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License (CC BY 4.0).<http://creativecommons.org/licenses/by/4.0/>**Open Access****Improving the Behavior of Indirect
Evaporative Cooler*****A B S T R A C T**

Indirect evaporative cooling is one of the technologies currently used to build highly efficient air conditioning systems and low power consumption. A computer program was created to predict the effectiveness of an indirect-evaporation cooling system which operates based on Maisotsenko cycle (M-cycle) to determine the environmental conditions and proper system design. Several variables that affect the performance of the system have been studied; the amount of volumetric flow of air ranged from (1050 cfm) to (1550 cfm) for the dry side, and changing from 700 cfm to 1200 cfm from the wet side. With respect to design variables, the length of the channel changed from 50 cm to 100 cm. For environmental variables, the effect of changes in dry and wet temperature on system performance had been studied. The experiment was conducted in mid-June over 24 hours. The results showed that the best air supply provided for the best performance of the system is (1050cfm) for the dry side, while the wet side was (900 cfm), when the length of the channel is 80 cm. The results showed the possibility of applying this system in (Tikrit) because it is characterized by its hot and dry climate in the summer, as evaporative cooling efficiency increases in hot, dry climates

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Introduction

Evaporative air cooling systems have been used successfully in many applications such as cooling of residential and commercial buildings, glasshouses, vegetable stores, and other applications. Evaporative cooling systems have recently been a matter of interest to researchers because of their low energy consumption compared with other compressive cooling systems that rely on the principle of vapor compression (Freon and ammonia) that affect human health and the ozone layer [1], low cost of running and maintenance, and easiness of installation. Cooling methods are divided into three categories:

Direct evaporative cooling (DEC):

Direct evaporative cooling is used to reduce the temperature and increase air humidity by using the evaporative technique by changing water to vapor. In this process, the energy of the air does not change. The warm dry air is changed to cool wet air. The external air temperature is used to evaporate the water. (Fig. 1, line b-c) shows that the temperature has been reduced and relative humidity increased (Humidity) [2].

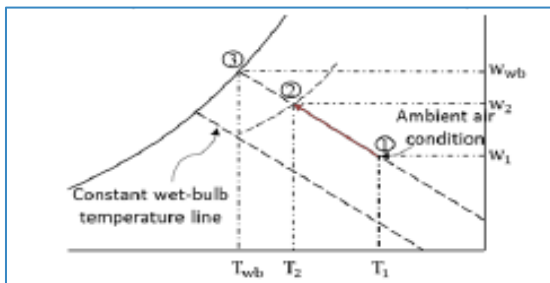


Fig.1: Illustrates the DE cooling procedure on psychrometric chart [3].

Some of the disadvantages of direct evaporative air coolers are:

- increase the moisture content of the cooled space.
- The cooling capacity is limited according to the temperature of the wet bulb of the surrounding air.

These disadvantages can be avoided by using indirect evaporative coolers.

Indirect evaporative cooling (IEC)

The basic idea of indirect evaporative coolers is to reduce air heat by transferring heat from the dry side to the wet side without contact between them, which leads to cool the air without increasing humidity. This characteristic distinguishes them from direct evaporative cooling systems. [4]. The researcher [5] assessed theoretically and experimentally the performance of an indirect evaporative cooling system based on M-Cycle for Greek weather conditions. less than 100% - 115%, while water consumption was in the range of (2.5 - 3) L kW/ h. Experimental results were presented for the study [6] of evaporative cooling system in different input air conditions (velocity, temperature and Humidity). The results were revealed that the wet bulb effectiveness ranged between (92 and 114%) during the day of summer in a humid and hot climate. A numerical study [7] on heat exchanger of indirect evaporative cooling unit was proposed a range of design parameters to achieve the greatest behavior of the system. These conditions were the velocity of the air inside the exchanger ranged of (0.3-0.5 m/s), height of the channel (6 mm), and concluded that under these British conditions, the system could give a wet bulb effectiveness up to (130 %). The cooling process is achieved through the passage of one part of air through the wet side, and the other part through the dry side of the heat exchanger. The air flow through the dry channel loses part of its heat to the wet side by evaporating the water molecules in the wet channels. moisture content constant as shown in (Fig. 2. line 1-2).

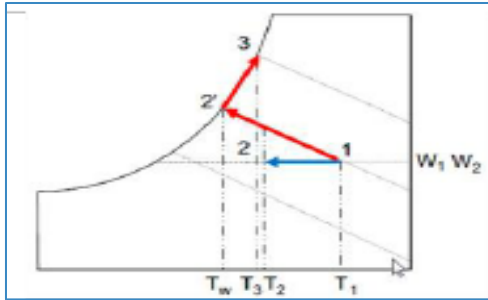


Fig.2: psychrometric chart of the Indirect evaporative cooling

Indirect evaporative cooling (Maisotsenko cycle)

Maisotsenko cycle is one of the types of evaporative cooling and is considered one of the renewable systems of indirect evaporative cooling and a modern scientific revolution, which reaches the temperature of cooling under the wet temperature of ambient air. Its invented by Valeri Maisotsenko [8]. It consists of two types of channels: (dry channels) through which the primary air was passed, and (wet channels) through which the secondary air is wetted with water. The heat exchanges between the two channels without mixing the air for the purpose of not increasing the moisture content of the air produced. The holes are located at the end of the flow channels as shown in figure (8). The produced air flows through the dry channels and conveys the heat to the wet channels. At the end, a portion of the air at the dry channels is delivered from the cooled air to the air-conditioned area. The remaining air is moved to adjacent wet channels where the cold air conveys the heat reasonably proportionately with the mixed air in the dry channel. It was found that the effectiveness of the wet bulb of this spaces is 130%, and the dew point effectiveness is up to (90%). In addition, a comparative study between an indirect cooling system for the M-Cycle between cross flow and orthogonal flow

showed that the opposite flow arrangement achieved a (20%) higher cooling capacity and (15-23%) higher than the dew point and wet bulb effectiveness respectively for the same engineering sizes and operating conditions.

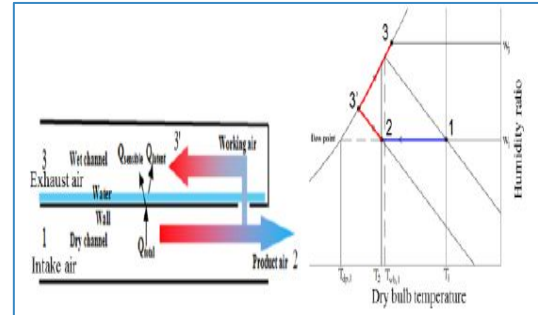


Fig.3: standard of work diagram and psychrometric chart of the Maisotsenko -IEC. [9]

equations

For the thermal balance of heat transfer and mass in the heat exchanger, assuming the following:

- No heat exchange with the surrounding.
- The heat is transferred from the dry side to the wet side without loss.
- Neglecting the thermal resistance of the heat exchanger channels, because the thickness of the channel is small (0.12mm), and the thermal conductivity is high.
- Neglecting the thermal resistance of the water layer that covers the surfaces of the exchanger plates of the wet channels, since the thickness of the water layer is very small (0.25 mm).

Mathematical Relationships

Based on the above assumptions, the heat balance of the heat exchanger will be as follows: [10]

$$q_1 + \dot{M}_{we} * Cp_w * Tw_i + q_2 = \dot{M}_{we} * H_{fg} + (\dot{M}_w - \dot{M}_{we}) * Cp_w * Tw_o \quad (1)$$

- A. Calculate the surface area of evaporation for heat Exchanger.

The area for flatness of the channel calculated by the following equations [11]:

- Calculating the surface area of the wet channel walls from the heat exchanger

$$A_{s1} = 2(w \times l) \quad (2)$$

- Calculating the surface area of the upper and lower part of the wet channel:

$$A_{s2} = 2(w \times h) \quad (3)$$

- Calculating the total surface area of evaporation of heat exchanger:

$$A_s = N_{wet} (A_{s1} + A_{s2}) \quad (4)$$

B. Calculating the cross section for channel for the exchanger:

$$A_{c.s} = (w \times h) \quad (5)$$

- C. Calculating the air velocity across the sides of the heat exchanger:

The average velocity of the air passing through the dry side of the exchanger can be calculated as follows:

$$Q_{dry} = V_{dry}^* \cdot A_{c.s} \quad (6)$$

$$V_{dry}^* = \frac{Q_{dry}}{A_{c.s} \cdot N_{dry}} \quad (7)$$

The average air velocity passing through the wet channels of the heat exchanger is calculated as follows: [11]:

$$A_s = A_o \quad \text{when } A_o: \text{ the free minimum Area.}$$

$$V_{wet}^* = \frac{Q_{wet}}{A_o} \quad (8)$$

- D. Calculation of cooling capacity [12]:

After the thermal content of the evaporation (H_{fg}), and the evaporated water rate ($\dot{M}_{we}$) are calculated, the cooling ability of the cooler is calculated (in Watt) as follows:

$$q_c = \dot{M}_{we} \cdot H_{fg} \quad (9)$$

- E. Calculation of the temperature dry bulb for Air outside the exchanger [12]:

After calculating the cooling ability of the indirect cooling phase, the dry temperature of the outside air on both sides of the heat exchanger is determined using the following relationship:

$$q_c = Q \cdot \rho_m \cdot C_{p_m} \cdot (TD_i - TD_o) \quad (10)$$

from the dry side of the heat exchanger:

$$TD_{od} = \left(\frac{(Q_{dry} \cdot \rho_m \cdot C_{p_m} \cdot TD_i) - q_c}{Q_{dry} \cdot \rho_m \cdot C_{p_m}} \right) \quad (11)$$

While the temperature from wet side is :

$$TD_{ow} = \left(\frac{(Q_{wet} \cdot \rho_m \cdot C_{p_m} \cdot TD_i) - q_c}{Q_{wet} \cdot \rho_m \cdot C_{p_m}} \right) \quad (12)$$

- F. Calculation of the wet bulb temperature [13]:

The wet bulb temperature is calculated according to the dry bulb temperature of air and its relative humidity as follow:

$$TW = TD * \text{atan}[0.151977 * (\phi\% + 8.313659)^{1/2}]$$

$$+ \text{atan}(TD + \phi\%) - \text{atan}(\phi\% - 1.676331) + 0.00391838 * (\phi\%)^{3/2} * \text{atan}(0.023101 * \phi\%) - 4.686035 \quad (13)$$

- G. Calculation of the amount of water evaporated through wet channel [13]:

$$\dot{M}_{wep} = \frac{\dot{V}_P \cdot A_{TP} \cdot (W_S - W)}{3600} \quad (14)$$

$$\dot{V}_w = (25 + 19 * \dot{V}_w^*) \quad (15)$$

- H. Calculation of effectiveness of the wet bulb temperature of the surrounding air [14]

$$\eta_{wb} = \left[\frac{TD_i - TD_{od}}{TD_i - T_{ow}} \right] * 100 \quad (16)$$

•

The main variables that have a significant effect on the cooling performance of the system operated by Maisotsenko cycle indirect evaporation, and the environmental conditions of the region of study during the summer were studied. The effect of changing the length of channels for the heat exchanger, and thermal and volumetric flow rate of the primary air passing through channels in the dry side of the heat exchanger.

Effect of change length of channel of heat exchanger

The length of channels of the heat exchanger has a significant effect on the performance of the cooling system. It has been studied for different lengths ranging from 50-100 cm with an incremental of (10 cm). Figure (4) shows the effect the change in case of air supplied during 24 hours of the day, when the lengths of the channels are changed and the other variables are fixed. When the channel (1 cm) is shown and the primary and secondary air flow rate at (1350 cfm) and (1000 cfm) respectively, with ambient air conditions changing as operational conditions, we notice that the curve starts to decrease until 4 am and then returns until 8 am to record the highest reading of the pipes with lengths (80,90,100 cm), then it decreases again to record the lowest temperature at 2 pm and then returns to increase again slightly and continuously until 11 pm. While we note that for lengths (50,60,70 cm), the curve starts to decrease to record the lowest temperature at the same time for the other lengths and then increases to record the highest temperature at 9 am, and then back down slightly in a semi-straight line until 5 pm, then decrease slightly in a semi-straight line to the last reading at 11 pm. The difference in temperature increase and decrease during the day is due to ambient air conditions of dry temperature and relative humidity. The temperature outside the channels depends on the temperature of the dry bulb of the surrounding air (the air entering the system). In general, increasing the length of the channel gives the best behavior of the cooler system, especially for times when the surrounding temperature is high, as shown by Figure (5) which shows the inverse relationship between the length of the tube and the

temperature of the fitted space, which took place at 2 pm. Increasing channel length means lowering the temperature.

Effect of volumetric volume of primary air

The volumetric flow rate is one of the important parameters effecting on the behavior of cooling system. Increasing the air flow rate means increasing velocity and thus affecting the amount of heat transferred between air and water.

The flow rate of primary air has been changed ranging between 1050-1450 cfm for the primary flow, and an incremental amount of 100 cfm at each change. The volumetric flow of secondary air at (1000 cfm) when studying the flow variation of primary air and stabilizing the flow to primary air at (1350 cfm), when studying the flow of secondary air. In addition, the other variables (80 cm, 2 cm and 0.5 cm) are the length, width and height of the channel respectively. Figure (6) shows the effect of the volumetric flow rate of primary air on the supplied air and wet bulb effectiveness along the day. The curve decreases until 5 am and then increases again until 10 am for volumetric flow rate (1050,1150,1250 cfm) respectively, and then decreases until it reaches the lowest amount at 2 pm, and then goes up again and continues until 11 pm. While the volumetric flow rate (1350,1450,1550 cfm) decrease until 6 am, and then increase until 10 am, and then go back again slightly and continuously like a straight line until the last reading. It should be noticed that, at the lowest rate of flow and at times when the ambient temperature is high, the temperatures are as low as possible. Due to

improved performance of evaporative cooling systems at such conditions. Figure (7) shows the relation between the volumetric flow rate and temperature at 2 pm, where the volumetric rate of flux is increased. The air is getting warmer. This is explained by the above. Low volumetric flow rate and other variables mean low air velocity and thus sufficient time for heat exchange, resulting in greater reduction of outgoing temperature.

In order to compare the theoretical and experimental results from the temperature of the treated space, figure 18 shows the comparison of the processed temperature of the indirect cooling, which is theoretically calculated and measured in experiment. It shows that there is a good agreement between theoretical and experimental results during 12 hours, especially at times when surrounding temperatures are low.

Correlation

In order to obtain an empirical correlation for the cooling capacity of the system as a function of the length of the channel, width of the channel, height of the channel, and velocity of the air, the statistical analysis program (SPSS) was used to obtain the following empirical formula:

$$q = 2.779 \times \frac{L^{1.143} W^{1.595} H^{-1.649}}{V^{1.740}} \quad (17)$$

The values of the studied variables (L, W, H, V) that achieve the correlation equation are in the following ranges:

$$50 < L < 100 \text{ cm}$$

$$1.1 < W < 2.6 \text{ cm}$$

$$0.2 < H < 0.7 \text{ cm}$$

$$0.31 < V < 3.81 \text{ m / s}$$

In order to compare the theoretical and experimental results of the temperature of the space fitted, it took the sixth month (June), because the variables studied theoretically were based on the environmental conditions of this month. Figure 11 shows the comparison of the supply air temperature of the indirect evaporative cooling, which is theoretically calculated and measured in practice. It shows that there is a good agreement between the theoretical and experimental results over 24 hours, especially at times when ambient temperatures are low, the highest deviation between the theoretical and experimental value was (4%).

Conclusions

The experimental and theoretical study was carried out to determine optimum performance of the indirect evaporative cooler. The results showed the following:

1. There is good compatibility between the results of the experimental model and the theoretical results, and the error does not exceed (4%).
2. The design dimensions of the heat exchanger channels are very important parameters to find the optimum performance of the system.
3. The wet bulb effectiveness increases with the increase of the dry bulb temperature of the supply air, and decreases with increasing the relative humidity.
4. The optimum secondary to primary air ratio depends on the ambient conditions, and its best value was 0.9.

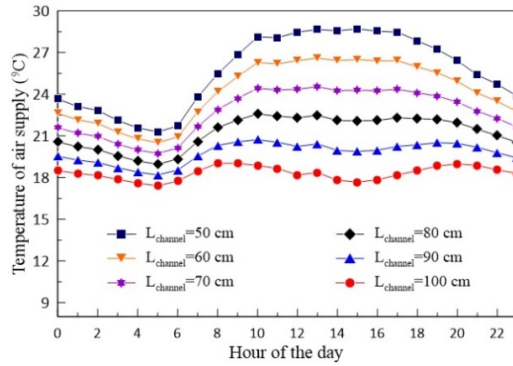


Fig.4: The effect of the length of the channels on the supply temperature on the other variables at ($h = 0.5$ cm $w = 2$ cm $Q_{dry} = 1350$ cfm $Q_{wet} = 1000$ cfm)

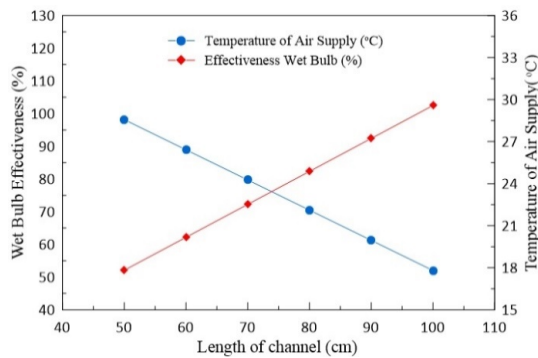


Fig.5: The relationship between length of channel and temperature of supply air and Wet Bulb Effectiveness processed at 2:00 pm

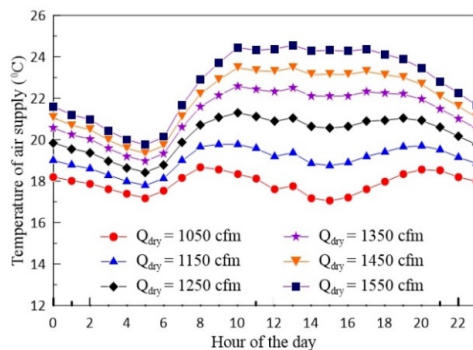


Fig.6: The effect of primary air volumetric flow rate on supply air temperature the other variables ($w = 2$ cm $l = 80$ cm $H = 0.5$ cm $Q_{wet} = 1000$ cfm).

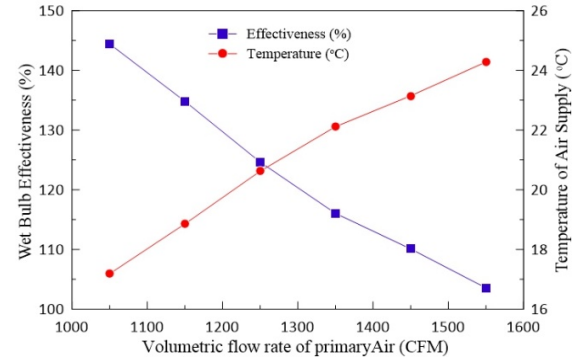


Fig.7: The effect of Volumetric flow rate on supply air at 2:00 pm when the other variables ($w = 2$ cm $l = 80$ cm $H = 0.5$ cm $Q_{wet} = 1000$ cfm)

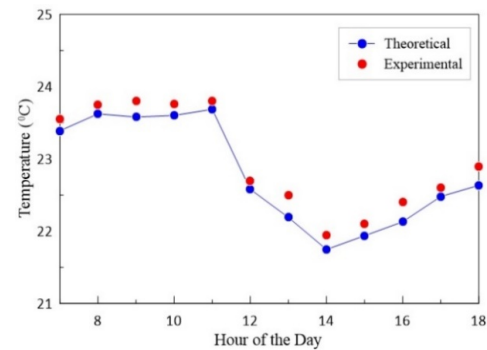


Fig.8: Comparison between experimental and theoretical results

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