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A Study on Thermal Effectiveness of Multi-Stages Evaporative Air Cooling

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Abstract

The air conditioning system performance is significantly affected by temperature rise which causes continuous increase in electricity consumption and pollution problems to environment. Evaporative cooling systems are characterized by their low energy consumption so that they represent successful potential alternatives to traditional vapor compression air conditioning systems. This study investigates the performance of multi-stages evaporative cooling systems experimentally and theoretically. The experimental set-up is mainly composed of two parts: indirect unit to decrease the air temperature and direct unit to moisturize the air. The system is installed and equipped with temperatures, humidity, and air velocity sensors. The experimental tests were run continuously to monitor the system performance at various weather conditions between 35 ºC to 50 ºC in June and July months. A mathematical model for the system components was developed and implemented in the Engineering Equation Solver (EES) program to simulate the performance of multi-stages evaporative cooling systems. The results showed that the heat flux Q_{flux} increases with the increase in the Reynolds number Re of inlet air, velocity fraction V_{AF} extracted air for sensible cooling, air temperature at the product-in $T_{ap,i}$, air velocity at the product-in $V_{ap,i}$, and the adiabatic efficiency Ef_{ad}. But, it is decreasing with increasing the spacing between the heat exchanger plates S and the relative humidity at the product-in $RH_{ap,i}$. Optimum performance was obtained with very small space between plates which was bout 5mm. Good agreement have been shown between experimental and predicted data, where the results. Uncertainty of experimental data was within the range 4.14 to 6.15.

Keywords: Direct evaporative cooling, indirect evaporative cooling; multi-stages.

الخلاصة: يتأثر أداء منظومة تكييف الهواء بشكل كبير بارتفاع درجات الحرارة والذي يسبب زيادة مستمرة في استهلاك الكهرباء ومشاكل التلوث البيئي. تتميز أنظمة التبريد التبخرية باستهلاكها القليل للطاقة بحيث تمثل كبدائل ناجحة لأنظمة تكييف الهواء التقليدية. ومع ذلك ، يتأثر نظام التبريد التبخيري بشدة بالأحوال الجوية التي تؤدي إلى تدهور أدائها. في الآونة الأخيرة ، برز التبخير المتعدد المراحل كحل واعد لهذه المشكلة. تبحث هذه الدراسة في أداء أنظمة التبريد التبخيري المتعددة المراحل عمليا ونظريا. مبدئيا، يتكون التبريد التبخيري متعدد المراحل من مرحلتين لتبريد الهواء والتي تتسم بالتبريد المحسوس والكامن. في التبريد المحسوس، يتم انتقال الحرارة بصورة غير مباشر باستخدام مبادل حراري مطلي دون إضافة الرطوبة. بعد التبريد المباشر، يتم ترطيب الهواء باستخدام مبرد تبخير تقليدي حيث يتم تبريد الهواء. يتم استخراج جزء من الهواء المبرد والمرطّب وإعادته ليتم استخدامه للتبريد غير المباشر. يتكون الجزء العملي بشكل أساسي من جزئين: وحدة غير مباشرة لخفض درجة حرارة الهواء ووحدة مباشرة لترطيب الهواء. يستخدم صمام للتحكم في تدفق الهواء في النظام. تم تثبيت النظام ومجهز درجات الحرارة والرطوبة وأجهزة استشعار سرعة الهواء.تم تشغيل الاختبارات التجريبية بشكل مستمر لمراقبة أداء المنظومة في مختلف الظروف الجوية بين ٣٥ إلى ٥٠ درجة سليليزية في شهري حزيران وتموز. تم استخدام برنامج لحل المعادلات الهندسية (EES) لمحاكاة أداء منظومة التبريد التبخيري المتعدد المراحل.كما تم الأخذ بنظر الاعتبار قياس سرعة الهواء ودرجة الحرارة والرطوبة للهواء الداخل. علاوة على ذلك ، تم دراسة تأثير المسافة الفاصلة بين لوحات المبادل الحراري على اداء المنظومة.

1. INTRODUCTION

Evaporative cooling is an environmentally friendly and energy efficient method for cooling buildings in hot and dry regions. Iraq climate country demands a variety of cooling systems to achieve optimized energy consumption, reduce emission, and provide summer comfort condition. Evaporative cooling is the process by which the temperature of a substance is reduced due to the cooling effect from the evaporation of water. Evaporative cooling system design is directly affected by dry bulb, wet bulb temperature, and relative humidity. Two principle methods of evaporative cooling are commonly used, the direct (DEC) and the indirect one (IEC). DEC is the simplest, oldest, and the most widespread form of evaporative air conditioning. This system typically uses a fan to draw hot outdoor air into a porous wetted medium. The water absorbs heat as it evaporates from the porous wetting medium, and the air thus leaves the DEC at a lower temperature. In fact, dry bulb temperature of the air reduces as it is moistened in this adiabatic saturation process. The principle underlying DEC is the conversion of sensible heat to latent heat. The wetted medium could be a porous wetted pad consisting of fibers, cellulose papers or a spray of water. In Indirect evaporative cooling IEC systems reduced temperature without raising indoor humidity. In indirect evaporative process air moisture content stays constant during temperature decreasing. As cooling of the outdoor air stream takes place by heat transfer across the heat exchanger walls, the outdoor air stream becomes cooler without an increase in its humidity. This process is called indirect and is mainly used in those applications where no humidity addition is allowed in the supply air, as well as no risks of pollution, as no mass exchange is permitted between the two air streams.

Two-stage indirect/direct evaporative coolers as showed in Figure 1 can cool air to lower temperatures than direct ("one-stage") evaporative coolers. In these coolers, a first-stage indirect evaporative cooler lowers both the dry bulb temperature (DBT) and wet bulb temperature (WBT) of the incoming air. After leaving the indirect stage, the supply air passes through a second stage direct evaporative cooler. In the direct evaporative cooler the air is exposed to water directly which allows it to cool and then air gets moist by losing its sensible heat to latent heat of vaporization of water.

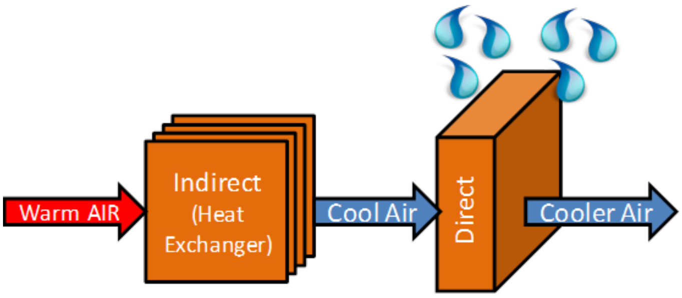

Figure 1: Indirect-direct evaporative cooling

Franco et al. [1] shows the evaporative cooling (EC) systems depend on the evaporation of water inside the greenhouse, delivering higher humidity and lower temperature. The change from fluid to vapor needs energy that is separated from the greenhouse air, cooling it and expanding its humidity. This realizes a change from sensitive heat to latent heat. They recommend a range of air velocities through the pad of 1 to $1.5\ \mathrm{m/s}$, at which the pressure drop was somewhere in the range of 3.9 and 11.25 Pa, upon the sort of pad and the water flow applied. Rogdakis et al. [2] investigated a comprehensive mathematical model based on sensible and latent heat transfer. The model predicts the efficiency of evaporative cooler under variable water supply condition and operation parameters. It was concluded that cooler water further reduces the temperature drop inside the cooler. G. Gohane et al. [3] studied two stages evaporative cooling known as "Indirect and direct "cooler. Here, in this cooling system, incoming hot air is first passed through the cooling pad that is heat exchanger when temperature of air to be decreases and this air is again passing through the cooling medium and then evaporated into the atmosphere. The results shown that the most advantage of is to be save water in 5or 6 days. It is energy consumption. Higher ventilation rates also help to lower the indoor temperature. The operational performance and effect variable of a counter-stream regenerative evaporative cooler (REC) was investigated by Duan et al. [4]. This was achieved by a dedicated test process. Temperature, flow rate and humidity of the air flows at the outlet, inlet and exhaust of the cooler were tested under different operational conditions. It shows that the wet-bulb effectiveness of the presented cooler extended from 0.55 to1.06. The wet-bulb effectiveness was fundamentally improved through decreasing intake air velocity and the cooling capacity and energy efficiency ratio (EER) of cooler that was quickly raised by means increasing intake air velocity.

A modern hybrid air-conditioning system consisting of indirect evaporative cooler IEC and mechanical cooling has been investigated by Chen et al. [5] proposed for IEC application in humid and hot places. The IEC of the air-conditioning system is utilized to pre-cool the incoming natural air for energy preservation. In this system, the dry and cool exhaust air from air-conditioned area was utilized as secondary air. But, because of the high humidity of the fresh air in humid places, condensation most likely occurs, that results in not only sensible cooling as well as dehumidification impact. Duan et al. [6] structured, created and experimentally tested a counter-flow regenerative evaporative cooler (REC) in research center. The REC's center mass and heat exchanger was manufactured utilizing stacked sheets made of high wicking evaporation and waterproof aluminium materials. The created REC system has more higher cooling performance compared to conventional IEC. The performance of a counter-flow REC prototype has been studied experimentally under various typical outdoor air conditions of China. The outcomes showed that for every single chosen locate, the REC could decrease 53– 100% of cooling load and 13– 58% of electrical energy consumption yearly. The performance of a system of indirect evaporative cooler with internal baffles studied by Kabeel et al. [7] of indirect evaporative cooler with internal baffles as air precooling unit and evaporative condenser. The test arrangement comprising of an indirect evaporative cooling with internal baffles pursued by an evaporative condenser contains the heated water serpentine pipes with outside thin film cotton layer.

Sharma and Darokar [8] developed the performance of two-stage indirect/direct evaporative cooling system. For this purpose, a two-stage evaporative cooling experimental setup consisting of an indirect evaporative cooling stage (IEC) followed by a direct evaporative cooling stage (DEC) was designed, constructed and tested. To achieve comfort conditions and power saving the system has been investigated with related excess water consumption. The results show that, this system has high potential to provide comfort conditions in regions where at present standalone direct evaporative coolers cannot provide comfort conditions. Despite wide variety of climatic conditions, it is found that IEC effectiveness varies between 55 and 61% and IEC/DEC effectiveness over a range of 108–111%.

2. ADVANTEGES AND DISADVANTEGES OF EVAPORATIVE COOLER

The main advantages of evaporative coolers are their low cost and high effectiveness, permitting a wide range of applications in the buildings, dwellings, commercial and industrial sectors. They can be applied in dry and hot climates. Evaporative air coolers which are used for air-conditioning in hot and dry climates, have considerably low energy consumption compared to refrigerated systems because they do not need any refrigerant. Disadvantage is the water consumption associated to the operation of these systems, which is a scarce resource in dry and hot climates.

3. TEMPERATURE CHANGE AND HEAT CAPACITY

One of the major effects of heat transfer (Q_{flux}) is temperature change: heating increases the temperature while cooling decreases it. We assume that there is no phase change and that no work is done on or by the system.

The Q_{flux} to cause a temperature change depends on the magnitude of the temperature change, the mass of the system, and the substance involved. (a) The amount of heat transferred is directly proportional to the temperature change. To double the temperature change of a mass, you need to add twice the heat. (b) The amount of heat transferred is also directly proportional to the mass. (c) The amount of heat transferred depends on the substance and its phase.

4. HEAT TRANSFER CALCULATION

The aluminium plates are an essential component used in the air condition cycle. It looks like an automobile radiator. It's usually found inside the conditioned space that needs to remove the heat. Cold supply air that exits in air conditioner has just transferred some heat energy to the system as it passes through the aluminium plates. The quantitative relationship between heat transfer and temperature change is given by [10]:

$$Q_{flux} = \rho \, Cp \, V_{ap,i} \, A \, (T_{ap,i} - T_{ap,o}) \tag{1}$$

The value of specific heat must generally be looked up in tables, because there is no simple way to calculate them. The specific heat capacity of air is $\mathbf{1012 \ J/(kg\,K)}$.

5. EFFECTIVENESS

The heat exchanger effectiveness $\boldsymbol{\varepsilon}$ is defined as the ratio of the actual to the maximum possible heat transfer [10]:

$$\varepsilon = \frac{1-\exp(-\mathrm{NTU}\,(1-Cr))}{1-Cr\,\exp(-\mathrm{NTU}\,(1-Cr))} \tag{2}$$

where

$$Cr = \frac{m_{ap}\, Cp_{ap}}{m_{aw}\, Cp_{aw}} \tag{3}$$

6. COEFFECIENT OF PERFORMANCE CALCULATION

The coefficient of performance COP specifies the ratio between the energy transfer of a heating pump and the total input of electrical energy. COP is a dimensionless number that expresses the efficiency of a heating system, a pump with a higher COP transfers more heat with an equivalent energy input.

$$COP = \frac{Q_{flux}}{Power} \tag{4}$$

7. UNCERTAINITY ANALYSIS

The uncertainty of a measured value referrers to the associated interval in which the expected true value should be found. The term "uncertainty" is reserved for the final results.

$$\sigma = \sqrt{\frac{\sum_{i=1}^{N}(x_i - x_m)^2}{N-1}} \tag{5}$$

where $\boldsymbol{x_m}$ is the mean value.

$$\Delta V_{ap,i,r} = \frac{\boldsymbol{t\, \sigma_{V_{ap,i}}}}{\boldsymbol{\sqrt{N}}} \tag{6}$$

$$\Delta T_{ap,o,r} = \frac{\boldsymbol{t\, \sigma_{T_{ap,o}}}}{\boldsymbol{\sqrt{N}}} \tag{7}$$

$$\Delta T_{ap,i,r} = \frac{\boldsymbol{t\, \sigma_{T_{ap,i}}}}{\boldsymbol{\sqrt{N}}} \tag{8}$$

$$\Delta Q_r^2 = (\Delta V_{ap,i,r}\, \frac{\Delta Q_{flux}}{\Delta V_{ap,i}})^2 + (\Delta T_{ap,o,r}\, \frac{\Delta Q_{flux}}{\Delta T_{ap,o}})^2 + (\Delta T_{ap,i,r}\, \frac{\Delta Q_{flux}}{\Delta T_{ap,i}})^2 \tag{9}$$

$$\Delta Q_f^2 = (\Delta V_{ap,i,f}\, \frac{\Delta Q_{flux}}{\Delta V_{ap,i}})^2 + (\Delta T_{ap,o,f}\, \frac{\Delta Q_{flux}}{\Delta T_{ap,o}})^2 + (\Delta T_{ap,i,f}\, \frac{\Delta Q_{flux}}{\Delta T_{ap,i}})^2 \tag{10}$$

$$\Delta Q_{flux} = \sqrt{\Delta Q_r^2 + \Delta Q_f^2} \tag{11}$$

The uncertainty

$$\pm \frac{\Delta Q_{flux}}{Q} * 100\% = \tag{12}$$

8. HEAT EXCHANGER SYSTEM

For this investigation, a counter-flow plated heat exchanger has been structured and created in the research center. The heat-exchanger (HX) model was stacked together using multiple thin aluminium flat plates. Plastic bars of thickness **1 cm** were used to separate the plates and to provide air flow passages. The location of air inlet and outlet for each plate were manipulated to guide air so that the counter flow between ambient (hot) and cold (humid) air can be insured. The whole HX module is divided into 8 groups. Each group contains two plates connected together in the opposite direction as shown in Figure 2.

Figure 2 Aluminium plates (side view) are wrapped together

The heat exchanger (HX) model operates in the following way: the product-in air would be dragged into dry channels from the bottom left side of the HX module by supply air blower as shown in the Figure 3.

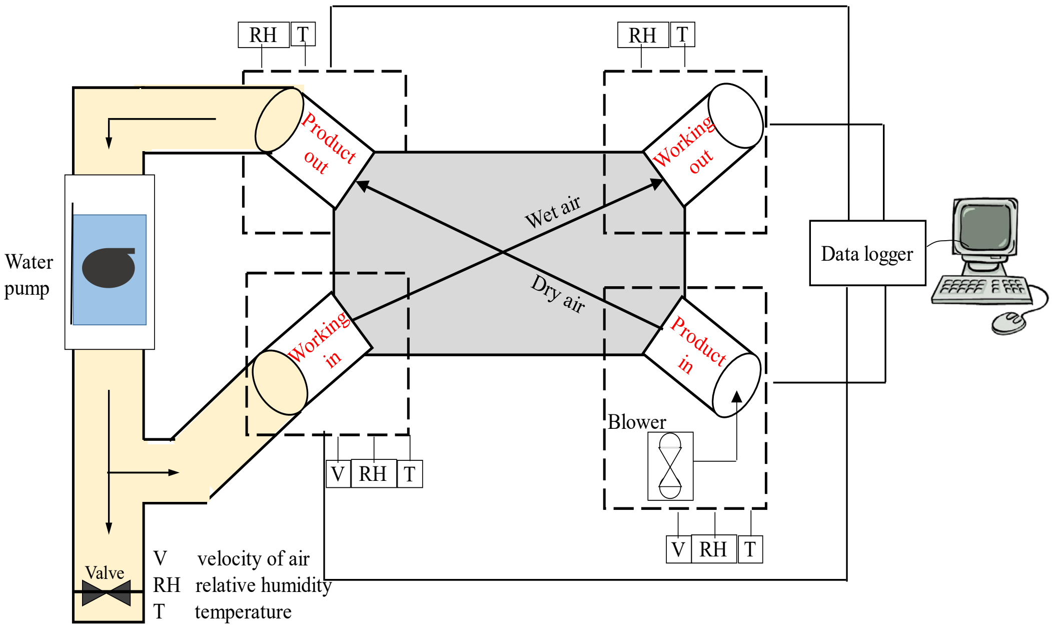

Figure 3 Schematic diagram of the heat exchanger set-up.

To evaluate performance of the proposed counter-flow evaporative cooler operating under various operation conditions, an experiment system was constructed as shown in Figure 4. The air flows through the channel and is divided into two parts at the end of channels. One part of the air stream, i.e. product-out of air, keeps moving at the same direction and is finally delivered to the space where cooling is required, and the other part of the air stream, i.e. working-in of air, is extracted from the product air and humidified though the evaporative cooler as shown in Figure 4.

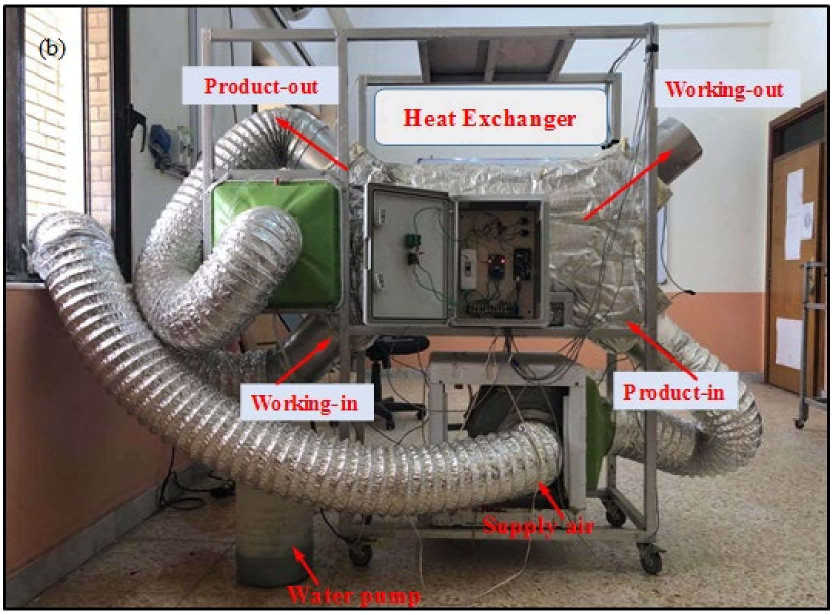

Figure 4 A real image of the experiment set-up.

9. MEASURING DEVICE

Figure 3 indicates the set-up for measuring the experimental parameters, including temperature (T) and relative humidity (RH) of product-in, product-out, working-in and working-out air streams. The air velocity (V) was measured at product-in and working-in streams.

9.1. Temperature and humidity sensors

As a general description, the digital-output temperature and relative humidity sensor/module DHT222 (DHT22 also named as AM2302) is used in the present work and is shown in Figure 5. Four sensors have been placed in four places product-in, product-out, working-in, and working-out.

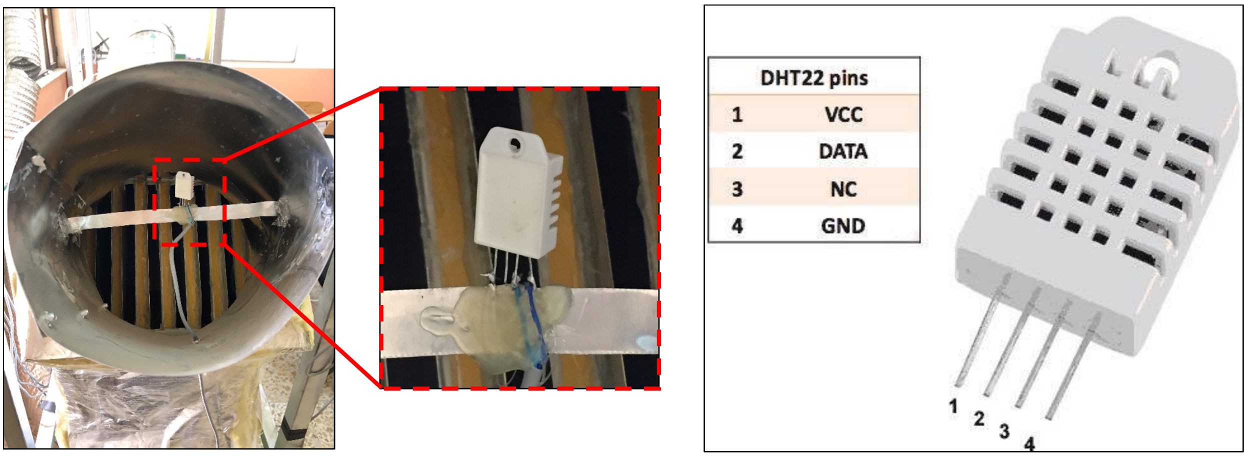

Figure 5 Air temperature and humidity sensor.

9.2.Velocity sensor

The air velocity sensor shown in Figure 6 was used in the present work. It indicates the air velocity in (m/sec). Two devices have been placed in two places: product-in, and working-in.

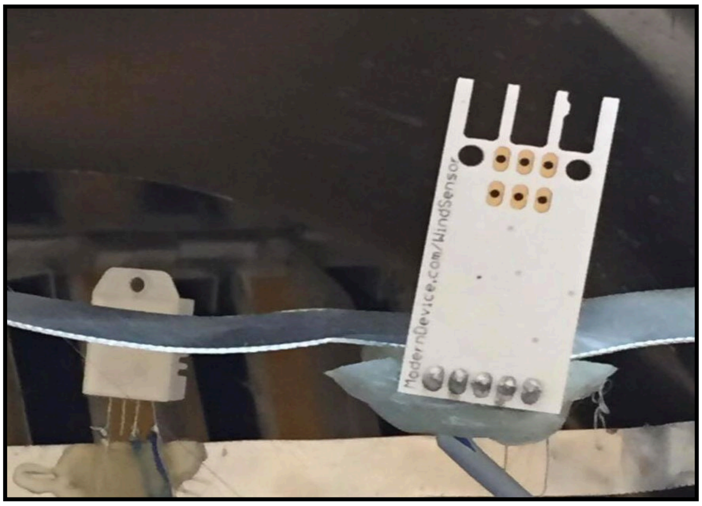

Figure 6 Air velocity sensor

10. EXPERIMENTAL TEST PROCEDURE

In each experimental test, the following were considered and measured:

1. Run the blower and water pump unit.
2. Monitor the air temperatures, relative humidity and velocity variations by the computer as shown in Figure7.

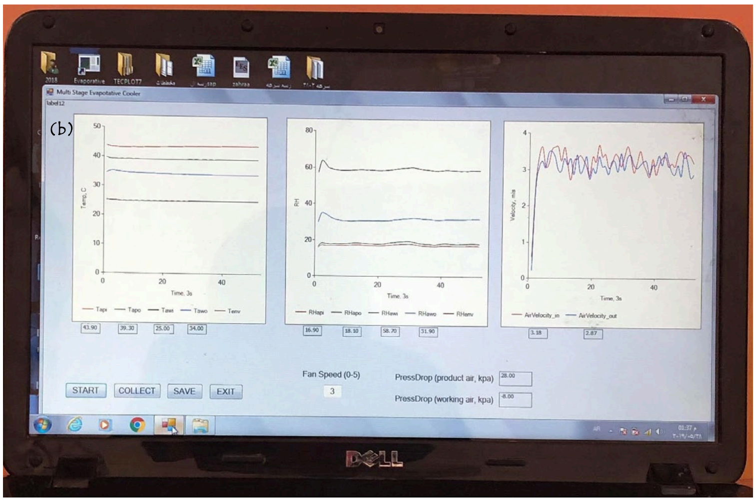

Figure 7 Monitoring the experimental data

3.Monitor the system power by the power meter.
4.After few minutes, if there is no variation in air temperatures, relative humidity, air velocity and the system power readings these readings will be recorded.
5.Read the air temperatures.
6.Read the air relative humidity
7.Read the air velocity.
8.Read the power consumed by the blower and water pump.
9.In the same manner, the cool air duct (outlet air duct) module includes a valve with different holes mounted in a long extension channel to control the released flow rate. The ratio of working-in to product-out airflow rate was varied by the valve.
10.Repeat the above steps starting from (2) to (9).

The experimental data that were recorded from (5) to (8) will be used to calculate the heat transferred to the air, Q_{flux} and the coefficient of performance, COP.

11. RESULTS AND DISCUSSION

Engineering Equation Solver (EES) is utilized to simulate the working as well as performance of direct-indirect evaporative cooling system considering mathematical relations. This software is a commercial software package used for solving non-linear equations. It provides many useful functions and equations for thermodynamics and heat transfer problems. A major feature of EES is the high accuracy and transport property database. In the following, the influence of several operating and design parameters on the system performance is presented.

This section concentrates on predicted data obtained from analytical analysis and experimental results. The results of the heat flux, the effectiveness of the heat exchanger, the sensible cooling, and latent cooling are considered to assess the system performance.

Figure 8 (a-d) demonstrate the relation between the heat flux (Q_{flux}), effectiveness of heat exchanger Ɛ, temperature difference ($t_{ap,i} - t_{ap,o}$) and temperature difference ($t_{ap,i} - t_{aw,i}$), respectively, as a function to the space (S) for different Reynolds number (Re) (362, 731, 1102, 1473) which correspond to value ($T_{ap,i} = 45C^o$, $RH_{ap,i} = 0.2$, $V_{ap,i} = 4\ m/s$, $Ef_{ad} = 0.95$) for all data, where $T_{ap,i}$ is the temperature measured at the product in duct, $RH_{ap,i}$ is the relative humidity measured at the product in duct, $V_{ap,i}$ is the air velocity measured at the product in duct, and Ef_{ad} is the adiabatic efficiency.

Figure 8a shows that Q_{flux} (which is calculated by Eq. (1)) decreases with the increase in the space between plates (S) and with the decrease of Re. in addition, at constant volumetric air flow fraction V_{AF}($V_{AF} = \frac{V_{aw,i}}{V_{ap,i}}$), the velocity of air ($V_{ap,i}$) is reduced when the space between the plates becomes larger and hence the amount of air becomes smaller which leads to reduction in the heat flux.

Figure [8b] shows that the heat exchanger effectiveness (Ɛ) decreases with the increase of spacing S because of the decrease in the heat flux Q_{flux}. Also, Ɛ is inversely proportional to Re of the working side. When the V_{AF} increases, the capacity ratio (C_r) decreases according to Eq. (3), so that will decrease the effectiveness Ɛ as you see in Eq. (2). Figure [8c] shows that the degree of sensible cooling ($t_{ap,i} - t_{ap,o}$) is directly proportional to Re because of the increases in the heat flux as you see in Figure 6a. In addition, the influence of S effects the temperature difference ($t_{ap,i} - t_{ap,o}$) much larger for values smaller than 0.02m when the effectiveness of heat exchanger is relatively high, as shown in Figure[6b].

Figure 8d shows that the degree of latent cooling $\left(t_{ap,i} - t_{aw,i}\right)$ is directly proportional to Re because of the increases in the heat flux. The interesting thing is that the temperature difference ($t_{ap,i} - t_{aw,i}$) in Figure 8d is higher than the temperature difference ($t_{ap,i} - t_{ap,o}$) in Figure 8c that is because of the air after it leaves the product-out it will go to the water supply (see Figure 3 for more clarifications) that decreases the air temperature more and that air will go the working-in with less temperature. Note that the $t_{ap,i}$ is constant.

Figure 9 (a-b) demonstrates the relation between the heat flux Q_{flux}, effectiveness of heat exchanger Ɛ, temperature difference ($t_{ap,i} - t_{ap,o}$) and temperature difference ($t_{ap,i} - t_{aw,i}$), respectively, as a function to the air velocity fraction $V_{AF}\left(V_{AF} = \frac{V_{aw,i}}{V_{ap,i}}\right)$ which correspond to value ($T_{ap,i} = 45\text{C}^o$, $RH_{ap,i} = 0.2$, $V_{ap,i} = 4\ \text{m/s}$) for all data. Figure (9a) shows that the $Q_{flux}$is increasing along with the V_{AF}. Because of keeping the air velocity in product-in $V_{ap,i}$constant and increasing $V_{aw,i}$, the heat flux inside the heat exchanger will increase. Also, when the adiabatic efficiency Ef_{ad} increasing the heat flux Q_{flux} is increasing too that is because of the humidity is increasing that will decrease the air temperature toward increasing the heat flux.

Figure 9b shows that the Ɛ of the heat exchanger is decreasing with increasing V_{AF}because of when the V_{AF} increases, the mass transfer in the working-in m_{aw} increases, too. That will decrease the Cr as it shown in Eq. (3). As Cr decreases, the effectiveness (Ɛ) is decreasing too. However, the effectiveness (Ɛ) is not affecting by Ef_{ad} that is because of there is no relation between them.

Figure 9c shows that when the $V_{AF}$is increasing, the temperature difference ($t_{ap,i} - t_{ap,o}$) is increasing because of the increase in the air velocity at the working-in that decrease the air temperature in the product-out. So, the temperature difference ($t_{ap,i} - t_{ap,o}$) is increasing along with the V_{AF}. Figure 9d shows that when the $V_{AF}$is increasing, the temperature difference ($t_{ap,i} - t_{aw,i}$) is increasing that is because of increasing of the air velocity in the working-in $V_{aw,i}$.

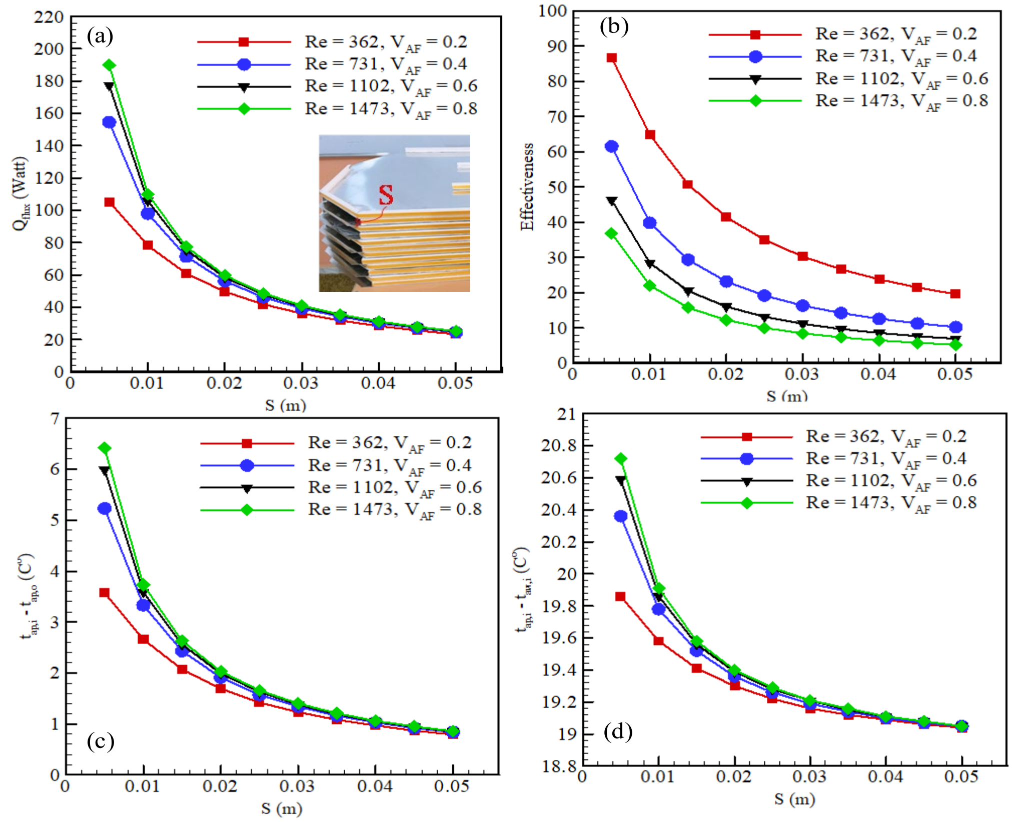

Figure 8 (a) Heat flux Q_{flux} (b) effectiveness of heat exchanger Ɛ (c) temperature difference $(t_{ap,i} - t_{ap,o})$ (d) temperature difference $(t_{ap,i} - t_{aw,i})$ as a function to the spacing between the plates S and Reynolds number.

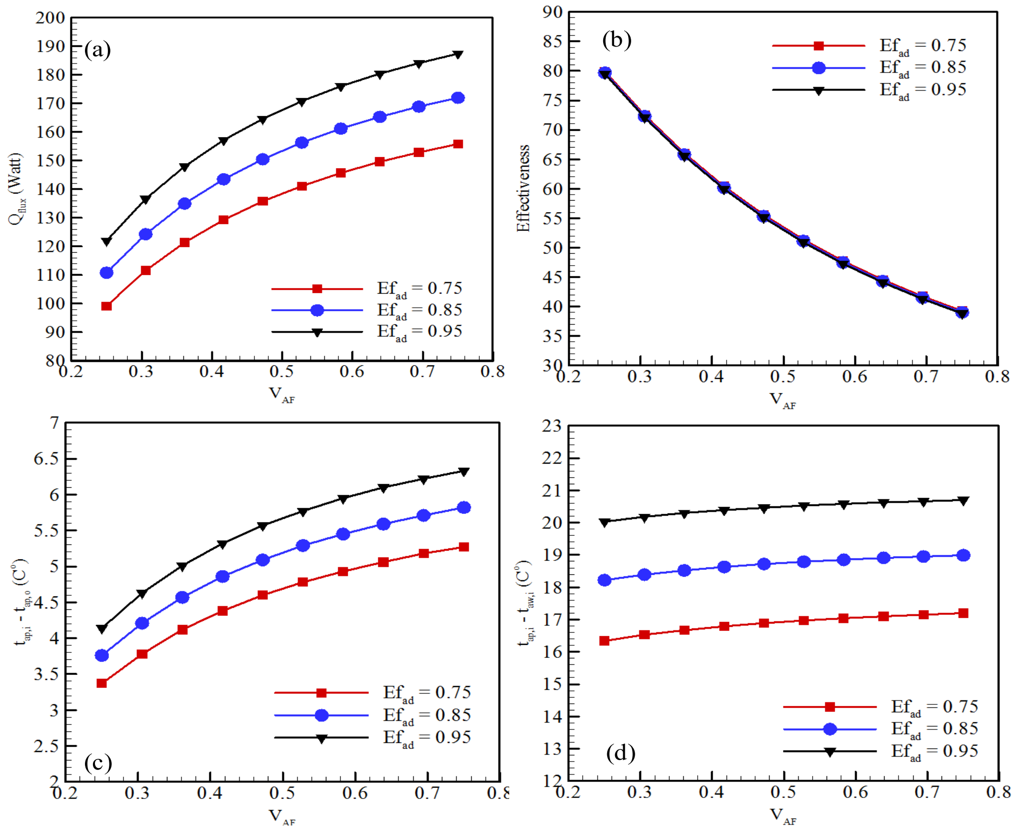

Figure 9 (a) Heat flux Q_{flux} (b) effectiveness of heat exchanger (c) temperature difference $(t_{ap,i} - t_{ap,o})$ (d) temperature difference $(t_{ap,i} - t_{aw,i})$ as a function to the velocity fraction V_{AF} and adiabatic efficiency Ef_{ad}.

Figure 10 (a-d) demonstrates the relation between the heat flux Q_{flux}, effectiveness of heat exchanger Ɛ, temperature difference $(t_{ap,i} - t_{ap,o})$ and temperature difference $(t_{ap,i} - t_{aw,i})$, respectively, as a function to the air velocity in the product-in $V_{ap,i}$ and the relative humidity $RH_{ap,i}$ which correspond to value ($T_{ap,i} = 45C^o$, $S =$ 0.005 m, $V_{AF} = 0.5$, $Ef_{ad} = 0.95$) for all data.

Figure 10a shows that the heat flux (Q_{flux}) is increasing with increasing the air velocity in the product-in $V_{ap,i}$ because of increasing the heat transfer between the plates. However, the heat flux (Q_{flux}) is decreasing with increasing the relative humidity $RH_{ap,i}$because of the density of moisture air decreases with increased humidity (water vapor is 2/3 the density of dry air), that will decrease the heat flux (see Eq.1).

Figure 10b shows that the effectiveness (Ɛ) is not affecting by $RH_{ap,i}$ that is because of there is no relation between them. Figure 10c shows that the degree of sensible cooling $(t_{ap,i} - t_{ap,o})$ is decreasing with increasing the relative humidity $RH_{ap,i}$ because of the density of moisture air decreases with increased humidity. Figure 10d shows that the degree of latent cooling $\left(t_{ap,i} - t_{aw,i}\right)$) is decreasing with increasing the relative humidity $RH_{ap,i}$ because of the density of moisture air decreases with increased humidity , that will decrease the latent cooling .

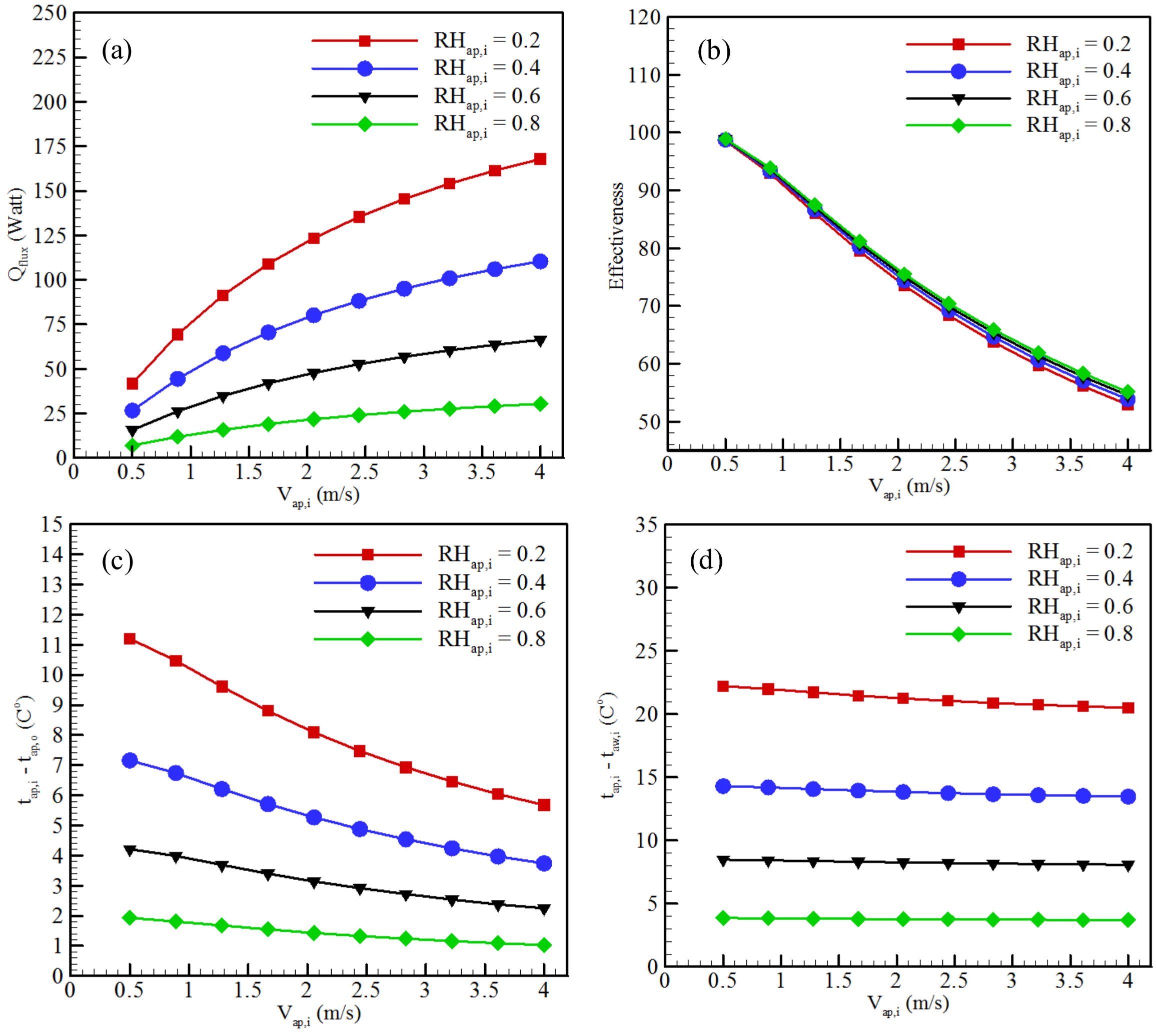

Figure 10: (a) Heat flux Q_{flux} (b) effectiveness of heat exchanger Ɛ (c) temperature difference $(t_{ap,i} - t_{ap,o})$ (d) temperature difference $(t_{ap,i} - t_{aw,i})$ as a function to the air velocity in the product-in $V_{ap,i}$ and the relative humidity $RH_{ap,i}$.

Figure 11 shows that the heat flux $Q_{flux}$is increasing with increasing the V_{AF} because of increasing air velocity, see Eq. 1.

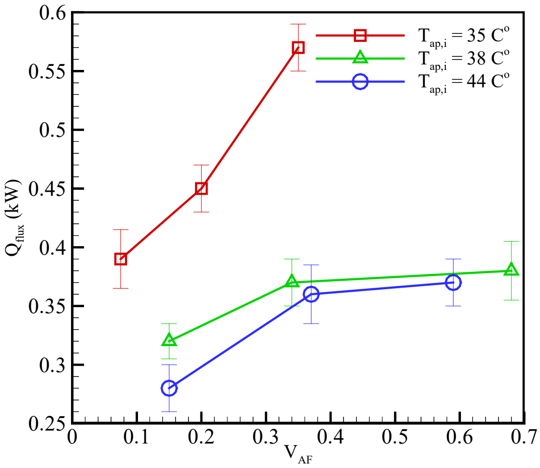

Figure 11 The experimental heat flux Q_{flux} results as a function to the air velocity fraction V_{AF} at three different air temperatures $T_{ap,i} = 35, 38,\ 44\ C^{o}$.

Figure 12 shows that the temperature difference $(t_{ap,i} - t_{ap,o})$ is increasing with increasing the V_{AF}. Because of increasing$V_{aw,i}$, the heat flux (Q_{flux}) inside the heat exchanger will increase, that will increase the temperature difference.

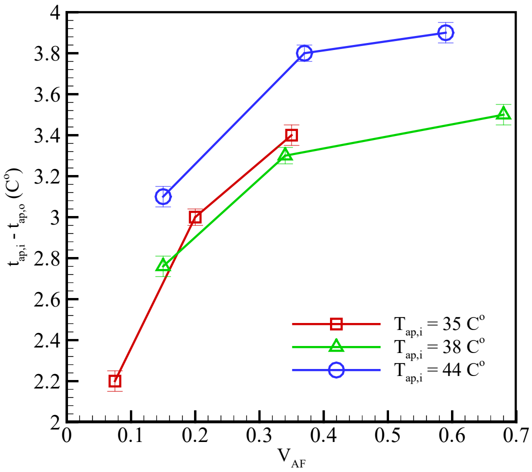

Figure 12 The experimental results of temperature difference $(t_{ap,i} - t_{ap,o})$ as a function to the air velocity fraction V_{AF} at three different air temperatures $T_{ap,i} = 35, 38,\ 44\ C^{o}$.

Figure 13 shows that the coefficient of performance COPis increasing with increasing the V_{AF}. Because of increasing $V_{aw,i}$, the heat flux (Q_{flux}) inside the heat exchanger will increase. So, the COP increasing with increasing the heat flux (Q_{flux}), see Eq. 4.

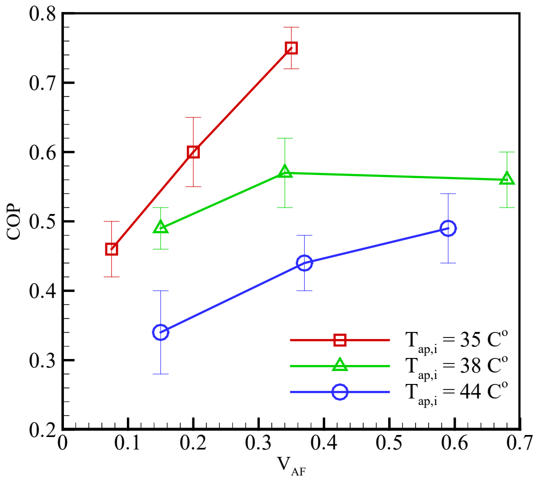

Figure 13 The experimental COP results as a function to the air velocity fraction V_{AF} at three different air temperatures $T_{ap,i} = 35, 38,\ 44\ C^{o}$.

Figure 14 shows that the temperature difference ($t_{ap,i} - t_{aw,i}$) is increasing with increasing the V_{AF}. Because of increasing$V_{aw,i}$, the heat flux (Q_{flux}) inside the heat exchanger will increase, that will increase the temperature difference.

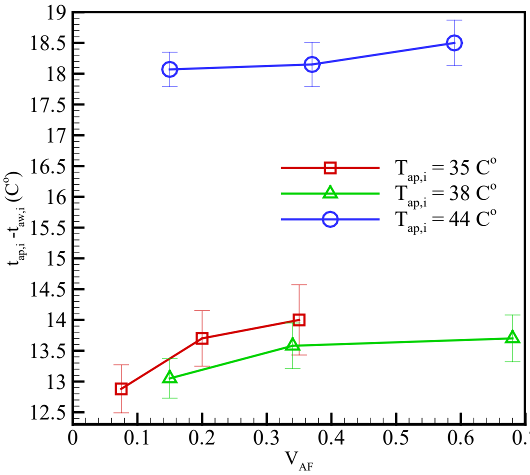

Figure 14 The experimental results of temperature difference ($t_{ap,i} - t_{aw,i}$) as a function to the air velocity fraction V_{AF} at three different air temperatures $T_{ap,i} = 35, 38,\ 44\ C^o$.

Figure 15 shows that there is a good agreement between the heat flux Q_{flux} and the air temperature in the product-out $T_{ap,o}$ experimental and theoretical since all the data are close to the dashed line that has been drawn by 45 degrees.

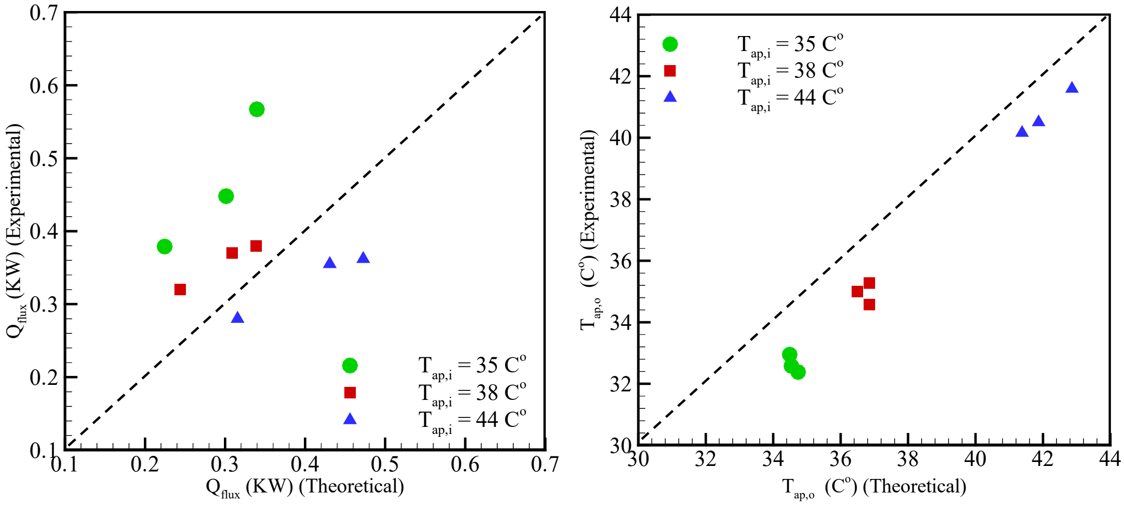

Figure 15 the error bar of h eat flux Q_{flux} and the air temperature in the product-out $T_{ap,o}$

12. CONCLUSIONS

The present work investigates the effect of using the direct and indirect evaporative cooling to reduce the air temperature .The indirect evaporative cooling takes place in a plated heat exchanger where the intake ambient air ($T_{ap,i}$) is sensibly cooled to the state point ($T_{ap,o}$) without adding moisture. The direct evaporative cooling is achieved by adding moisture to the sensibly cooled air while flowing through a wetted pad. From the results obtained, the concluding remarks can be summarized as follows:

1. Using multi stages evaporative cooling is effective for decreasing air temperature in summer season and can be used as potential alternative to vapor compression air conditioning systems.
2. The heat flux Q_{flux} is increasing with increasing the Reynolds number Re, velocity fractionV_{AF}, air temperature at the product-in $T_{ap,i}$, air velocity at the product-in $V_{ap,i}$, and the adiabatic efficiency Ef_{ad}. But, it is decreasing with increasing the spacing between the aluminium plates S and the relative humidity at the product-in $RH_{ap,i}$.
3. The degree of sensible cooling ($t_{ap,i} - t_{ap,o}$) and the degree of latent cooling $\left(t_{ap,i} - t_{aw,i}\right)$ are increasing with increasing the Reynolds number Re, velocity fraction V_{AF}, air temperature at the product-

in $T_{ap,i}$, and the adiabatic efficiency Ef_{ad}. But, it is decreasing with increasing the spacing between the aluminium plates S, the relative humidity at the product-in $RH_{ap,i}$, and the air velocity at the product-in $V_{ap,i}$.
4. The effectiveness ε is decreasing with increasing the Reynolds number Re, velocity fraction V_{AF}, air velocity at the product-in $V_{ap,i}$, and the spacing between the aluminium plates S. But, it is not affecting by the relative humidity at the product-in $RH_{ap,i}$, air temperature at the product-in $T_{ap,i}$, and the adiabatic efficiency Ef_{ad}.
5. Optimum performance was obtained with very small space between plates which was bout 5mm.

Nomenclature	***Abbreviations***
Q_flux: Heat flux (Watt)	IEC: Indirect evaporative cooling
ρ: Density of air (kg/m^3)	EC: Evaporative cooling
Cp: Specific heat capacity (J/(kg K))	DEC: Direct evaporative cooling
V_(ap,i): Air velocity at product-in duct (m/s)	RH: Relative humidity
A: Duct cross-section area (m^2)	HX: Heat exchanger
T_(ap,i): Air temperature at product-in duct (C)	
T_(ap,o): Air temperature at product-out duct (C)	
ε: Effectiveness	
m_ap: Mass of air at product duct (kg/s)	
m_aw: Mass of air at working duct (kg/s)	
N: Number of samples	
σ: Standard deviation of the samples	

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