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# Combined evaporative air cooler and refrigeration unit for water purification and performance enhancement of air cooling system

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## Abstract

Experimental and theoretical studies of hybrid system combined an evaporative air cooler and refrigeration unit are presented. The aim of the current study is to enhance performance of the evaporative air cooler and reduce moisture content in the outlet air as well as to produce fresh water. Heat and mass transfer for wetted pad in the air cooler and the evaporator of the refrigeration unit are formulated and simulated under Basra climate conditions in (May, June, July, and August). Different controlling parameters like (inlet temperature, relative humidity, evaporator coil temperature, frontal air velocity and thickness of the wetted pad are studied). The evaporative air cooler used in the experimental rig. was of (2061) cfm volumetric flow rate with dimension (0.72m, 0.72m, 0.85m). It has a selector of two air velocities (low and high) the minimum and maximum frontal air velocities are (0.5 m/s - 4 m/s) respectively. The refrigeration unit consist of a Compressor of 1/3hp capacity (AC 220 volts, R134a refrigerant). The evaporator coil has one row, 18 tube of 24.5 cm length, and outside diameter of 0.9 cm. The experimental results show a decrease in outlet temperature by (1-3℃) and the fresh water production increases with increasing, humidity ratio, relative humidity, and decreasing of the coil temperature. A comparison between the experimental and theoretical work shows a good agreement*[*

***Keywords:*** evaporative air cooler, refrigeration unit, productivity, moisture content, evaporator coil.

## 1. Introduction

The improving in living criterions, and increasing in the population and cheap electricity in some regions such as North Africa and the Middle East led to an increase in the using of air conditioning systems in the world **[1]**.

Recently most of the buildings and workplaces use conventional air conditioning systems which depend on vapour compression refrigeration system. These systems spend a substantial amount of power and they may be harmful to the environment and causes the suffering to the residents from low thermal performance in hot climate conditions **[2]**.

Evaporative cooler systems consider as low-cost cooling systems so that is it's highly demanded by the majority of the population. This kind of systems considers as one of the oldest principles of air conditioning known to Man. the most widespread system of household cooling found in a barren area is air cooling system by the evaporation of water. The popularity of evaporative cooling systems in many places is because it's low primary and operational cost compared to refrigerate cooling systems **[3]**. There are many disadvantages of the evaporative cooler, one of the moisture levels in the conditioned space could be higher. This study is trying to combine evaporative air cooler with a small refrigeration unit to decrease moisture content in conditioned and enhance performance of the evaporative air cooler.

**Nomenclature**

| Symbol | Description | Unit |
|---|---|---|
| $A$ | Surface area | $m^2$ |
| $A_o$ | Total air-side surface area of the tube and the fins | $m^2$ |
| $A_f$ | Total fin surface area | $m^2$ |
| $A_e$ | Total fin surface area of evaporator | $m^2$ |
| $cp_h$ | Specific heat of moist air | J/(kg. K) |
| $cp_u$ | Specific heat of the dry air | J/(kg. K) |
| $cp_v$ | Specific heat of the humid air | J/(kg. K) |
| $rp$ | Packing fraction of the pad | $m^2/m^3$ |
| $h$ | Enthalpy | J/kg |
| $h_c$ | Convective heat transfer coefficient | $W/m^2$ °C |
| $hfg$ | Latent heat of water vaporization | J/kg |
| $h_m$ | Mass transfer coefficient between air and water. | m/s |
| $SQ_l$ | Latent heat transfer rate | W |
| $SQ_s$ | Sensible heat transfer rate | W |
| $T$ | Temperature | °C |
| $\omega_a$ | Air humidity ratio | $kg_v/kg_{d.a}$ |
| $\omega_{sat}$ | Saturated air humidity ratio | $kg_v/kg_{d.a}$ |
| $\omega_s$ | Saturated air humidity ratio at Ts | $kg_v/kg_{d.a}$ |
| $\omega_{a,po}$ | The humidity ratio of the air leaving | $kg_v/kg_{d.a}$ |

| | | |
|---|---|---|
| $\dot{m}_f$ | Water mass flow rate | kg/s |

**Greek symbols**

| Symbol | Description | Unit |
|---|---|---|
| Ɛ1 | Effectiveness of direct evaporative cooler | |
| $\rho$ | Density of air | kg/m³ |
| $\eta$ | efficiency | _ |
| $\Omega$ | standard extended surface parameter | – |
| $\xi$ | integrated factor | _ |
| $\delta$ | Thickness of pad | m |

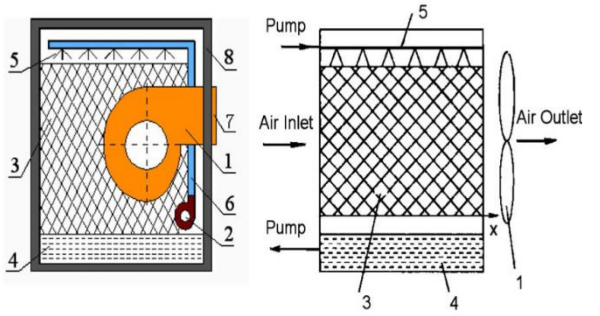
**Subscripts**

| Symbol | Description |
|---|---|
| i | Inlet |
| o | Outlet |
| a | Air |
| w | Wet-bulb temperature |
| e | Evaporative cooler |
| s | Surface of evaporator coil |
| p | Evaporator coil |
| $f$ | fin |

An overview of the relevant available literature Dai et al, **[4]** studied direct evaporative cooler through the flow was used the honeycomb as a packing material and then investigated. The system was expected to create a good internal environment in the arid area. The result of the analysis shows the length of the air duct is ideal, leading to the lowest temperature. Improved process parameters can increase the performance of the system. The flow rates of the mass of water feeder, in the case of typical conditions, can increase the relative humidity by 50% and lower the temperature of air to 9 degrees and can also improve performance. Jose et al, **[5]** studied in the summer of the Brazilian city, direct evaporative cooler. They developed a mathematical model of the direct evaporative cooling system for heat exchange equations. The empirical results were presented that occurred on the direct evaporative cooler and they used experimental results to determine transmission direct convection and compare it with the mathematical model. Kulkami et al, **[6]** suggested the theoretical performance of the different materials of cooling pads for evaporator cooler. Materials that were examined, rigid cellulose, high corrugated polyurethane density, corrugated paper and asbestos fibers. They noted that the efficiency of saturation was increased with a decrease in the flow rate of the air mass. It has also seen greater saturation efficiency if the material has high wetted surface. Yu and Chan, **[7]** used mist pre-cooling to decrease the inlet air temperature of chiller equipment. Mist pre-cooling is used to improve the (COP) and decreasing the power consumption. A hotel in Hong Kong used mist pre-cooling to enhance energy consumption. Their results show around 18% decrease in power consumption when used mist pre-cooling of air entering to the chiller. Jain et al, **[8]** studied a hybrid system of direct evaporative cooler with air conditioning (AC), to reduce the annual expenditure of electricity with a similar level of comfort. Four different applications of construction are located in four different cities in India. The hybrid system can be a financially attractive option especially in high-density offices, cinema, theatre, and lobby applications. The hybrid system is more attractive to build with higher cooling loads. Therefore, this hybrid system will be used to meet high load locations for cooling in the appropriate climatic conditions. Cost analysis has been completed for the four cities in India with a minimum payback period of (3.6) years at Akola and maximum of (6.0) years at Indore. Oday Kadhem, **[9]** studied a hybrid system consisting of two parts, an evaporative part and a compressive part, in order to contribute to the reduction of electrical energy. There are a number of variables that effect on the performance of the evaporative cooling, such as the mass flow rate of water, the inlet water temperature, the size of the pad and the air mass flow rate. The results show that the pre-cooled water has some low effect on the system performance as the increase in the effectiveness about (5% -10%). Shailendra and Rajput, **[10]** studied a dew point evaporative- vapour compression based combined air condition system for thrift perfect human comfort condition at low working cost. They also unite the system with conventional vapour compression air conditioner on the basis of cooling load cooling coil 100% worked on fresh air assumption. The saving load on the cooling was found maximum with a value (60.93%) at (46C) and (6g/kg) specific humidity the average monthly power saving (192.31kwh) for a dry and hot condition. And (124.38kwh) for moderate humid and dry condition. Therefore it could be a better alternative for moderate humid and dry climate with the period payback of (7.2) years. Nada et al, **[11]** studied theoretical investigation of the performance of the proposed integrated air conditioning system (AC). And the systems of removing moisture and moisturizing water desalination, as well as the purpose of providing energy from the air conditioning system. While at the same time benefiting from the system in the production of fresh water for air conditioning systems. Sang et al, **[12]** suggested develop a liquid desiccant and dew point evaporative cooling assisted 100% outdoor air system (LDEOS), and to estimate its energy savingipotentials. A quasi-dynamic simulation model of (LDEOS) containing a dew point evaporative cooler model was developed to compare the energy performance of (LDEOS) against a typical variable air volume (VAV) system. Simulation results presented that (LDEOS) could save (12%) of primary energy compared to a conventional (VAV), while 100% of outdoor air was conditioned and supplied to space without using vapour compression technology. Peng et al, **[13]** studied the numerical model of a novel outside evaporative cooling liquid desiccant dehumidifier (OECD). Was developed and the effects of inlet parameters, including the relative humidity of the dehumidified air and inlet temperature and evaporative cooling air as well as the inlet mass flow rate of solution and so, on the device performances were investigated in this study. The result expression, the moisture removal rates of (OECD) were increased by (14.0%-18.0%) (31.1%-101.5%) compared to the non-evaporative cooling dehumidifier and the adiabatic dehumidifier respectively an inlet temperature of the solution increased from (31 to 42ºC). Whereas the dehumidification rate was only decreased by about (1.6%) with an increase in the inlet temperature of LiCl solution from( 24 to 44°C). In a review of the related information on the basics, research and development of some evaporative cooling technologies and their applications on buildings. Discussed reduce moisture content in space condition by desiccant based evaporative cooling systems, such as indirect evaporative air cooler or combined direct and indirect evaporative air cooler and enhance the performance of the different types of an evaporative air cooler. After the deep literature review, no work has been found like the work proposed in this paper to determine reduced the moisture content indirect evaporative air cooler by used refrigeration unit, which on operating together under dry and hot conditions produces humid and cold air to the human comfort condition at reasonably low cost. With less electric power consumption as compared to the conventional air conditioning system.

## 2. System Description

The process of transfer liquid water to air is called direct evaporative cooling, which leads to lower its temperature. Figure (1) shows a typical direct evaporative cooler. In this system, the outside air was drawn by a fan through a wetted pad the cooled air was circulated through the building. Due to evaporation of water which leads to increase moisture content and a decrease in air temperature give a suitable energy requisite. The latent heat gain is balanced out the sensible heat loss of air due to the adiabatic process in which the moisture content of the air was increased. The cooling path is illustrated on a psychrometric chart with a typical mathematical value of the performance in figure (2) **[14]**. Vapor compression refrigeration cycle is the most widely used cycle for refrigerators, air conditioning systems, and heat pumps. The common vapor compression refrigeration cycle has four main components that are (condenser, expansion valve or capillary tube, evaporator and the compressor) **[15]**. The major function of a refrigerator is to create a cold region by rejecting heat to the ambient as illustrated in figure (3).




1 fan 2 pump 3 pad material 4 water basin 5 water distribution system 6 water supply pipe 7 air outlet 8 frame

Fig. (1) Sketches of the inner configuration and the working principle of the direct evaporative cooler **[17]**.

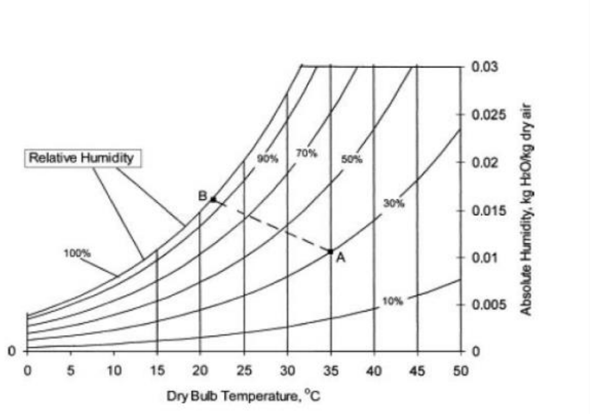



Fig. (2) Cooling path for direct evaporative cooler **[16]**.

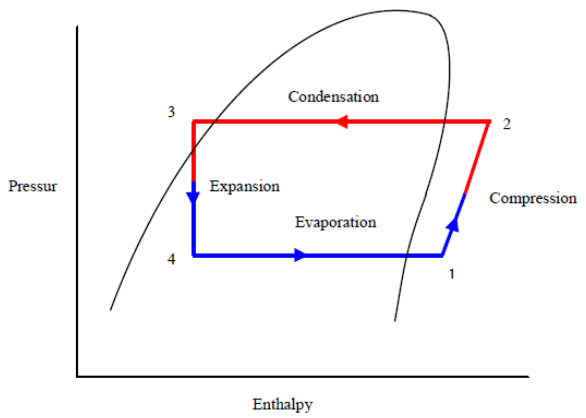


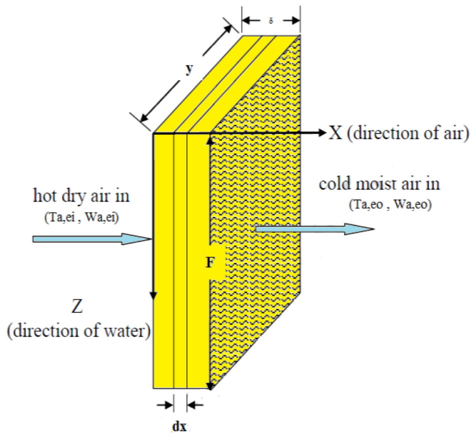

Fig. (3) Typic al vapour compression refrigeration cycle **[20]**.

### 3. Theoretical analysis

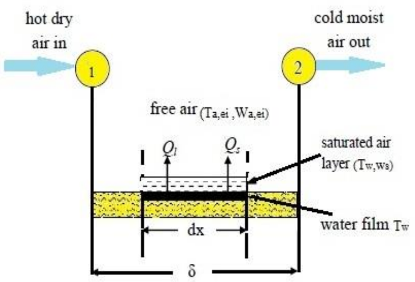
This work presents a theoretical analysis of evaporative cooling systems applicable in different fields.

All parameter of the evaporative air cooler and optimal design are taken, to evaluate the performance of the evaporative cooling system and to reach the optimum conditions. In this study, a theoretical analysis for a hybrid system which consists of an evaporative air cooler and a vapour compression system will be explained. Initially, the analysis of each system alone will be introduced, then the interaction between the two systems is illustrated. The following assumption are used to simplify the solution of the proposed model:

1. The wetted pad is fully and uniformly.
2. The thermal propereties of water and air are constant.
3. The humidification process is adiabatic
4. The analysis is conducted in a steady-state mode.
5. The sensible heat removed and latent heat gained to air are equal.
6. The surface temperature of the evaporator coil is equal to the refrigerant temperature.
7. The compression process in refrigeration cycle is isotropic.




(a)




(b)

Fig. (4) Schematic of direct evaporative cooler.

## 3.1 Theoretical Analysis of the Evaporative Cooling Unit

The amount of water evaporating ($S\dot{m}_v$) from pad element (dx) may be decided by Fick's law **[17]**:

$$S\dot{m}_v = m_a.dw = h_m.\rho\,(\omega_{sat} - \omega_a)dA \quad (1)$$

Where $dA$ is the surface area of element :

$$dA = F.y.dx.rp \quad (2)$$

Substitute equation (2) into (1) get:

$$\dot{m}_a\,.\delta w_a = h_m.\rho(\omega_s - \omega_a).F.y.dx.rp \quad (3)$$

from Fig. (4) get boundary condition and integrate equation (3) gets **[19]**:

$$\int_{\omega_{ei}}^{\omega_{eo}} \frac{d\omega_a}{(\omega_{sat}-\omega_a)} = \frac{h_m.\rho.F.y.rp}{\dot{m}_a}\int_0^{\delta} dx \quad (4)$$

Integrating of equation (4) yields :

$$\frac{\omega_{a,eo}-\omega_{a,ei}}{\omega_{sat}-\omega_{a,ei}} = 1 - e^{\frac{-h_m.A}{\dot{m}_a}} \quad (5)$$

$$SQ_t = \dot{m}_a dh_a = SQ_s + SQ_l \quad (6)$$

$$\dot{m}_a.dh_a = h_c(T_{a,ei} - T_w)dA + h_m.hfg(\omega_{sat} - \omega_{a,ei}) \quad (7)$$

The enthalpy ($h_a$) of moist air can be inscribed as follows **[18]**:

$$h_a = cp_u.T_{a,ei} + \omega_{a,ei}(hfg + cp_v.T_{a,ei}) \quad (8)$$

Substitute equation (8) into (7) get:

$$\dot{m}_a(cp_u + \omega_{a,ei}.cp_v)dt_a = [h_c + h_m.cp_v(\omega_{sat} - \omega_{a,ei})](T_{a,ei} - T_w)dA \quad (9)$$

The specific heat of moist air can be inscribed as follows **[18]**:

$$cp_h = cp_u + \omega_e.cp_v \quad (10)$$

Relationship of Lewis number that relays both mass and heat transfer coefficients **[22]**.

$$\frac{h_c}{h_m} = \rho.cp_h.\left(le^{\frac{2}{3}}\right) \quad (11)$$

Where

" $le$ is the Lewis number".

preparations equation (9) and Substitute equation (10,11) in (9) to get:

$$\dot{m}_a.cp_h.dt_a = h_c\left[1 + \frac{cp_v}{cp_h.\rho.\left(le^{\frac{2}{3}}\right)}(\omega_{sat} - \omega_e)\right](T_a - T_w)dA \quad (12)$$

The second term in the square brackets approaching to zero compared to 1, then equation (12) becomes **[19]**:

$$\dot{m}_a.cp_h.dt_a = h_c(T_{a,ei} - T_w)F.y.dx.r \qquad (13)$$

from Fig.(4) get boundary condition and integrate equation (13) get:

$$\int_{T_{a,ei}}^{T_{a,eo}} \frac{dT_a}{T_a - T_w} = \frac{h_c.F.y.rp}{\dot{m}_a.cp_h}\int_0^{\delta} dx \qquad (14)$$

Introducing the definitions of total convective heat transfer area $A = F.y.\delta.rp$ and air mass flow rate $\dot{m}_a = A.\rho.u$ in the below equation **[19]**:

$$\frac{T_{a,eo} - T_{a,ei}}{T_w - T_{a,ei}} = 1 - e^{\frac{-rp.\delta.h_c}{cp_h.\rho.u}} \qquad (15)$$

$$T_{a,eo} = T_{a,ei} + (T_w - T_{a,ei})\left(1 - e^{\frac{-rp.\delta.h_c}{cp_h.\rho.u}}\right) \qquad (16)$$

The effectiveness ($\varepsilon$ ) can be written as follows **[23]**:

$$\mathcal{E} = 1 - e^{\frac{-rp.\delta.h_c}{cp_h.\rho.u}} \qquad (17)$$

## 3.2 Theoretical Analysis of the Evaporator Model

The air pass over a cold coil in the air handling unit which is below the dew point temperature of the air. This causes some of the water vapor in the air to condense out onto the coil where it is drained away. The humidity ratio of the air leaving the evaporator will be lower than that of the entering air to the evaporator **[20]**:

$$SQ_t = SQ_s + SQ_l = h_c\, A_e(T_{a,eo} - T_s)\eta_s + h_m\, A_e\, \rho_a(\omega_{a,eo} - \omega_s)\eta_s.h_{fg} \qquad (18)$$

Where: $\omega_s$ is the humidity ratio of saturated air at $T_s$.

$$SQ_t = \frac{h_c\, A_e\, \eta_s}{cp_h} = \left[(cp_h\, T_{a,eo} + \omega_{a,eo}\, h_{fg}) - (cp_h T_s + \omega_s\, h_{fg})\right] \qquad (19)$$

Then,

$$H_{a,e} = (cp_h T_{a,eo} + \omega_{a,eo}\, h_{fg}) \qquad (20)$$

$$H_s = (cp_h T_s + \omega_s\, h_{fg}) \qquad (21)$$

Rearranging and Substitute from equation (19,20) and (21), gets **[20]**:

$$\dot{m}_a dH_a = \frac{h_c\, A_e\, \eta_s}{cp_h} = (H_{a,e} - H_s) \qquad (22)$$

Equation (22) is integrated assuming that the enthalpy of the saturated air at the tube wall temperature is constant along the air cell. With this assumption, the evolution of the enthalpy of the air becomes exponential. Then the solution for enthalpy becomes **[20]**:

$$\int_{H_{a,eo}}^{H_{a,po}} \frac{dH_a}{(H_a - H_s)} = \frac{h_c\, \eta_s}{cp_h \dot{m}_a}\int_{x=0}^{x=A_e} dAe \qquad (23)$$

$$\frac{H_{a,po} - H_s}{H_{a,eo} - H_s} = e^{\left(\frac{-h_c\, A_e\, \eta_s}{cp_h \dot{m}_a}\right)} \qquad (24)$$

$$H_{a,po} = H_s + (H_{a,eo} - H_s)\left(e^{\left(\frac{-h_c\, A_e\, \eta_s}{cp_h \dot{m}_a}\right)}\right) \qquad (25)$$

the air temperature $T_{a,po}$, can be obtained by integration equation (26)

$$SQ_s = h_c\, A_e(T_{a,eo} - T_s)\eta_s \qquad (26)$$

Also,

$$SQ_s = \dot{m}_a\, cp_h dT_a \qquad (27)$$

$$\dot{m}_a\, cp_h dT_a = h_c\, A_e(T_a - T_s)\eta_s \qquad (28)$$

$$\int_{T_{a,eo}}^{T_{a,po}} \frac{dT_a}{(T_a - T_s)} = \frac{h_c\, \eta_s}{cp_h \dot{m}_a}\int_0^{A_e} dAe \qquad (29)$$

Equation (27) is inetegrated by assuming that the temperature of the sateurated air at the wall tube temperature is constant. The solution for temperature becomes:

$$\frac{T_{a,po} - T_s}{T_{a,eo} - T_s} = e^{\left(\frac{-h_c\, A_e\, \eta_s}{cp_h \dot{m}_a}\right)} \qquad (30)$$

$$T_{a,po} = T_s + (T_{a,eo} - T_s)\left(e^{\left(\frac{-h_c\, A_e\, \eta_s}{cp_h \dot{m}_a}\right)}\right)$$

(31)

The value of the outlet humidity ($\omega_{a,po}$) can be found follows as:

$$SQ_l = SQ_t + SQ_s = \dot{m}_a(H_{a,eo} - H_{a,po}) - cp_h \dot{m}_a(T_{a,eo} - T_{a,po}) \qquad (32)$$

$$\omega_{a,po} = \omega_{a,eo} - \left[\frac{SQ_l}{\dot{m}_a\, h_{fg}}\right] \qquad (33)$$

The removal of moisture occurs when the air is cooled down below its dew point temperature, the production of water can be founed as follows **[11]**:

$$\dot{m}_f = \dot{m}_a(\omega_{a,eo} - \omega_{a,po}) \qquad (34)$$

The total surface efficiency of the fin $\eta_s$ can be descript as below **[21]**:

$$\eta_f = \frac{tanhtanh\,(\Omega\, N_e\, \xi)}{\Omega\, N_e\, \xi} \qquad (35)$$

$$\eta_s = \left[1 - \frac{A_f}{A_o}(1 - \eta_f)\right] \qquad (36)$$

## 4. Experimental Work

The experimental apparatus and the procedure to get the experimental data is described in this work. The details of the device apparatus with sketches and photographic in addition to the tools that used for the measurement.


Fig. (5) Photograph illustrates of the hybrid system.

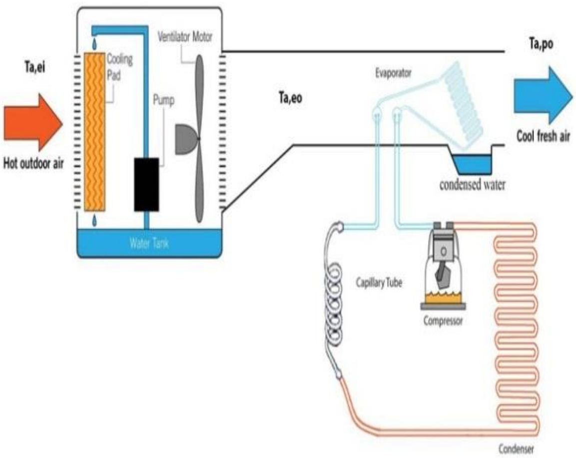



Fig. (6) Schematic diagram of apparatus working.

## 4.1 Experimental Procedure

**Case 1: Direct evaporative air cooler without refrigeration system**

Firstly, the evaporative air cooler was used without refrigeration system, and the following measurements were taken :

1. The air temperature and relative humidity of inlet and outlet are measured using four digital thermometers and hygrometers.
2. The air velocity is measured by using anemometer, at three points upper , mid and bottom of the duct.
3. A four prop digital thermometer was used to measure the temperatures of the outlet air.
4. A Clamp meter was used to measure the electric current of the system.

**Case 2 : Direct evaporative air cooler with refrigeration system**

The evaporative air cooler is used with the refrigeration system and the following measurements were taken:-

1. Air temperature and relative humidity of the inlet and outlet are measured using four digital of thermometer and hygrometer.
2. A four prop digital thermometer was used to measure the temperatures of the outlet air.
3. One digital infrared thermometer used to measure the surface temperature of the evaporator coil.
4. The condenser and evaporator pressures measured by using two pressure gauges, first one was fixed at the beginning of the condenser to measure the high pressure, and the second was fixed at the end of the evaporator to measure the low pressure.

**Table (1)** Specifications of the evaporative air cooler system

<table>
<tr><th>NO.</th><th>Component</th><th colspan="3">Specification</th></tr>
<tr><td>1</td><td>evaporative air cooler</td><td colspan="3">2061cfm,<br>0.72m length, 0.72m width, 0.85m height</td></tr>
<tr><td rowspan="3">2</td><td rowspan="3">Blower motor (two speed)</td><td colspan="3">220v, 50Hz , 1Ph</td></tr>
<tr><td>rpm</td><td>1280</td><td>1240</td></tr>
<tr><td>Current</td><td>1.6 A</td><td>1.1 A</td></tr>
<tr><td>3</td><td>blower</td><td colspan="3">0.25m diameter, and 0.25m length</td></tr>
<tr><td>4</td><td>water pump</td><td colspan="3">AC , 220 v, 50 Hz, 1/60Hp, 0.5 A, 2400 rpm<br>flow rate 20 L/min, 2 m max of head</td></tr>
<tr><td>5</td><td>Pad material</td><td colspan="3">0.5m width, 0.62m height, 0.07m thickness</td></tr>
<tr><td>6</td><td>water basin</td><td colspan="3">0.72m length, 0.72m width, 0.09 m height</td></tr>
</table>

**Table (2)** The basic specifications of the vapour compression refrigeration unit:

| NO. | Component | Specification |
|---|---|---|
| 1 | Compressor | 1/3HP, AC 220 volts, 50 Hz, R134a |
| 2 | Condenser | 2 row, 40 tube, 31.5cm length, 0.9cm diameter |
| 3 | Capillary tube | 1.5m length, |
| 4 | Evaporator | 4.41m length, 0.9 diameter |
| 5 | Fan of condenser | 1500rpm, AC 220-volts, 50 Hz, 0.2A |

## 5. Results and Discussion

The results of theoretical and experimental study for four months (May, June, July and August). The study consists the effect of many parameters on the effectiveness of the evaporative air cooler such as wetted pad thickness, relative humidity, inlet air temperature velocity of frontal air, and evaporator surface temperature. All calculations are performed using Matlab (9.0.2) programs.
The steady includes two cases as follows:

| | |
|---|---|
| **Case 1** | without refrigeration unit |
| **Case 2** | with refrigeration unit |

## 5.1 Theoretical Study

Fig. (7) indicates the variation of effectiveness ($\varepsilon 1$) with wetted pad thickness ($\delta$) at one value of inlet relative humidity ($RH_{a,ei} = 0.3$), in case1 for the theoretical study. Its noted that the effectiveness increases while increasing the wetted pad thickness, because the outlet air temperature ($T_{a,eo}$) decreases with the increase in wetted pad thickness due to increase in the contact time between the air and the wetted pad. The effectiveness at (1.5 m/s), (33 ℃) inlet temperature and (0.05 m) wetted pad thickness, is found to be (0.73).

Fig. (8) shows the effect of wetted pad thickness ($\delta$) on the production of water ($m_f$) at different values of relative humidity ($RH_{a,ei}$) (0.3, 0.35, 0.4), in case 2 for the theoretical study. It can be seen that the production of water increase with the increase the wetted pad thickness (dx). because increase the contact time between the air and the wetted pad. and when the relative humidity increases the productivity increases. Due to increase in specific humidity of air.

Fig. (9) shows the effect of wetted pad thickness ($\delta$) on outlet temperature ($T_{a,eo}$) at different values of frontal air velocity ($u$) (0.5 - 1.5 - 2.5 m/s), in case 1 for the theoretical study. Its noted that the outlet temperature decreases with the increase in the wetted pad thickness, and also decreases with the decrease in the frontal air velocity. because of lower velocity increase the contact time between the air and the wet pad that leads to dicrease in outlet temperature. The outlet air temperature reach a steady state at pad thickness (30cm). It seen that the temperatures are closed to the saturation temperature.

Fig. (10) shows the effect of velocity of frontal air (u) on production of water ($m_f$) at different values of relative humidity ($RH_{a,ei}$) (0.30, 0.35), in case 2 for the theoretical study. Its noted that the production of water increases with the increase in relative humidity, also it increases with decrease in velocity of frontal air. Because the mass flow rate on evaporator coil is high, that leads to increase in temperature of evaporator coil surface that lead to decrease in productivity. The productivity is found to be (0.122 kg/hr.) at (0.3) relative humidity, (1 m/s) velocity of frontal air. While at (0.35) at the same velocity of frontal air is found to be (0.228 kg/hr.).

Fig. (11) shows the effect of inlet humidity ratio of moist air ($\omega_{a,ei}$) on production of water ($m_f$) at different values of frontal air velocity (u) (1- 1.5- 2- 2.5 m/s), in case 2, for the theoretically study. Its noted that The production of water increases with the increase in humidity ratio of moist air, also it increases with the decrease in velocity of frontal air. This can be attributed to the increase of air relative humidity with increasing inlet humidity ratio of moist air at constant ($T_{a,ei}$). The productivity is found to be (0.51 kg/hr) at (0.014) humidity ratio of moist air and (2.5 m/s) frontal air velocity. While is found to be (1 kg/hr.) at (2.5 m/s) frontal air velocity and (0.02) humidity ratio of moist air.

Fig. (12) shows the influences of inlet temperature ($T_{a,ei}$) on the outlet temperature ($T_{a,eo}$) at different values of relative humidity ($RH_{a,ei}$) (0.15, 0.2, 0.25, 0.3), in case 1 for the theoretical study. Its noted that for a given constant inlet temperature of (40℃) the outlet is (27.7℃) at 0.2 relative humidity while it became (29.8) when the relative humidity was increased to (0.3). Hence at higher relative humidity no considerable cooling effect can be expected from direct evaporative cooling.

Fig. (13) show the influences of inlet temperature ($T_{a,ei}$) on outlet temperature ($T_{a,po}$) at different values of relative humidity ($RH_{a,ei}$) (0.15, 0.2, 0.25, 0.3), in case 2 for the theoretical study. Its noted that for a given constant inlet temperature of (40℃) the outlet is (26℃) at (0.2) relative humidity while it became (27.7℃) when the relative humidity was at (0.2) in figure (12). The reason of the decrease in the outlet temperature can be attributed to the high cooling of the evaporator coil.

Fig. (14) shows the influences of inlet temperature ($T_{a,ei}$) on outlet relative humidity ($RH_{a,eo}$) at different values of frontal velocity (u) (1 - 1.5 - 2 - 2.5 m/s), in case 1 for the theoretical study. Its noted that the outlet relative humidity decreases with the increase in inlet temperature, also it increases with the decrease in the frontal velocity of air. This is due to higher contact time between air and the wetted pad which, in turn, increase the outlet relative humidity. The outlet relative humidity is found to be (0.604) at (1.5 m/s) velocity of frontal air and (40 ℃) inlet temperature. While at (2 m/s) in same inlet temperature is found to be (0.5804).

Fig. (15) show the influences of inlet temperature $(T_{a,ei})$ on outlet relative humidity $(RH_{a,po})$ at different values of frontal velocity (u) (1 - 1.5 - 2 - 2.5 m/s), in case 2 for the theoretical study. Its noted that the outlet relative humidity decreases with the increase in inlet temperature, also it increases with the decrease in the frontal velocity of air. This is due to higher contact time between air and the wetted pad which, in turn, increase the outlet relative humidity. The outlet relative humidity is found to be (0.654) at (1.5 m/s) velocity of frontal air and (40 ℃) inlet temperature. While in same condition the outlet relative humidity is found to be (0.604) shown figure (14). The reason of the increase in the outlet relative humidity is due to the productivity process.

## 5.2 Experimental study

Fig. (16) shows effect of velocity of frontal air on production of water $(m_f)$ at (36.75℃) inlet temperature $(T_{a,ei})$ and (0.239) inlet relative humidity $(RH_{a,ei})$ for both experimental and theoretical study, case 2. It is noted that the production of water decreases with increase in the velocity of frontal air because of increasing the surface temperature of the evaporator coil. The productivity for the theoretical and experimental study at velocity of (2 m/s) is respectively found to be (0.0588 kg/hr.) (0.072 kg/hr.). At (3 m/s) frontal air velocity for theoretical and experimental study the productivity is found to be (0.0229 kg/hr.) and (0.048 kg/hr.) respectively.

Fig. (17) shows the influences of velocity of frontal air (u) on outlet relative humidity $(RH_{a,eo})$ at (36.75℃) inlet temperature $(T_{a,ei})$ and (0.239) inlet relative humidity $(RH_{a,ei})$ for both experimental and theoretical, case 1. Its noted that the outlet relative humidity decreases with the increase in velocity of frontal air. Because the contact time between the air and the wetted pad is lower that leads to decrease in the outlet relative humidity. The outlet relative humidity is found to be (0.638) at (1.5 m/s) velocity of frontal air for theoretical study . While in same the condition the outlet relative humidity is found to be (0.682) for experimental study. Although at (3 m/s) velocity of frontal air for theoretical and experimental study is found to be (0.595) and (0.642) respectively.

Fig. (18) shows the influences of velocity of frontal air (u) on outlet relative humidity $(RH_{a,po})$ at (36.75℃) inlet temperature $(T_{a,ei})$ and (0.239) inlet relative humidity $(RH_{a,ei})$ for both experimental and theoretical, case 2. Its noted that the outlet relative humidity decreases with the increase in velocity of frontal air. This is due to lower contact time between air and the wetted pad which, in turn, decrease the outlet relative humidity. The outlet relative humidity is found to be (0.684) at (1.5 m/s) velocity of frontal air for theoretical study . While the outlet relative humidity is found to be (0.741) for experimental study. whereas at figure (17) at the same conditions is found to be (0.638) for theoretical study and (0.682) for experimental study. The reason of increase in the outlet relative humidity due to productivity process.

Fig. (19) shows the effect of frontal air velocity (u) on outlet temperature $(T_{a,eo})$ at (36.75℃) inlet temperature $(T_{a,ei})$ and (0.239) inlet relative humidity $(RH_{a,ei})$ for both experimental and theoretical, case 1. Its noted that the outlet temperature increases with the increase in velocity of frontal air. Because the contact time between the air and the wetted pad is fast that leads to increase in the outlet temperature. The outlet temperature is found to be (26.10℃) at (1.5 m/s) velocity of frontal air for theoretical study. While at the same conditions the outlet temperature it found (25.9 C) for experimental study. Whereas at (3 m/s) velocity of frontal air for theoretical and experimental study is found to be (26.901℃) and (26.7℃) respectively.

Fig. (20) shows the effect of frontal air velocity (u) on outlet temperature $(T_{a,po})$ at (36.75℃) inlet temperature $(T_{a,ei})$ and (0.239) inlet relative humidity $(RH_{a,ei})$ for both experimental and theoretical, case 2. Its noted that the outlet temperature increases with increase in velocity of frontal air. Because the contact time between the air and the wet pad is fast that leads to increase in the outlet temperature. The outlet temperature is found to be (24.68 C) at (1.5 m/s) velocity of frontal air for theoretical study. While at same the conditions the outlet temperature is found to be (24.1℃) for experimental study. whereas in figure (19) at the same conditions is found to be (26.10℃) for theoretical study and (25.9℃) for experimental study. The reason of decreases in the outlet temperature is due to the high cooling of the evaporator coil.

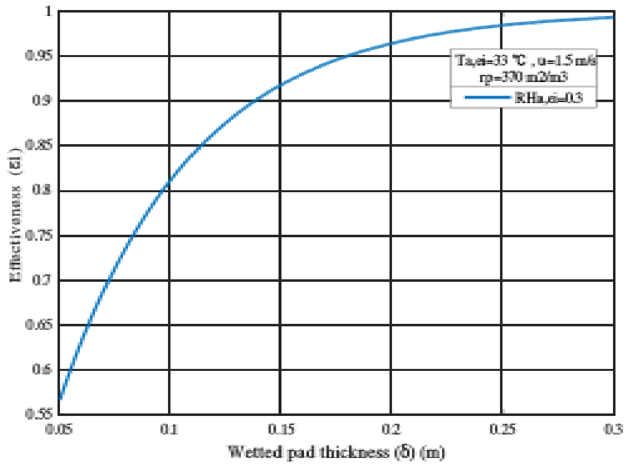


**Fig. (7)** Effect of wetted pad thickness on effectiveness, case 1.

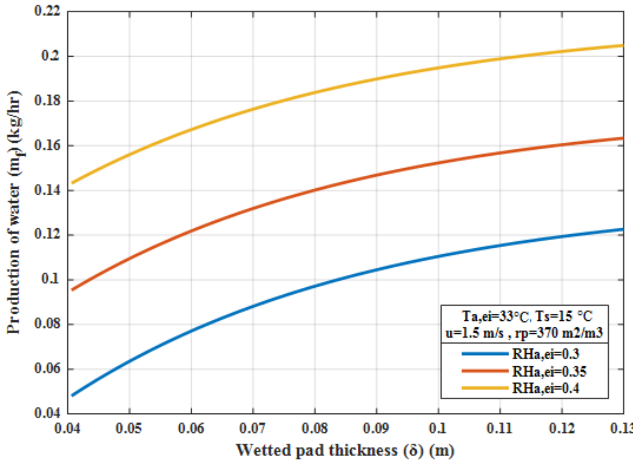



**Fig. (8)** Effect of wetted pad thickness ($\delta$) on production water at different values of relative humidity, case 2.

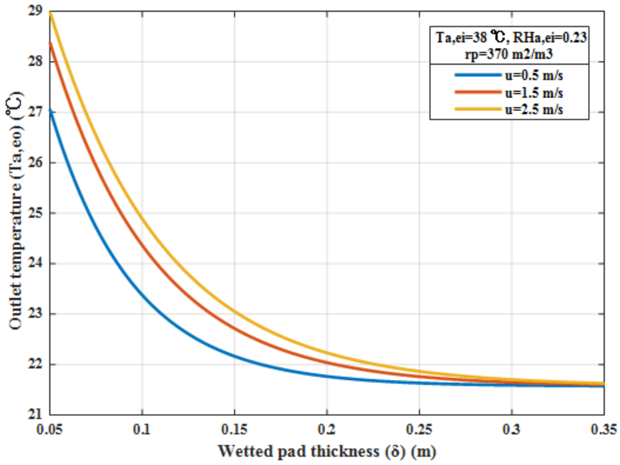



**Fig. (9)** Effect of wetted pad thickness ($\delta$) on outlet temperature ($T_{a,eo}$) at different values of frontal air velocity ($u$), case 1.

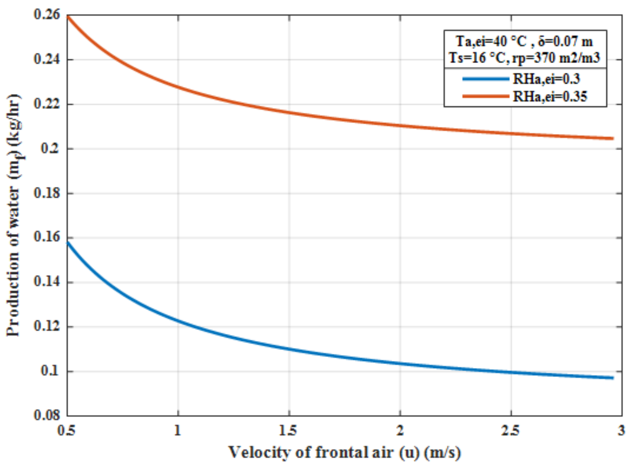



**Fig. (10)** Variation of production water with ( $u$) at different values of ($RH_{a,ei}$), case 2

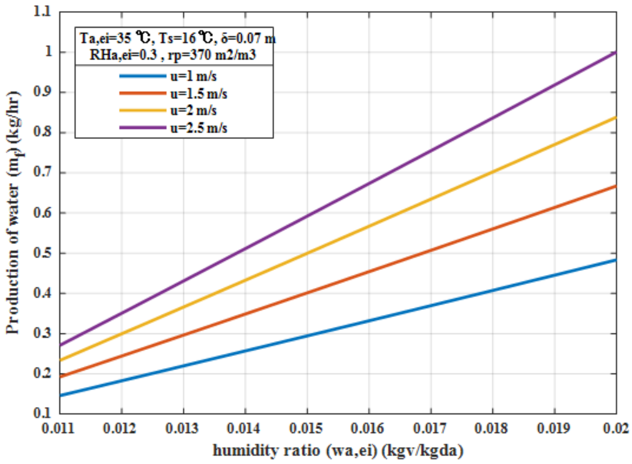



**Fig. (11)** Effect of ($\omega_{a,ei}$) on production of water at different values of frontal air velocity (u), case 2.

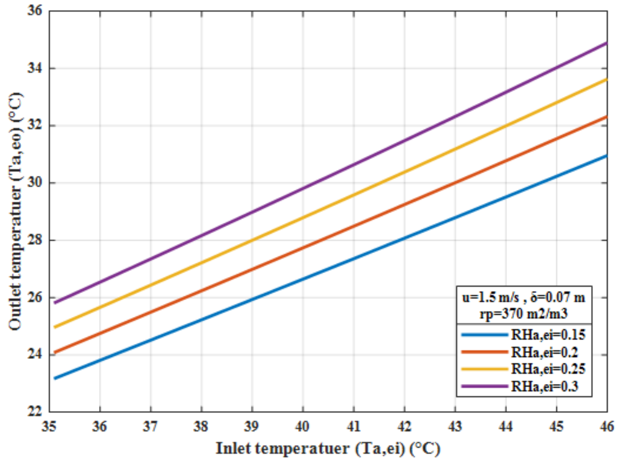



**Fig. (12)** Variation of ($T_{a,eo}$) with ($T_{a,ei}$) at different values of ($RH_{a,ei}$), case 1.

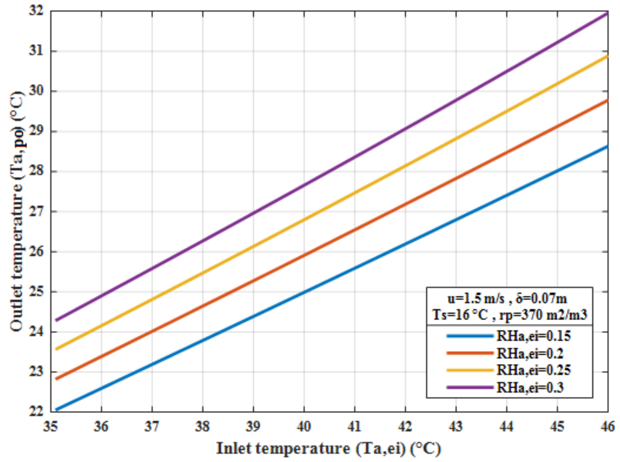



**Fig. (13)** Variation of ($T_{a,po}$) with ($T_{a,ei}$) at different values of ($RH_{a,ei}$), case 2.

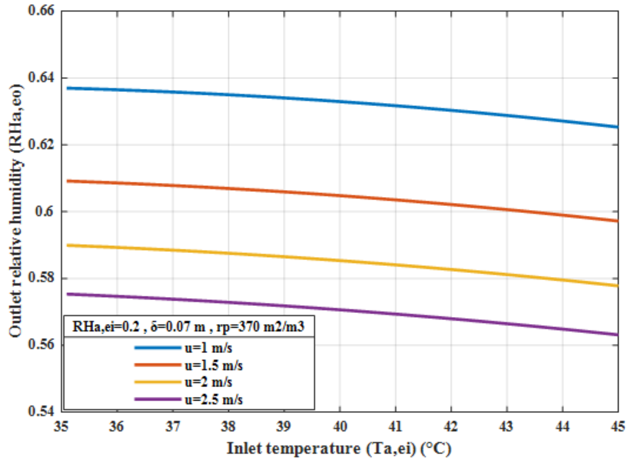



**Fig. (14)** Influences of inlet temperature on outlet relative humidity at different values of frontal velocity, case 1.

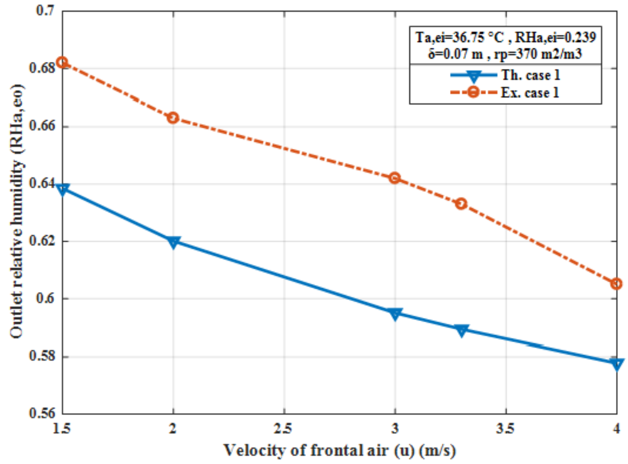



**Fig. (17)** Influences of (u) on outlet relative humidity ( $RH_{a,eo}$) of at both experimental and theoretical, case 1.

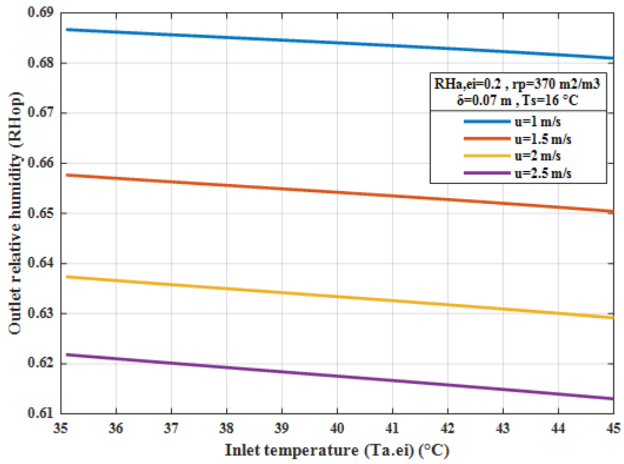



**Fig. (15)** Influences of inlet temperature on outlet relative humidity at different values of frontal velocity, case 2.

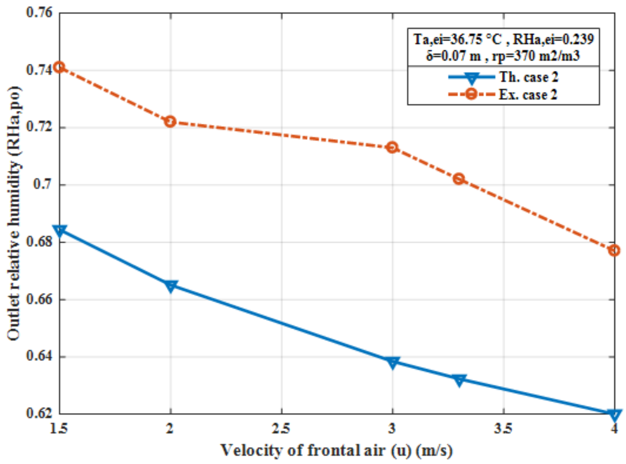



**Fig. (18)** Influences of (u) on outlet relative humidity ( $RH_{a,po}$) of at both experimental and theoretical, case 2.

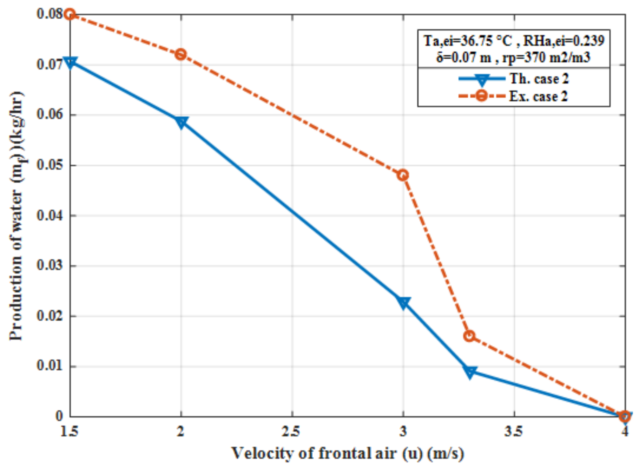



**Fig.(16)** Influences of (u) on production of water at both experimental and theoretical, case 2.

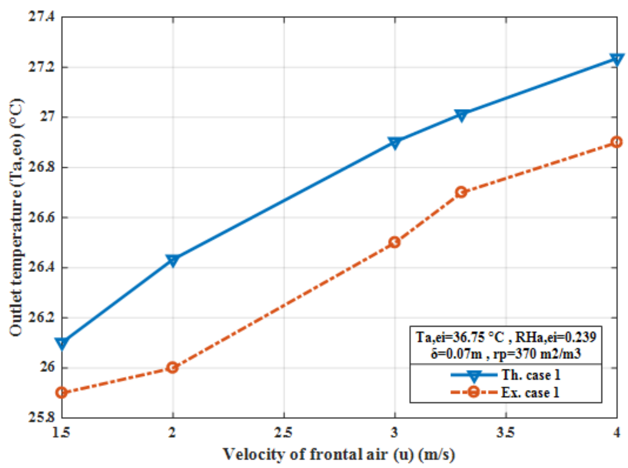



**Fig. (19)** Influences of (u) on outlet temperature ($T_{a,eo}$) at both experimental and theoretical, case 1.

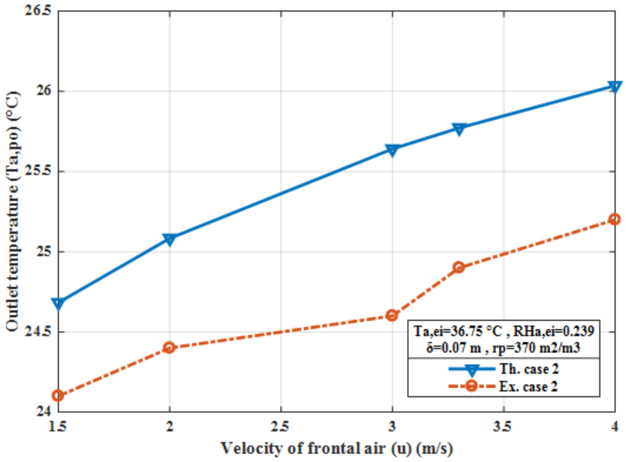



**Fig. (20)** Influences of (u) on outlet temperature ($T_{a,po}$) at both experimental and theoretical, case 2.

## 6. Conclusion

### 6.1 Theoretical Study

1. Effectiveness of the evaporative air cooler increase with increase wetted pad thickness.
2. The optimum thickness of wetted pad was 30cm.
3. The productivity increases with the increase of wetted pad thickness at constant evaporator coil temperature. When the wetted pad thickness (0.07m) and (0.12m) and the inlet relative humidity (0.3) the productivity was (0.088 kg/hr.) (0.12 kg/hr.) respectively.
4. The productivity increases with increasing inlet relative humidity and decreasing the frontal air velocity.
5. The productivity increases with the increase of humidity ratio. It was (0.51 kg/hr.) and (1 kg/hr.) at humidity ratio (0.014) and (0.02) respectively.
6. Using refrigeration unit decreases the outlet temperature by about (0.5_1.7°C°C), and decrease the humidity ratio by (0.0004-0.0008)

### 6.2 Experimental study

1. The productivity decreases with increase of the frontal air velocity because the evaporator coil temperature increases.
2. The maximum productivity was found to be (0.32 kg/hr.) at (43 C) inlet temperature and (3 m/s) frontal air velocity.
3. The productivity increases with the increase of inlet temperature. It was (0.2 kg/hr.) and (0.3 kg/hr.) at inlet temperature (35.5°C) and (43°C) respectively.
4. Using refrigeration unit decreases the outlet temperature by about (1.5_3°C), and decrease the humidity ratio by (0.0006-0.001).

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