INTERNATIONAL JOURNAL OF RESEARCH IN AERONAUTICAL AND MECHANICAL ENGINEERING

ISSN (ONLINE): 2321-3051

Vol.3 Issue 12,

December 2015

Pgs: 131-141

Performance Analysis of Direct Evaporative Cooling in Indian summer

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Abstract:

In this paper analysis of DEC on the basis of heat and mass transfer is presented. Pads of different thickness are used for finding the variations in some important parameters like wet bulb saturation effectiveness, COP, Nu, NTU, HTC, CC and water consumption with varying face velocities. It is observed that with the increase in the thickness of cooling pads, wet-bulb saturation effectiveness increases and with the increase in the face velocity wet-bulb saturation effectiveness decreases. Higher temperature drops are observed with high WBD. It is also observed that CC of the system is higher with high WBD.

Key words: Direct Evaporative Cooling, Saturation Efficiency, cooling capacity, Pad Materials, Water Use.

1. INTRODUCTION

It is believed that evaporative cooling was first used in ancient Egypt, around 1000 A.D. During that period evaporative cooling was used for preservation of food items against hot weather conditions. For cooling the inside space, walls were integrated with water chutes. These concepts of preservation of food items and space cooling were spread in the other hot and arid regions of the world.

In case of evaporative cooling part of sensible heat of hot air is converted into the latent heat of water which is responsible for the evaporation of water. For the evaporation of water to take place spray of water, porous fibres pads, cellulose papers etc. are used.

On the basis of contact of water and air, evaporative cooling system can be classified in two categories, first one is direct evaporative cooling (DEC) system and the second one is indirect evaporative cooling system (IEC). In case of DEC direct contact of air and the water takes place. And in case of IEC there is always a film or sheet of matter which allows only the sensible heat of hot air to transfer from one side to another side. These two modes of evaporative cooling systems can be combined to produce greater level of cooling effect. These modes can also be used in the form of hybrid cooling systems.

Evaporative cooling technology has many advantages like supply of fresh air, use of water in place of CFCs as working fluid, ease of manufacturing, lesser power requirements, easy maintenance and in maintaining suitable level of humidity in drier regions. Evaporative cooling has many environmental benefits which include reduction of CO_2 and CFC/HCFC emissions.

2. LITERATURE REVIEW

Many researchers analyzed the evaporative cooling systems in depth and published their works. Watt [1] pioneered the publication of direct and indirect evaporative cooling. A general mathematical model is developed by Halasz [2] to explained the evaporative cooling process used in many devices. Camargo et al. [3] presented the principles of DEC and IEC systems. They also presented the mathematical model of evaporative cooling systems. Koca et al. [4] have presented procedure for testing DEC media. They concluded that performance of cooling media depends upon pad thickness, pad angle, pressure drop and air velocity. They also concluded that the performance of the cooling pad can be analyzed in terms of wet bulb cooling effectiveness and pressure drop across the pads. With the help of integration method Dai et al. [5] solved the governing equations of DEC. They concluded that for the improvement of performance, length of air channel of honeycomb paper, feed water and air mass flow rates should be optimized. Kruger [6] suggested that in humid places like Maracaibo DEC system is not effective. A wind tunnel method for finding the performance of cooling pads is developed by Liao et al. [7]. Liao et al. [8] analyzed the performance of pressure drop across the cooling pads and wet bulb saturation efficiency DEC on the basis of face velocity and the thickness of pads. Performance analysis of evaporative cooling pads made up of jute, luffa and palm is presented by Al-sulaiman [9]. Gunhan et al. [10] presented the suitability of volcanic tuff, pumice stones and greenhouse shading net as wetted pad media for DEC. Performance analysis of DEC pads made up of wood wool, stainless steel wire mesh, khus and coconut coir is presented by Khond [11]. Dzivama et al. [12] presented the performance of DEC pads of charcoal, jute, ground sponge and stem sponge. For the preservation of fruit and vegetable articles, Anyanwu [13] presented the performance of a porous DEC. Kulkarni et al [14] reported the performance of jute fiber ropes as DEC media as jute ropes have large wetted area. Kulkarni et al [15] presented the theoretical performance of DEC pads of corrugated paper, rigid cellulose, high density polythene packing and aspen fiber. Heidarinejad [16] presented a performance of a DEC combined with a ground circuit. He showed that this system provides high wet bulb cooling effectiveness. A.Joudi [17] reported that IEC system can provide thermal comfort conditions for most time of operation period with high effectiveness. To improve the performance of the MVC system, IEC is used to pre-cool the supply air by Delfani [18]. Brown [19] invented IDEC system, on the basis of his model many researchers studied the coupled IDEC systems. Heidarinejad [20] experimentally found that IDEC systems can provide thermal comfort conditions in those places where DEC could not provide thermal comfort conditions due to high wet bulb temperature. Kulkarni et al [21] analyzed theoretically the performance of IDEC. In DEC stage aspen fiber and rigid cellulose pads in rectangular, semi-hexagonal and semi-cylindrical shapes are used. Kulkarni et al [22] experimentally analyzed the performance of cooling pads made up of aspen fiber, rigid cellulose and wood wool. Pads of rectangular, semi-cylindrical and semi-hexagonal shapes are used.

DIRECT EVAPORATIVE COOLIG (DEC)

As hot and humid air at temperature T passes over the wet surface at temperature T_s heat transfer takes place due to temperature difference. Mass transfer also takes place due to difference in the absolute humidity W_s of the air close to the wet surface and absolute humidity of flowing air W. Sensible heat transfer through a small elementary area dA is given by

$$dQ_s = h_c dA(T_s-T) \qquad (1)$$

where h_c is the convective heat transfer coefficient,

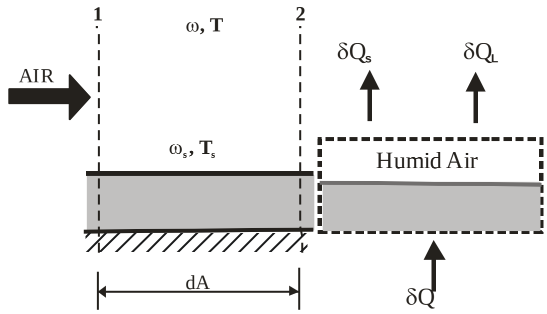

Fig. 1. Schematic Direct Evaporative Cooler

As we know that the value of heat transfer coefficient h_c is determined from the value of *Nu*. We also know that the rate of vapour transfer between the air close to the wet surface and bulk air flow is given by

$$dm_v = h_m \rho dA(w_s - w) \quad (2)$$

where h_m is the convective mass transfer coefficient.

On the basis of conservation of energy latent heat transfer at the air-liquid interface is given by

$$\delta Q_L = \delta Q - \delta Q_s = h_{Lvs} dm_v \quad (3)$$

where h_{Lvs} is the sp. enthalpy of vaporization. On combining equations 1, 2 and 3 we get the heat flow

$$\delta Q = \left[h_c(T_s - T) + \rho_a h_{Lvs} h_m (w_s - w)\right] dA \quad (4)$$

The eq. 4 represents the sum of heat transferred due to temperature and absolute humidity difference.

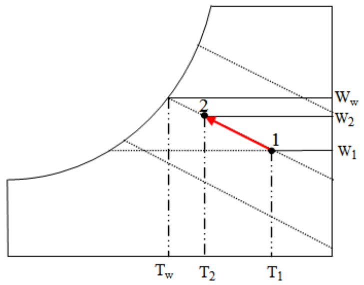

Fig. 2. DEC process on the psychometric chart

As the total sp. enthalpy of the mixture is due to the effect of the individual enthalpies [21]. So,

$$h_s - h = (h_{sa} - h_a) + (w_s h_{vs} - w h_v) \quad (5)$$

Considering vapour and air as perfect gases we get

$$h_s - h = c_{pu}(T_s - T) + h_{vs}(w_s - w) \quad (6)$$

So $$T_s - T = \frac{(h_s - h) - h_{vs}(w_s - w)}{c_{pu}} \quad (7)$$

Considering density of moist air equal to the density of dry air and on combining the equations 4 and 7 and dropping small terms we get

$$\delta Q = \frac{h_c dA}{c_{pu}}\left[\left(h_s - h\right) + \frac{(w_s - w)}{R_{Le}}(h_{Lvs} - R_{Le} h_{vs})\right] \qquad (8)$$

Total heat transfer is

$$\delta Q = \frac{h_c dA}{c_{pu}}\left(h_s - h\right) \qquad (10)$$

Flow of sensible heat

$$\delta Q_s = m_a c_{pu} dT \qquad (11)$$

Rearranging equations 1 and 11 we get

$$h_c dA(T_s - T) = m_a c_{pu} dT \qquad (12)$$

On integrating the equation 12 we get

$$\frac{h_c}{m_a c_{pu}} \int_0^A dA = \int_{T_1}^{T_2} \frac{dT}{(T_s - T)} \qquad (13)$$

$$1 - \frac{T_1 - T_2}{T_1 - T_s} = \exp\left(-\frac{h_c A}{m_a c_{pu}}\right) \qquad (14)$$

Wet bulb saturation effectiveness of DEC is given by

$$\varepsilon = \frac{T_1 - T_2}{T_1 - T_s} \qquad (15)$$

$$\varepsilon = 1 - \exp\left(-\frac{h_c A}{m_a c_{pu}}\right) \qquad (16)$$

Correlation of *Re, Pr, l and* l_s provides the Nusselts Number [24]

$$Nu = 0.01\left(\frac{l_c}{l}\right)^{0.12} \mathrm{Re}^{0.8}\, \mathrm{Pr}^{0.33} \qquad (17)$$

where l is the thickness *of* pad and l_c is the characteristic length.

Characteristic length is give by

$$l_c = \frac{V_p}{A} \qquad (18)$$

where A is the total wetted area and V_p is the vol. occupied by the wetted media.
Cooling capacity of DEC is given [1].

$$Q_{cc} = m_a c_{pu}(T_1 - T_2) \qquad (19)$$

COP for DEC is the ratio of the rate of heat transferred to power consumed

$$COP = \frac{\dot{Q}_{Cooling}}{Power\ Used} \qquad (20)$$

3. Variation in Saturation Efficiency

Wet-bulb saturation efficiency of DEC decreases with increase in the air mass flow rate.

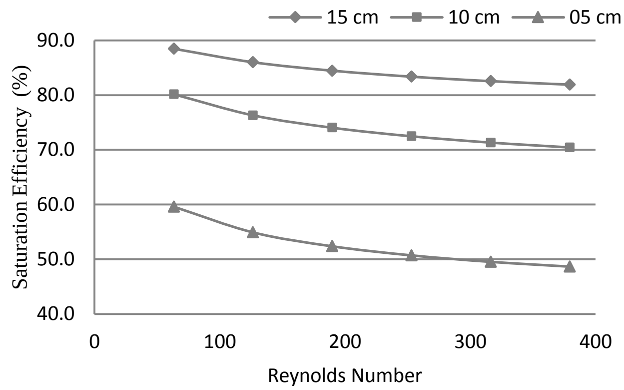

Fig. 3. Variations of saturation efficiency of pads of different thickness with Reynolds number.

This is expected because with increase in air mass flow rate, air has lesser contact time with water layer causing less evaporation of water. Saturation efficiency of DEC increases with increase in the thickness of cooling pads. This is also expected because with increase in the thickness of the cooling air gets greater contact time with water layer causing higher evaporation of water. For 5 cm thick pads saturation efficiency varied from 48.7 to 59.6%, for 10cm thick pads saturation efficiency varied from 70.4 to 80.1% and for 15cm thick pads saturation efficiency varied from 81.9 to 88.5%. The overall variation in the saturation efficiency is from 48.7 to 88.5%.

4. Outlet Temperature and Temperature Drop of air

The saturation efficiency has direct impact on outlet temperature of air. With the increase in air velocity and decrease in the thickness of pads, temperature drop decreases and the corresponding outlet temperature of air increases from 23.6 ^{0}C to 27.8 ^{0}C when inlet temperature is 33.4 ^{0}C and similarly outlet temperature of air increases from 23.9^{0}C to 31.2^{0}C when inlet temperature is 40.2^{0}C. Temperature drop ranges from 5.5 to 10.0^{0}C when inlet temperature is 33.4^{0}C and from 8.9 to 16.2^{0}C when inlet temperature is 40.2^{0}C.

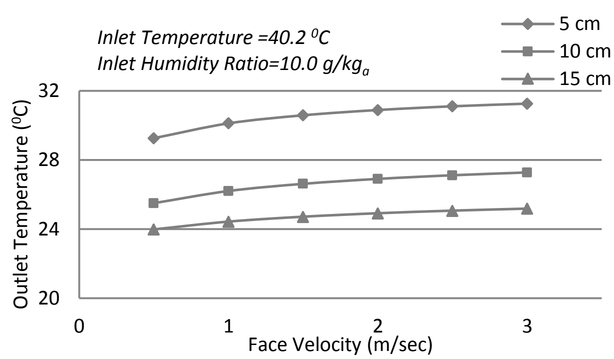

Fig. 4. Variations of outlet temperature of air with face velocity.

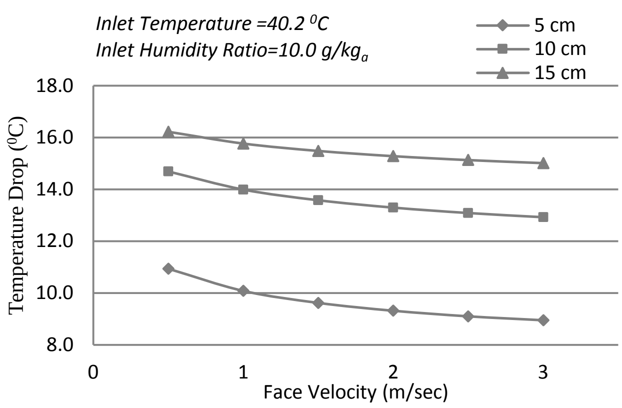

Fig. 5. Variations of temperature drop of air with face velocity.

5. Variation in the Nusselt Number (*Nu*)

In the present study it is found that the Nusselt Number (*Nu*) varied with the thickness of and Reynolds number. For 5cm thick pads *Nu* varied from 1.67 to 7.01, for 10 cm thick pads *Nu* varied from 1.54 to 6.45 and for 15 cm thick pads *Nu* varied from 1.47 to 6.14. The overall variation in the *Nu* is from 1.47 to 7.01.

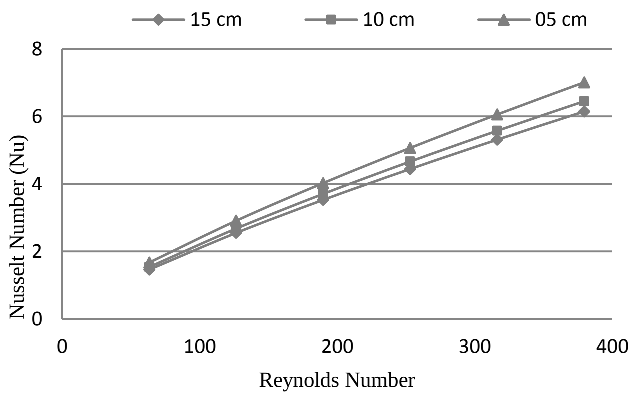

Fig. 6. Variations of Nusselt Number (Nu) with Reynolds Number (Re).

6. Variation in the Number of Transfer Units (*NTU*)

In the present study it is found that the Number of Transfer Units (*NTU*) varied with the thickness of pads and Reynolds number. For 5 cm thick pads *NTU* varied from 0.71to 1.01, for 10 cm thick pads *NTU* varied from 1.30 to 1.86 and for 15 cm thick pads *NTU* varied from 1.86 to 2.66. The overall variation in the *NTU* is from 0.71 to 2.66.

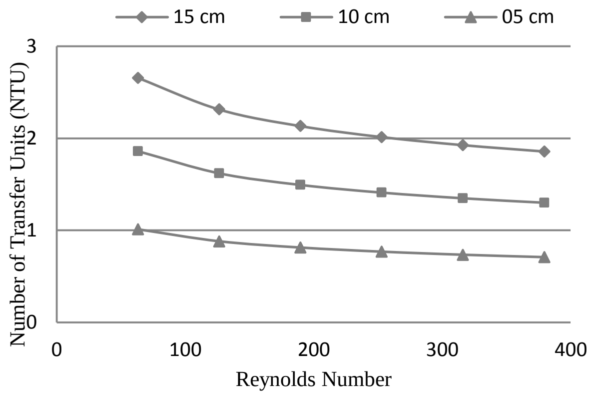

Fig. 7. Variations of Number of Transfer Units (NTU) with Reynolds Number.

7. Variation in Convective Heat Transfer Coefficient (hc)

In the present study it is found that the heat transfer coefficient varied with the thickness of pads and Reynolds number. For 5 cm thick pads heat transfer coefficient varied from 22.14 to 92.84 KJ/m^2-K, for 10 cm thick pads heat transfer coefficient varied from 20.37 to 85.43 KJ/m^2-K and for 15 cm thick pads heat transfer coefficient varied from 19.41 to 81.37 KJ/m^2-K. The overall variation in the heat transfer coefficient is from 19.41 to 92.84 KJ/m^2-K.

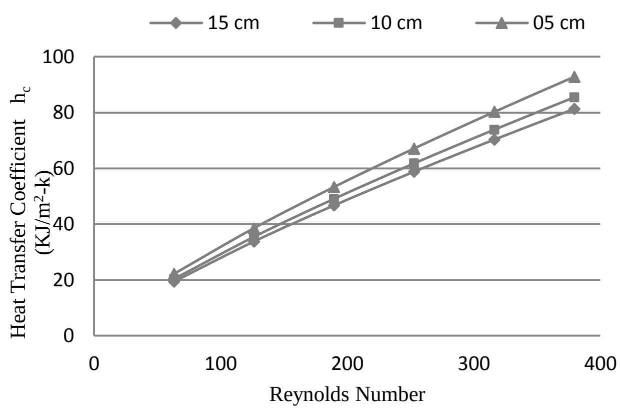

Fig. 8. Variations of Heat Transfer Coefficient (h_c) with Reynolds Number

8. Variation in Cooling Capacity (CC)

In the present study it is found that the cooling capacity varied with the thickness of pads, Reynolds number and the inlet dry-bulb temperature of air. The inlet temperature range is 33.4 to 40.2^{0}C.

In case of 33.4 ^{0}C inlet dry-bulb temperature the overall cooling capacity varied from 7523 to 24590 *KJ/hr*. For 5 cm thick pads cooling capacity varied from 5146 to 24590 *KJ/hr*, for 10 cm thick pads cooling capacity varied from 6831 to 35313 *KJ/hr* and for 15 cm thick pads cooling capacity varied from 7523 to 40966 *KJ/hr*.

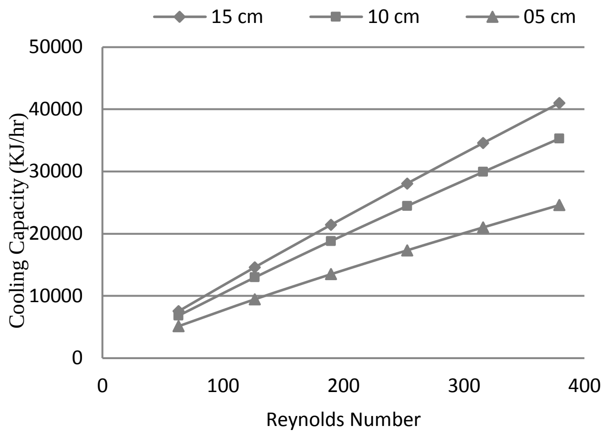

Fig. 9. Variations of Cooling Capacity of pads of different thickness with Reynolds number at inlet temperature range is 33.4 .

In case of 40.2 ^{0}C inlet dry-bulb temperature overall cooling capacity varied from 12163 to 39753 *KJ/hr*. For 5 cm thick pads cooling capacity varied from 8319 to 39753 *KJ/hr*, for 10 cm thick pads cooling capacity varied from 11044 to 57089 *KJ/hr* and for 15 cm thick pads cooling capacity varied from 12163 to 66228 *KJ/hr*.

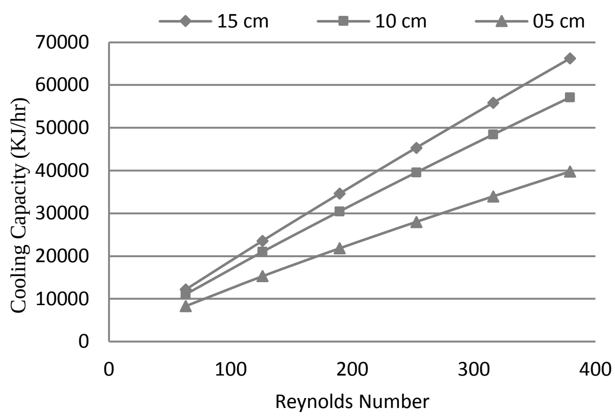

Fig. 10. Variations of Cooling Capacity of pads of different thickness with Reynolds number at inlet temperature 40.2^{0}C..

9. Variation in water consumption

In the present study it is found that the water consumption rate varied with the thickness of pads and face velocity. For 5 cm thick pads water consumption rate varied from 0.77 to 3.71 g/s, for 10 cm thick pads water consumption rate varied from 1.04 to 5.49 g/s and for 15 cm thick pads water consumption rate varied from 1.15 to 6.39 g/s. The overall variation in the water consumption rate is from 0.77 to 6.39 g/s.

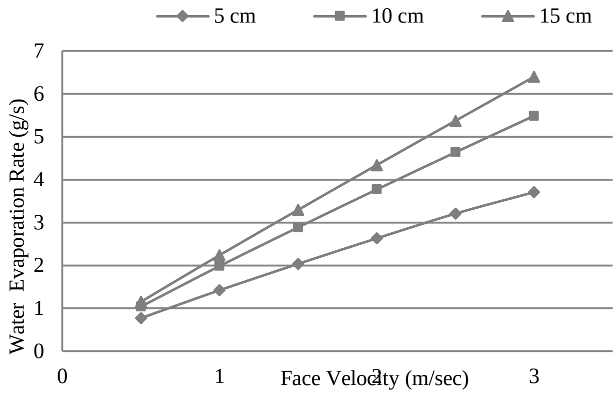

Fig. 11. Variations of saturation effectiveness of pads of different thickness with Reynolds number.

10. Coefficient of Performance (COP)

In the present study it is found that the *COP* varied with the thickness of and face velocity. For 5 cm thick pads *COP* varied from 10.2 to 32.5, for 10 cm thick pads COP varied from 13.6 to 46.7 and for 15 cm thick pads *COP* varied from 14.9 to 52.9. The overall variation in the water *COP* is from 10.2 to 52.9

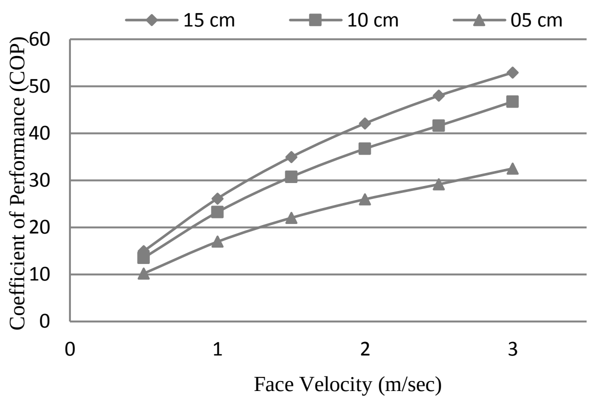

Fig. 12. Variations of Coefficient of Performance (COP) with face velocity.

11. Conclusion

After decades of research and development in the field of evaporative cooling optimization of system and process design could not be achieved. This is due to high dependency of the evaporative cooling process on the ambient parameters. Evaporative cooling technology is environmental friendly. This has the potential to minimize the use of CFCs/ HCFCs presently being used in the MVC systems. In this analysis it is observed that with the increase in the thickness of cooling pads wet-bulb saturation effectiveness increases and with the increase in the face velocity wet-bulb saturation effectiveness decreases. Higher temperature drop is observed when WBD is high. It is also observed that cooling capacity of the system is higher when WBD is high, face velocity high and pads are thicker.

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