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Research Article

Effect of Operating Parameters on the Performance of Direct Evaporative Cooler

Abdul Kareem R. Abed[*]

Middle Technical University, Engineering Technical College, Baghdad -Iraq

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Abstract

Evaporative air cooler is an effective device in cooling private houses in Iraq, and used commonly in this country for many reasons, the mean one is due to the lack in power supply. An evaporative cooler of 5000 cfm capacity (8500 m³/hr.) was studied under extremely hot Iraqi weather. The outdoor temperature was varied from 30 to 60oC, while the outdoor relative humid was varied from 10 to 60%. The results showed that Relative humidity affect both effectiveness and exergy efficiency more than another key variables, the exergy of water circulated in evaporator cooler was spend in destruction the exergy of hot air. The more destruction in exergy through evaporative cooler gives the higher exergy efficiency. The exergy efficiency of the cooler was deteriorated significantly in hot and humid weather.

Keywords: *Evaporative cooler, Effectiveness, exergy destruction, Second law efficiency.*

1. Introduction

Evaporative coolers are commonly used in Iraq due to the lack in power supply, therefore most of private houses in this country contains more than one cooler. Accordingly, the consumption of electrical power and water by his devices are hug. So, many works are required to improve the performance of evaporative cooler. A simple methodology in design house hold desert cooler was introduced by (Kachhwaha, *et al*, 2010):, this method was used to predict the performance of evaporative medium, pad thickness and height for cooler. Modelling and optimization analysis for cooling system in the building were introduced by (Taufiq *et al*, 2007). . Exergy technique has been used in them work. They found that this method is well suited for analyzing thermodynamic model and identified exergy losses of air conditioning application in a building. (Musa, 2008) studied evaporative cooling processes comprising of direct, indirect, combined direct and indirect and also desiccant based types. A theoretical studies covering the heat and mass transfer as well as psychometric associated with evaporative cooling were. The basic principles of the direct evaporative cooling process for human thermal comfort, was presented by (Camargo, *et. al*, 2009) experimental results were introduced also. (Gómez, *et. al*, 2010) introduce the thermodynamic basis for saturation process in evaporative air cooler. The work extended to study legionnaire's disease in commercial evaporative. A heat and mass transfer mathematical model was developed by (Bisoniya, *et. al*, 2011), the model was used to simulate the properties of indirect evaporative cooler. The work was supported by experimental wok, The theoretical and experimental results were compared and analyzed. The theoretical model can be used to predict the performance of modified indirect evaporative cooler.

In this work, an energy and exergy analysis of a direct evaporative air cooler was introduced under Iraq conditions. The performance of evaporative cooler of 5000 cfm capacity (8500 m³/hr) was studied for different inlet dry bulb temperature and different relative humidity of air.

2. Thermal analysis

The heat transfer in evaporative cooler consist of two types, the first was sensible heat that transferred from hot air to cold water by convection, and the second is latent heat transfer due to mass transfer from saturated air to moist air. The total heat transfer in cooler can be written as:

$$Q_t = Q_s + Q_l \tag{1}$$

While the convective heat transfer air depends on the convective heat transfer coefficient and temperature difference between surface and air stream temperature.

$$Q_s = h_c . A . (T_a - T_w) \tag{2}$$

*Corresponding author: **Abdul Kareem R. Abed**

The latent heat transfer, depends on the evaporated water mass and latent heat of evaporation as follow (Kachhwaha, *et al*, 2010):

$$Q_l = m_v . h_{fg} \quad (3)$$

The mass of vapour produced can be calculated as follows:

$$m_v = h_m . A . (\omega_{sat} - \omega_a) \quad (4)$$

Substitute equations (2, 3 and 4) into equation (1) yields

$$Q_t = h_c . A . (T_a - T_w) + h_m . A(\omega_{sat} - \omega_a) . h_{fg} \quad (5)$$

The enthalpy of moist and dry air can be written as function of dry bulb temperature for moist air, While for saturated air it can be written as a function of circulated water bulk temperature. (Baker, *et al*, 1961)

$$h_a = c_{pa} . T_a + \omega_a . \left(h_{fg} - c_{pv} . T_a\right) \quad (6)$$

$$h_{a,sat} = c_{pa} . T_w + \omega_{sat} . \left(h_{fg} - c_{pv} . T_w\right) \quad (7)$$

The solution of equations (5, 6 and 7) are as follows(Stoecker, *et al* 1958):

$$T_a - T_w = \frac{(h_{sat} - h_a) - h_{fg} . (\omega_s - \omega)}{c_{pm}} \quad (8)$$

Where c_{pm} is specific heat of the air-water vapor mixture (kJ/kg K), and can be written as:

$$c_{pm} = \left(c_{pa} + \omega . c_{pv}\right) \quad (9)$$

Substitute equations (8) and (9) into equation. (5) and neglecting the term $(\omega_s - \omega)$ yields the final equation of heat transfer in evaporative cooler (Baker, *et al*, 1961):

$$Q_t = \frac{h_c . A}{c_{pm}} . (h_{sat} - h_a) \quad (10)$$

The effectiveness of the evaporative cooler is found from the following equation:

$$\epsilon = \frac{T_{a,i} - T_{a,e}}{T_{a,i} - WBT} \quad (11)$$

The total mass of evaporated water is calculated as follows:

$$\dot{m}_w = \dot{m}_a . (\omega_e - \omega_i) \quad (12)$$

3. Exergy analysis

Figure 1 shows the evaporative air cooler (EAC) as a control volume, it can be seen from the figure that the exergies entering EAC are the moist air exergy and makeup water exergy, while the exergies leaving control volume are the saturated air exergy as well as the exergy associated with water vapour.

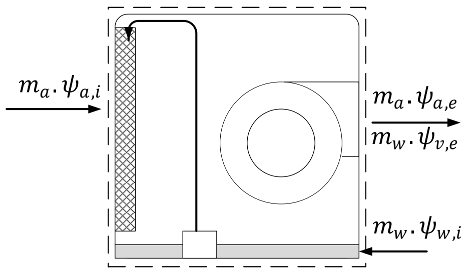

Fig. 1 Exergy flow through evaporative cooler

Tus the exergy destruction in EAC can be written depending on the exergy balance equation, as follows (Claus, *et al* 2013):

$$\frac{d\phi}{dt} = Q . \left(1 - \frac{T}{T_o}\right) - W_{C.V} + P_o \frac{dV}{dt} + \sum \dot{m}_i . \psi_i - \sum \dot{m}_e . \psi_e - \psi_{dest.} \quad (13)$$

Since study state condition and adiabatic saturation are assumed through EAC, then, the exergy destruction through EAC can be written as:

$$\psi_{dest.} = \dot{m}_a . \psi_{a,i} + \dot{m}_w . \psi_{w,i} - \dot{m}_a . \psi_{a,e} - \dot{m}_w . \psi_{v,e} \quad (14)$$

The exergy efficiency for EAC is related to exergy enter the C.V and the exergy destruction as follows (Taufiq *et al*, 2007)

$$\eta_{exe.} = 1 - \frac{\psi_{dest.}}{\dot{m}_a . \psi_{a,i} + \dot{m}_w . \psi_{w,i}} \quad (15)$$

The exergy of water is:

$$\psi_w = (h_w - h_o) - T_o(s_w - s_o) \quad (16)$$

The exergy of water vapour leaving EAC is (Marletta, *et al*, 2010):

$$\psi_v = c_w . (T_w - T_o) - T_o . c_w . \ln\left(\frac{T_w}{T_o}\right) - T_o . c_w \ln(\phi_o) \quad (17)$$

While the exergy of air consist of three terms, namely, chemical, thermal and mechanical exergy, the last term is neglected usually, since this term depends on pressure loss through EAC (Kloppers, *et al*,2005).

The chemical exergy of air can be written as:

$$\psi_{Chem.} = R_a . T_o \left[(1 + 1.68 . \omega) . \ln\frac{1 + 1.68 . \omega_o}{1 + 1.68 . \omega} + 1.68 . \omega . \ln\frac{\omega}{\omega_o}\right] \quad (18)$$

While, the thermal exergy is:

$$\psi_{the.} = c_{pm} . T_o . \left(\frac{T_a}{T_o} - 1 - \ln\frac{T_a}{T_o}\right) \quad (19)$$

Then, the exergy of air is the summation of chemical and thermal exergies as follows:

$$\psi_a = \psi_{\text{the.}} + \psi_{\text{Chem.}} \quad (20)$$

4. Results and discussion

The performance of EAC was studied under arid and normal climates, the range of outdoor dry bulb temperature was in the range of 35 to 50°C in step of 2.5°C, while the relative humidity was varied from 10 to 60% in step of 10%.

Figure 2 shows the effect of entering air DBT on the leaving air DBT from EAC for different values of outdoor relative humidity (RH), it can be seen from the figure that at low relative humidity the cooler gives low DBT of leaving air, this because that, dry air can contain more water vapour than moist or saturated air. As the relative humidity reaches 60% the difference between inlet and outlet DBT reduces significantly.

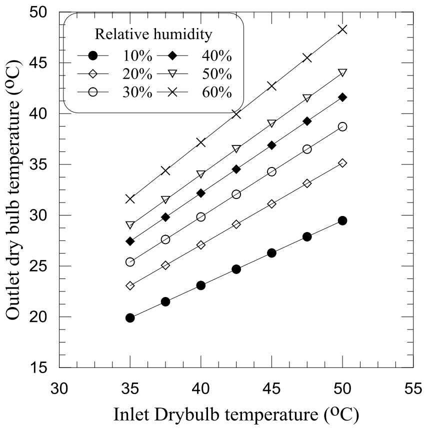

Fig. 2 Outlet DBT vs inlet DBT at different relative humidity

This effect reflects on the WBT as shown in Figure 3, it can be seen from the figure that at 60% RH and 60°C DBT the WBT of leaving air approach entering DBT thus entering air approaches saturation conditions.

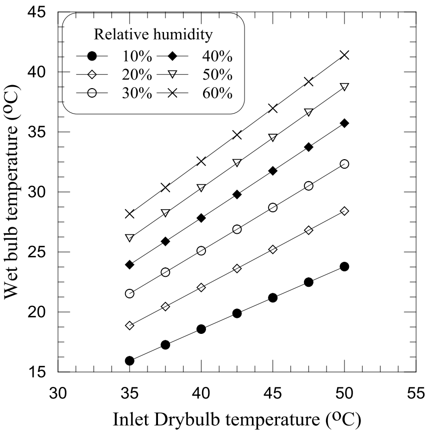

Fig. 3 Outlet DBT vs WBT at different relative humidity

The effectiveness of such case is very low as shown in Figure 4, it can be seen from the figure that EAC operate at low RH gives higher effectiveness for different inlet DBT of air, while, when the RH reaches 60% the effectiveness of the cooler reduces significantly for all values of RH.

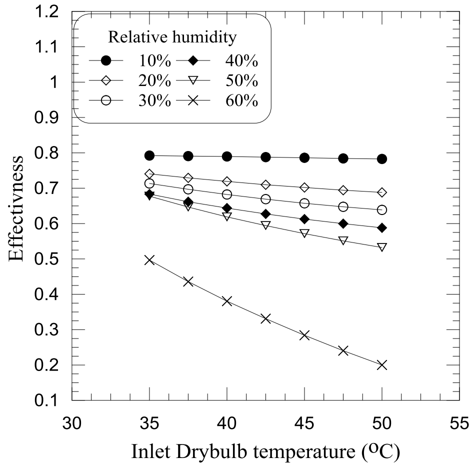

Fig. 4 Evaporative cooler effectiveness vs inlet DBT for different relative humidity

Therefore the use of evaporative cooler under such conditions is limited. The EAC consumption of water related with cooler effectiveness as shown in Figure 5, it can be seen from the figure as the DBT close to WBT the ability of air to contain water vapour reduces, hence less water is evaporated. Thus, evaporative cooler works at high ambient relative humidity consumed less water even DBT is low. while, at low relative humidity the consumption of water increases rapidly.

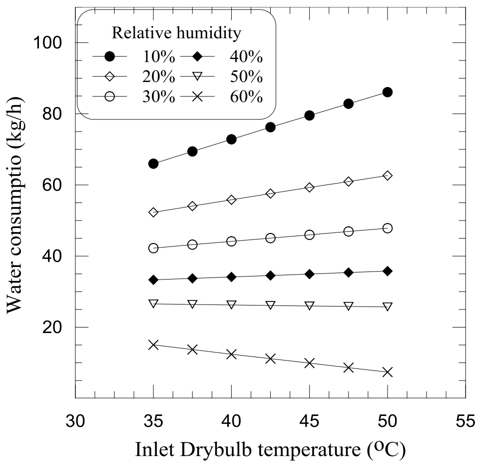

Fig. 5 Water consumption vs inlet air DBT

Also, the figures shows that the consumption of water increases with increasing of DBT , due to high sensible heat transfer between air and water. Figure 6 shows the variation of inlet and outlet moisture when the outdoor relative humidity of 10 and 60%, it can be seen from the that, the moisture content increases with increasing of both outdoor DBT and relative humidity.

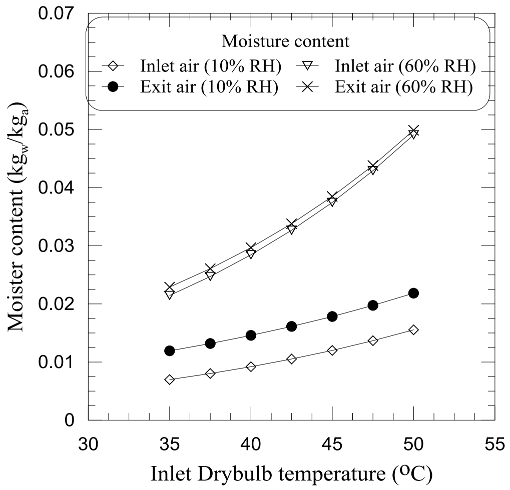

Fig. 6 Moisture content vs inlet DBT

The difference between inlet and outlet air moisture content shows a significant value when the outdoor RH is 10%, while, as the relative humidity increases to 60 the air equipped less amount of water vapour, which is reflect negatively on the evaporative cooler effectiveness, as mentioned previously. As the entering DBT increases the destruction in exergy through cooler increases also, as shown in Figure 7.

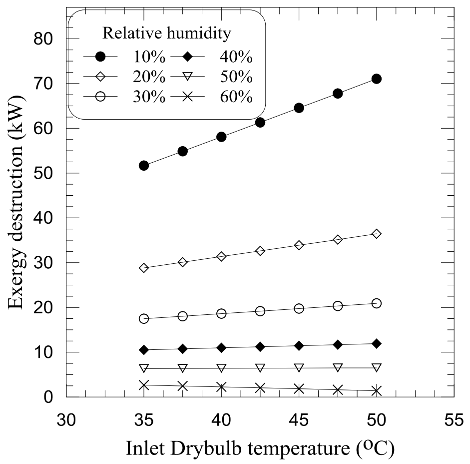

Fig. 7 Effect of inlet DBT on the exergy destruction

This is due to the increasing in entropy generation that caused by formulation of water vapour. This phenomena, gives a clear impression that the exergy of water is consumed in destroyed the exergy of air. Thus, a more exergy destruction means a lower exit DBT. Therefore, the exergy efficiency of evaporative cooler increases with the increasing of exergy destruction, as shown in Figure 8. The effect of relative humidity on the outlet leaving air DBT and WBT, as well as, water consumption is shown in Figure 9, when the inlet DBT equals 40°C. It can be seen from the figure that increases in relative humidity tend to increase both DBT and WBT of exit air and reduces water consumption. The increases of outlet DBT means less effectiveness and less exergy efficiency as shown in Figure 10. Also, the figure shows a significant deterioration in exergy efficiency when the inlet air conditions of 60°C DBT and 60% RH.

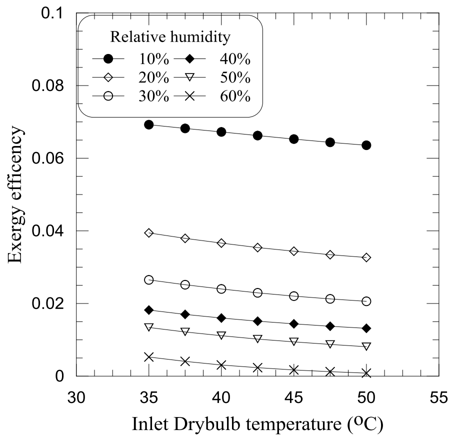

Fig. 8 Exergy efficiency vs inlet DBT

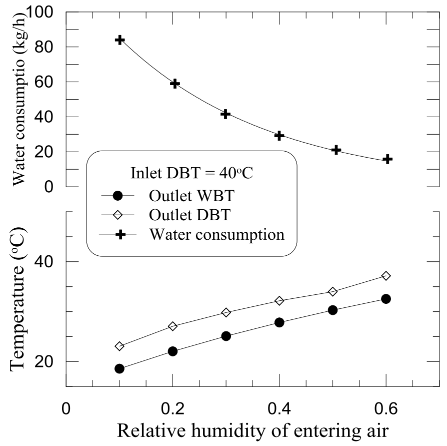

Fig. 9 Effect of RH on the outlet DBT and WBT of leaving air, as well as water consumption

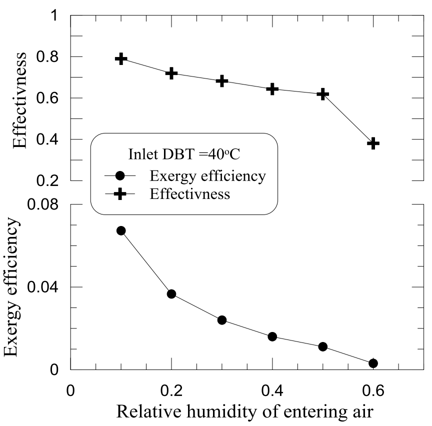

Fig. 10 Effect of RH on the effectiveness and exergy efficiency of the cooler

Conclusions

From above it can be conclude that:'

1- Relative humidity was the dominated factor that affecting cooler effectiveness.

2- The exergy of water is consumed in destruction the exergy of air through the evaporative cooler.
3- As the exergy destruction through evaporative cooler increases the exergy efficiency of cooler increases also.

Nomenclature

A	Area	m^2
c_{pa}	Specific heat of air at constant pressure	kJ/kg. K
c_v	Specific heat of vapour at constant pressure	kJ/kg. K
H	Enthalpy	kJ/kg
h_m	mass transfer coefficient	kg/m^2s
$\dot{m}_i$	Mass flow rate	Kg/s
P	Pressure	kPa
Q	Heat transfer	kW
Ra	Air gas constant	kJ/kg .K
S	Entropy	kJ/kg. K
t	Time	s.
T	Temperature	^oC / K
V	Volume	m^3
W	Work	kJ/kg
Greek Symbols		
η_{exe}	Exergy efficiency	
Φ	Relative humidity/ control volume exergy	--/ kJ/kg
Ψ	Exergy	kJ/kg
ω	Moisture content	kg_w/kg_a
Subscripts		
A	Air	
C.V.	Control volume	
Chem.	Chemical	
Dest.	Destruction	
E	Exit	
Fg	Fluid gas	
i	Inlet	
l	Latent	
o	Dead state	
S	Sensible	
Sat	Saturated	
T	Total	
th.	Thermal	
V	Vapour	
W	Water	

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