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Experimental and Theoretical Study of Dew Point Evaporative Cooling System Suitable for Erbil Climate

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Evaporative cooling is an efficient natural process. In this study a dew point evaporative cooling system has been designed, and constructed. A dew point evaporative cooler comprised of combination of multi-stage sensible water to air heat exchangers and evaporative media. In this study the performance of the dew point evaporative cooling system is studied as theoretical and experimental work which is carried out at Erbil Polytechnic University. This work includes the estimation of the effect of impact factors on the system. Based on the conducted experimental and numerical analysis, it is concluded that the performance of the cooling system significantly depends on the; circulating air mass flow rate to the pad, water spray mass flow rate, dry bulb temperature, humidity ratio of incoming air and using return air when evaporative media is having insufficient air. The performance of the system depends on both the effectiveness of water to air heat exchangers and evaporative media. The wet bulb effectiveness of the system is ranged from (45% to 73%) for a single stage. Using return air from cooled zone improves the wet bulb effectiveness. Exhaust humid air can be used in economizer for precooling incoming air. The economizer improves the wet bulb effectiveness of the system from (73% to 93%) with using return air from cooled zone. The ASHRAE guidelines for thermal comfort recommend (20°C to 24°C) in the winter and (23°C to 27°C) in the summer and a relative humidity (RH) of (30% to 60%) is recommended.

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Evaporative cooling is an efficient natural process. In this study a dew point evaporative cooling system has been designed, and constructed. A dew point evaporative cooler comprised of combination of multi-stage sensible water to air heat exchangers and evaporative media. In this study the performance of the dew point evaporative cooling system is studied as theoretical and experimental work which is carried out at Erbil Polytechnic University. This work includes the estimation of the effect of impact factors on the system. Based on the conducted experimental and numerical analysis, it is concluded that the performance of the cooling system significantly depends on the; circulating air mass flow rate to the pad, water spray mass flow rate, dry bulb temperature, humidity ratio of incoming air and using return air when evaporative media is having insufficient air. The performance of the system depends on both the effectiveness of water to air heat exchangers and evaporative media. The wet bulb effectiveness of the system is ranged from (45% to 73%) for a single stage. Using return air from cooled zone improves the wet bulb effectiveness. Exhaust humid air can be used in economizer for precooling incoming air. The economizer improves the wet bulb effectiveness of the system from (73% to 93%) with using return air from cooled zone. The ASHRAE guidelines for thermal comfort recommend (20℃ to 24℃) in the winter and (23℃ to 27℃) in the summer and a relative humidity (RH) of (30% to 60%) is recommended.

Key words: ***Dew point evaporative cooler, cooling tower and heat exchanger, Dew point temperature, Evaporative cooling.***

NOMENCLATURE

α_{fi}	Surface area per unit volume, m^{-1}
A	Area, m^2
c	Capacity Ratio
C_p	Specific heat, $kJ/kg.°k$
D	Diameter, m
G	Mass velocity, $kg/m^2.s$
h	Convection heat transfer, kJ/kg.k
h_d	Mass transfer Coefficient, $kg/m^2.s$
i	Enthalpy, kJ/kg
i_{fgwo}	Latent heat of evaporation, kJ/kg
i_{masw}	Enthalpy of saturated air, kJ/kg
m_a	Air mass flow rate, kg/s
m_{a_in}	Inlet air mass flow rate, kg/s
m_{a_supply}	Mass flow rate of supply air, kg/s
Me	Merkel Number

m_w	Water mass flow rate, kg/s	T_{w_in}	Inlet water temperature in pad, °C
m_{w_in}	Inlet water mass flow rate, kg/s	T_{w_out}	Outlet water temperature in pad, °C
m_{w_spray}	Water spray mass flow rate, kg/s	$T_{wb\_incoming}$	Incoming air wet bulb temperature, °C
Q	Heat transfer, kW	U	Overall heat transfer coefficient, $W/m^2.°k$
$Rh_{\_supply}$	Relative humidity of supply air, %	W	Power, kW
T	Temperature, °C	W	Humidity ratio, kg_{water}/kg_{dryair}
t	Thickness of pad, m	w	Width of pad, m
T_{a_in}	Inlet air temperature in pad, °C	$W_{\_in}$	Inlet air humidity, kg_{water}/kg_{dryair}
T_{a_out}	Outlet air temperature in pad, °C	W_{sw}	Humidity ratio of saturated air, kg_{water}/kg_{dryair}
$T_{db\_incoming}$	Incoming air dry bulb temperature, °C	ε	Effectiveness, %
T_{db_supply}	Dry bulb temperature of supply air, °C	ΔT	Temperature difference, °C
$T_{w\_in\ tank}$	Water temperature in tank, °C		

SUBSCRIPTS

a	Air	HX	Heat exchanger
c	Sensible heat, cold stream	m	Latent
ci , co	inlet and outlet of cold stream in HX	ma	Moist air
db	Dry bulb	x	Water vapor
dp	Dew point	w	Water
hi , ho	inlet and outlet of hot stream in HX	wb	Wet bulb

ABBREVIATIONS

Btu	British thermal unit	e-NTU	Effectiveness-Number of Transfer Unit
COP	Coefficient of Performance	HVAC	Heating, Ventilation and Air Conditioning
CT/HX	Cooling Tower and Heat exchanger	HX/EC	Heat exchanger and Evaporative Cooler
DEC	Direct Evaporative Cooling	IEC	Indirect Evaporative Cooling
DPEC	Dew Point Evaporative Cooling	Le_f	Lewis Factor
EC	Evaporative Cooling	REC	Regenerative Evaporative Cooling

1 INTRODUCTION

Evaporative cooling is suitable for Erbil environment due to high dry bulb temperature and low relative humidity of the ambient air, and considering that it requires less power consumption. Evaporative cooling is a physical phenomenon in which evaporation of a liquid (usually water) into surrounding air cools an object or a liquid in contact with it. As the liquid turns to a gas, the phase change absorbs heat. Technically, this is called the "latent heat of evaporation". There are some barriers of using evaporative cooling (EC), such as; high humidity ratio, water consumption and accessing of high supply air temperature. While heating ventilation and air conditioning (HVAC) system has some drawbacks, such as; high power consumption, using refrigerant is harmful to the environment when it's released to the atmosphere and it scores very low on the air change scale. The proposed system is good compromise between (EC) and (HVAC). The dew point evaporative cooling system with circulating air is modified version of indirect evaporative cooling system, which sensibly cools the air, keeping the same humidity ratio of the air, just by using water evaporation process. The water temperature can approach the wet bulb temperature of entering air. The system is installed in Erbil Polytechnic University, for performance investigation with Erbil climate. The minimum temperature of the supply air is equal to the wet bulb temperature of ambient air for direct evaporative cooling (DEC) and indirect evaporative cooling (IEC), usually it can't achieve thermal comfort, especially for humid regions because of inability to dehumidify air. (DEC) has better efficiency than (IEC) but it adds moisture to the air, which may cause discomfort; (DEC) can be used only when cooling and humidification is required, while (IEC) has the same humidity ratio. Theoretically, supply air temperature with this system (Dew point evaporative cooler) can approach the dew point temperature of the incoming air.

Many researchers have studied the evaporative cooling. In this section the important results and conclusions of the works have been summarized as follows; Heidarinejad et al. [1] investigated the performance of (IEC/DEC) systems. A wet bulb effectiveness value for the combination system is in the range of (1.08 to 1.11) which is higher than the wet bulb effectiveness of a single (IEC) system that was between (0.55 and 0.61). Elberling [2] tested the performance of a cross-flow (DPEC) unit based on the Maisotsenko cycle (M-cycle). The effects of inlet air dry/wet bulb temperatures, fan speed, airflow rate on the performance of cooling unit including effectiveness, cooling capacity, power consumption, energy efficiency and water evaporation rate were investigated experimentally. The test results showed that the wet-bulb effectiveness of the unit varied from (81% to 91%) over all test conditions, which is nearly (20-30%) higher than that of the typical (IEC) systems. Zhao et al. [3] found that the intake air velocity, amount of primary air extraction, and the dimensions of the heat exchanger flow passages were the primary factors affecting the overall effectiveness of the cooling system, while the temperature of the feeding water was not an important factor. Guo and Zhao [4] investigated the thermal performance of a cross-flow heat exchanger in (DPEC) system by analyzing the effects of primary and secondary air velocities, channel width, inlet relative humidity and wettability of plate. Their study suggested that smaller channel width, lower inlet relative humidity of the secondary air, higher wettability of plate and higher ratio of secondary to primary air, can improve effectiveness. Jun Xiong et al. [5] studied the (REC), The circulation volume of secondary air is one of the main factors impacting the effectiveness of (REC) system. The cooling effect was tested under different circulation volumes of secondary air in summer. They conclude that a proper range of the circulation volume is in range of (0.4 to 0.6). Jiang and Xie [6] developed a novel indirect evaporative chiller which can be used in buildings with the chilled water at a temperature closing on the dew point temperature of the outdoor air. It composed of the air-to-water heat exchanger and the air-to-water counter-current padding tower. Simulations were carried out to analyze the performance of the chiller, which include output water temperature, cooling efficiency, and the coefficient of Performance (COP). The results showed that the output water temperature provided by the prototype chiller was around (14–20℃), which was below the wet-bulb temperature and above the dew point of outdoor air. The (COP) of the prototype chiller was about (9). More than (40%) of energy can be saved by replacing conventional air conditioning systems (COP≈2-3) with this type of chiller.

2 DEW POINT EVAPORATIVE COOLING THEORY

The dew point evaporative cooling system can be developed by coupling a cooling tower with a sensible water to air heat exchanger. A system

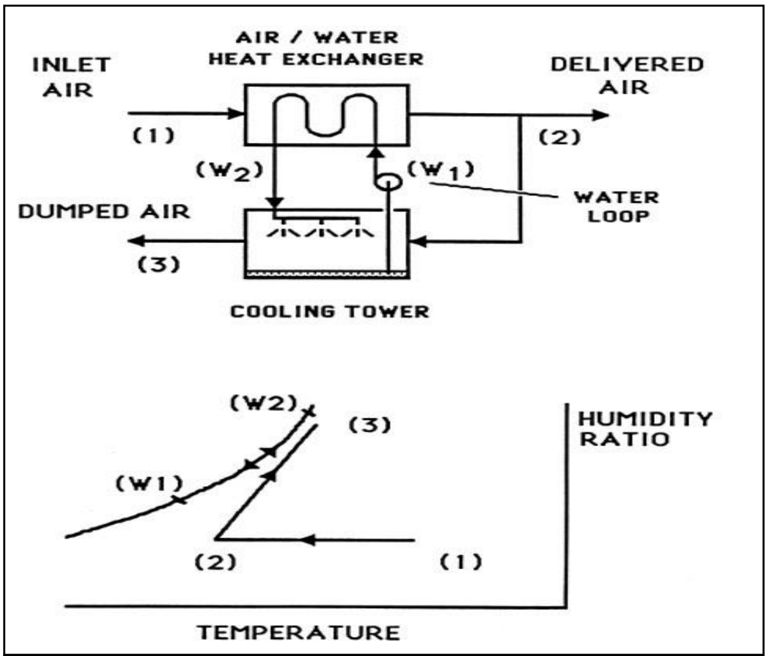


Figure (1) Schematic diagram and process representation of a cooling tower/heat exchanger air

schematic and process representation of this cooler is shown in Fig. (1). In this process, the inlet air at state (1) is sensibly cooled to state (2) while at the same time; the chilled tower water is warmed from state (w_1) to (w_2), the air at state (2) is split into two streams. One is delivered as conditioned air to the building zone and the other is routed through the cooling tower. The portion of the inlet air that is channeled through the cooling tower is humidified and warmed to state (3) and then dumped to the ambient. And warm water returns to the cooling tower at state (w_2), and the water cools in tower due to water evaporation then cooling tower delivers the water at state(w_1) to the heat exchanger. The Cooling Tower/Heat Exchanger (CT/HX) air cooler clearly has great potential for air conditioning applications. It has superior temperature reducing and cooling capacity effectiveness for any value of fraction delivered. The only possible drawback to this system, in comparison to the other evaporative coolers, is its complexity. This cooler uses a cooling tower instead of an evaporative cooler. It also requires a water circulating loop to couple the two components. A smart control system would be required to maintain the cooler near its optimum operating point. One further caution regarding this cooler is that even though the temperature effectiveness is very high, this cooler is still not able to dehumidify the inlet air stream, it would have to be used in conjunction with a dehumidifier to successfully meet a total air conditioning load [9].

3 THEORETICAL WORK

3.1 Mathematical model for water to air heat exchanger

The effectiveness–NTU method greatly simplified heat exchanger analysis. This method is based on a dimensionless parameter called the heat transfer effectiveness ε, defined as [7].

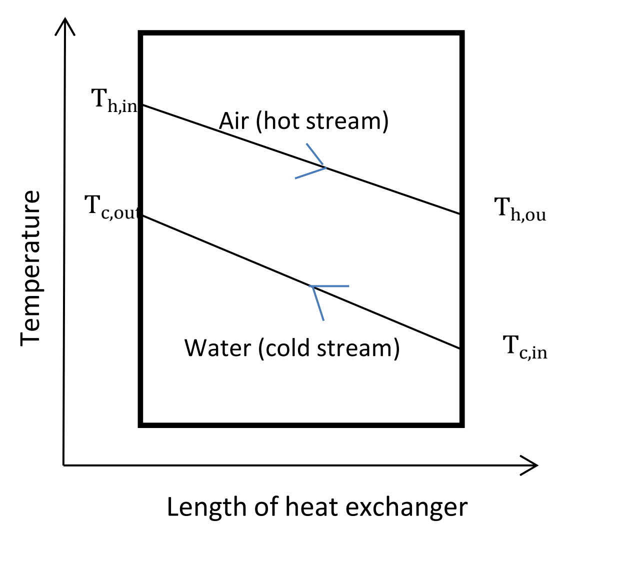


Figure (2) Temperature distribution in counter flow heat exchanger

$$c = \frac{Q_{actual}}{Q_{maximum}} = \frac{\text{actual heat transfer rate}}{\text{maximum possible heat transfer rate}} \quad (1)$$

$$Q_{actual} = C_c\,(T_{c,out} - T_{c,in}) = C_h\,(T_{h,out} - T_{h,in}) \quad (2)$$

$$\Delta T_{max} = T_{h,in} - T_{c,in} \quad (3)$$

$$Q_{max} = C_{min}\,(T_{h,in} - T_{c,in}) \quad (4)$$

$C_{min} = (C_h = m_h C_{p,h})$ or $(C_c = m_c\, C_{p,c})$

$$NTU = \frac{U\,A_s}{C_{min}} = \frac{U\,A_s}{(\dot{m}\,C_p)_{min}} \quad (5)$$

$$c = \frac{C_{min}}{C_{max}} \quad (6)$$

$$\epsilon = \text{function}\ \left(\frac{U\,A_s}{C_{min}}, \frac{C_{min}}{C_{max}}\right) = f\,(NTU, c) \quad (7)$$

3.2 Mathematical model for wet aspen pad (heat and mass transfer)

To evaluate the performance of the evaporation media for cross flow, Merkel's theory is applied to the pads [8].

Assumptions of the Merkel's theory are:

The Lewis factor relating heat and mass transfer is unity. This assumption has small influence but affects results at low ambient temperature.

The air exiting the tower is saturated with water vapor and it is characterized only by its enthalpy. This assumption regarding saturation has a negligible influence above an ambient

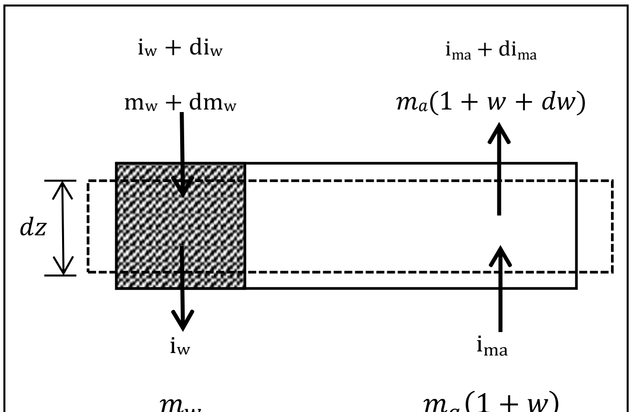


Figure (3) Control volumes of the counter flow temperature of 20°C but it is significant at lower temperatures.

The reduction of water flow rate by evaporation is neglected in the energy balance. This energy balance simplification has a greater influence at elevated ambient temperature.

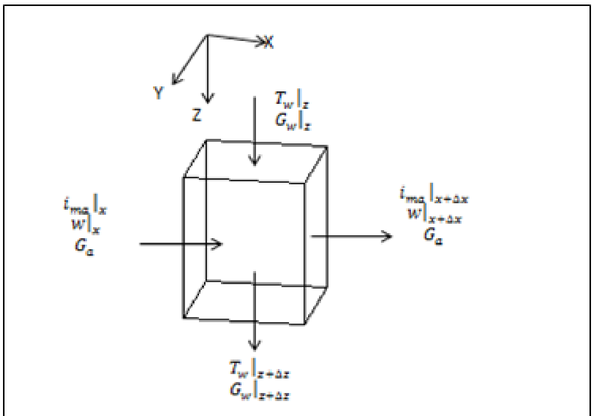


Figure (5) Control volumes of cross flow fill

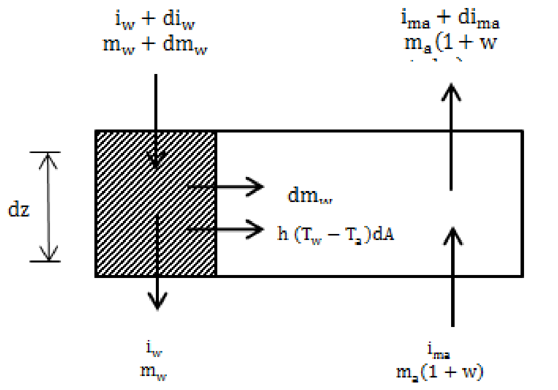


Figure (4) Air side control volume of the fill

Governing equations for wet pad are;

$$\frac{\partial G_w}{\partial z} = -G_a \frac{h_d a_{fi}}{G_a}(w_{sw} - w) \tag{8}$$

$$\frac{\partial w}{\partial x} = \frac{h_d a_{fi}}{G_a}(w_{sw} - w) \tag{9}$$

$$\frac{\partial i_{ma}}{\partial x} = \frac{1}{G_a}\frac{\partial q}{\partial x} = \frac{h_d a_{fi}}{G_a}[i_{masw} - i_{ma} + (Le_f - 1)\{i_{masw} - i_{ma} - i_v(w_{sw} - w\}] \tag{10}$$

$$\frac{\partial T_w}{\partial z} = \frac{G_a h_d a_{fi}}{C_{pw} G_w G_a}\left[\begin{array}{c}(w_{sw} - w)C_{pw}T_w - (i_{masw} - i_{ma}) - \\ (Le_f - 1)\{i_{masw} - i_{ma} - (w_{sw} - w)i_v\}\end{array}\right] \tag{11}$$

$$Me_m = \frac{h_d a_{fi}}{G_a} \quad \text{(Merekel number from the mathematical model)} \tag{12}$$

$$Me = \frac{G_a}{G_w} Me_m \tag{13}$$

$$Me = \frac{h_d a_{fi} L}{G_w} \tag{14}$$

A mathematical model is required to design and compute the performance of the system. The system consists of the heat exchanger (water to air heat exchanger) and the heat and mass exchanger (wet aspen pad). A mathematical model for both parts is required to accomplish the design of the system. In this section the Merkel's theory applies to the wet pad, a mathematical model for the wet aspen pad achieved by solving governing equations (8, 9, 10, 11, 12, 13, and 14). The Merkel number is a non-dimensional coefficient of performance, there are some unknown parameters in (Me), and it can be calculated with trial solution in the model.

4 EXPERIMENTAL WORK

The dew point evaporative cooler with circulating air in a form of combination between sensible water to air heat exchanger and cooling tower has been designed, constructed and tested in Erbil Polytechnic University. It is a modified indirect evaporative cooler. The system consists of two main parts, which are water to air compact sensible heat exchanger with the dimensions (60*40*6) cm, and evaporative cooling media (aspen pads) with dimensions (60*60*15) cm. Three layers of aspen pad were used. The incoming (ambient) air was sensibly cooled through water to air heat exchanger, utilizing the cooled water which is pumped from atmospheric tank placed underneath the evaporative media. Part of the cooled air was diverted to the room. The flow rate of this air was controlled by using duct dampers and variable speed fan. The other part of the cooled air was diverted to the evaporative media after been mixed with return air from the room. The ratio of mixing was controlled by the duct dampers and variable speed fan. Water was pumped from the atmospheric tank to the heat exchangers and then pumped to evaporative media to be cooled again. The water was sprayed on wet pad and cool air passed through the pad, some water evaporates and goes into the air, the air will be humid and cool. Because high humidity causes discomfort, it was dumped to outdoor. The water molecules that evaporated had higher kinetic energy than water remain, that is why remaining water can be cooled even if both incoming air and water have higher temperatures. Water and air flow rates were measured by rotameter and anemometer respectively. Water flow rate was regulated via globe valves. Also the temperature of air and water were measured by resistance temperature detector (RTD). The system can theoretically approach the dew point temperature of the incoming air, which is the lowest temperature that can be achieved by evaporative cooling without using vapor compression refrigeration system.

Figure (6 a) Picture of the dew point evaporative cooling system

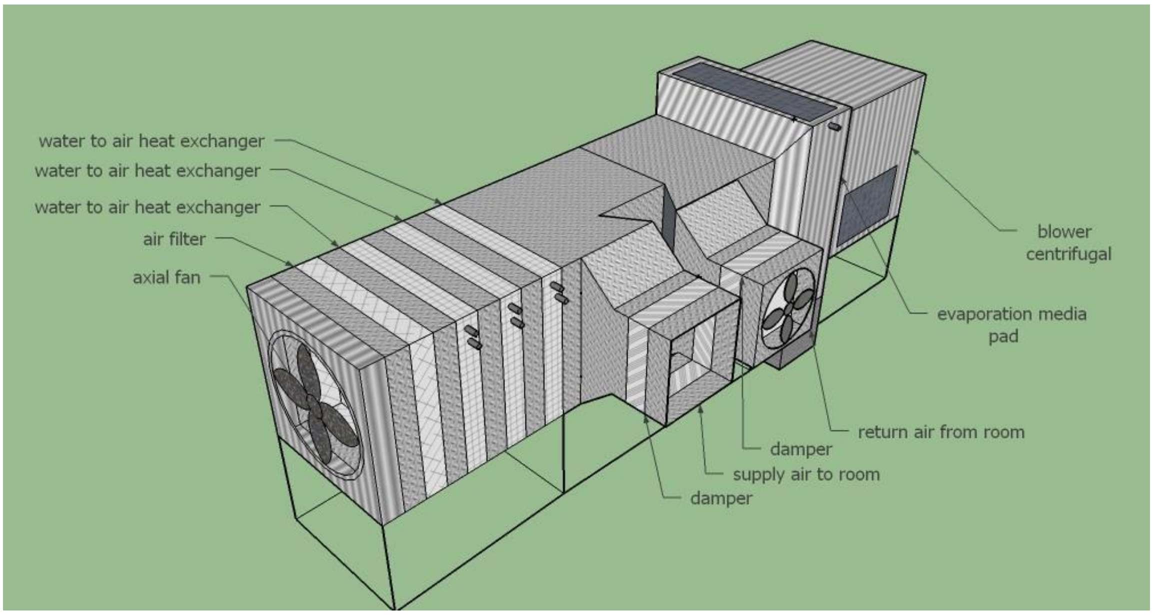


Figure (6 b) schematic of dew point evaporative cooling system

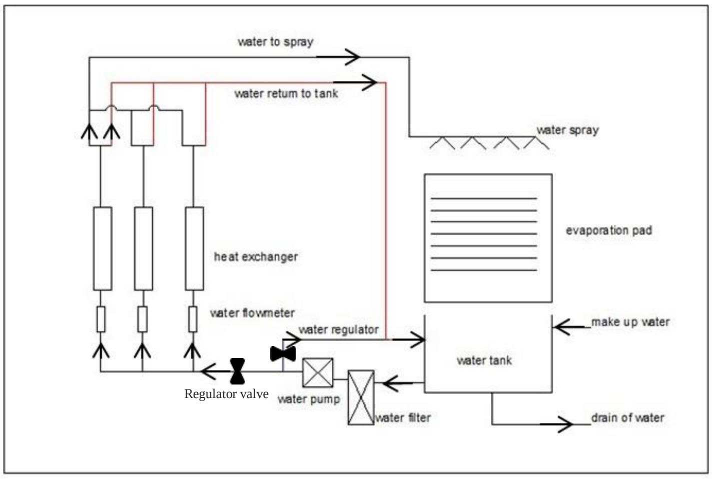
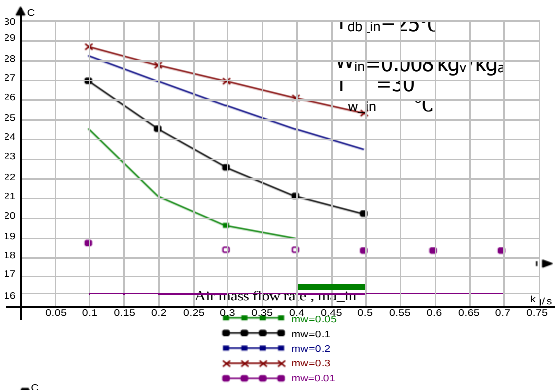


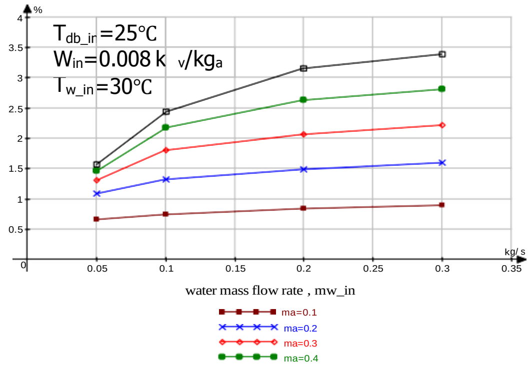
Figure (6 c) water flow diagram of the dew point evaporative cooling system

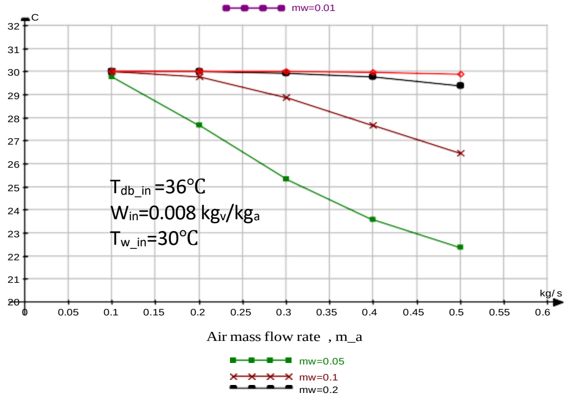
5 RESULTS AND DISCUSSION

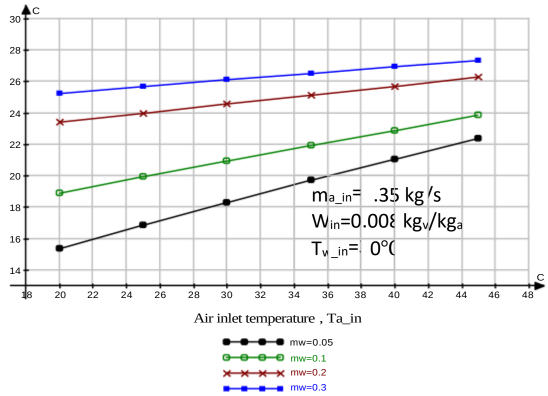
Analyzing theoretical and experimental results in section (5), the effect of all important parameters on water evaporation through the pad is studied and results shown in section (5.1).

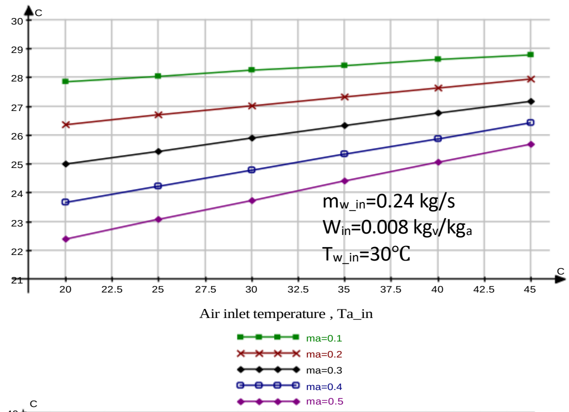
5.1 Theoretical results of the heat and mass balance in wet pad

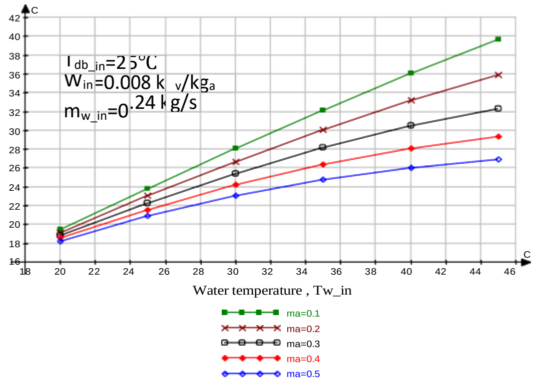

mw=0.05
mw=0.1
mw=0.2
mw=0.3
mw=0.01

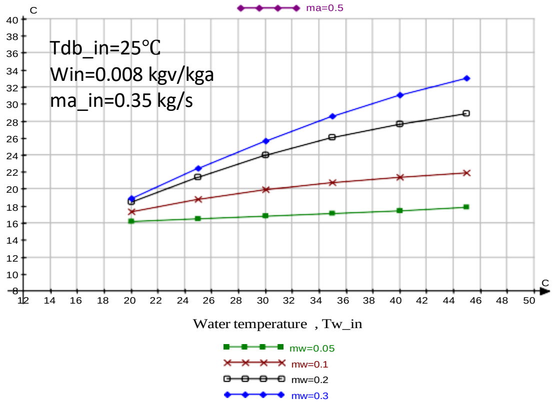

T_{db_in}=25°C
water mass flow rate , mw_in
ma=0.1
ma=0.2
ma=0.3
ma=0.4

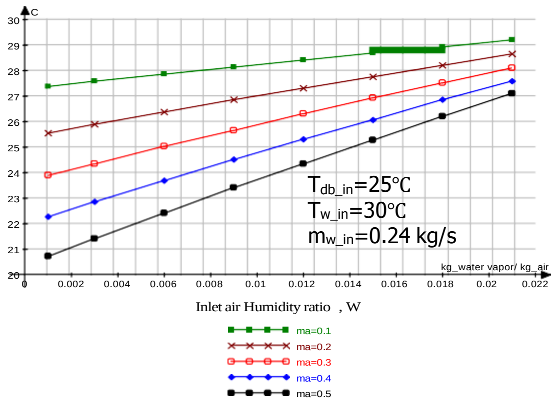

T_{db_in}=36°C
W_{in}=0.008 kg_v/kg_a
T_{w_in}=30°C
Air mass flow rate , m_a
mw=0.05
mw=0.1
mw=0.2


Air inlet temperature , Ta_in
mw=0.05
mw=0.1
mw=0.2
mw=0.3


m_{w_in}=0.24 kg/s
W_{in}=0.008 kg_v/kg_a
T_{w_in}=30°C
Air inlet temperature , Ta_in
ma=0.1
ma=0.2
ma=0.3
ma=0.4
ma=0.5


Water temperature , Tw_in
ma=0.1
ma=0.2
ma=0.3
ma=0.4
ma=0.5


Tdb_in=25°C
Win=0.008 kgv/kga
ma_in=0.35 kg/s
Water temperature , Tw_in
mw=0.05
mw=0.1
mw=0.2
mw=0.3


T_{db_in}=25°C
T_{w_in}=30°C
m_{w_in}=0.24 kg/s
kg_water vapor/ kg_air
Inlet air Humidity ratio , W
ma=0.1
ma=0.2
ma=0.3
ma=0.4
ma=0.5

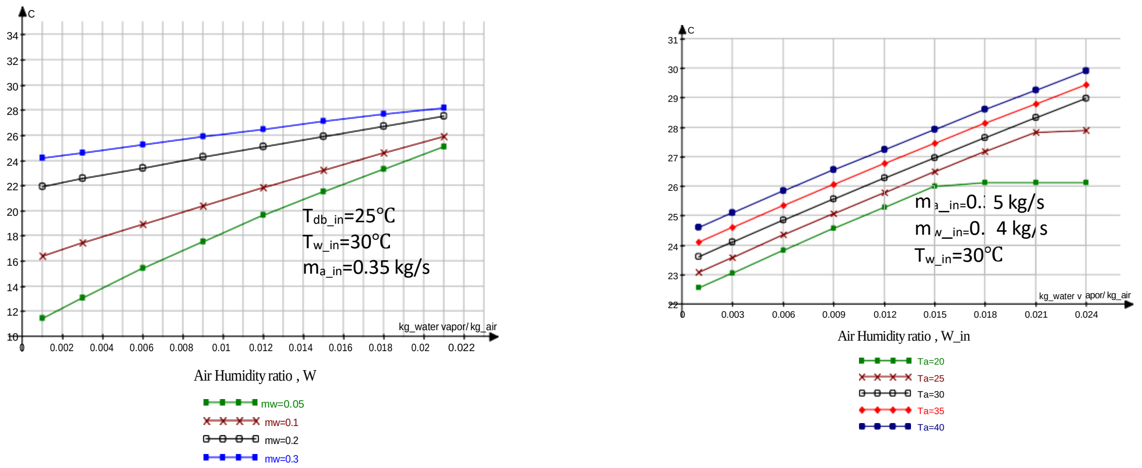


Figure (7) Effect of thermodynamic parameters on heat and mass transfer in wet pad

5.2 Experimental results of the system

All the experimental results are shown in section (5.2). The effect of important parameters is experimentally shown in this section.

5.2.1 Effect of supply air mass flow rate on the system

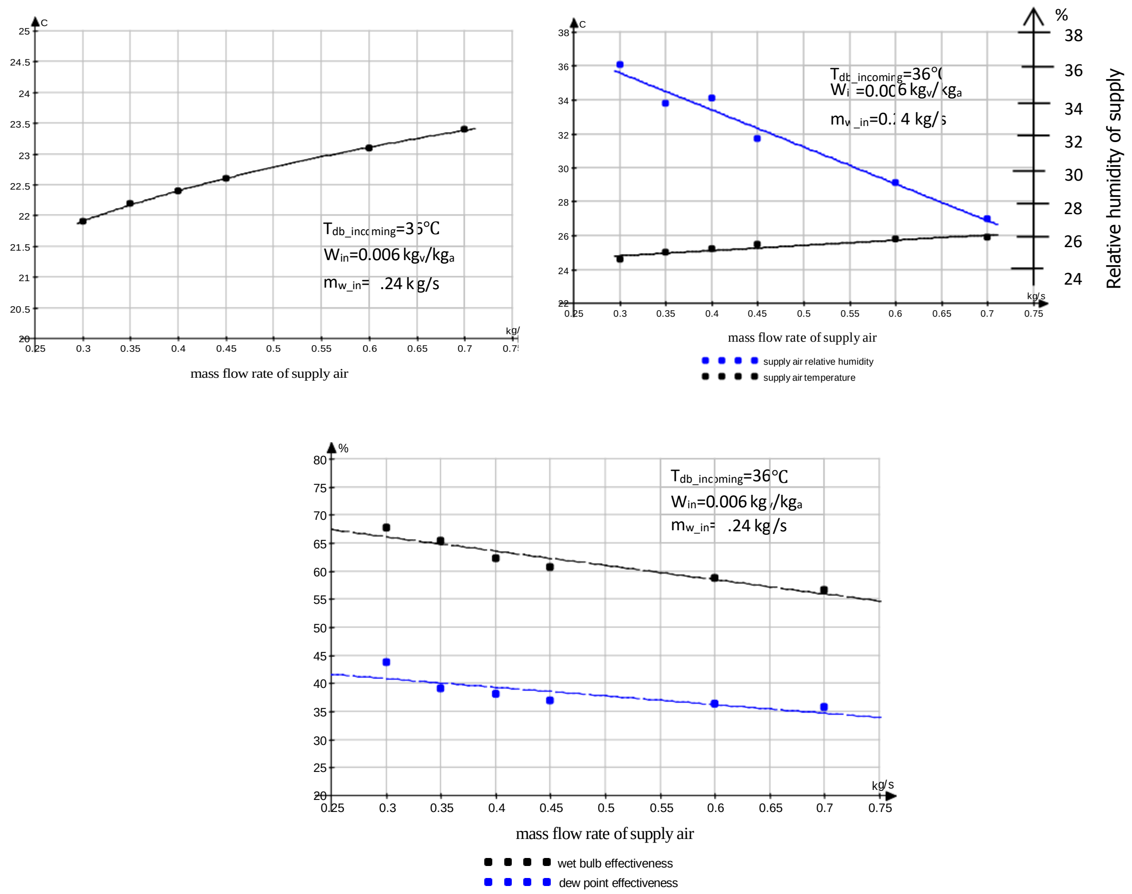


Figure (8) Effect of supply air mass flow rate on the system

5.2.2 Effect of water spray mass flow rate on the system

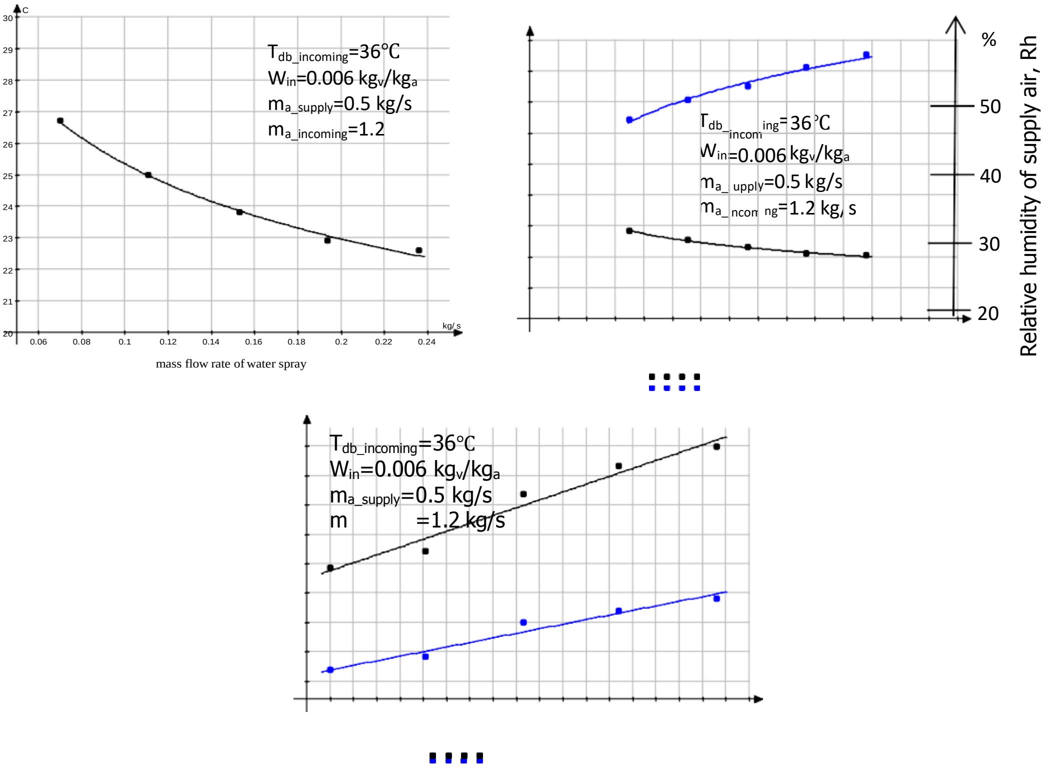


Figure (9) Effect of water spray mass flow rate on the system

5.2.3 Effect of incoming air dry bulb temperature on the system

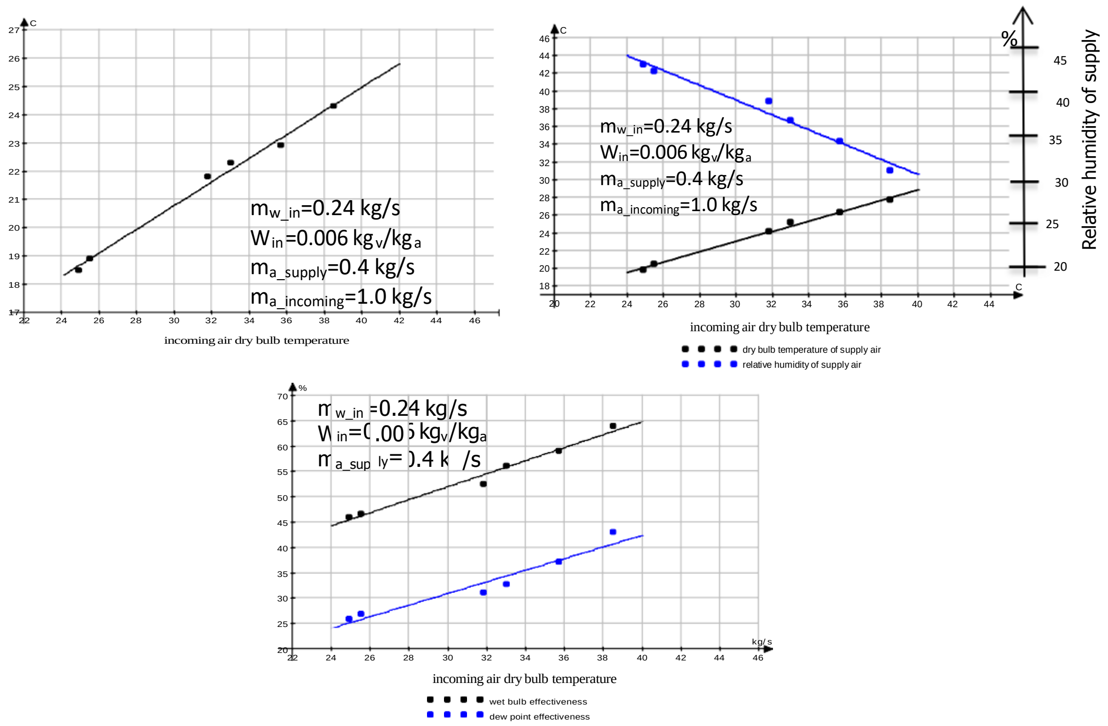


Figure (10) Effect of incoming air dry bulb temperature on the system

5.2.4 Effect of dry bulb temperature of incoming air on wet bulb effectiveness with different supply air mass flow rate, and effect of using return air

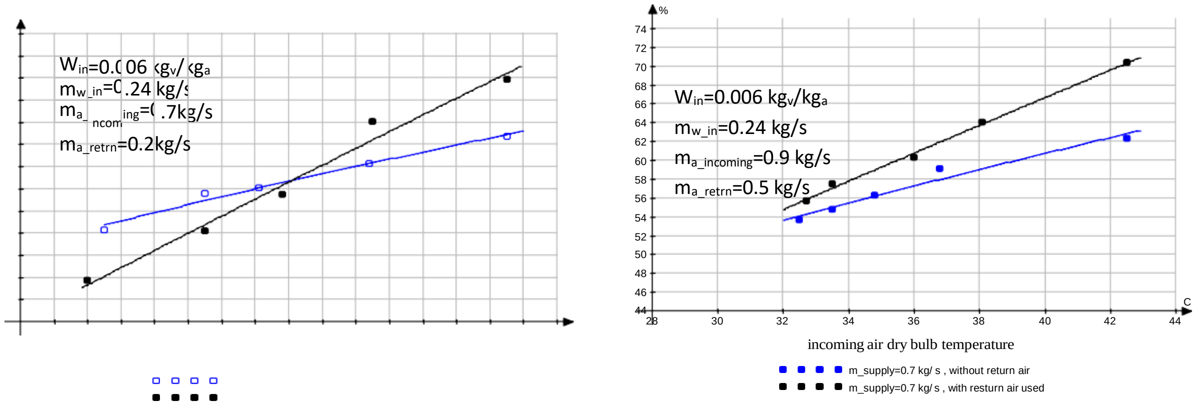


Figure (11) Effect of using return air on the system at different dry bub temperature

5.2.5 Effect of supply air mass flow rate on wet bulb effectiveness at different incoming air dry bulb temperature, and effect of using return air

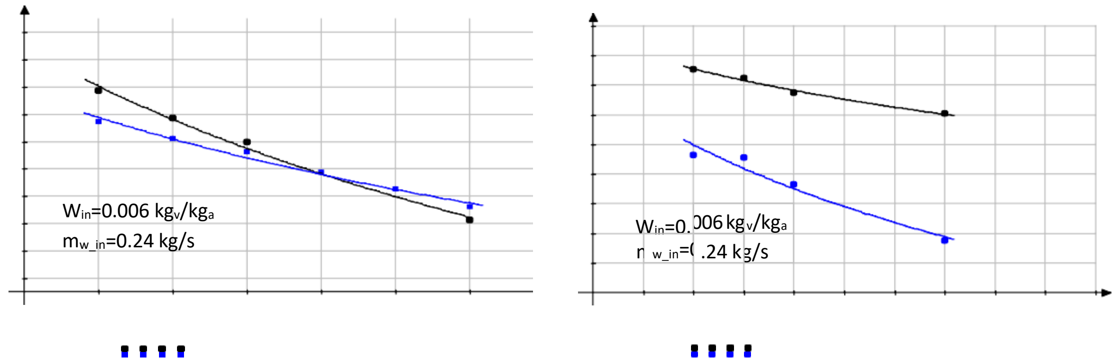


Figure (12) Effect of using return air on the system at different supply air

5.3 Experimental and theoretical results of the system

A theoretical and experimental result is analyzing and comparing in this section, the figures shows the deviation of the model with experimental system.

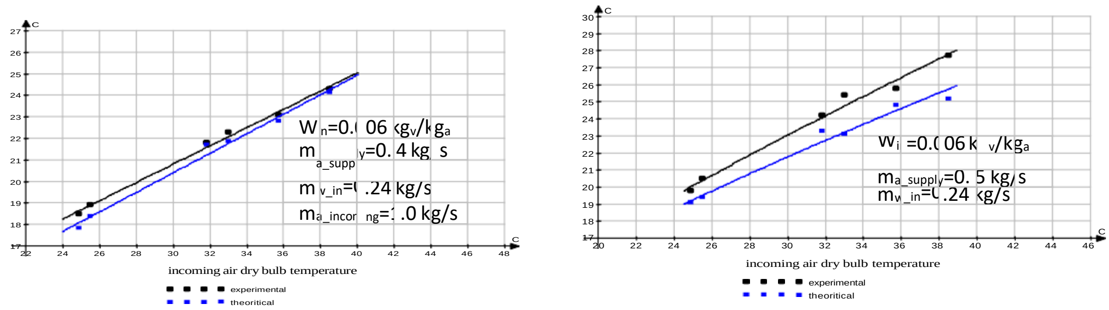


Figure (13) Comparison between theoretical and experimental results

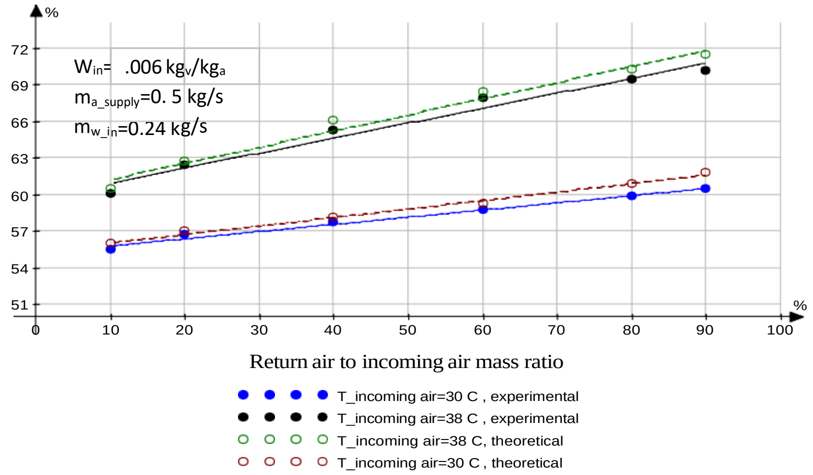


Figure (14) Effect of return/incoming air ratio on the system effectiveness

5.4 Theoretical results of the system

All the theoretical result is analyzing in this section, controlling some parameter is difficult in the system to get results experimentally, so the effect of these parameters is taken theoretically.

5.4.1 Effect of humidity ratio of incoming air on the system

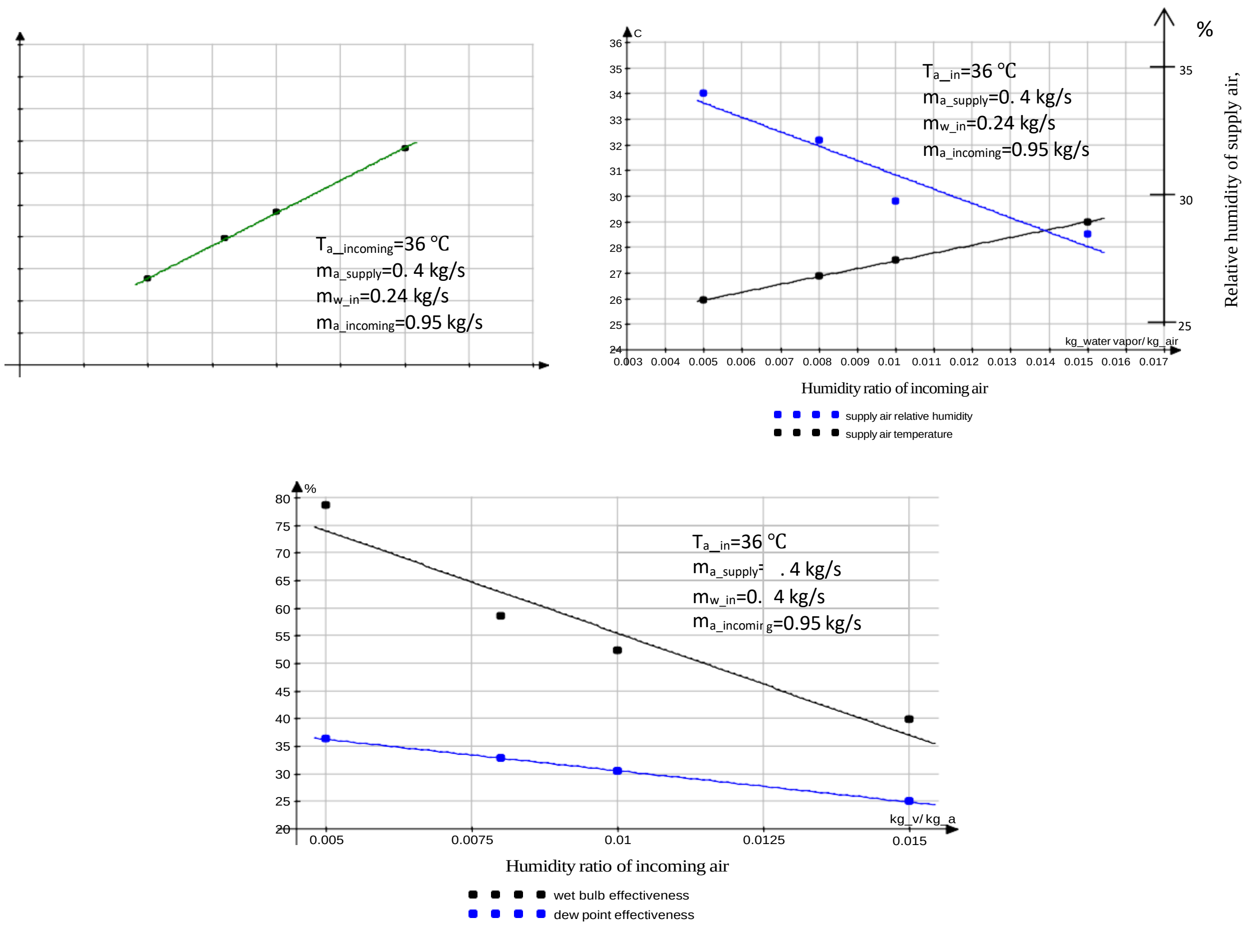


Figure (15) Effect of incoming air humidity ratio on the system

5.4.2 Effect of using air to air heat exchanger as an economizer in the system

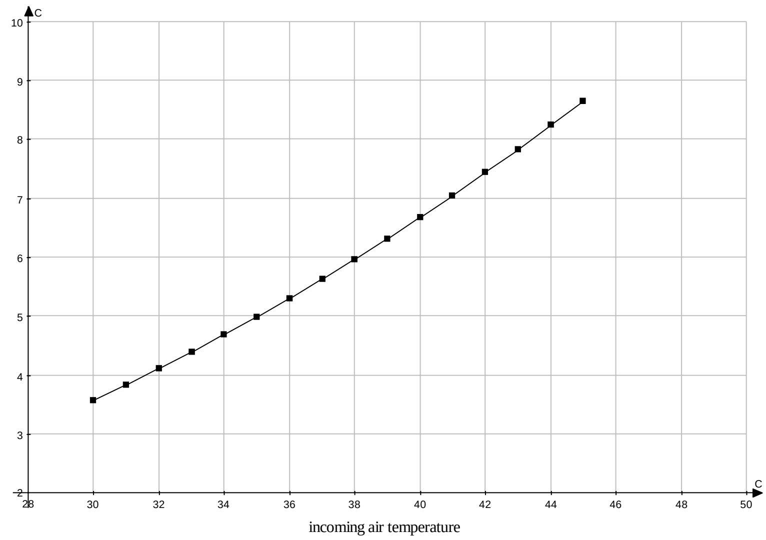


Figure (16) Temperature reduction with using an economizer

5.4.3 Effect of using both return air from the room and air to air heat exchanger as an economizer in the system

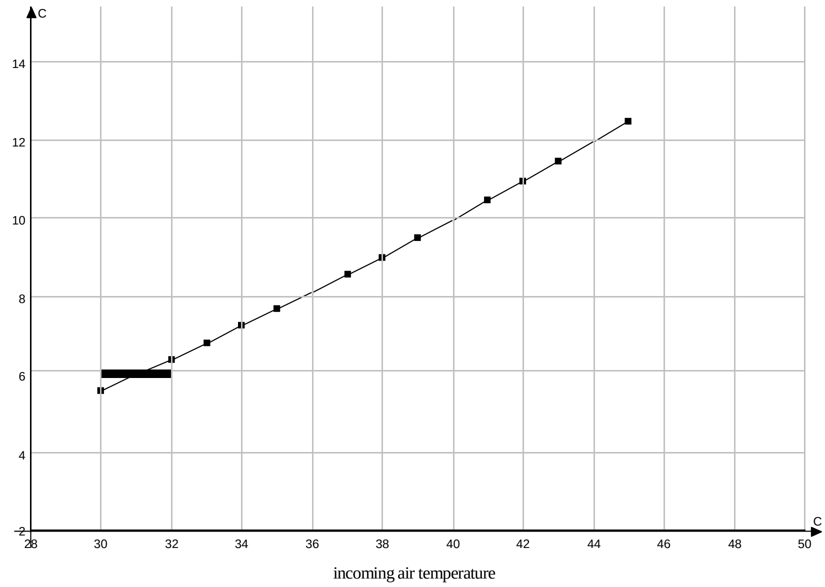


Figure (17) Temperature reduction with using return air and economizer

Theoretical results in section (5.1) show that, the rate of water evaporation increases and the air outlet temperature decrease with increasing water mass flow rate in the pad. The water outlet temperature increase with increasing; inlet air temperature, inlet water temperature and inlet air humidity ratio but the water outlet temperature decrease with increasing inlet air mass flow rate to the pad. When the air reaches saturation condition, the evaporation process will stop because the air cannot hold water anymore. Experimental results in section (5.2) shows that, with increasing the supply air mass flow rate in the system the water temperature in the tank increases and this increasing in water temperature will cause decrease in the system effectiveness. Also it shows that, with increasing the water spray mass flow rate, the water temperature decrease and system effectiveness increase. With increasing incoming air temperature the water temperature increase and system effectiveness decrease. Using return air will increase system effectiveness when there is insufficient air in the pad. The experimental and theoretical results are compared in section (5.3). In section (5.4), the theoretical results of the system indicate that, with increasing inlet air humidity ratio the water temperature in the tank increase and system effectiveness decrease. Also it shows that, the using air to air heat exchanger as an economizer will decrease the water temperature in the tank. Therefore, adding economizer to the system and using return air will increase the system effectiveness.

6 CONCLUSIONS

A theoretical model has been used to test the performance of the system which concludes;

- Experimental and theoretical results are in good agreement as presented graphically in section (5) with (4% to 10%) deviation.
- Using the return air from the cooled zone can improve the wet bulb effectiveness by (9%).
- Using return air from the room in the system depends on both, the incoming air dry bulb temperature and the supply air mass flow rate in the system. When incoming air temperature is (30℃), using return air, improves the performance of the system only when supply air mass flow rate is higher than (0.5 kg/s). But when incoming air temperature is (36℃), using return air, improves the performance of the system for all range of the supply air flow rate.
- Using exhaust humid air of the system (air leaving the evaporative media) in air to air heat exchanger as an economizer can improve the wet bulb effectiveness by (10%).
- Using both return air from the cooled zone and economizer for exhaust humid air from discharge of evaporative media can improve the wet bulb effectiveness in total by (21%) for (45℃) of incoming air temperature.
- Experimentally, the system achieves wet bulb effectiveness up to (73%) with using return air.
- The wet bulb effectiveness of the system can be improved to (93%) by using both the return air from the cooled zone and the economizer for precooling incoming air via exhaust humid air.
- The dry bulb temperature of the supply air in this system ranges from (21℃ to 28℃) with (30% to 50%) of relative humidity for a range of (25℃ to 45℃) of incoming air, without using the economizer.
- The results are within the thermal comfort criteria. The (COP) of the system is (8), while the (COP) of conventional air conditioning cooling system is (2 to 3).

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