = x_o$ are used.

$$\varepsilon = 1 - \exp(-\frac{h_c.F_p.x_o}{v_a.\rho_a.cp_a}) \quad (10)$$

For turbulent flow, h_c may be calculated through Nusselt number [6].

$$Nu = \gamma.\mathrm{Re}^{0.8}.\mathrm{Pr}^{1/3} \quad (11)$$

where γ is a constant depending on pad material and pad configuration like Fp.

Furthermore, the best fit for the relationship between h_c and v_a for a kind of pad material named CELdek7090 module (made by Minters Company, Switzerland) is obtained as shown in Eq.

$$h_c = b.v_a^n \quad (12)$$

where b = 42.9, n = 0.65. The two factors of b and n may change with the pad material and pad configuration.

2.2 *Analysis of water temperature:*

Consider a vertical evaporating pad Fig.2, with water flowing from the top to a bottom tray; the air flow is normal to the pad. In common with the variation of the temperature of water along the x axis (direction of flow of air) has been neglected in this work on account of the much larger heat capacity of water as compared to that of air. However, the temperature of water on z (the direction of flow of water the vertical) has been taken into account. Considering an element of pad of thickness dz. Fig.2, the energy balance of water may be expressed as

$$\dot{m}_w.c_w.\frac{dT_w}{dz}.dz+Q_l.dz+Q_s dz=0 \qquad (13)$$

For unit Lewis number the heat transfer per unit area, associated with the mass transfer is given by Tiwari [7]

$$q_l=13*10^{-3}h_c(P_w-\phi.P_a) \qquad (14)$$

It is important to note that the evaporating surface in the volume element (x_0 y_0 dz) is (F_p. x_0 y_0. dz).

$$\begin{aligned}Q_l&=q_l.F_p.x_o.y_o\\&=13*10^{-3}h_c(P_w-\phi.P_a).F_p.x_o.y_o\end{aligned} \qquad (15)$$

Further,

$$Q_s=h_c(T_w-T_a).F_p.x_o.y_o \qquad (16)$$

In the range of temperature of interest, the saturation vapor pressure of water (tabulated by Tiwari [7]) can be represented to a very good approximation by

$$P=R_1.T^2+R_2.T+R_3 \qquad (17)$$

Where

R_1=6.42 N m^{-2} °C^{-2}, R_2=−135.4 N m^{-2} °C^{-1} and R_3=2516 N m^{-2}.

From Eqs. (13) - (17) we obtain:

$$\frac{dT_w}{d\zeta}=A_1.T_w{}^2+B.T_w+C \qquad (18)$$

Where

$$\zeta=[\frac{h_c.F_p.x_o.y_o}{\dot{m}_w.c_w}].z \quad , \quad \alpha=\frac{h_c.F_p}{\rho_a.v_a.cp_a}$$

K_1= (1-exp (-α. x_o)) /(α x_o), A_1=0.083, B= (1.76- K_1),

C=0.083 ϕ T^2_{a1}+T_{a1} (K_1-1.76ϕ)+32.7(ϕ-1),

Integrating Eq. (18) obtains [2]:

$$\frac{T_w(\zeta)-(B/2A_1)-C1}{T_w(\zeta)-(B/2A_1)+C1}=\beta.\exp(-2A_1.C1.\zeta) \qquad (19)$$

Where

$$\beta=\frac{T_{wo}-(B/2A_1)-C1}{T_{wo}-(B/2A_1)+C1},$$

$$C1=\sqrt{(B/2A)^2+C/A_1}$$

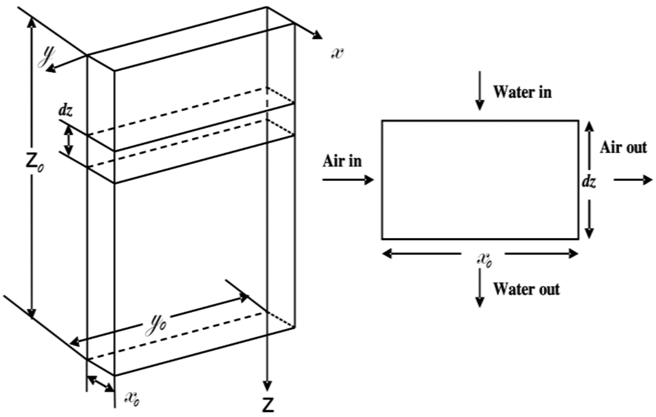


Fig. (2): Pad profile and elemental volume [2]

From Eq. (19); the exit temperature of water flowing through the cooler pad are

$$T_w(\zeta_o)=(\frac{B}{2A_1})+C1\{\frac{1+\beta\exp(-2A_1C1\zeta_o)}{1-\beta\exp(-2A_1C1\zeta_o)}\} \qquad (20)$$

The mean temperature (over z) of water flowing through the cooler pad are

$$\langle T_w(\zeta)\rangle=\frac{B}{2A_1}+(\frac{1}{A_1\zeta_o})\ln\{\frac{\exp(-2A_1C1\zeta_o)-\beta}{1-\beta}\}-C1 \qquad (21)$$

From Eq. (13) and Eq. (21) the average exit temperature of air is given by:

$$\langle T_a(\zeta)\rangle=\langle T_w(\zeta)\rangle+\{T_{a1}-\langle T_w(\zeta)\rangle\}\exp(-\alpha.x_o) \qquad (22)$$

2.3 *Analysis of the basin Temperature*:

If the heat transfer during the flow of water from the basin to the top of the pad is negligible, the temperature of the water in the basin and at the top of the evaporating pad may be taken as the same, T_0. (by circulation of water or otherwise), the energy balance equation for the water in the basin is

$$q_{stg.}=q_{in}-q_{out} \qquad (23)$$

$$M_t.c_w\frac{dT_o}{dt}=\dot{m}_w.c_w.T_w(\zeta_o)-\dot{m}_w.c_w.T_o \qquad (24)$$

From Eq. (20) and Eq. (24)

$$\frac{dT_o}{dt}=-\frac{\dot{m}_w}{M_t}[T_o-\frac{B}{2A_1}-C1\{\frac{1+\beta\exp(-2A_1C1\zeta_o)}{1-\beta\exp(-2A_1C1\zeta_o)}\}] \qquad (25)$$

2.4 *The Hybrid Systems*:

In this system, water is pumped up to the upper basin which represents evaporator of a small refrigeration unit Fig.3, water is pre-cooled before dripped into the wetted pad. Water temperature can be estimated as follows. The coefficient of performance is given as:

$$COP = \frac{Q_r}{w_r} \tag{26}$$

From Fig. (4), the energy balance can be written as:

$$Q_r = Q_w \tag{27}$$

While, the heat rejected from water is given by:

$$Q_w = \dot{m}_w . cp_w (T_b - T_{w2}) \tag{28}$$

Then

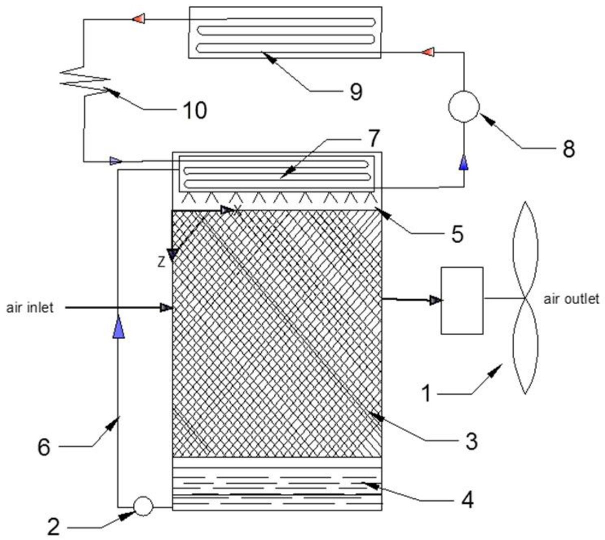
$$Q_r = COP * W \tag{29}$$

Where

$$W = I * V \tag{30}$$

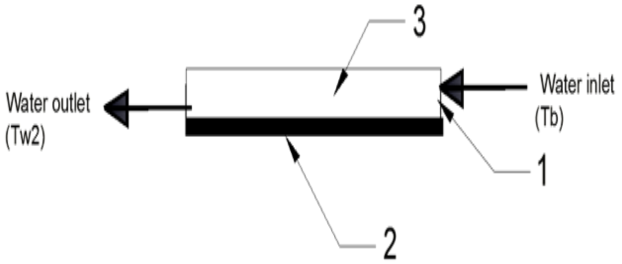
Sub Eqs. (29) and (30) in Eq. (28) yields

$$T_{w2} = T_b - \frac{COP * I * V}{\dot{m}_w * cp_w} \tag{31}$$



1-fan. 2- water pump. 3-pad material. 4-water basin. 5-water distribution system. 6-water supply pipe. 7-Evaporator.8-Compressor.9-Condenser.10-Capillary tube.

Fig. (3) Schematic diagram of the working apparatus.



1- Secondary basin. 2- Evaporator coil. 3- Thin water.

Fig. (4) Schematic diagram of the Secondary basin (evaporator).

3. Experimental Work

Fig.5 is shows the Integrated system of evaporative air-cooler and a small refrigeration unit. The rig is build to study the effect of precooling of water on the air cooler effectiveness. The compressor and condenser of refrigeration unit is located at the top cover of the air cooler box. While the evaporator is fixed inside the box and close to the top cover. In this system, water is pumped up to the upper basin where pre cooled. Before dripped onto wetted pad.

Evaporative air cooler is box of three vertical wetted pad with dimension (0.62×0.49×0.04) m. inlet and outlet temperature, relative humidity and air velocity are measured using digital thermometer and turbometer respectively. The inlet and out water temperature are also measured.

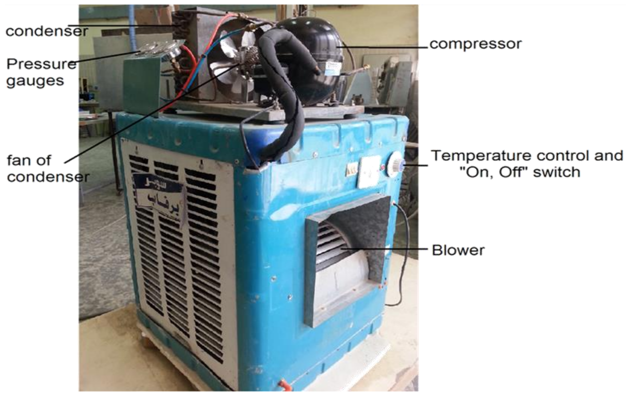


Fig. (5) components of hybrid system of evaporative air cooler and refrigeration unit.

4. Results and Discussion:

Comparison of theory with experiment (validation of model), The steady includes two cases as follows:

Case 1	without pre-cooling water
Case 2	with pre-cooling water

Fig.6 shows a decrease the temperatures of outlet air and the temperature of water at basin with time for the theoretical and experimental studies, the temperature of air and water reach a steady state after about 15 minutes from

the beginning. **Fig.7** illustrates the variation of temperature with time. It shows a decrease in the outlet air temperature with time, in the case 2, it was found that the temperature of the outlet air and water falling down on the pad, reach steady state at short time (about7minutes) experimentally while both air and water temperature required more time to reach steady state theoretically. **Fig.8** shows the variation of effectiveness with time. It can be seen that an increase of effectiveness with time for the theoretical and experimental study in case 1. The effectiveness reach a steady state after about 15 minutes from the beginning. **Fig.9** shows the variation of effectiveness with time for the theoretical and experimental work. In case 2 the effectiveness increased with time. It reach a steady state after about 10 minutes from the beginning. **Fig.10** shows effect of outlet air temperature vs. time for different values of air velocity (1m/s, 1.43m/s and 2 m/s), at case 1 for the theoretically study. It can be seen that the outlet air temperature increased with increasing air velocity because of higher velocity decreased the contact time between the air and the wet pad and increase of cooling load on the air cooler. Also, increasing of the mass flow rate of air. **Fig.11** shows outlet air temperature vs. time for different values of pad thickness (40mm, 100mm and 150mm) in case 1. It can be seen that the outlet air temperature decreased with increasing of pad thickness but the period of reaching the steady state condition takes longer time due to the higher mass of the pad. **Fig.12** shows water temperature vs. height of pad material for different values of air velocity (1m/s, 1.43m/s and 2 m/s) in case 1 for the theoretically results. It can be seen that when the velocity increased the water temperature is decreased with height (z-axis) of pad. **Fig.13** shows outlet air temperature vs. thickness of pad for different values of air velocity (1m/s, 1.43m/s and 2 m/s) in case 1 for the theoretical results. It can be seen that when the pad thickness increases, outlet air temperature is decreased. Its noted that the outlet air temperature reach steady state at thickness of (30cm), and the temperatures are very closed to wet bulb temperature. **Fig.14** shows no significant effect of the inlet air temperature on the effectiveness because the effectiveness depends on the relative humidity and ΔT is approximately constant.

Fig.15 shows the effectiveness increased with increase of the mass flow rate of water, also it is increased with decrease of air velocity (1m/s, 1.43m/s and 2 m/s), in case 1 for the theoretical results. because of increasing the cooling load. the effectiveness reach steady state at mass flow rate of water about (0.05 kg/s).

5. Comparison with other works:

The outlet air temperature and evaporative air cooler effectiveness of the present work are compared with works by Fouda and Melikyan [8], Sodha and Somwanshi [2]. The comparison shows good agreement. As shown in **Fig. (16 and 17)**.

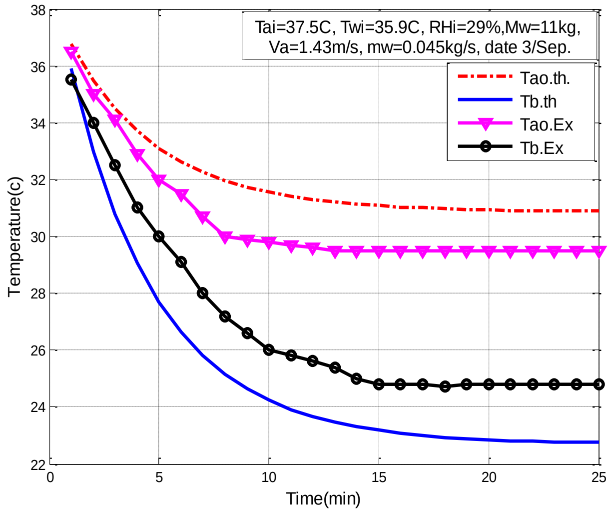


Fig.(6)Time variation of air and water temperature **Case 1**

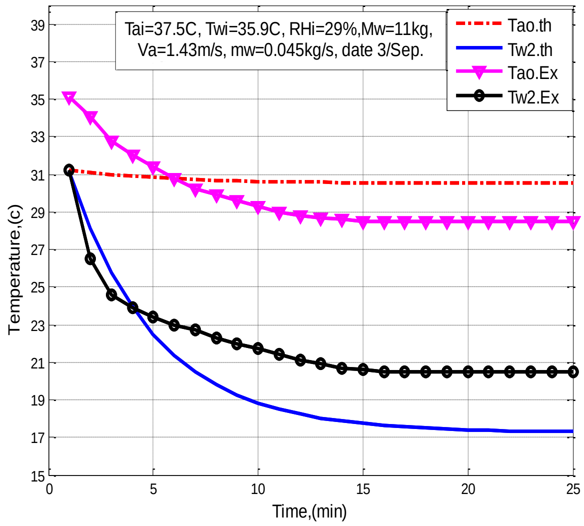


Fig.(7) Time variation of air and water temperature **Case 2**

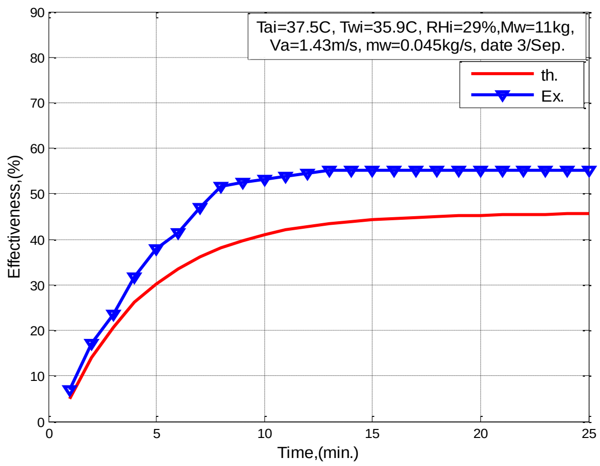


Fig.(8) Time variation of effectiveness **case 1**, theoretical and experimental.

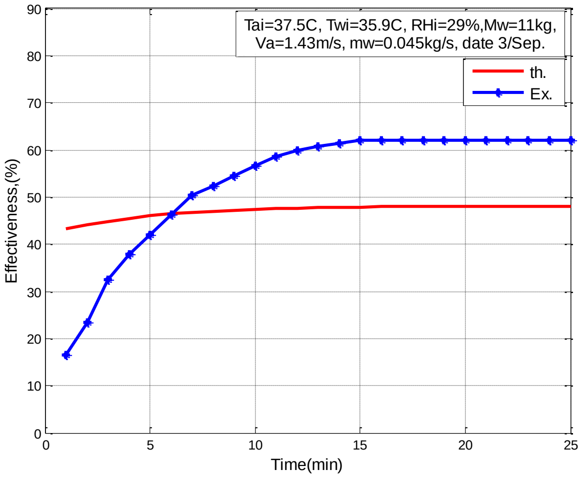


Fig.(9) Time variation of effectiveness **case 2**, theoretical and experimental.

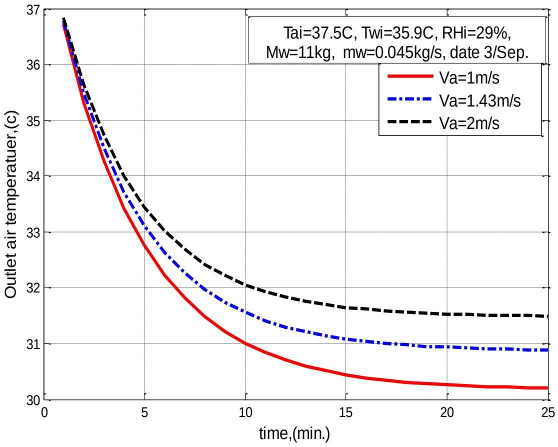


Fig.(10) Time variation of outlet air temperature at different values of frontal velocity, **case 1.**

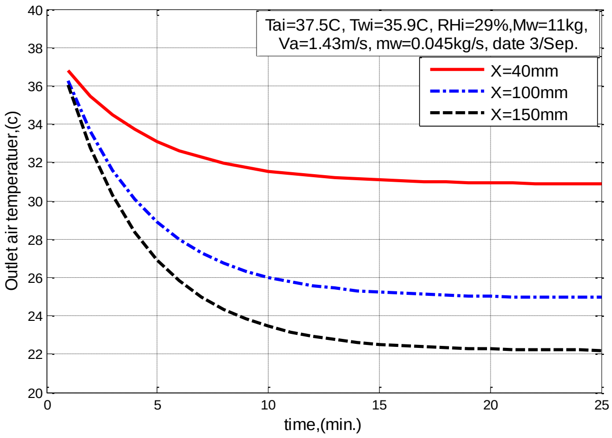


Fig.(11) Time variation of outlet air temperature at different values of pad thickness, **case 1.**

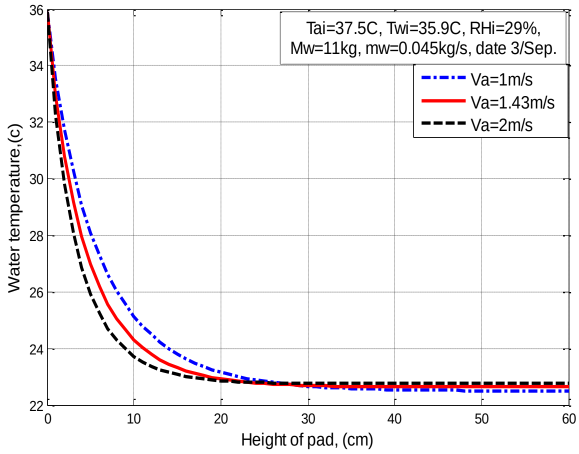


Fig.(12) variation water temperature with height of pad material at different values of frontal velocity, **case 1.**

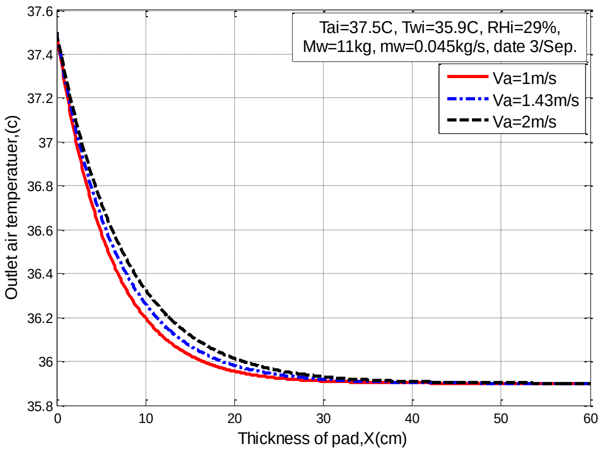


Fig.(13) variation outlet air temperature with thickness of pad material at different values of frontal velocity, **case 1.**

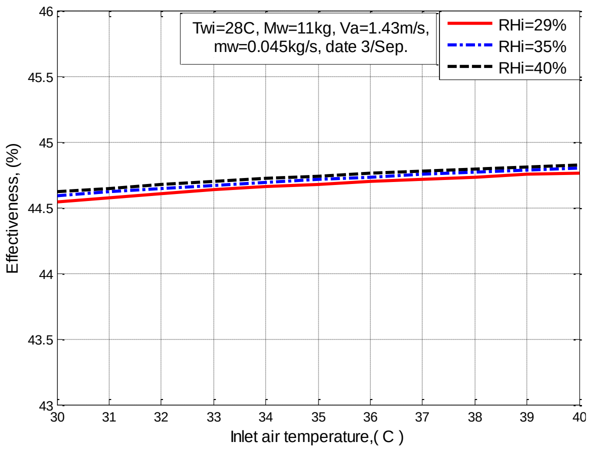


Fig.(14) Influences of inlet air dry-bulb temperature on effectiveness at different values of relative humidity**, case1.**

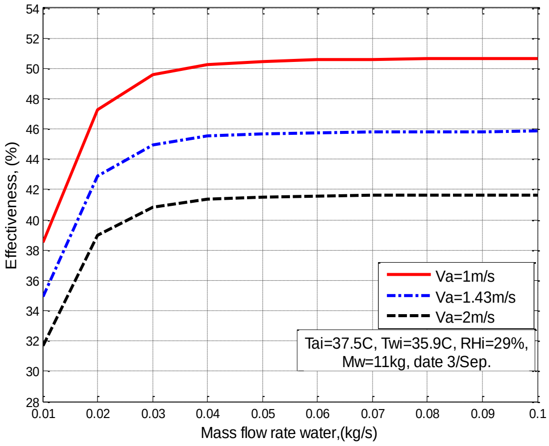


Fig. (15) Effect of mass flow rate of water on effectiveness at different values of frontal velocity, **case 1**.

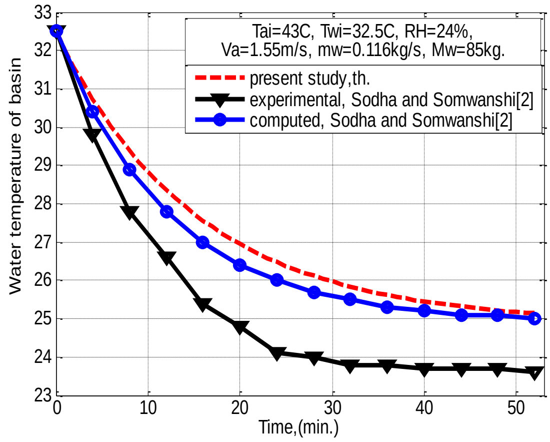


Fig.(16) Comparisons the water temperatures of experiment and computed, Sodha and Somwanshi [2] with the present study.

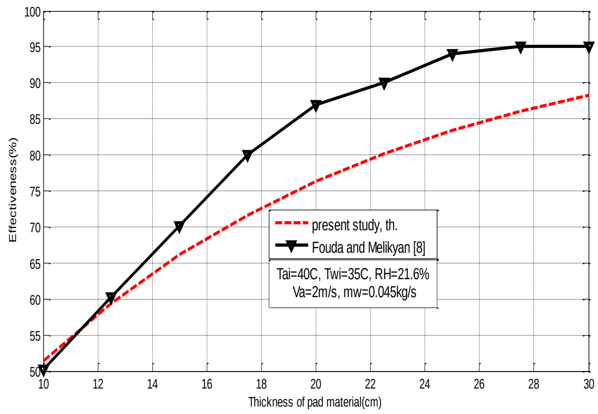


Fig. (17) Comparison of effectiveness, predicted by the present study with Fouda and Melikyan [8].

Conclusions:

The following conclusions are found from this study:

1.Effectiveness decreases with the increase in mass flow rate of air.

2.The best thickness of wetted pad was 30cm.

3.The best mass flow rate of water is (0.05kg/s).

4.When hybrid system is used, the effectiveness is improved about (5-10%) especially in humid environment.

5.When hybrid system is used, the time steady state decrease for the temperature of air.

Nomenclature

c_{pa} Specific heat of air at constant pressure ($J.kg^{-1}.°C^{-1}$).
c_w Specific heat of water ($J.kg^{-1}.°C^{-1}$).
COP Coefficient of performance
F_p Packing fraction of the pad (m^2m^{-3}).
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}).
Q_r Rate of heat rejected from basin water (W).
Q_l latent heat transfer rate (W)
Q_s sensible heat transfer rate (W)
T_a Temperature of the inlet atmospheric air (°C).
h_c Convective heat transfer coefficient between air and water ($Wm^{-2}\,°C^{-1}$).
I Current (A)
h_m Mass transfer coefficient between air and water ($kg.m^{-2}s^{-1}$).
m_w Mass flow rate of water ($kg.s^{-1}$).
M_t Mass of basin water (kg).
T_b temperature of water in basin (°C).
t Time(s).
V Volt (v)
v_a Velocity of air (ms^{-1}).
W Work (W)
x_0 Thickness of pad (m).
y_0 Breadth of pad (m).
z_0 Height of pad (m).
ϕ Relative humidity of air.
ρ_a Density of air (kgm^{-3}).
Ɛ Effectiveness (%).
F_P Packing fraction of the pad(m^2/m^3)
Le Lewis number

Subscripts

1 Inlet
2 Outlet
a Air
b Basin
i Initial
in Inlet
o outlet
r Refrigerant
s Saturated
wb wet bulb
w Water

References

[1] **G. Heidarinejad, M. F. Farahani and S. Delfani,** "Investigation of a hybrid system of nocturnal radiative cooling and direct evaporative cooling", Building and Environment, 45, pp. (1521–1528), 2010.

[2] **M. S. Sodha and A. Somwanshi**, " Variation of Water Temperature along the Direction of Flow: Effect on Performance of an Evaporative Cooler", Journal of Fundamentals of Renewable Energy and Applications, 2, pp.(1-6), 2012.

[3] **J.K.Jain and D.A. Hindoliya,** "Development and Testing of Regenerative Evaporative Cooler **",** International Journal of Engineering Trends and Technology- Volume 3 Issue 6- 2012.
[4] **R. Boukhanouf, A. Alharbi, H. G Ibrahim and M. Kanzari,** "Investigation of A Sub-Wet Bulb Temperature Evaporative Cooler For Buildings", Sustainable Building Conference, 2013.
[5] **J. R. Camargo, C. D. Ebinuma and S. Cardoso,** "A mathematical model for direct evaporative cooling air conditioning system", Engenharia Termica, pp.(30 - 34), 2003.
[6] **F.P. Incropera, D.P. DeWitt,** "Fundamental of Heat and Mass Transfer", fourth ed., John Wiley & Sons, New York, 2002.
[7] **G. N. Tiwari,** "Solar Energy Fundamentals, Design Modelling and Applications", Narosa Publishing House, New Delhi, 2002.
[8] **A. Fouda and Z. Melikyan,**" A simplified model for analysis of heat and mass transfer in a direct evaporative cooler", Applied Thermal Engineering, 31, pp. 932-936, (2011).
[9] **J.M. Wu, X. Huang and H. Zhang,**" Theoretical analysis on heat and mass transfer in a direct evaporative cooler", Applied Thermal Engineering, 29, pp. 980-984, (2009).
[10] **J.M. Wu, X. Huang and H. Zhang,**" Numerical investigation on the heat and mass transfer in a direct evaporative cooler", Applied Thermal Engineering, 29, pp. 195–201, (2009).