Performance Enhancement by Using Wet Pad in Vapor Compression Cooling System

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Abstract. Vapor compression cooling systems are widely used in hot and dry climates where the atmospheric temperature in summer reaches around 48 °C and the relative humidity can be below 20%. These cooling systems normally use environmentally friendly gases that do not harm the ozone layer. These systems work with high gas pressure, which consumes high energy and leads to a low coefficient of performance when compared to cooling systems that do not use an environmentally friendly gas. Theoretical and experimental studies have been conducted to find a suitable solution for the performance improvement of this type of cooling system. A possible solution is to lower the temperature of the air before it enters the condenser of the compression cooling system by passing it through a wet pad. The water evaporates due to the latent heat that is withdrawn from the wet pad. The air temperature is reduced to about 11 °C before it enters the condenser of the compression cooling system. This enhancement increases the cooling capacity of the vapor compression cycle by nearly 20% and increases the coefficient of performance by 15%, in addition to reducing the consumed electricity up to 15%. In conclusion, the proposed method of adding a wet pad can significantly enhance the performance of the vapor compression cooling system.

Keywords: *cooling pad; direct evaporative; performance; vapor compression cooling system; wet pad.*

1 Introduction

Evaporative cooling technologies, for example conventional vapor compression systems, have been improved in the last decade [1-4]. In desert weather, direct evaporative coolers are used to control air temperature. The operation principle of these systems is water evaporation to cool the air stream directly [5-9]. In this process, the air passes through a wet pad in the direct evaporative cooling (DEC) system and then proceeds to the condenser of the vapor compression cooling system. DECs are essentially plastic or perforated metal boxes that contain one or more wet pads. The wet pads are made of a porous material [9-11]. Water is pushed continuously through them, starting at the upper edge, in

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order to keep them wet [6,12]. Air is blown through the wet pads by generator fans in the vapor compression cycle. The water is sprayed on the top edges of the pad and then moves down by gravity and capillarity action [13-18]. A water pump re-circulates the water from a basin to the upper surface of the wet pad to maintain the operation. The processed air passes through the cooler's evaporative (wet) pad, hence being humidified and cooled. A new design with high commercialization potential for an integrated evaporative cooling system in the condenser of a window type air conditioner was investigated experimentally in [19]. Before passing over the condenser of the air conditioner, the air passes through two cooling pads that are located on both sides of the air conditioner. Water is continuously injected into the cooling pads to keep them wet. The results showed that the coefficient of performance was increased by about 55% and power consumption was decreased by approximately 16%. Also, the results showed that the proposed design would experimentally improve the heat capacity of the air conditioner.

The performance and power consumption of an evaporative cooler used to cool the condenser of a vapor compression cycle cooler was investigated in [20]. The results showed that the performance was improved significantly compared with an air-cooled condenser. This improvement led to an increase in the ambient temperature. Also, the power consumption was decreased by nearly 20% and the total improvement of the performance was around 50%. It was found that an increase in the ambient air temperature decreased the coefficient of performance of the air-cooled condenser considerably, while it had a much smaller effect in increasing the performance of an ordinary evaporative cooled air condenser. Alotaibi, *et al.* [21] experimentally investigated an evaporative cooling design of an air conditioner using an air-cooled condenser with water atomization. The results showed that the coefficient of performance increased on average by about 13%, while the power consumption decreased on average by about 11%. Also, the thermodynamic characteristics of the proposed system were considerably improved [22].

Kassai, *et al.* [23] studied the optimization of energy consumption of a DC refrigerator and a test facility was installed in the introduction room of a national refrigeration company. Due to the imperfect data provided by the manufacturer, the PID controller of the DC refrigerator had to be tuned by setting the correct proportional, integral and derivative parameter values to achieve optimal energetic operation. The combined effects of the electronic expansion valve, scroll compressor operation and correct experimental settings of the PID controller in the DC refrigerator resulted in around 62.4% energy savings compared to a traditional on-off controlled appliance under the same operational conditions in the same cold store [24,25].

An experimental study of the relation between coefficient of performance and condenser inlet dry bulb temperature with and without evaporative cooling was performed by Jain & Hindoliya [26]. The results showed that there was an inverse relationship between the coefficient of performance and the condenser inlet dry bulb temperature. Also, the results showed that using evaporative cooling led to a reduction in power consumption of up to 14.3% and the coefficient of performance increased on average by about 6.1%.

An investigation of the effect of water and air flow rate on cooling capacity and saturation efficiency was performed experimentally by Khobragade & Kongre [27]. They used different cooling pad materials, such as wood-wool, cellulose and khus-grass material. The cooling capacity and saturation efficiency were studied for a 4 inch thick cooling pad material. It was also calculated for varying water flow rates (60 to 100) cm^3/hr and varying air flow rates (0.25 to 0.45) m^3/s. The results showed that the cooling capacity increased with the increase of air flow rate. The obtained values were (1.1 to 6.72) kW for different pad materials. The khus-grass material gave the lowest saturation efficiency value (about 40.13%), while the cellulose material gave the highest saturation efficiency value (about 92.8%). Several studies have investigated hybrid systems in a hot and humid tropical climate to improve the performance of the vapor compression cooling system. The present study investigated a hybrid cooling system in a hot and dry climate. Cooling pads were used to reduce the inlet air temperature of the condenser section and enhance the performance of the vapor cooling system.

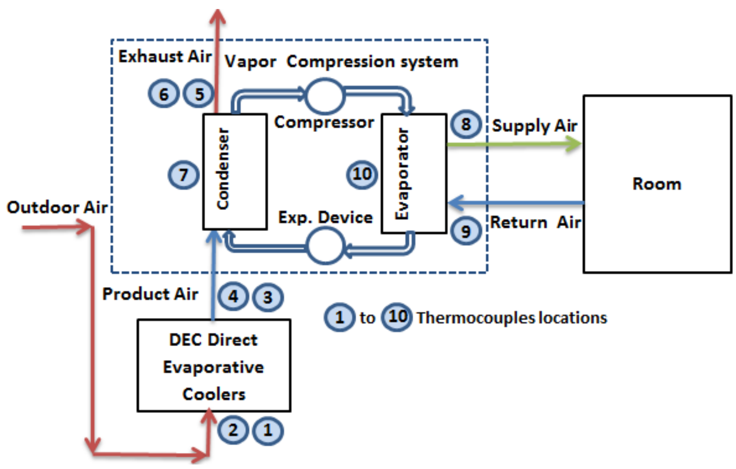


Figure 1 Schematic drawing of vapor compression cooling system with direct evaporative cooler.

2 Experimental Set-up and System Assumptions

A commercially available vapor compression cooling system was improved by adding a regenerative direct evaporative cooler that fit in front of the condenser of the vapor compression system. This evaporative (wet pad) cooler consisted of a galvanized plate made into a rectangular perforated cabinet with (1.41 m) width, (1.04 m) height and (0.07 m) thickness. The packing material (wet pad) consisted of wood wool with a specific surface area of 503 m^2/m^3 and was put inside the perforated cabinet. A separate pump was used to supply water from the water tank to the water distributor installed in the upper part of the evaporative wet pad system. Ten thermocouples were fitted and then connected to a data logger to measure the dry and wet bulb temperatures at various points (before and after the wet pad, the condenser and the evaporator of the vapor compression system and the supplied water) as shown in Figure 1.

The following assumptions were used in this study:

1. Working fluid properties are constant throughout the vapor compression cycle.
2. Heat transfer (by conduction) along the condenser tube is neglected.
3. The heat losses in the pipes are negligible.
4. The liquid exiting from the condenser is saturated.
5. The vapor exiting from the evaporator is saturated.
6. The expansion valve works as an adiabatic cycle.
7. The pressure drop inside the pipes is negligible.

3 Mathematical Modeling of Direct Evaporative (Wet Pad) Cooler and Vapor Compression Cooling System.

The total wet area of the pad material was calculated using the following Eq. (1) [28,29]:

$$A_{TP} = A_P * V_P \tag{1}$$

where

(A_P) = wet pad area per cubic meters m^2/m^3.
(V_P) = wet pad volume, calculated in Eq. (2) as follows:

$$V_P = h_P \times w_P \times t_P \tag{2}$$

The velocity with which the air passes through the wet pad was calculated in Eq. (3) as follows:

$$v_{FP} = \frac{V_p}{A_{FP}} \tag{3}$$

where AFP is the wet pad face area, calculated in Eq. (4) as follows:

$$A_{FP} = h_P \times w_P \tag{4}$$

The amount of water that evaporated from the wet pad ($\dot{M}_{wep}$) was calculated in Eq. (5) as follows:

$$\dot{M}_{wep} = \frac{\exists_P \times A_{TP} \times (W_S - W)}{3600} \tag{5}$$

where ($\exists$p) represents the evaporation coefficient, calculated in Eq. (6) using the following imperial formula:

$$\exists_P = (25 + 19 \times V_{FP}) \tag{6}$$

The moisture content leaving the wet pad (outlet) was calculated in Eq. (7) as follows:

$$W_{op} = W_i + \left(\frac{\dot{M}_{wep}}{\dot{m}_{ad}}\right) \tag{7}$$

The air cooling capacity of the wet pad was calculated in Eqs. (8) and (9) as follows [30]:

$$Q_{cp} = \dot{M}_{wep} \times H_{fg} \tag{8}$$

$$TD_{op} = \left(\frac{(\dot{V_p} \times \rho_m \times Cp_m \times TD_{ot}) - q_{cp}}{\dot{V_p} \times \rho_m \times Cp_m}\right) \tag{9}$$

The relative humidity of the air outlet from the wet pad ($\emptyset_{op}$) was calculated in Eq. (10) as follows:

$$\emptyset_{op} = \left[\frac{P_B \times W_i}{(0.622 + W_i) \times P_{sop}}\right] \tag{10}$$

The effectiveness of the wet bulb (η_{wbp}) was calculated in Eq. (11) as follows [30]:

$$\eta_{wbp} = \left[\frac{TD_{ot} - TD_{op}}{TD_{ot} - TW}\right] \times 100 \tag{11}$$

The solution of the vapor compression cooling system can be found based on the energy analysis. The energy balance for the evaporator ($Q_L = Q_{4-1}$) as shown in Figure (2) can be described in Eq. (12) as follows [31]:

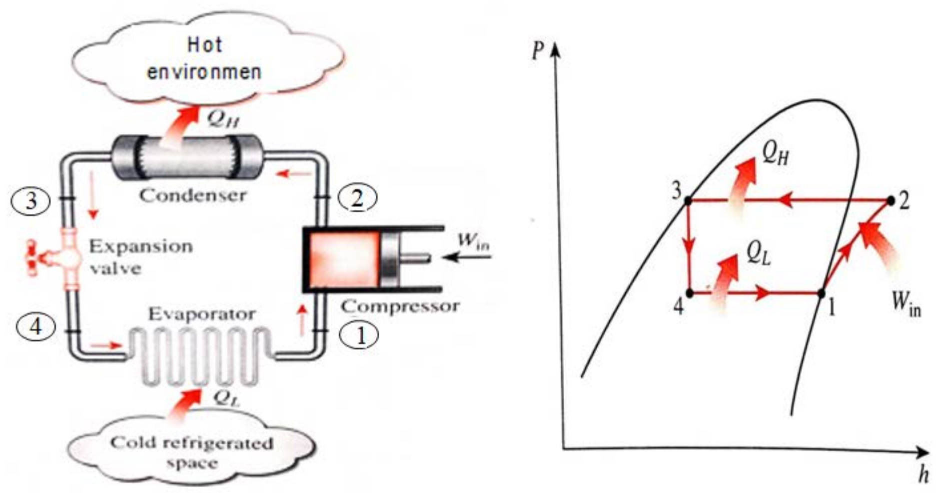
$$Q_{4-1} = \dot{m}_f (h_1 - h_4) \tag{12}$$

The energy balance for the compressor (W_{1-2}) can be written in Eq. (13) as follows:

$$W_{1-2} = \dot{m}_f (h_1 - h_2) \tag{13}$$

The energy balance for the condenser ($Q_H = Q_{2-3}$) can be written in Eq. (14) as follows:

$$Q_{2-3} = \dot{m}_f(h_3 - h_2) \tag{14}$$



(a) Refrigeration cycle (b) Schematic diagram for the refrigeration cycle

Figure 2 Schematic drawing for the operation of the vapor cooperation system.

The energy balance for the expansion valve (1-4) can be written in Eq. (15) as follows:

$$h_1 = h_4 \tag{15}$$

The coefficient of performance of the vapor compression cycle COP [32,33] was calculated in Eq. (16) as follows:

$$COP = \frac{\dot{m}_f(h_1 - h_4)}{\dot{m}_f(h_2 - h_1)} = \frac{(h_1 - h_4)}{(h_2 - h_1)} \tag{16}$$

4 Results and Discussion

An experimental rig was built to test the thermal performance of the vapor compression cycle. The test rig was fully instrumented to enable the measurement of ambient temperature, dry and wet bulb temperatures and air flow rates during the whole day. The measurement readings were registered each day (24 hours) for two consecutive months (July and August).

Figures 3 to 6 show the variation of dry and wet bulb temperatures and the relative humidity of the ambient air for each hour during the day for Tikrit city during the summer season and (July and August) respectively. Figures 3 and 5 show a significant difference between dry air bulb temperature and wet air bulb temperature. This difference was large during the period from 9 AM to 4 PM. The dry bulb temperature had the highest value during this period. It is also noted that the dry bulb temperatures in August were always higher than the July temperatures.

Figures 4 and 6 show the hourly air relative humidity variation during the whole day in July and August. The highest values of air relative humidity during July and August were 34% and 31% respectively. These values were obtained during the first hours of the morning, when the air temperature was relatively low. The air relative humidity decreased sharply after sunrise, when the dry bulb temperatures began to increase. Generally, the climatic conditions were always relatively dry. Therefore, the direct evaporative cooling system would be activated during the day hours.

The histogram (bar chart) presented in Figures 7 and 8 represents the number of operating hours at different ambient temperatures in July and August, respectively. When the ambient temperature was higher than 40 °C, the temperature on the high pressure side, which represents the condenser of the vapor compression cycle, reached the set conditions and the wet pad started working. When the cooling pads work, the temperature of the air that enters the condenser starts decreasing due to the effect of the direct evaporative cooler, hence the number of operating hours will shift to the left. In a hot and dry desert climate, the working hours at temperatures higher than 40 °C account for more than 30% of the day hours during July and 40% during August. This gives a good impression of the importance of direct evaporative cooling for lowering the temperature of the air entering the condenser of the vapor compression cycle.

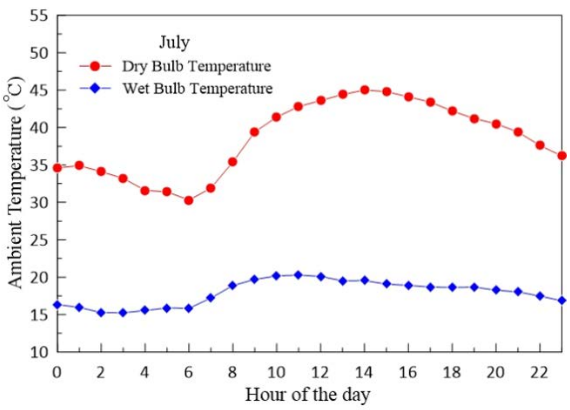


Figure 3 Ambient temperature during one day in July.

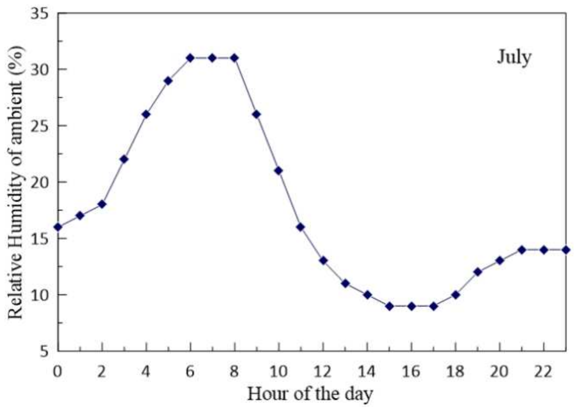


Figure 4 Variation of relative humidity during one day in July.

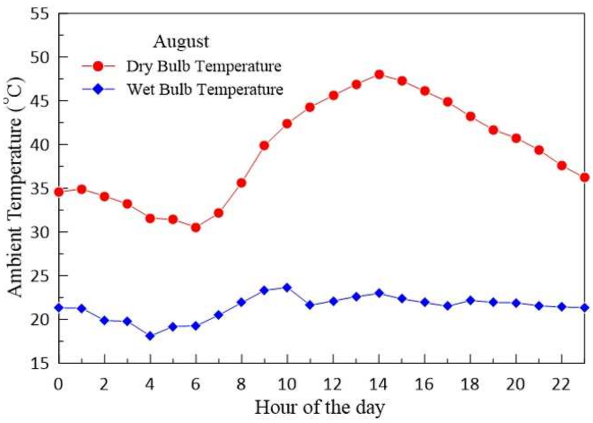


Figure 5 Ambient temperature during one day in August.

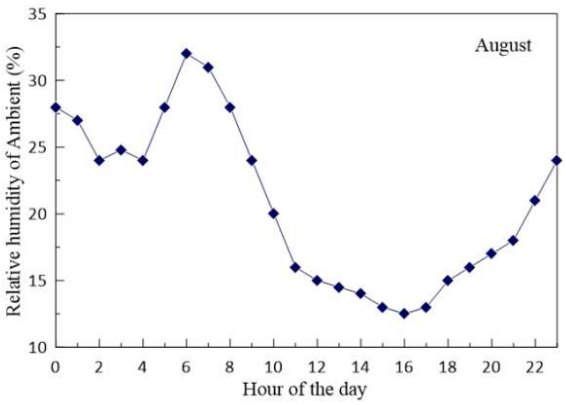


Figure 6 Variation of relative humidity during one day in August.

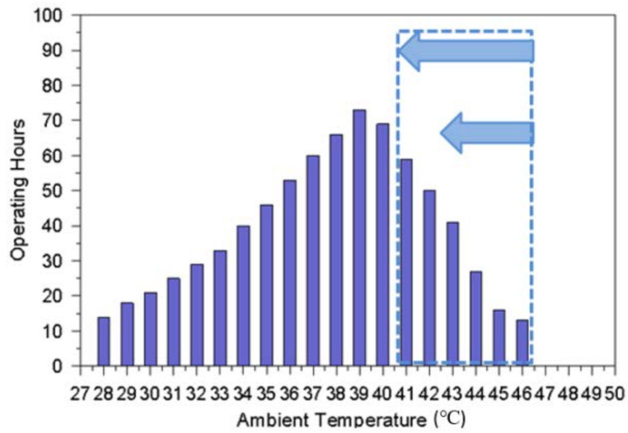


Figure 7 Variation of the number of operating hours with ambient temperature in July.

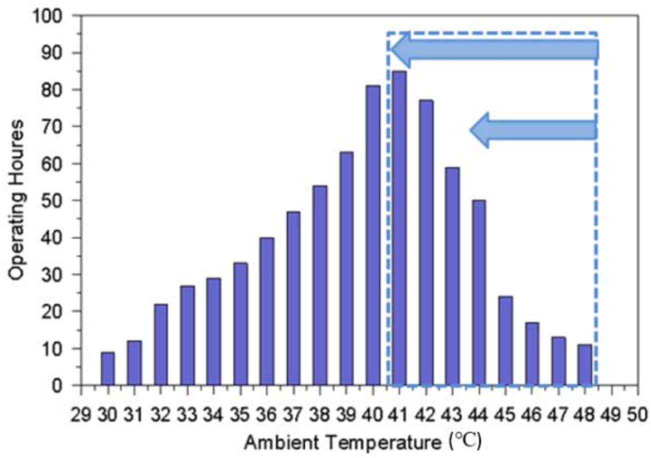


Figure 8 Variation of the number of operating hours with ambient temperature in August.

Figure 9 shows the variation of the water evaporating rate with relative humidity of the main air that entered the wet pad at different dry bulb temperatures for the system design conditions. The results show that decreasing the water evaporating rate caused an increase in the relative humidity of the air for various inlet dry bulb temperatures in a linear manner. The decrease of the water evaporating rate with increasing relative humidity happens when the difference between the vapor pressure and the saturation pressure of the air on the wet pad is low; this leads to a decrease in the water evaporating rate, which depends on the evaporation of the water in the cooling pads. This gives a clear indication that when the relative humidity increases, the water evaporating rate decreases. Thus, in the phase of cooling by direct evaporation in the wet pad, the evaporating rate of the water decreases due to the increasing relative humidity at different dry bulb temperatures.

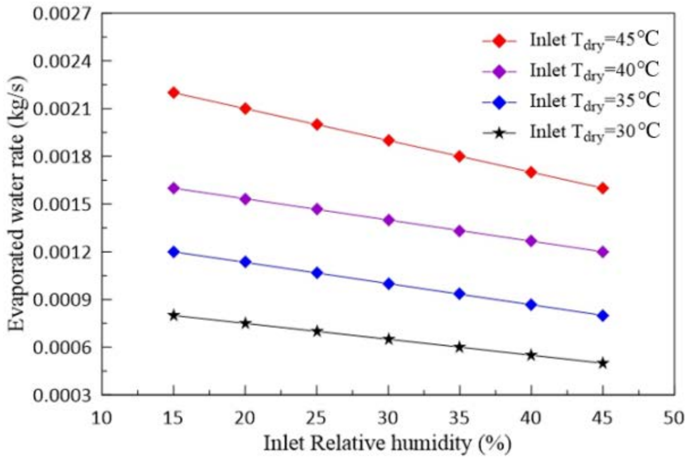


Figure 9 Effect of relative humidity on water evaporation rate.

The cooling capacity variation with the relative humidity of the main air that entered the wet pad at different dry bulb temperatures is shown in Figure 10. The results show that decreasing the cooling capacity caused an increase of the air relative humidity in a linear manner at various inlet dry bulb temperatures. The decrease in the cooling capacity and the increase in the relative humidity ratio depend on the evaporation of the cooled water in the wet pad. Cooling in the wet pad depends on the evaporation of the water from the total surface area of each wet pad, which represents the latent heat that is needed to evaporate the sprayed water, which is cooled by the wet pad and the air. This quantity of heat is the same as the quantity of latent heat that dissipates from the outer surface area of the wet pad.

On the other hand, if the relative humidity of the air on the wet pad is low, this will lead to an increase of water evaporation, which in turn leads to an increase in the amount of heat that is withdrawn from the wet pad, which depends on the evaporation of the cooling water from the external surface of the wet pad and the relative humidity of the inlet air. This leads to an increase in both water evaporation and latent heat quantity with a decrease in relative humidity. This gives a clear indication that when the relative air humidity increases, the amount of cooling capacity is decreased, thus in the phase of cooling by direct evaporation in the wet pad, the cooling capacity is decreased by the increased relative humidity at different dry bulb temperatures.

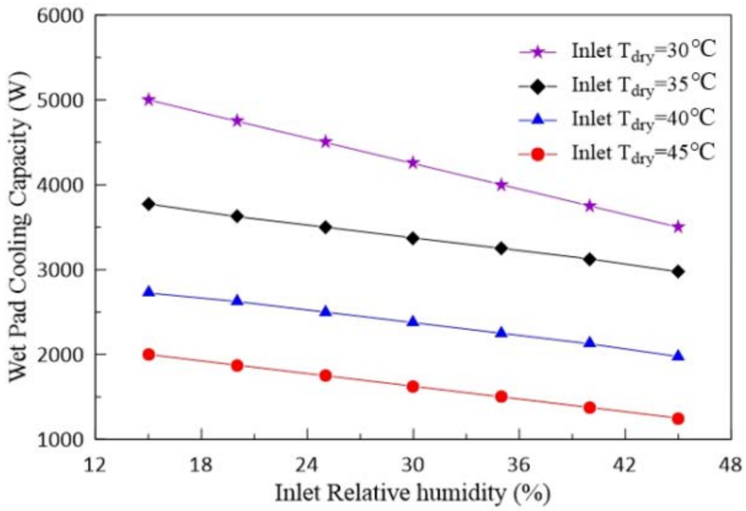


Figure 10 Effect of relative humidity on the cooling capacity of the cooling pad.

Figure 11 shows the variation of wet bulb effectiveness with relative humidity of the main air that enters the wet pad at different dry bulb temperatures for the system design conditions. The results show that an increase in wet bulb effectiveness caused an increase in air relative humidity in a linear manner at various dry bulb inlet temperatures. Increasing the air relative humidity caused an increase in wet bulb effectiveness. This happens due to the large difference between the wet bulb temperature and the air dew point on the wet pad, which leads to an increase in wet bulb effectiveness, which depends on the water evaporation in the wet pad. This gives a clear indication that when the relative humidity increases, the wet bulb effectiveness also increases, thus in the phase of cooling by direct evaporation, the wet bulb effectiveness in the wet pad increases due to the relative humidity increase at different dry bulb temperatures.

Figure 12 illustrates the variation of the coefficient of performance (COP) with the heat capacity of the vapor compression cycle at different ambient temperatures. It can be seen that the coefficient of performance (COP) increased with a decrease in the heat capacity of the vapor compression cycle. It also decreased, generally, with the increase of the ambient temperature. The effect of the cooling pads helps to decrease the outside temperature, hence it contributes to the increase of the coefficient of performance (COP). Its optimum enhancement occurred at a ratio of around 15%.

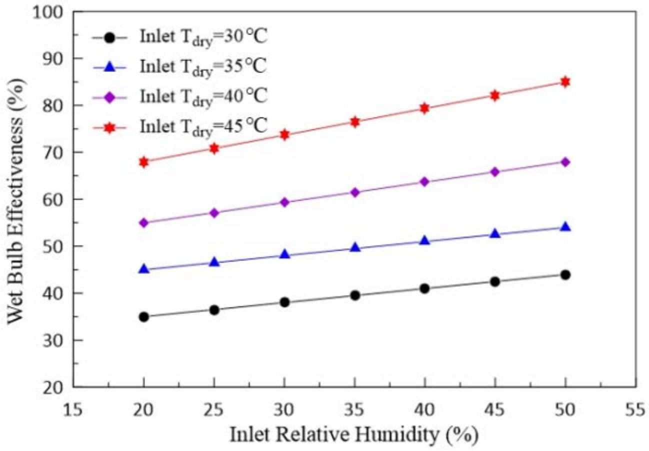


Figure 11 Effect of inlet relative humidity on wet bulb effectiveness.

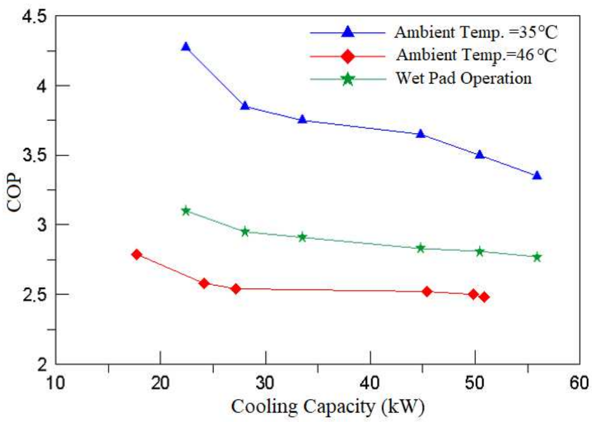


Figure 12 Effect of relative cooling capacity on coefficient of performance (COP).

5 Conclusions

In this study, a hybrid direct evaporative cooler and vapor compression cooling system was tested to design and evaluate the optimum performance under different operating conditions. A computational model was developed that was validated against real experimental data. The performance of the DEC unit under dry desert climate conditions was studied. The coefficient of performance of the vapor compression cooling cycle can be increased and the energy consumption can be reduced by using a wet pad as a pre-cooling unit in a dry desert climate.

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Nomenclature

A_p	Wet pad area per cubic meters m^2/m^3
A_{FP}	Face area of the wet pad m^2
A_{TP}	Total surface area of the wet pad m^2
c_{pm}	Heat capacity of the wet pad
h	Enthalpy kJ/kg °K
h_P	Wet pad height (m)
H_{fg}	Saturated enthalpy kJ/kg °K
$\dot{M}_{wep}{}^{\cdot}$	Water evaporated from the wet pad
$\dot{M}_{wep}$	Mass flow rate of the evaporated water kg/s
$\dot{m}_{ad}$	Mass flow rate of the dry air kg/s
$\dot{m}_f$	Mass flow rate of the working fluid kg/s
Q_{cp}	Cooling capacity of the wet pad kW
Q_L	Cooling capacity of the vapor compression cooling system kW
Q_H	Power rejected from the condenser of the vapor compression cooling system kW
v_{FP}	Velocity of the air that passes through the wet pad
V_P	Wet pad volume m^3
$\dot{V_p}$	Flow rate across the wet pad m^3/s
P_B	Barometric pressure kPa
P_{sop}	Outlet vapor pressure kPa
t_P	Wet pad thickness (m)
TD_{op}	Dry bulb temperature of the outlet air leaving the wet pad °C
TD_{ot}	Dry bulb temperature of the outside air around the wet pad °C
w_P	Wet pad width (m)
W	Moisture content
W_i	Moisture content of air inlet the pad
W_{op}	Moisture content of air outlet the pad
W_s	Saturated moisture content
W_{1-2}	Work of the compressor kW

Greek Symbols

$\exists_p$	Evaporation coefficient
$\emptyset_{op}$	Relative humidity of the air outlet from the wet pad
ρ_m	Density of the wet pad kg/m^3
η_{wbp}	Wet bulb effectiveness

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