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Experimental Investigation of Using Evaporative Air Cooler for Winter Air-Conditioning in Baghdad

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Abstract

This paper presents an efficient methodology to design modified evaporative air-cooler for winter air-conditioning in Baghdad city as well as using it for summer air-conditioning by adding a heating process after the humidification process. Laboratory tests were performed on a direct evaporative cooler (DEC) followed by passing the air on hot water through heat exchanger placed in the coolers air duct exit. The tests were conducted on the 2nd of December /2011 when the ambient temperature was 8.1°C and the relative humidity was (68%). The air flow rate is assumed to vary between 0.069 to 0.209 kg/s with constant water flow rate of 0.03 kg/s in the heat exchanger. The performance is reported in terms of effectiveness of DEC, saturation efficiency of DEC, outlet temperature of air and cooling capacity. Heat transfer rate in heat exchanger mode is also estimated. The paper presents the mathematical development of the equations of thermal exchanges through DEC and HE. Prediction of air condition that exits o this system show that the present system could bring the air stream to a comfortable winter zone .

Key words: *Evaporative air cooler, Winter air-conditioning, Heat transfer, Heat exchanger.*

1. Introduction

Refrigerated air conditioners can be used for both winter and summer air-conditioning, but their energy consumption is high. Sometimes, partially effective systems yield the best results in terms of comfort and cost. Evaporative air conditioning systems are inexpensive and offer an attractive alternative to the conventional summer air conditioning systems in places, which are hot and dry. Since the conventional evaporative air cooler (EAC) is not suitable for winter air-conditioning because it would lower the temperature to an inconvenient level, the present work aims to investigate the possibility of modifying the system by equipping it with a heating element. The average temperature for Iraq varies from higher than 48 ºC (120 Fahrenheit) in July and August to below freezing in January. The long-term averages of monthly climatic conditions in Iraq are summarized in Figure 1.The required winter air-conditioning process in this work consists of: an evaporative cooling process which preformed through the using the EAC and a sensible heating process can be achieved by passing hot water through a heat exchanger inserted into the EAC's outlet air duct. The literatures regarding the use of evaporative cooling and conventional (EAC) for summer air conditioning are many [1-4], but the literatures that deal with using EAC for winter air – conditions are few [5-6]. The present work presents the results of a laboratory tests that assessed the effectiveness of the modified evaporative air cooler EAC for cold winter in Baghdad. The test was conducted in the morning of 2 nd of December when the ambient temperature was 8.1ºC and humidity ratio was about 68%. The laboratory tests were performed by passing hot water through a heat exchanger placed in the delivery duct of air cooler. The flow rate of the hot water was constant while different air mass flow rate were taken. These tests showed that the system could bring the air stream to a comfortable condition. This paper develops a mathematical model for both a direct evaporative cooling (DEC) system and heat exchanger (HE) process. A test rig was designed and fabricated to collect experimental data in the Air Conditioning Laboratory at the University of Technology, Mechanical Engineering Department, in Bagdad.

2. Experimental work

2.1. Direct Evaporative Cooling

The principle underlying direct evaporative cooling is the conversion of sensible heat to latent heat. Non-saturated air is cooled by heat and mass transfer increases by forcing the movement of air through an enlarged liquid water surface area for evaporation by utilizing blowers or fans. Some of the sensible heat of the air is transferred to the water by evaporating some of it. The latent heat follows the water vapor and diffuses into the air [8].

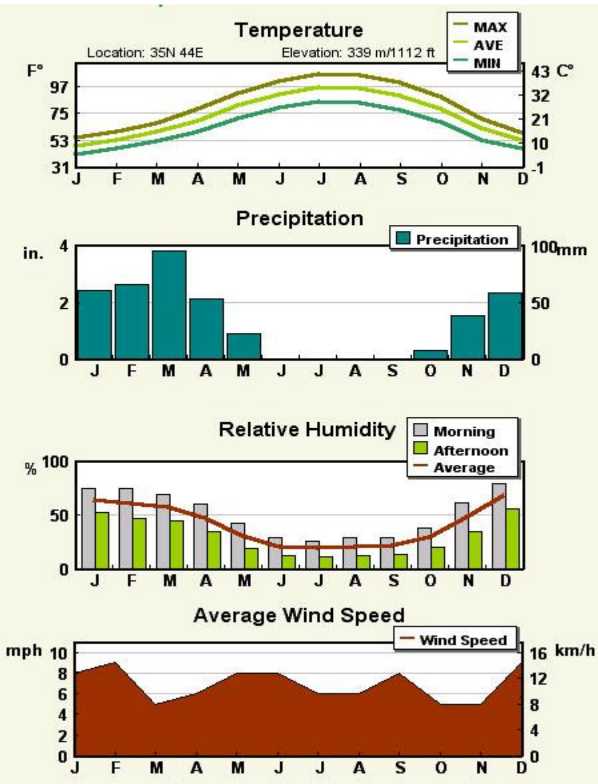

Fig. 1. Climate Conditions in Iraq [7].

Figure 2 shows a schematic direct evaporative cooling system, where water is running in a loop and the makeup water entering the sump to replace evaporated water must be at the same adiabatic saturation temperature of the incoming air. In a DEC, the heat and mass transferred between air and water decreases the air-dry bulb temperature (DBT) and increases its humidity, keeping the enthalpy constant (adiabatic cooling) in an ideal process. The minimum temperature that can be reached is the thermodynamic wet bulb temperature of the incoming air. The effectiveness of this system is defined as the rate between the real decrease of the DBT and the maximum theoretical decrease that the DBT could have if the cooling were 100% efficient and the outlet air were saturated. Practically, wet porous materials or pads provide a large water surface in which the air moisture contact is achieved and the pad is wetted by dripping water onto the upper edge of vertically mounted pads.

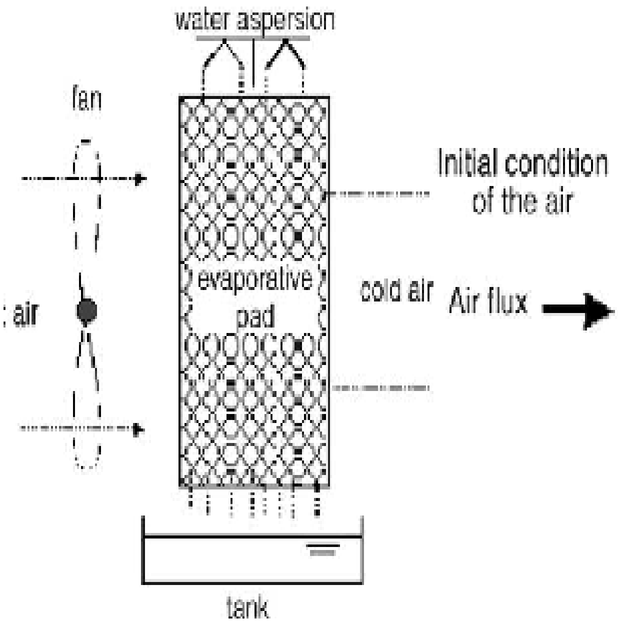

Fig. 2. Direct Evaporative Cooling (DEC).

2.2. Heat Exchanger

A space radiator of which a water-to-air compact heat exchanger was used as a compact heat exchanger in this work.. In a space radiator, heat is transferred from the hot water flowing through the radiator tubes to the air flowing through the closely spaced thin plates outside attached to the tubes. In compact heat exchangers, the two fluids usually move perpendicular to each other, and such flow configuration is called cross-flow. Both fluids are unmixed in a space radiator. The radiator has 20 tubes of internal diameter 1.02 cm and a wall thickness 0.9 mm. The length of the heat exchanger in the direction of the air flow is 65 cm. the tubes are in a closely spaced platefinned matrix of (0.25×0.25) m, as shown in Figure 3.

The radiator was placed at the front opening of the air-cooler duct. Two valves were placed before and after the heat exchanger. The hot water is coming from a storage tank. These systems heat and store water in a tank so that hot water is available at any time. Hot water enters the tubes at 90ºC at a rate of 0.03 kg/s. As hot water is drawn from the top of the tank, cold water enters the bottom of the tank and is heated. The heating source in this work is electricity. Radiator is designed to heat the air stream coming from the DEC. it do this by drawing cold air in at the bottom, warming the air as it passes over the radiator, and discharging the heated air at the top as shown in Figure 4. This sets up convective loops of air movement within a space.

Fig. 3. A Space Radiator.

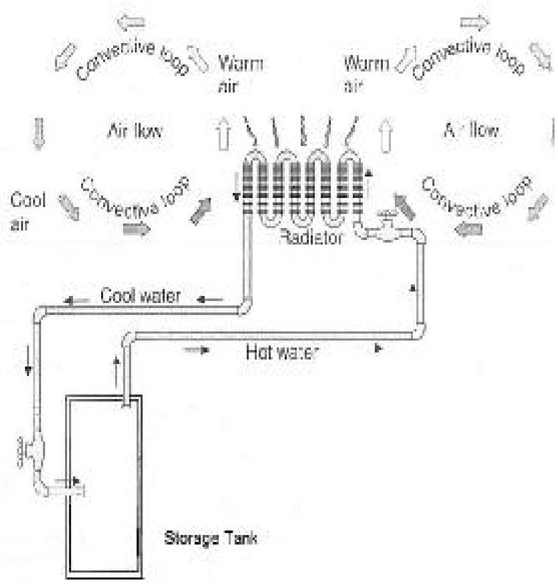

Fig. 4. Convective Loops of Air Flow from Radiator.

3. The Experimental Setup

An experimental study was performed during the winter at the laboratory of air conditioning and refrigeration of the Mechanical Engineering Department in the University of Technology in Baghdad on the second of December/2011.

The test rig shown in Figure 5 consists of a variable speed fan blows air through a 254 mm square duct of a direct evaporative air cooler. The air-cooler was run for an adequate time to reach steady state. The cold outdoor air is first filtered and then is brought in contact with the wetted surface. The equipment utilizes an evaporative pad with 230×230 ×150 mm, and provides about 370 m^2 of evaporative surface area per cubic metre of media [9],providing a wetted area equal to 2.93 m^2 in the pad. The air gets cooled due to simultaneous transfer of sensible and latent heats between air and water. The cooled and humidified air is then heated by using the heat exchanger (radiator), which increase the sensible heat. The hot water is coming from` an insulated storage tank. The top valve was then opened fully, allowing the hot water to run through the heat exchanger. The flow of water was controlled by adjusting the bottom valve's opening.

The heat exchanger was placed at the front opening of the air-cooler duct supplying to the conditioned space. The measuring instruments included a normal thermometer to measure the inlet and outlet of hot water temperature and an electronic humidity-and-temperature meter to measure the properties of: ambient air which represents the evaporative cooler inlet, evaporative cooler outlet and a heat exchanger outlet which represent the conditioned space.

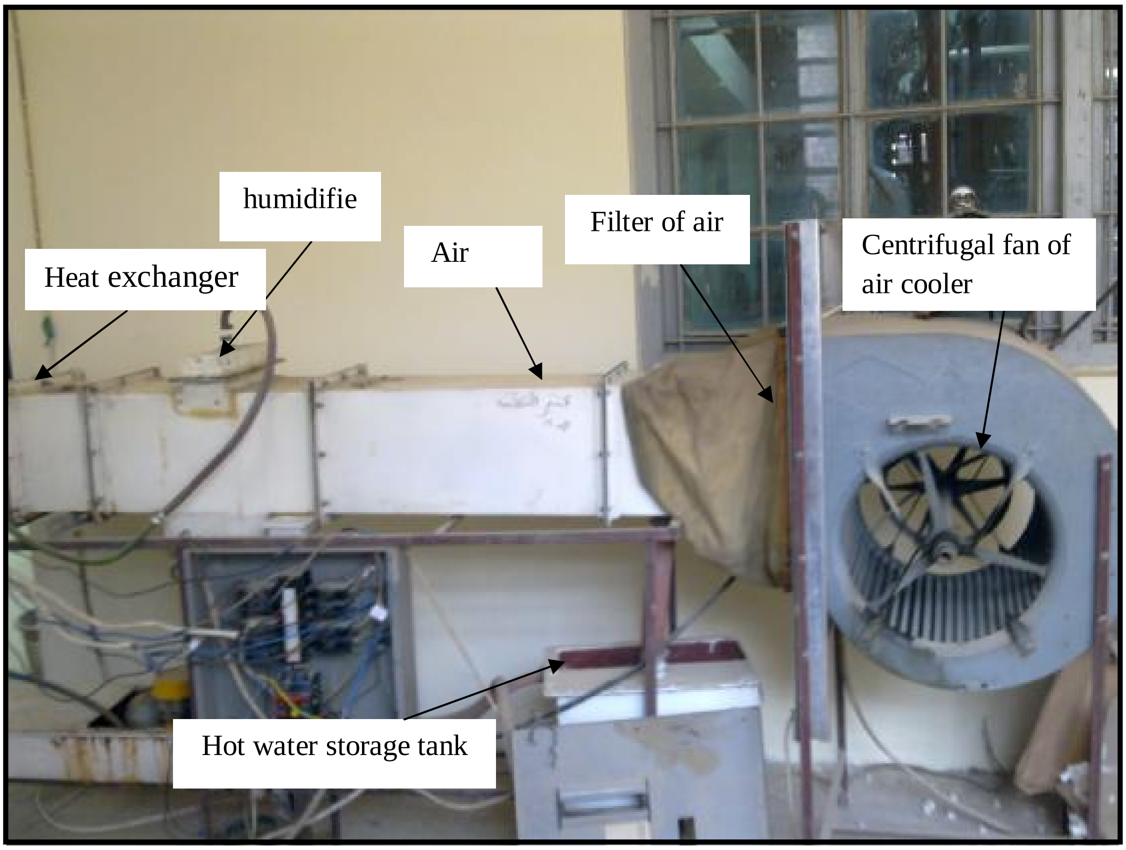

Fig. 5. The test rig.

Table 1 shows the results of the test conducted on the 2nd of December when the ambient temperature (T_1) was 8.1^{o}C and the relative humidity (RH_1) was 68%, the hot-water temperature was 90^{o}C for different mass flow rates of air.

Table1,
The Experimental Data for Inlet Conditions and Properties for DEC and HE.

Inlet Conditions	**T_1=8.1°C , RH_1=68% , T_{w1}=5°C**					
ma(kg/s)	**T_2(°C)**	**$R.H_2$(%)**	**T_{w2}(°C)**	**T_3(°C)**	**$R.H_3$(%)**	**T_{w3}(°C)**
0.069	5.2	95.2	4.8	21.2	42.5	13
0.095	5.7	94.8	5.1	21.0	42.1	12.8
0.139	5.9	94.4	5.3	20.8	41.7	12.4
0.177	6.3	93.7	5.5	20.3	41.3	12.1
0.209	6.6	93.1	5.7	19.8	41.0	12.0

4. Mathematical Model

4.1. Direct Evaporative Cooling Analysis

In this work the humid air is considered as a mixture of two gases: the dry air and water vapour. Considering the humid air flow close to a wet surface, according to Figure 6, the heat transfer will occur if the surface temperature Ts is different from the draft temperature T. If the absolute humidity (concentration) of the air close the surface w_s is different from the humidity of the draft w a mass transfer will also occur. The elementary sensible heat is [8] :

$$\delta Q_s = h_c\, dA(T_s - T) \qquad \ldots(1)$$

The h_c coefficient is determined from the Nusselt number (Nu) expressed as a function of the Reynolds number (Re) and Prandtl number (Pr). In a similar way the rate of water vapour transfer dm_V between the draft and the air close to the surface will be :

$$dm_v = h_m\, dA(w_s - w) \qquad \ldots(2)$$

By analysis of the interface air–liquid, the latent heat δQ_L is determined by the energy conservation law.

$$\delta Q_L = \delta Q - \delta Q_s = h_{LVS} dm_v \qquad \ldots(3)$$

Rearranging Eqs. (1)–(3), the total differential heat flow is :

$$\delta Q = [h_c(T_s - T) + \rho_w h_{LVS} h_m (w_s - w)]dA \qquad \ldots(4)$$

Eq. (4) indicates that the total heat transfer is a result of heat transfer due to temperature difference and due to the difference of the absolute humidities. These two potentials can be combined by the Lewis relationship so that the total heat flow will be expressed by a single potential that is the enthalpy difference between the air close to the wet surface and the air free current. Using the specific enthalpy of the mixture as the sum of the individual enthalpies [10] gives:

$$h_s - h = (h_{sa} - h_a) + (w_s h_{VS} - w h_v) \qquad \ldots(5)$$

With the hypothesis that air and vapour are perfect gases it follows that:

$$hs - h = c_{pu}(T_s - T) + h_{vs}(w_s - w) \qquad \ldots(6)$$

Where the humid specific heat is:

$$C_{pu} = C_{pa} + wC_{pv} \qquad \ldots(7)$$

In the standard environmental conditions C_{pa}=1006 J/ kg K and C_{pv}=1805 J/kg K. Therefore [12]:

$$T_s - T = [(h_s - h) - h_{VS}(w_s - w)]/C_{pu} \qquad \ldots(8)$$

Combining Eqs. (4) and (8) gives:

$$dQ = \frac{h_c dA}{C_{pu}}\{(h_s - h) + \frac{(w_s - w)}{Le}(h_{LVS} - Le h_{VS})\} \qquad \ldots(9)$$

Where Le is the Lewis relationship, a dimensionless number expressed as:

$$Le = h_c / h_m C_{pu} \rho \qquad \ldots(10)$$

In the above deduction the density of the humid air was approximated by the density of the dry air. Taking the Lewis relationship as being unitary, gives $(h_{LVS} - h_{VS}) \approx h_{Ls}$. It is also verified that the term $(w - w_s)h_{Ls}$ is usually negligible in the presence of difference of the specific enthalpies $(h_s - h)$,so that only the first term inside brackets is significant. In the same way, the total heat flow is caused by the difference of specific enthalpies of the air and of the saturated air close to the wet surface and is given by:

$$\delta Q = h_c dA\,(h_s - h)/C_{pu} \qquad \ldots(11)$$

The sensible heat transferred is:

$$\delta Q_s = m_a C_{pu} dT \qquad \ldots(12)$$

Therefore, by combining Eq. (12) with Eq. (1) gives:

$$h_c dA(T_s - T) = m_a C_{pu} dT \qquad \ldots(13)$$

This can be integrated, resulting in:

$$1 - \frac{T_1 - T_2}{T_1 - T_s} = \exp\left(-\frac{h_c A}{m_a C_{pu}}\right) \qquad \ldots(14)$$

The effectiveness of direct evaporative cooling equipment is defined as:

$$e = \frac{T_1 - T_2}{T_1 - T_s} \qquad \ldots(15)$$

$$e = 1 - \exp\left(-\frac{h_c A}{m_a C_{pu}}\right) \qquad \ldots(16)$$

Analyzing Eq. (15) it is verified that an effectiveness of 100% corresponds to air leaving the equipment at the wet bulb temperature of entrance. This requires a combination of large area of heat transfer and a high heat transfer coefficient and low mass flow. It is also observed that the effectiveness is constant if the mass flow is constant since, it controls directly and indirectly the value of the parameters on the Eq. (16).

The flow rate of supply air should be such that when released in to the conditioned space, it should be able to maintain the space at satisfactorily condition, and offset the sensible and latent heat losses. The amount of supply air required (m_a) can be obtained by using the following equation:

$$m_a = \rho_a u A_w \qquad \ldots(17)$$

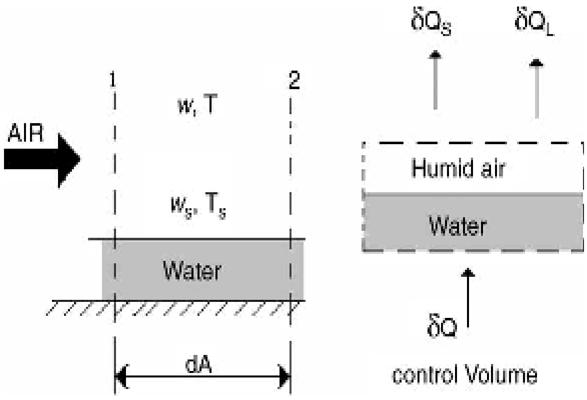

Fig. 6. Schematic Direct Evaporative Cooler.

Heat transfer coefficient estimation

Dowdy and Karabash [11] presents a correlation to determinate the convective heat transfer coefficients in a rigid cellulose evaporative media:

$$Nu = 0.1\left(\frac{l_c}{l}\right)^{0.12} \mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3} \qquad \ldots(18)$$

In DEC state, rectangular, pad of rigid cellulose material are considered as cooling media. Wetted surface area of cellulose material is assumed to as 370 m^2/m^3 [12].
Total wetted surface area of each pad is given by:

$$A_w = V_p\, A_s \qquad \ldots(19)$$

The characteristic dimension of the pad is:

$$l_c = \frac{V_p}{A_w} \qquad \ldots(20)$$

This parameter is called characteristic length and is used to calculate the Reynolds (Re) and Nusselt (Nu) numbers.

$$\mathrm{Re} = \frac{u l_c}{n} \qquad \ldots(21)$$

So the convective heat transfer coefficient can be calculated from:

$$Nu = \frac{h_c l_c}{k} \qquad \ldots(22)$$

The following air properties: k= 0.0263 W/m °C; Pr=0.708; C_{pu}=1033 J/kg °C and $\nu=15.8\times10^{-6} m^2/s$ at the inlet air temperature are used.
Table 2 shows the resulting convective heat transfer coefficient for several air velocities calculated from Eq. (22).

Table 2,
Convective Heat Transfer Coefficient for Several Air Speeds.

m_a(kg/s)	u(m/s)	Re	$h_c(W/m^2.C^o)$
0.069	1.1	191	35.47
0.095	1.5	261	45.46
0.139	2.4	417	66.26
0.177	2.9	504	77.11
0.209	3.3	573	85.44

4.2. Heat Exchanger Analysis

The characteristics of fluids contribute to a fundamental property of heat exchangers is the heat transfer rate (Q). The heat transferred to the colder fluid(air) must equal that transferred from the hotter fluid(water), according to the following equation [13]:

$$Q = [m_a \times C_p \times (T_{out} - T_{in})]_{cold} = -[m_w \times C_{pw} \times (T_{out} - T_{in})]_{hot} \qquad \ldots(23)$$

The rate of heat transfer in this radiator from the hot water to the air is determined from an energy balance on air flow,

$$Q = [m_a\, C_p\, (T_{in} - T_{out})]_{air} \qquad \ldots(24)$$

The tube-side heat transfer area is the total surface area of the tubes, and is determined:

$$A_o = n\pi D_o\, L \qquad \ldots(25)$$

Knowing the rate of heat transfer and the surface area, the overall heat transfer coefficient on the air side can be determined from:

$$U_o = Q/\, A_o\, \Delta T_{lm} \qquad \ldots(26)$$

Where ΔT_{lm} is the log mean temperature difference for the counter-flow arrangement:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln \dfrac{\Delta T_1}{\Delta T_2}} \qquad \ldots(27)$$

Where;

$$\Delta T_1 = T_{h,\,in} - T_{c,\,out} \qquad \ldots(28)$$

$$\Delta T_2 = T_{h,\,out} - T_{c,\,in} \qquad \ldots(29)$$

The overall heat transfer coefficient of the out-side air is calculated from Eq.(26).

5. Results and Discussion

Two modes of test were conducted namely: evaporative cooling process through DEC and heating process through H.E.

5.1. Saturation Effectiveness for DEC Mode

The effectiveness of DEC for different air mass flow rate is shown in Figure 7. In direct cooling mode, saturation efficiency ranges from 80.1 % to 68.6 % for air mass flow rate of 0.069 to 0.209 kg/s for rectangular pad shape. It decreases for increase in air mass flow rate because at higher velocities, air has lesser contact time with water causing less evaporation.

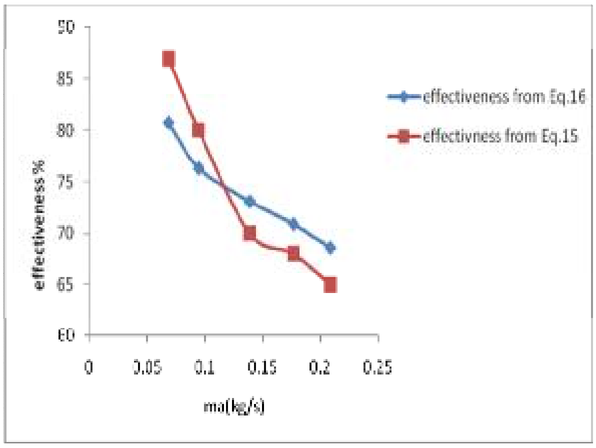

Fig. 7. Variation of Saturation Effectiveness .

5.2. Cooling Capacity for DEC Mode

Cooling capacity in direct cooling mode can be estimated in the following manner [14].

$$Q_c = m_a C_{pu}(T_2–T_1) \qquad \ldots(30)$$

Cooling capacity for different air mass flow rate is shown in Figure 8 . It depends on mass flow rate of air and temperature drop. DEC cooling capacity ranges from 221 to 370 W for mass flow rate. The general trend is that cooling capacity increases with air mass flow rate. The values are compared well with those obtained in the literature [12].

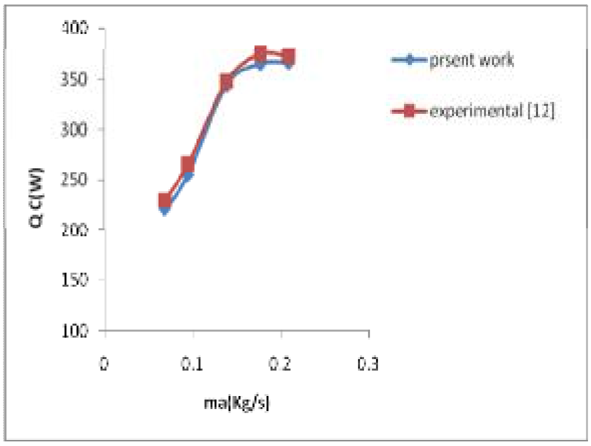

Fig. 8. Variation of Cooling Capacity.

5.3. Heat Transfer Rate and the Overall Heat Transfer Coefficient for HE Mode

Both the heat transfer rate of HE mode calculated from Eq.(24) using the air side and the overall heat transfer coefficient calculated from Eq.(26) are shown in Table 3 for different air mass flow rate. The values of heat transfer rate (Figure 9) and the overall heat transfer coefficient of the air side are increased with increasing the mass flow rate of air for constant water flow rate in the radiator.

Table 3,
Heat Transfer Rate and Overall Heat Transfer Coefficient for Several air Mass Flow Rate.

m_a(kg/s)	Q(W)	$U_o(W/m^{-2}C^{o-1})$
0.069	1140.4	40.4
0.095	151.3	57.3
0.139	2139.4	75.8
0.177	2559.7	90.7
0.209	2849.8	101.1

5.4. Outlet Air Dry Temperature from HE Mode

The outlet dry temperature from the radiator is decreased with increasing the air mass flow rate as shown in Figure10 because air has lesser contact time with hot water in the radiator for the same inlet temperature of water. The values of these temperatures showed that the present system succeed to achieve comfortable condition for air flow for winter air-conditioning in Baghdad.

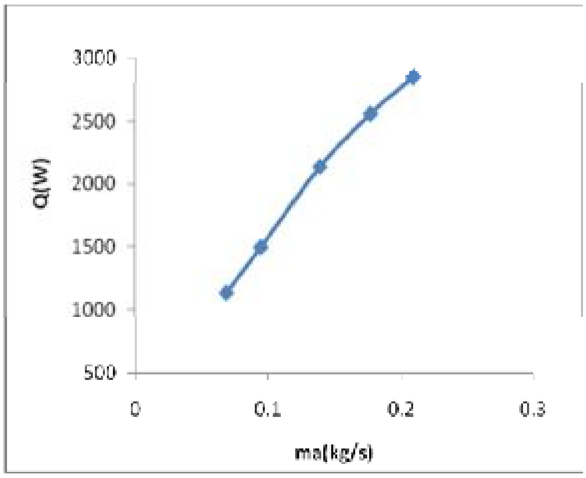

Fig. 9. Variation of Heat Transfer Rate.

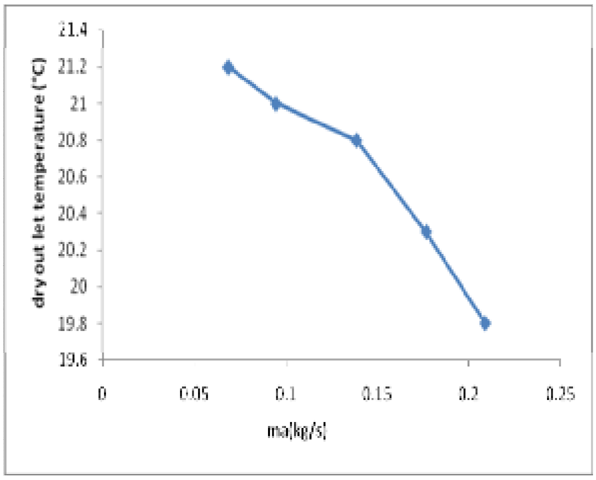

Fig. 10. Variation of Dry Outlet Temperature.

5.5. Thermal Comfort: Pyschometry and Thermodynamics

The psychometric analysis provides more details on the actual process of changing the DBT and RH values, during the winter air conditioning process. This analysis not only provides the thermal comfort range but also, describes the actual implementation associated efficiencies. Changing the inlet air condition from state 1 to state 3 can be done through an evaporative cooling system, followed by sensible heating process shown schematically in Figure 11, for m_a=0.095 kg/s along with its psychometric trace. Following the psychometric trace; the ambient air enters through DEC (state 1) (T_1=8.1ºC &RH=68%) under adiabatic cooling and humidification process (1→2) and air leaves the DEC at state (2) (T_2=5.2ºC &RH=97%).

Then the cold air enters to the heat exchanger (radiator), heat is added to the air through the radiator (2→3). In this analysis the heat exchanger will be used to achieve the temperature and relative humidity for air at state (3) (T_3=20.2ºC &RH=42%), which raises the DBT and reduces RH, in order to achieve a more comfortable condition. The warm air exit from heat exchanger (state 3) is bringing the cold air to winter comfort zone. In the same manner, presented the states on psychometric chart for different air mass flow rate.

The electricity consumption of this air-cooler is considerably lower than that required by a refrigerated system. The cost of water consumed by the evaporative air coolers adds only a minimal cost to their operation. A study performed at the University of Arizona, that compared the combined electrical and water consumption of evaporative coolers with the electrical consumption of central refrigerative air conditioners, found that that the typical evaporative cooler consumed about 1500 kWh of electricity per summer, costing about $150 [15]. The cooler’s water consumption added an average of $54 to a municipal water bill over the course of the summer, giving an electricity-and-water total of $204. By comparison, the central air conditioners consumed an average of 6000 kWh of electricity per summer, or about $600. The $400 saved annually by the evaporative cooler makes it an attractive option not only for the residents of Arizona but for all families across the hot country.

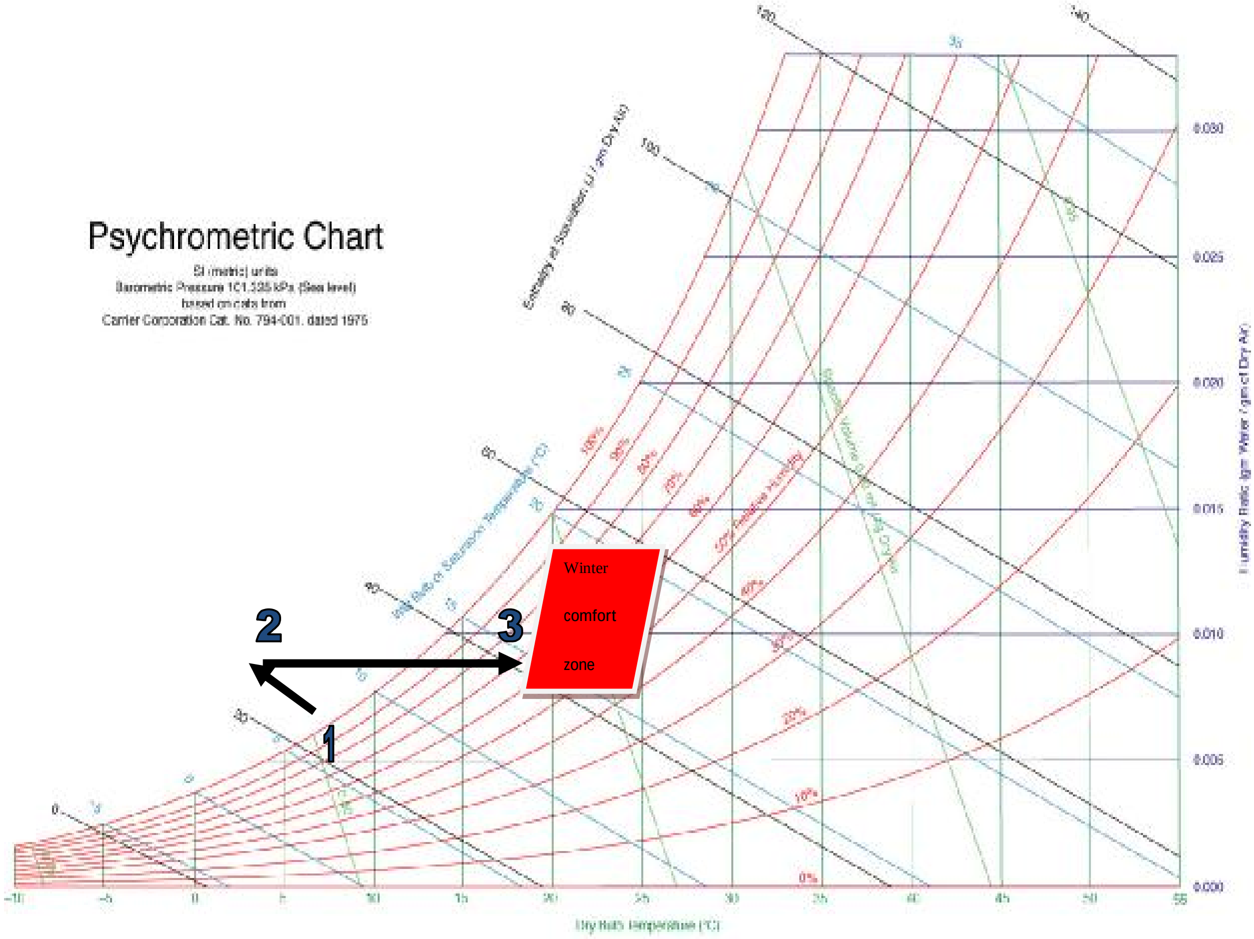

Fig. 11. Representation of the Experimental Result.

6. Conclusion

In the present work, the required winter air-conditioning process consists of an evaporative cooling process followed by a sensible heating process which can be achieved through the use of direct evaporative air cooler followed by heat exchanger placed into the DEC outlet. The Psychometric and thermodynamic analyses showed that the test of the present system could bring the air stream to the winter thermal comfort zone. The data acquired by experiment were analyzed by plotting the curves between various performance parameters. The theoretical model can be used to predict the performance of modified indirect evaporative cooler. The proposed system maks a good use of DEC in winter as well as in summer because the evaporative cooler makes it an attractive option across hot weather areas. Also the present system can be used not only in Baghdad city, but in other cities which have the same weather.

List of Symbols

A	Area of the heat transfer surface; total wetted surface area (m^2)
C_{pa}	Constant pressure specific heat of the dry air (J /kg.K)
C_{pu}	Specific heat of the humid air (J /kg. K)
C_{pv}	Constant pressure specific heat of the vapour (J /kg. K)
h_a	Specific enthalpy of the air (J/kg)
h_c	Convective heat transfer co-efficient (W/ m^2.°C)
h_{LVS}	Specific enthalpy of vaporization of the water at surface temperature (J /kg)
h_m	Mass transfer co-efficient (kg/ m^2. s)
hsa	Specific enthalpy of the leaving air (J /kg)
h_v	Specific enthalpy of the vapour (J/ kg)
h_{VS}	Specific enthalpy of the vapour at surface temperature (J /kg)
l	Pad thickness (m)
l_e	Characteristic length (m)
L	Heat exchanger length (m)
Le	Lewis relationship (dimensionless)
m_a	Air mass flow (kg/ s)
m_v	Mass flow of the water vapour (kg/ s)
Nu	Nusselt number (dimensionless)
Pr	Prandtl number (dimensionless)
Qc	Cooling capacity of evaporative cooler (W)
Q	Heat transfer rate of heat exchanger (W)
Re	Reynolds number (dimensionless)
RH	Relative humidity of air (%)
T	Dry bulb temperature of air (°C)
T_s	Surface temperature (°C)
u	Air speed (m/s)
Uo	Overall heat transfer coefficient (W/m^2.C°)
V_p	Volume occupied by the evaporative pad (m^3)
w	Absolute humidity of the draft (kg_w/kg_{air})
w_s	Absolute humidity of the air close the surface (kg_w /kg_{air})

Greek symbols

ε	cooling effectiveness (dimensionless)
ΔT_{lm}	log mean temperature difference
ρ	density (kg/ m^3)

Abbreviations

EAC	evaporative air cooler
DEC	direct evaporative cooling
DBT	dry bulb temperature
HE	heat exchanger

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دراسة عملية باستخدام مبردة الهواء التبخيرية لغرض تكييف الهواء شتاءاً في بغداد

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الخلاصة

هذا البحث يقدّم نموذج بسيط وفعّال لتطوير مبردة هواء لغرض استخدامها للتدفئة في فصل الشتاء وبما يلائم المناخ في مدينة بغداد، ويتم ذلك بإضافة عملية تسخين بعد عملية الترطيب. تم تصميم وبناء منظومة لهذا الغرض واخذ القراءات المختبرية، حيث تتكون هذه المنظومة من مبردة هواء مرتبطة بمجرى للهواء ويّبت في نهايته مبادل حراري. تم أجراء التجارب في الثاني من شهر كانون الثاني لسنة 2011 حيث كانت درجة حرارة الهواء 8.1°C والرطوبة النسبية 68% ومعدل تدفق الهواء الداخل للمنظومة تتغير بين 0.069 to 0.209 kg/s مع ثبوت تدفق الماء للمبادل الحراري 0.03 kg/s. تم الحصول على النتائج على هيئة فعالية مبردة الهواء، كفاءة مبردة الهواء، درجة حرارة الهواء الخارج من المنظومة وسعة التبريد وكذلك تم حساب انتقال الحرارة خلال المبادل الحراري . ان هذا البحث يقدم نموذج رياضي لمعادلات التبادل الحراري خلال مبردة الهواء والمبادل الحراري. على ضوء النتائج التي تم الحصول لظروف الهواء عند خروجه من المنظومة، فقد أثبتت هذه المنظومة فعاليتها وكفاءتها حيث تم الحصول على ظروف مريحة للهواء لفصل الشتاء.