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licensed under the Creative
Commons Attribution International
License (CC BY 4.0).<http://creativecommons.org/licenses/by/4.0/>**Open Access****Three Stages of Indirect Evaporative Cooling:
Experimental and Theoretical Evaluation Study*****A B S T R A C T**

Economical energy, reduction of cost and utilization of clean energy are required to meet the human needs. Evaporative cooling units are considered as a solution for these require -cements by transforming such systems into technologies that meet these needs. The equipment's cost, installation and oper-acting costs are simple and low compared with refrigeration systems. An effective design is obtained by employing three stages settlement that equipped with a cross flow heat exchan-ger,direct and indirect evaporative coolers. In order to assess the design performance, a program code is developed. Flow and design parameters namely, air flow rate, piping length and diameter are studied. In addition to that the inlet air-dry bulb temperature at several different time duration over day is studied. The study was conducted in Tikrit University, Iraq (34.35N;43.37E).Readings are recorded in June, July and end of August for two days(24hours a day). The results show that, saturation of direct evaporative cooler effectiveness varies in the range,67%-96% and overall effectiveness of the unit varies in the range,80%-120%.It is provided that the system is efficient in dry and hot areas, and an improvement in the performance of the current design is achieved successfully.

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Nomenclatures

$C_{p_{air}}$	Heat capacity of the air [kJ/kg K]
H_{fg}	Saturated enthalpy [kJ/kg K]
$\dot{M}$	Mass flow rate of the water that evaporated from the wet pad
Q_{coil}	Cooling capacity of coil [kW]
Q_{ct}	Cooling capacity of tubular heat exchanger [kW]
Q_{cp}	Cooling capacity of wet pad [kW]
$\dot{V}_{air}$	Flow rate across the tubular heat exchanger [m ³ /s]
$\dot{v}_{air}$	Flow rate across the wet pad
ρ_{air}	Density of the air entering the wet pad [kg/m ³]
η_{wbHE}	Wet bulb effectiveness of the tubular heat exchanger
η_{wbp}	Wet bulb effectiveness on the cooling pad
η_{wbT}	Wet bulb effectiveness on the three-stage system
T_o	Dry bulb temperature of the inlet air that entering through the coil (radiator) [°C]
T_A	Dry air Temperature of bulb that leaving the wet coil and entering the tubular heat exchanger [°C]
T_B	Dry air Temperature of bulb that leaving the tubular heat exchanger and entering the wet pad [°C]
T_C	Dry air Temperature of bulb that leaving the wet pad [°C]
T_W	Dew point [°C]

Introduction

The dramatic demand for the air conditioning consume a huge quantity of electrical energy which has a negative impact on the environment along with the increase of the emission of the greenhouse gases [1]. The evaporative cooling technology is based on using an evaporated water that spread around to absorb heat from the surrounding air causing a cooling down, and then conveys the absorbed heat back to the air as a latent

heat. And as a result, the air is moistened, and some of air properties could be hardly changed i.e, sensible heat is converted to latent heat. Evaporative cooling systems differ from the traditional systems in terms of non-using of compressors, coil chiller, condenser, towers for cooling or totally isolated piping system and high operational cost. Initial evaporative systems and operating costs are less than 50% and 33% of the total cost respectively. Maintenance costs are minimal, using low carbon and more economical feasible methods for building air conditioning in hot and dry climates. There is no need for airtight structure such as air conditioning to operate at maximum efficiency.

According to American Society of Heating, Refrigerating and Air conditioning Engineers (ASHRAE), in warm areas, many industries suffer from several problems such as, diseases, low productivity and increased absenteeism among laborers. Evaporative cooling is the proper solution for resolving this heat problems and much contribution for improving the efficiency. The earliest idea for using the evaporative cooling emerged ancient Egyptians and Roman as pioneers. They have used wet mats to cool the indoor air that pass through the hanged them over doors and windows [2]. There are some limitations in using the evaporative coolers such as, very high humidity which cause decreasing the capability of the evaporative cooler and accelerate corrosion that leads to a short life for the electronic equipment's. The evaporative cooling system's efficiency increases as the air temperature increases and its humidity decreases. Therefore, in dry and hot regions, evaporative cooling system save large energy in the traditional air conditioning systems. Evaporative coolers systems are used successfully in cooling and moistening

residential and commercial buildings, greenhouses, farms, electronic manufacturing plants, computers rooms and textile industries. An increasing of air temperature, a decrease in relative humidity and impurities of the airflow in the system of the power plant gas turbine compressors. And this is solved by using an evaporative cooler units to improve the thermal efficiency [3]. To overcome such problems of humidity and cooling system, it is focused on using the indirect and two-stages evaporative cooling system to reduce the energy consumption by (60% - 75%) of the current traditional systems. Variation of the cooling capacity and saturation efficiency of the air and water flow rate was investigated by Khobragade and Kongre [4]. They have reported that cellulose made material gives 92.8% saturation efficiency, considered the highest one, while Khus-Grass material gives the lowest saturation efficiency, 40.13%. A potentially commercialized new design of an evaporative cooling in the condenser window-air-conditioner type. This is explored by Hajidavalloo et al. [5]. They have indicated that the suggested design decreases that consumed power and increase the performance coefficient by 16% and 55%. Hajidavalloo et al. [6] have showed that power consumption and performance of the evaporative cooling air condenser have been improved significantly in comparison with the air-cooled condenser. However, it was found that the evaporative cooled condenser is less sensitive to increase of the air temperature. Alotaibi, et al. [7] had experimentally investigated the design of the evaporative cooling in the air-cooled condenser with water atomization. As a result, power consumption decreases by 11% while the performance coefficient increases by 13%. Tianwei Wang et al. [8] have showed that the dry bulb temperature of the condenser inlet is

inversely proportional to the coefficient of performance (COP) that utilizes the evaporative cooling condenser to pre-cool the air. However, COP was found that to increase temperature from 6.1% to 18% resulting a 14.3% percentage power reduction.

There are two methods of evaporative cooling; direct and indirect evaporative cooling. The efficiency of each method depends on the extent to which the dry-bulb temperature of the air supply exceeds the wet-bulb temperature. Direct evaporative coolers are made of metal cubes or plastic boxes with large flat vertical air filters, called "pads". Specifications and process of the direct evaporative cooling are presented in [9]. Figure 1 shows the schematic diagram of the processed air in a drip-type Direct Evaporative Cooler (DEC). In DEC equipment, the evaporated water converts the sensible heat to a latent heat, which leads to decrease in both of the leaving air and process air temperatures. However, air humidity increases. The relative humidity of the processed air could reach up to 60% with 80% saturation efficiency in dry climate conditions. Moist conditions usually result in as much as 80% relative humidity.

Abbreviations and Acronyms

Practical work and Experimental process

The experiment was conducted during the summer time (June, July and August). Three different flow rates were used. Steady state conditions were attained after a period of the experimentation time. Thermocouples were used to record both dry, bulb and wet bulb temperatures of the ambient air. The cooled air temperature was recorded before and after the heat exchange. Also, both the collected and the exit water temperatures were recorded. Air flow rates were measured

using hot wire anemometer. For a high precision purpose, all the tests twice.

Indirect Evaporative Cooling (IEC)

To evaluate the performance of DEC/IEC system at ambient conditions, an experimental setup was developed as shown in Fig. 2. In IEC, the main function of heat exchanger is to separating the primary air from the water. IEC involves exchanging heat between two air streams. A heat exchanging wall is used for this purpose. The supplied air (working air) passes through the wet side, while the outlet air (product air) exit through the dry side. The wet side absorbs heat from the dry side by water evaporation process then the dry side is cooled. The wet air stream conveys the latent heat while the dry air stream conveys the sensible heat without any moisture added to the produced air. The indirect evaporative coolers is based on the principles of decreasing the sensible heat of the air without influencing the humidity content. Like the conventional IEC unit elements, the present design consists two heat exchangers, coil and cross types, water tank, water pipe lines, pump and small fan

Experimental of IEC set up

The cooler consists a rectangular galvanized aluminum cabinet, 1.21 m length, 0.914 m width and 1.07 m height. The main components of the experimental set up are, 15 inch (0.381 m) fan, 15 inch (0.381 m) exhaust fan, air ducts, duct for the returned exhausted air, cross flow plate heat of the exchanger, water tank and water pump (1.5 m head). A separate pump is used to supply water to the heat exchanger from the tank. A number of thermocouples are fitted and connected to a data logger to that used measure the dry and wet bulbs temperatures at many points as shown in Fig. 3.

The DEC unit consists an exhaust fan and a recirculating pump that used to drip water on the upper side of the pad. Wood wool is used as a packing material a 45 cm length, 60 cm widths and 3 cm thickness, as shown in Fig. 1.

The specific surface area packing material is , $503 \text{ m}^2/\text{m}^3$ and an equivalent diameter, of 0.0093 m. All the specifications are presented in Table 1.

Heat Exchanger (Radiator type)

The first heat exchanger type used to reduce the working fluid temperature before entering the second heat exchanger. This is achieved by the liberation of the sensible heat without changing the humidity ratio. The first stage of the experiments set consists a heat exchanger of coil flow radiator. This exchanging is a rectangular aluminum duct of $80 \times 56 \times 4 \text{ cm}$ dimensions. A schematic diagram of the heat exchanger and the other stages are shown in Fig. 4. Two streams pass through it, cold-water stream passes through horizontal pipes and upstream air passes across the outer surface. The draw load from the removed air represents about 34% of the total cooling load. The drawn load from the air that passes through the external fins of the heat exchanges the water inside the tubes, which flows over the outer fins of the wet side of the second heat exchanger. The hot water evaporates easily since it flows contacting with the external dry air and absorbs the latent heat from the fins. Then, the pipe walls through which the cooled air passes from the first stage cools the air again and removes about 30% of the total cooling load. The air comes from the dry side of the second heat exchanger, which is cool and dry, passes over the moist pad it is subjected to a third cooling process to remove the left cooling load (approximately 36% of the total cooling load).

IEC Heat Exchanger (Tubular type)

H. Xiang, W.Y.J.U.Y.X.Z. Xinli, X. Mingyuan [10] have worked on a simplified design of the tubular IEC. In this design, the primary air is separated from the water when the secondary air and water films are mixed in horizontal planes. Huang et al. [11] have come up with a mathematical model of a tubular IEC which is presented by Peterson [12] an (experimental study). The experimental results also

indicated that the optimum secondary/primary air flow rate ratio was (0.6–0.8) [13]. A schematic drawing, psychometric chart and concept of working principles of typical heat exchanger of wet-bulb temperature IEC system are shown in Fig's. 4 and 6. It comprises several pairs of, side to side, channels: wet conduits of the secondary air and dry conduits of the primary air supply. A common configuration of such exchanger includes bundle of tubes are mounted on a cylindrical or rectangular shell, where the primary air passes inside the tubes and secondary air flows in a perpendicular direction to the primary air, while water is sprayed over the tubes surface. Thus, it offers more uniformed water films over the tubes and less pressure losses compared to plate-type IEC. Heat transfer occurs between the two working fluids during a heat conductive plate; therefore, the supplied air is cooled sensibly without any additional moisture, while the latent heat of the vaporized water plays the master role in heat transfer mechanism between air and water that passing in a wet passages.

Mathematical modeling

To develop a balances for the thermal and mass of the proposed cooling system, the following hypotheses were employed;

1. There is no thermal exchange with the outer circumscription.
2. Heat is transmitted completely from the dry to the wet side without any loss.
3. Thermal resistance of the heat exchanger plates is neglected because of low thickness of plates (0.12 mm) and high thermal conductivity.
4. Thermal resistance of the water layer that covers the surfaces of the exchanger plates that supply by the wet channels is neglected, since its thickness is very low, not exceed 0.25 mm.

Mathematical relations

The cooling capacity of the first stage coil (radiator) is calculated as follows [15]:

$$Q_{coil} = \dot{v}_{air} \times \rho_{air} \times C_{p_{air}} (T_o - T_A) \quad (1)$$

The cooling capacity of the second stage tubular cross flow heat exchanger is calculated as follows [15],

$$Q_{ct} = \dot{V}_{air} \times \rho_{air} \times C_{p_{air}} (T_A - T_B) \quad (2)$$

The cooling capacity of the third stage-cooling pad is calculated as follows[15],

$$Q_{cp} = \dot{M}_{wep} \times H_{fg} \quad (3)$$

The dry bulb temperature of the air outlet from the second stage tubular cross flow heat exchanger T_B is calculated as follows:

$$T_B = \frac{\dot{V}_{air} \times \rho_{air} \times C_{p_{air}} \times T_A - Q_{ct}}{\dot{v}_{air} \times \rho_{air} \times C_{p_{air}}} \quad (4)$$

The dry bulb temperature of the air outlet from the third stage cooling pad T_D is calculated as follows:

$$T_C = \frac{\dot{V}_{air} \times \rho_{air} \times C_{p_{air}} \times T_B - Q_{cp}}{\dot{v}_{air} \times \rho_{air} \times C_{p_{air}}} \quad (5)$$

The effectiveness of the wet bulb is calculated as follows [16]:

- a. For a heat exchanger

$$\eta_{wbHE} = \left[\frac{T_A - T_B}{T_A - T_w} \right] \times 100 \quad (6)$$

- b. For a cooling pad

$$\eta_{wbp} = \left[\frac{T_B - T_C}{T_B - T_w} \right] \times 100 \quad (7)$$

- c. For the whole system

$$\eta_{wbT} = \left[\frac{T_A - T_C}{T_A - T_w} \right] \times 100 \quad (8)$$

Sample of calculations is presented in Table 2.

$$Q_{coil} = 0.566 \times 1.1 \times 1.005(32.8 - 28.5) \\ = 2.7 \text{ kW}$$

$$Q_{ct} = 0.566 \times 1.1 \times 1.005(28.5 - 24.8) \\ = 2.3 \text{ kW}$$

$$Q_{Cp} = 1.2 \times 2.4342 = 2.9 \text{ kW}$$

$$\eta_{wbHE} = \left[\frac{28.5 - 24.5}{28.5 - 13} \right] \times 100 = 27\%$$

$$\eta_{wbp} = \left[\frac{24.5 - 22.7}{24.8 - 13} \right] \times 100 = 18\%$$

$$\eta_{wbp} = \left[\frac{28.5 - 22.7}{28.8 - 13} \right] \times 100 = 40\%$$

Results and discussion

Effect of air temperature

For the steady state condition and as shown in Fig.5 are recorded. The supplied air (inside the dry channel) was reduced from 40 °C to about 24 °C (drop down 16 °C) with 35% relative humidity. The volume flow are for the dry and wet channels are 1250 ,1500 cfm, respectively. Cooling capacity at different dry bulb temperatures, relative humidity conditions, wet bulb temperature and dew point are determined to adjust the wet bulb effectiveness and performance of the evaporative cooler. Cooling capacity at different dry bulb temperatures and relative humidity conditions are shown in Fig. 7. It shows that the cooling capacity is obviously impacted by dry bulb temperature and relative humidity of ambient air and cooling capacity is inversely proportional to the relative humidity. The total calculated cooling capacity was about 8200 W/m² at 40°C and 35% relative humidity. Moreover, the results indicated that the temperature of the supplied air was reduced from 40 °C to 30 °C and relative humidity 35% and airflow rate, 1250 cfm, then reduced furthermore to 24 °C after passing through the cooling pad.

Sample of test data of three months is presented in Table 3.

Effect of volumetric flow rate of air

Variation of the air volumetric flow rate with time inside (30 cm length and 1.5 cm diameter) tube of heat exchanger at atmospheric air temperature and relative humidity are presented in Fig's. 8 and 9. The results showed that the air temperature outside the wet and dry side begin to rising slightly as the atmosphere temperature increases after the sunrise and reach the maximum temperature after two hours the midday. Then drop down gradually to atmospheric temperature before sunrise of the next day. On the other hand, the difference between the entering and existing temperature air of the wet and dry sides is the highest value when the ambient air temperature is high. The results also show that the optimum airflow rate that gives a significant drop in the temperature of the wet or dry sides is 750 cfm. Since cooling depends on the evaporation rate of the water that is sprayed on the outer fins of the wet side and the amount of the produced heat to evaporate the water in the heat exchanger, the temperature decreases as air velocity decreases. However, this decrease become less as more water evaporated till the water the approaches saturation state. In addition, reducing the air speed requires a large evaporation area that implies a large size system. In general, thermal equilibrium depends on the climatic conditions of the region, the design conditions of the first stage (both the dry and wet sides), and evaporative cooling of the second stage. Among different variables in a wide range, the following variables are observed and monitored

1. The temperature of the fresh supplied air to the room (the air space) and its relative humidity.
2. The amount and speed of the supplied air to the designated air space.
3. The dimensions of the heat exchanger and system in general.
4. The amount of air and its speed that passing through the pipes in the dry side of the heat exchanger.
5. The amount of air and its velocity that passing through the fins of the wet side of the heat exchanger.
6. Pressure drop through the dry side pipes and pressure drop through the wet filling and fins of the heat exchanger.
7. The electrical power of the propellers (mainly through the dry and wet sides and secondly charge through the wet side of the heat exchanger).
8. The amount of the evaporated water through the wet side of the heat exchanger and wet charge after the dry side.
9. The amount of air and its speed that passing through the pipes of the dry side of the heat exchanger.
10. The amount of the air and its velocity passing through the fins of the wet side of the heat exchanger.
11. The electrical power of the propellers (mainly through the dry and wet sides and secondly charge during the wet side of the heat exchanger).
12. The amount of the evaporated water through the wet side of heat exchanger and wet charge after the dry side.

Effect of pipe diameter

Figure 10 illustrates the effect of the pipe diameter, in the range, (0.5-3.0) cm, on the outlet bulb dry temperature. The temperature profile shown decrease of its value at the period (12 am to 6 am) where minimum value was recorded, then increase to the maximum values at 2 pm, value then decrease again slightly and continue decreasing until 11 pm. Comparison with Fig.

5, indicate that there is a reduction in the air temperature that began at 12 am to 4 am, then increasing until 9 am then after, decreasing and recording its lowest value at 2 pm. This is attributed to the fact of the effectiveness of the evaporative air coolers which would enhance at highest temperatures due to the decrease of the wet bulb temperature as the atmospheric temperature increases. This leads to an increase of the evaporative cooling efficiency. Also, it is observed that the lowest outlet air temperature is obtained where one cm diameter tube heat exchanger is used and a highest value was obtained 3 cm diameter. This may attribute to the fact that the pipe diameter is inversely proportional to air flow rate of air pass over the heat exchanger as shown in Fig. 5. The decrease of the outlet air temperature is caused by the increase of heat exchanging surface area due to forced convection. At the same time, increasing the pipe diameter with constant volumetric flow rate does not allow air (near the pipe center to exchange heat with the walls). This observed in this study that is an optimum values of pipe diameter, below this value an increase of the pressure drop was occurred accompanied with a drag in airflow.

Effect of pipe length

The pipe length of the heat exchanger is another factor, which is studied to show how it affects the effectiveness of the cooling system. Different lengths were selected namely 20, 25, 30, 35, 40 and 45 cm as shown in Fig. 11. It shows that the predicted temperatures profiles are the same at all pipe lengths with a slight change in 45 cm pipe length. This is due to increasing the evaporation surface area.

Performance of the cooling system

Since the air temperature is very high and the humidity is very low in the study area. Therefore using three stages evaporative system is very important. **Figure 7** shows the effect of the relative humidity on the total

cooling capacity at different inlet temperatures. The results showed that in the first stage air temperature through the dry side of the heat exchanger was reduced by 5 °C. In the second stage, the air temperature through the dry side of the heat exchanger was reduced by 7 °C at constant moisture content, while the relative humidity remains low and below the normal limits of the standard comfort conditions. Also, in the third stage, the air temperature must be reduced at least 6 °C, with 50% of the relative humidity. The results showed that the supplied air is fairly acceptable and very close to the comfort conditions. In addition, the humidification cooling of one stage system with the use of the air coolers is mostly used in the low humidity areas like present study area. However, the use of evaporative cooling especially in July and August, because their temperatures are the highest and their humidity is the lowest along the year. The lowest obtained temperature was 8 °C, during these two months the supplied air temperature high and uncomfortable. Therefore, it is found that the addition of second stage is necessary in order to reduce the dry side temperature of the heat exchanger makes the atmosphere more comfortable and similar to that during in May and June, which are considered as mild conditions. Figure 12 displays the variation of the air supply temperature over single day in June, July and August. It is noticed that the change is related to the atmosphere temperature. The results also showed that the temperatures difference between the of supplied air and the atmosphere was more than 16 °C. Figure 13 represents a comparison between the experimental (measured the results and the predicted values of the supply air temperatures). The effect of the relative humidity of the air entering the system on the total cooling capacity at different dry temperatures and design parameters, is shown in Fig. 7. It shows a decrease in the total cooling capacity with the increase of the relative humidity of air in a linear manne, at various inlet dry temperatures. The decrease in the total cooling capacity with the increase of the relative humidity ratio, on both stages,

this is affected by the evaporative cooling of the water in both stages. Cooling, in the first stage, depends on the evaporation of the water from the surface area of the heat exchanger, which represents the latent heat that is needed to evaporate the water that sprayed on the external fins of the heat exchanger pipe, to cool it. Meanwhile, the cold air that passes through the pipes cool its and internal surface. This heat amount is the same as the latent heat that dissipated from the outer surface of the pipes. On the other hand, if the air relative humidity on the wet side is low, more water evaporates and this leads to an increase of amount of the withdrawn heat in the second stage of the system. The same behavior is observed in the third stage, which depends on the evaporative cooling of water of the external surface of wet pad and the relative humidity of the dry air. This leads to an increase in both quantities, of evaporated water and the latent heat with a decrease in the relative humidity. This gives a clear perception that the amount of the cooling capacity is increased in all stages, the first (the presence of the first heat exchanger) the second (the presence of the second heat exchanger) and the third (the cooling phase by direct evaporation) by increasing the relative humidity. Thus increasing the total cooling capacity and decreasing the supply temperature by increasing the relative humidity to an acceptable range, (50%-60%) at different dry bulb temperatures.

Effectiveness and Validation

The total effectiveness of the cooling system is illustrated in Fig. 14. The effectiveness increases as the air temperature of the environment increases, due to the increase of the difference between the upstream and downstream temperatures. In contrast, if the difference between the upstream air temperature and wet air temperature decreases, then this would leads to an increase in the wet effectiveness. A comparison between the experimental and theoretical results is presented in Fig. 13. It shows a good consistency between the

results. And the theoretical model is capable for predicting the effectiveness of the evaporative cooler with a an error of less than 5%.

Conclusions

In this study, three stages, direct and indirect evaporative air-cooling systems, were investigated in order to design and evaluate their performance at different operating conditions. The study comes with the following! The effect of the reduction in the volumetric flow rate, which leads to a low outlet temperature. Increasing efficiency and cooling capacities related to the increase in the dry bulb air temperature. The effect of pipe diameter variation on the system performance is small, whereas, the variation of the pipe length has a great impact on the system effectiveness. It is also noticed that an increase in the pipe length would leads to an increase in the thermal effectiveness. The optimum design conditions for the hot and dry climate (the study area) are $Q_{dry} = 1500$ cfm, $Q_w = 1250$ cfm, $D_t = 1.5$ cm and $L_t = 45$ cm. It is concluded that the proposed system is efficient in dry and hot areas.

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