Int. J. of Thermal & Environmental Engineering
Volume 6, No. 1 (2013) 15-20

Experimental Performance of a Direct Evaporative Cooler Operating in Kuala Lumpur

Abdulrahman Th. Mohammad[a,b], Sohif Bin Mat[a], M.Y. Sulaiman[a], K. Sopian[a], Abduljalil A. Al-abidi[a]

[a]*Solar Energy Research Institute, University Kebangsaan Malaysia, 43600 Bangi, Selangor, Malaysia*
[b]Dept. of Mechanical Engineering, Baqubah Technical Institute, Foundation of Technical Education, Baghdad, Iraq

Abstract

The present work presents an experimental investigation on the performance of a direct evaporative cooler in hot and humid regions. The experimental study is based on weather data from Kuala Lumpur, Malaysia. The direct evaporative cooler consists of a cellulose pad with a surface area per unit volume ratio of 100 m^2/m^3. The performance of the evaporative cooler is evaluated using the output temperature, saturation efficiency, and cooling capacity. The output temperature of the air varies between 27.5°C and 29.4°C, while the cooling capacity is between 1.384 kW and 5.358 kW.

Keywords: Evaporative cooler, Saturation efficiency, cooling capacity.

1. Introduction

Air conditioning identifies the conditioning of air for maintaining specific conditions of temperature, humidity, and dust level inside an enclosed space[1].Evaporative cooling is widely used in hot and dry regions. Two common types of evaporative cooling systems are the direct and indirect systems. In the direct evaporative cooler (DEC), the air comes into direct contact with water. The direct evaporative cooling system adds moisture to the cool air, while an indirect evaporative cooling system (IEC) provides only sensible cooling to the processed air without any addition of moisture. Therefore, the indirect evaporative system is more attractive than the direct evaporative system. However, the cooling effectiveness is generally low.

Many studies have dealt with the performance of direct evaporative cooling. In India, Kulkarni and Rajput [2] made a theoretical performance analysis of direct evaporative cooling. Different materials were considered in the analysis; rigid cellulose, high density polythene, aspen fiber and corrugated paper material. The results of the analysis showed that the aspen fiber material had the highest efficiency of 87.5%, while the rigid cellulose material had the lowest efficiency at 77.5%. The outlet temperature and cooling capacity varied between 28.8°C and 26.5°C and 13408 KJ/h and 56686 KJ/h for the two materials, respectively. Kachhwaha and Suhas [3] designed, fabricated, and predicted the performance of an evaporative medium. The pad thickness and height were achieved for maximum cooling. Chenguang and Agwu[4] evaluated the effect of speed and the dry-bulb temperature of frontal air, and the temperature of the incoming water on cooling performance of a direct evaporative cooling combined with a wetted medium. The results of the analysis showed that direct evaporative cooling efficiency decreased with frontal air velocity and incoming water temperature, and increased with frontal air dry-bulb temperature. A simplified mathematical model was used by Fouda and Melikyan [5] to discuss the heat and mass transfer between the air and water in a direct evaporative cooler. A comparison between the model results and the experimental results was presented. The results indicate that during a steady state condition, the cooling efficiency is decreased by increasing the inlet frontal air velocity, and increased by increasing the pad thickness. This is because the contact surface between water and air is increased. Moien et al.[6] studied a two-stage cooling system that consisted of a nocturnal radiative unit, a cooling coil, and an indirect evaporative cooler. The investigation was conducted in weather conditions for the city of Tehran. The results showed that the first stage of the system increased the effectiveness of the indirect evaporative cooler. Also, the regenerative model provided the best comfort conditions. Dai and Sumathy[7] developed a mathematical model to predict and discuss the interface temperature of falling film in a cross-flow direct evaporative cooler. Analysis results indicated that the system performance could be improved by optimizing the mass flow rates of the feed water and processed air, as well as the different dimensions of the pad. Wu et al.[8] proposed a

[*] Corresponding author. Tel.: +60173672679
Fax: + 60389214593; E-mail: abd20091976@gmail.com

DOI: 10.5383/ijtee.06.01.003

simplified cooling efficiency based on the energy balance analysis of air to analyze the heat and mass transfer between air and water film in the direct evaporative cooler. The analysis showed that the frontal air velocity and thickness of the pad module are two key factors influencing the cooling efficiency of a direct evaporative cooler. A model of the dew point evaporative cooling system was developed by Riangvilaikul and Kumar [9] to simulate the heat and mass transfer processes under various inlet air conditions and the influence of major operating parameters. The model was used to optimize the system parameters and to investigate the system effectiveness when operating under various inlet air conditions. This paper aims to verify the experimental results of a direct evaporative cooling system operating under hot and humid climate conditions.

2. Experimental setup

The direct evaporative cooling unit that is used for this study consists mainly of an exhaust fan at the end of the unit and a re-circulating pump to drip water on the upper side of the pad. As shown in Fig. 1, the air enters the pad in a horizontal configuration. The evaporative cooler is made of stainless steel sheet; its height, length, and width are 80 cm, 80 cm, and 80 cm, respectively, and it has a sump with dimensions, 66 cm long, 20 cm width, and 25 cm depth. A cellulose pad is used as the packing material with a length of 71 cm, width of 71 cm and thickness of 5 cm, as shown in Fig.1. The packing is made of cellulose sheets with a specific surface area of 100 m^2/m^3 and an equivalent diameter of 0.009 cm.

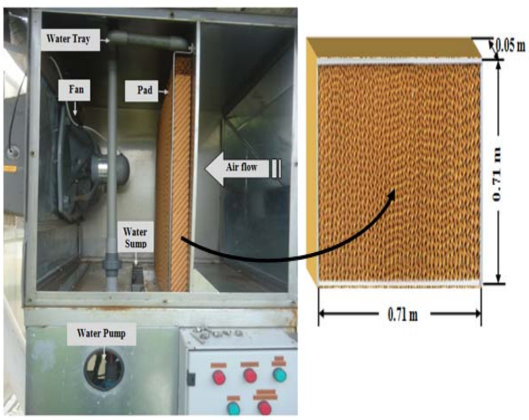

Fig.1. Photograph of a direct evaporative cooling and its pad

3. Measurements and instrumentation

Experimental tests are carried out to evaluate the performance of the direct evaporative cooling unit. To measure the air temperature and relative humidity at inlet and outlet points of the evaporative cooling unit, two thermocouples of type K with an accuracy of ±0.1°C and two humidity sensors (type TR-RH2W) with an accuracy of ± 3% RH are installed, respectively. One AM-402 anemometer is installed at the inlet point of the evaporative cooling unit to measure the air velocity. The mass flow rate of air entering the evaporative cooling unit is determined by using the air velocity and the cross-section area of the inlet duct. One Li-cor-2005Z Pyrometer is used to record the solar radiation in the test location. All these sensors are coupled with a PC system to store the data on a computer.

An ADAM-4018 data logger was used to collect sensor data. This data-acquisition system has a capability of 2 single-ended (6 differential) voltage channels. Various single channel input are available in ADAM data acquisition; thermocouple, mV, V, and mA. The data acquisition system samples data every ten minutes from 2 thermocouple sensors, 2 humidity sensors, 1 anemometer and 1 pyrometer.

4. Performance parameters

A direct evaporative cooler (DEC) is a simple air-conditioning system widely used in dry and hot regions. The air and water are in direct contact; the hot, dry air passes over a wet pad's surfaces; the air will lose its sensible heat, thereby reducing its temperature. The performance parameters of DEC are calculated based on the following relation:

Saturation efficiency is the rate between the real decreasing of the dry bulb temperature and the maximum theoretical decreases (dry bulb temperature would be equal to the wet bulb temperature of the inlet air) as seen by Camargo et al. [10] and [11].

$$\varepsilon_{DEC} = 1 - \exp\left(h_c A_w \big/ m_a Cp_a \right) \tag{1}$$

Fig.2 shows the process of evaporating cooling on a psychometric chart; this process is seen to be a constant wet bulb temperature process.

The dry bulb temperature of the outlet air can be calculated as: Watt and Brown[12].

$$T_2 = T_1 - \varepsilon_{DEC}\left(T_1 - T_{wb1}\right) \tag{2}$$

Cooling capacity is given by:

$$Q_c = m_a Cp_a \left(T_1 - T2\right) \tag{3}$$

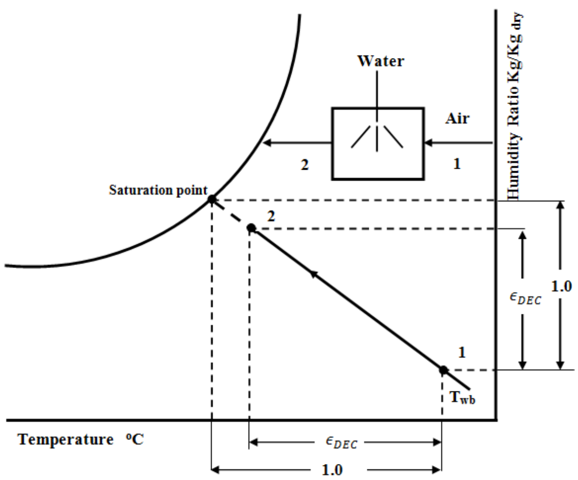

Fig. 2. Process of evaporating cooling on a psychometric chart

5. Results and Discussion

The values of the experimental test; dry-bulb temperature of the inlet and outlet air, solar radiation, saturation efficiency, and cooling capacity are shown in Table 1.

During experimentation, the inlet air conditions were as follows: the dry bulb temperature 29.6°C to 36°C, the air relative humidity 42.5% to 81.1%, and the solar radiation 307.4 to 903.7 W/m^2. The experimental weather data of Kuala Lumpur is presented in Fig. 3. The average air velocity during the experiments was 5.5 m/sec and the water mass flow rate was 10 l/min. Fig. 4 represents the variation of the dry bulb temperature between inlet air temperature and outlet air temperature, and it is noticed that it was possible to obtain the maximum temperature difference up to 7.6°C at a relative humidity of 42.5 %, and the minimum temperature difference up to 2.1°C at a relative humidity of 81.1%.

The variation of cooling capacity with the time of test is shown in Fig. 6. It is seen that the cooling capacity varies between 1.3 KW and 5.3 KW. The maximum cooling capacity is recorded at the time 1:50 PM, while the minimum cooling capacity is recorded at the time 8:30 AM. From the results, the cooling capacity is based on the drop in the dry bulb temperature of air. Therefore it increases with the increase in the difference between the dry bulb temperatures at a constant air mass flow rate.

The outlet dry-bulb temperature has a direct effect on the saturation efficiency of the evaporative cooler. The variation of saturation efficiency with time is shown in Fig. 5, the efficiency ranges from 77.3% to 63.5%. The maximum saturation efficiency was recorded at the time 11:40 AM, with a dry-bulb temperature and relative humidity of 34.8°C and 52.7%, respectively.

Table 1. Experimental data and the performance parameters

TIME (MINUTE)	T_1 (^OC)	T_{WB} (^OC)	T_2 (^OC)	SOLAR RADIATION W/M^2	EFFICIENCY (%)	Q_C (KW)
8:30	29.8	27.07	27.7	307.4	76.923	1.3847
8:40	29.6	26.63	27.5	347.5	70.707	1.4631
8:50	30.5	27.51	28.2	385.5	76.923	1.5452
9:00	30.3	27.11	28.2	393.5	65.830	1.4108
9:10	30.8	27.27	28.4	428.6	67.988	1.5527
9:20	31.2	27.16	28.4	435.4	69.306	1.9856
9:30	31.5	27.25	28.7	478	65.882	1.9508
9:40	32.4	27.74	29.1	539	70.815	2.2581
9:50	33.1	28.04	29.4	425	73.122	2.6239
10:00	32.4	26.45	28.4	454.8	67.226	2.8864
10:10	33.1	26.83	28.7	583	70.175	3.2845
10:20	33.4	27.02	28.9	531.3	70.532	3.0233
10:30	33.6	27.42	29.1	489.3	72.815	3.1912
10:40	33.6	27.55	29.4	641.9	69.421	2.9785
10:50	34.3	26.67	29.1	784.6	68.152	3.7523
11:00	34.8	26.96	29.1	804.7	72.704	3.4040
11:10	34.5	26.74	28.9	691.4	72.164	4.3893
11:20	35.0	27.11	29.1	706	74.778	4.3309
11:30	35.7	27.04	29.1	630.3	76.212	4.5984
11:40	34.8	26.53	28.4	855.2	77.388	4.5386
11:50	35.5	26.48	29.4	867.1	67.627	4.4777
12:00	35.0	25.98	28.7	787.5	69.844	4.4677
12:10	35.1	26.11	28.2	802.7	76.751	4.8074
12:20	35.0	26.15	28.4	721.8	74.576	4.6805
12:30	34.8	24.89	28.5	823.7	63.572	4.6245
12:40	35.2	25.82	28.6	884.9	70.362	4.6805
12:50	35.3	25.82	28.3	871	73.839	4.8771
1:00	35.5	25.81	28.4	903.7	73.271	5.0351
1:10	35.5	25.61	28.4	852.5	71.789	5.2117
1:20	35.7	25.79	28.2	666.4	75.681	5.3187
1:30	36.0	25.28	28.4	722.1	70.895	5.1060
1:40	36.0	25.33	28.7	510.5	68.416	5.1769
1:50	35.7	26.24	28.4	335.3	77.167	5.3585
2:00	35.2	25.59	28.2	443.8	72.840	5.1383
2:10	35.7	26.59	28.9	500.4	74.643	4.9915
2:20	35.7	26.02	28.9	411	70.247	4.9915
2:30	36.0	27.07	29.1	420.1	77.267	4.3781
2:40	35.2	26.02	28.4	359.3	74.074	4.7377
2:50	35.5	26.27	28.7	336.5	73.672	4.7377
3:00	35.3	26.11	28.4	342	75.081	4.6357

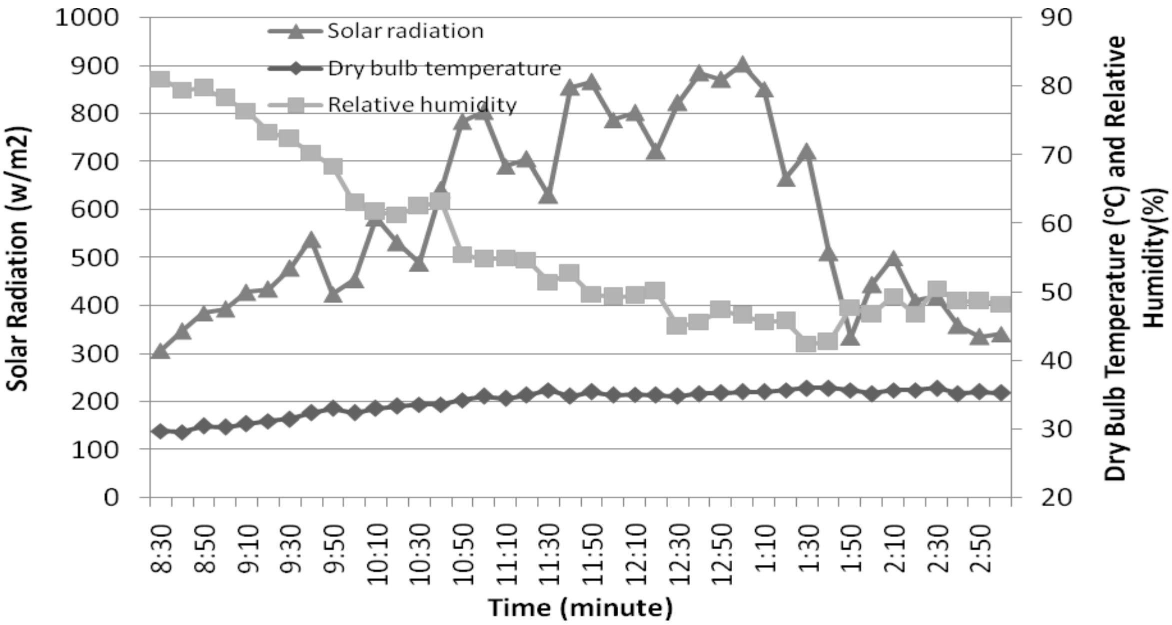

Fig. 3. Variation solar radiation, dry-bulb temperature, and relative humidity with time

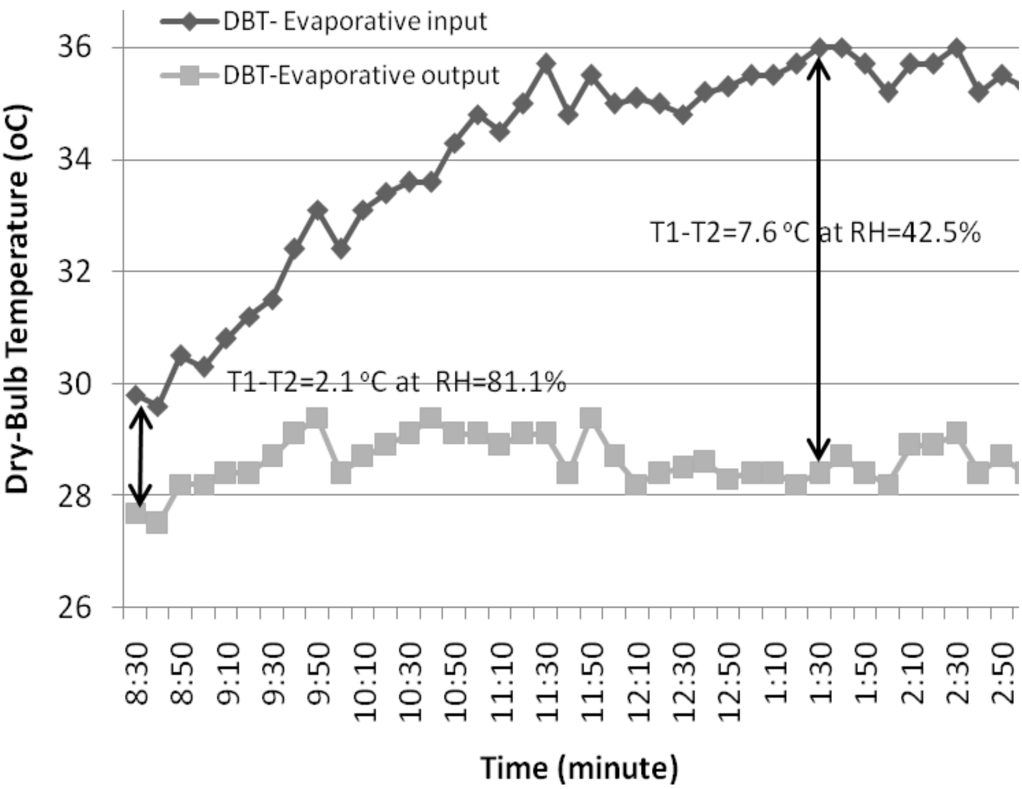

Fig. 4. Variation inlet and outlet dry-bulb temperatures with time

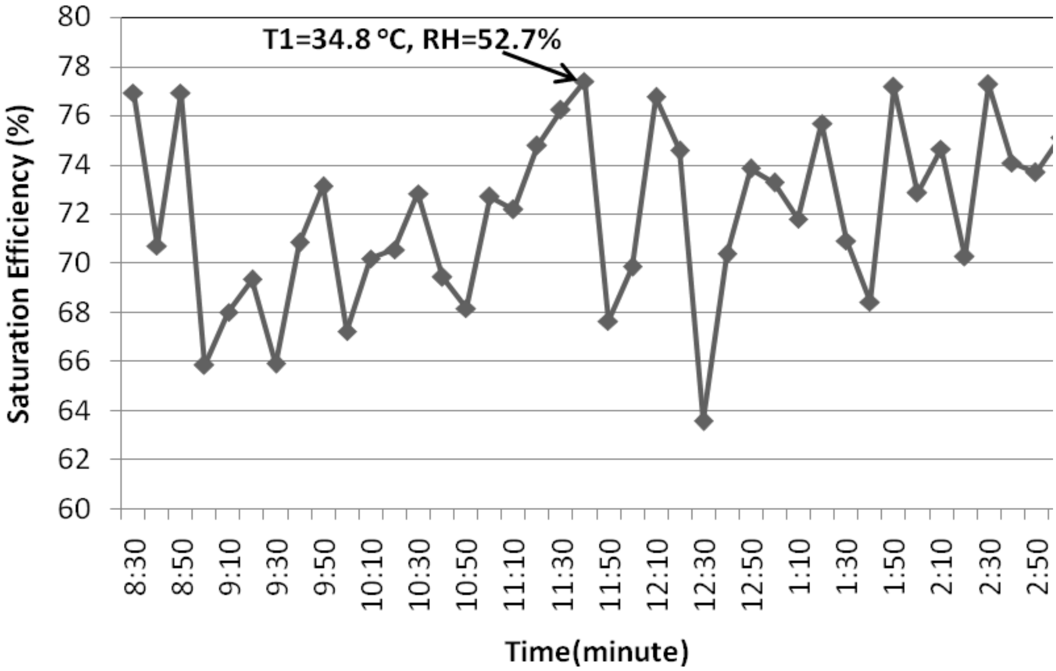

Fig. 5. Variation of saturation efficiency with time

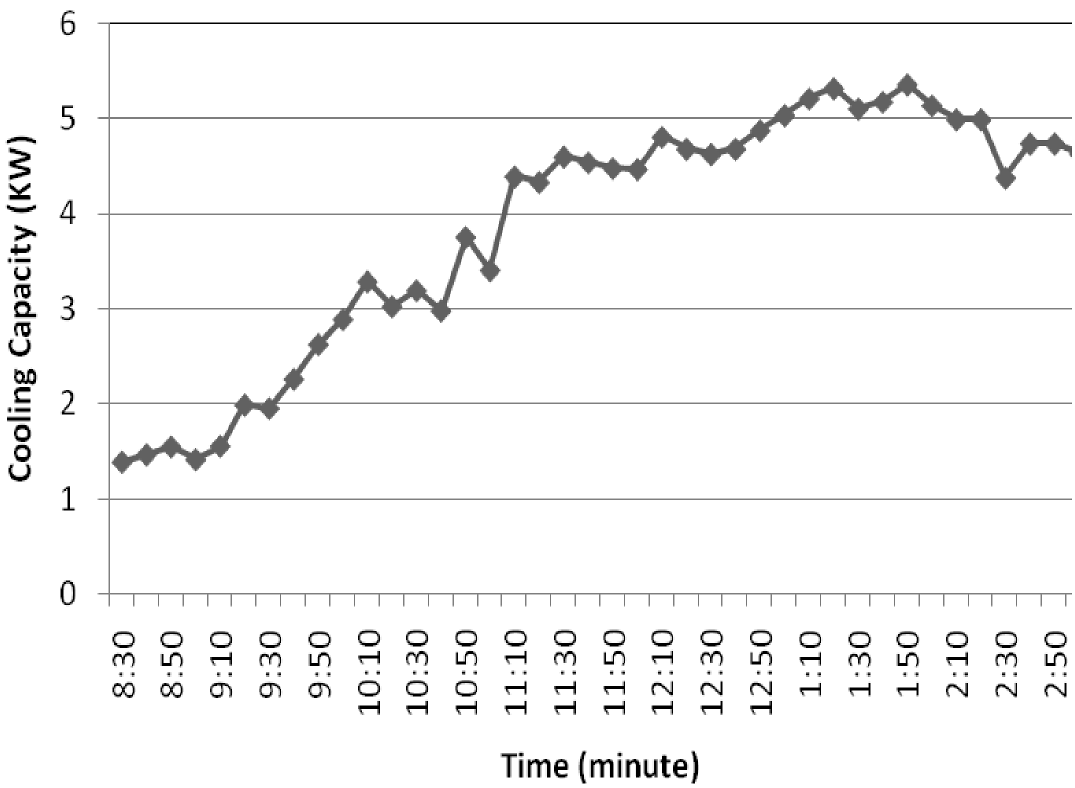

Fig. 6. Variation of cooling capacity with time

6. Conclusions

An experimental study was carried out to evaluate the performance of a direct evaporative cooler in humid regions. From the analysis of the experimental data, the following conclusions can be summarized:

- Experiments indicate that a dry bulb temperature decrease up to 7.6°C is obtainable by using a direct evaporative cooling unit in hot and humid regions.
- Saturation efficiency varies from 77.3% to 63.5 % and cooling capacity ranges from 1.384KW to 5.358 KW.
- A direct evaporative cooler may be made to work in humid places like Malaysia by drying the air before the evaporative process using the desiccant dehumidification concept.

Nomenclature

A_w	Wetted area, m^2
Cp_a	Specific heat of air, J/kg K
h_c	Heat transfer coefficient, W/m^2K
m_a	Air mass flow rate, kg/sec
Q_c	Cooling capacity, kW
T_1	Evaporative inlet dry bulb temperature, ^oC
T_2	Evaporative outlet dry bulb temperature, ^oC
T_{w1}	Evaporative inlet wet bulb temperature, ^oC

Greek symbols

ε_{DEC}	Evaporative saturation efficiency (%)

Subscripts

DEC	Direct evaporative cooler
DBT	Dry bulb temperature
IEC	Indirect evaporative cooler
RH	Relative humidity

Acknowledgement

The authors are grateful to the staff of Solar Energy Research Institute/University Kebangsaan Malaysia for their help.

References

[1] Essam E. Khalil. Energy-Efficiency in Air Conditioned Buildings: The Green Buildings Dream. Int. J. of Thermal & Environmental Engineering. 2011; 2: 9-18

[2] R. K. Kulkarni and S. P.S. Rajput. Comparative performance of evaporative cooling pads of alternative materials. IJAEST.2011; 10: 239-244.

[3] S. S. Kachhwaha and Suhas Prabhakar. Heat and mass transfer in direct evaporative cooler. J. Sci. Ind. Res.2010; 69: 705-710.

[4] Chenguang Sheng and Agwu Nnanna. Empirical correlation of cooling efficiency and transport phenomena of direct evaporative cooler. Appl. Therm. Eng. 2012; 40: 48–55.

[5] A.Fouda and Z. Melikyan. A simplified model for analysis of heat and mass transfer in a direct evaporative cooler. Appl. Therm. Eng. 2011; 31: 932-936.

[6] Moien Farmahini, Ghassem Heidarinejad and Shahram Delfani. A two-stage system of nocturnal radiative and indirect evaporative cooling for conditions in Tehran. Energy Build. 2010; 42: 2131–2138.

[7] Y.J. Dai and K. Sumathy. Theoretical study on a cross-flow direct evaporative cooler using honeycomb paper as packing material. Appl. Therm. Eng. 2002; 22: 1417–1430.

[8] J.M. Wu, X. Huang and H. Zhang. Theoretical analysis on heat and mass transfer in a direct evaporative cooler. Appl. Therm. Eng.2009; 29: 980–984.

[9] B. Riangvilaikul and S. Kumar. Numerical study of a novel dew point evaporative cooling system. Energy Build. 2010; 42: 2241–2250.

[10] J. R. Camargo, C. D. Ebinuma and J. L. Siveria. Experimental performance of a direct evaporative cooler operating during summer in Brazilian city. Int. J. Refrig. 2005; 28: 1124-1132.

[11] J. R. Camargo, E. Godoy Jr. and C. D. Ebinuma, 2005. An evaporative and desiccant cooling system for air conditioning in humid climates. J. Braz. Soc. Mech. Sci. & Eng. 2005; XXVII: 243-247.

[12] Watt R.J. and Brown W. K. Evaporative air conditioning hand book, third ed., the Fairmane Press Inc.: Liburn GA, 1994; pp.185-189.