EXPERIMENTAL INVESTIGATION OF HEAT AND MASS TRANSFER IN AN EVAPORATIVE COOLER MADE OF POROUS CERAMIC

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Evaporative cooling systems have intrinsic important qualities: improve the air quality, have low energy consumption, low cost of maintenance and production, and it is inert to the atmosphere. Direct evaporative cooling can be applied in regions where wet bulb temperatures do not exceed 25°C. In these study prototypes were developed with porous ceramic material acting as medium to establish mass and heat transport. Dry bulb temperatures, relative humidity and velocity air stream had been measured to estimate evaporative cooling efficiency, air mass flow rate, evaporated water flow rate, sensible heat rate, heat and mass transfer coefficients, on direct evaporative cooling process. Maximum evaporative cooling efficiency of 90,9 % and maximum power cooling of 780 W/m^2 has been achieved. Dry-bulb temperature drops off 6,7 °C for a couple 33,5 °C, 54 %, respectively inlet temperature and relative humidity. Results indicated 0,5 kg/h of average water consume. Values of heat and mass transfer coefficients are in good agreement with results of another searchers presented in literature.

Keywords: *evaporative cooling, heat and mass transfer, porous ceramic*

1. INTRODUCTION

Evaporative cooling systems have intrinsic important qualities: improve the air quality, have low energy consume, low cost of maintenance, production and do not attack the atmosphere. It is used about 90% of the residences in the West Texas Region and 4,5 million residences throughout the United States (Paschold, 2003). Evaporative cooling has been useful in hybrid systems to combat sensible heat rate saving energy consumption (Davis, 1989). In Ahmedabad-India have been achieved 36% savings in plant expenditure and 64% less energy consumption compared to conventional air conditioning (Ibrahim, 2003). Already in Brazil, due to climatic condition, evaporative systems show potential for application over northeast, southeast, middle east and south regions. To the north and coast regions hybrid systems could be reducing substantially the electric energy consumption (Mathaudhu, 2000). Givoni (1992) references wet bulb temperature of 24°C as limit to apply efficiently Direct Evaporative Cooling, Liao (2002) 25°C and Lomas (2004) 22°C.

Basically evaporative systems can be classified into Direct Evaporative Cooling – DEC, Indirect Evaporative Cooling – IEC. DEC has a wet bulb temperature as limit condition to cooling the air stream. DEC can be sub-divided basically in the porous media and spray. There are porous media coolers made by cellulose corrugated sheets (Paschold, 2003), aspen wood pad, fiber membrane (Jonhson, 2003), PVC sponge mesh (Liao, 2002), ceramic materials (Ibrahim, 2003), natural fiber (de Olveira, 2007). Investigations examined cooling efficiency, equations of heat and mass transfer and non-dimensional numbers. Those results are based on experimental measurements of air velocity, relative humidity, temperature and static drop pressure. Add to these measurements, thermodynamic properties and psychometric chart for water vapour are fundamentals. Results of all those papers indicate that DEC of porous exchanger has a very large application for thermal comfort on commercial, residential and industrial buildings. The studies in spray airflow systems explore parameters effects, droplet size, flux, and velocity, for example, in order to achieve maximum heat flux. Several many experimental and numeric models have been proposed (Kachhwaha, 1998). Spray systems are best applications in large volume places, small and middling cultures and food storage: droplets can moisten surfaces. Another kind of DEC includes ultra sonic vibration that induces the cavitations of liquid and enhancing water evaporation. Effects of ultrasonic waves on heat and mass transfer are investigated. Results indicate an ideal column of liquid to a maximum mass transfer. The better frequency of boiling is associates to the resonance frequency of vessel (Kim, 2004; Park, 1987). IEC arrangement useful water-cooling from DEC process acting as secondary refrigerant. These arrangements allow obtaining cooling temperature bellow of wet bulb temperature and close to the dew point. In fact, the inlet dew point can be approached (Hsu, 1989).

In this paper, porous ceramic materials act as medium of mass and heat transport, in DEC process. The objective is to obtain the design information of the influence of air flow velocity on the behavior of the heat and mass transfer process in prototypes made by porous ceramic material acting like evaporative cooler.

2. MODEL

Direct evaporative cooling process (DEC) is similar to the fundamental adiabatic saturation concept (Van Wylen, 1995). In it process the temperature of the air stream decrease while absorbs humidity. The sum of latent plus sensible energies is zero. Note below the heat and mass exchanger model as presented in Figure 1. Inside this cooler a little pump makes the water flow from a down reservoir to higher level. For gravity water fill the porous ceramics cylindrics elements. One portion of water bleeds off, while another evaporates.

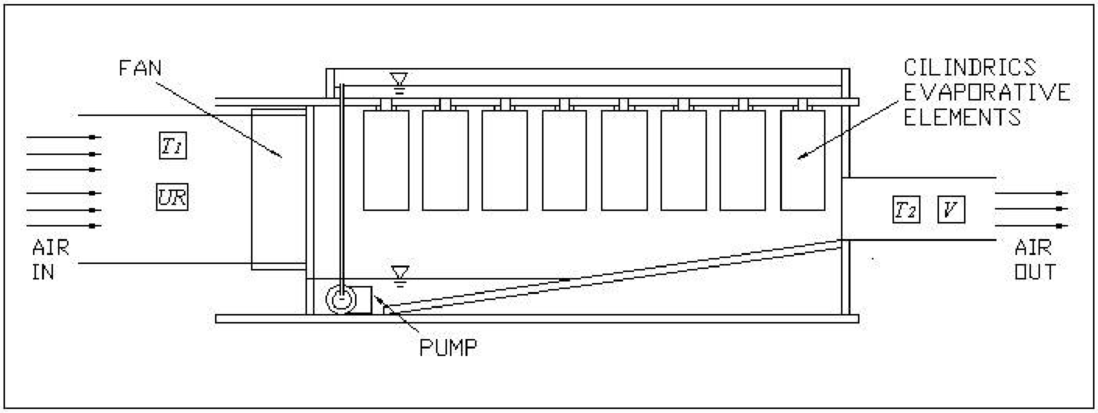

Figure 1. Evaporative cooler model: chamber test.

The cooling power from the evaporative exchanger is given by:

$$\dot{q}_{sen\,s,as} = \dot{m}_{as}.c_{pas}.(T_2 - T_1) \tag{1}$$

Where $\dot{m}_{as}$ (kg/s) is the dry air mass flow rate, C_{Pa} (kJ/kg.K) is the specific heat of dry air, T_1 (°C) and T_2 (°C) are respectively the inlet and outlet dry-bulb temperatures. The dry air mass flow rate can be as a found function of velocity V(m/s), area A (m^2) and density ρ_2 (kg/m^3):

$$\dot{m}_{as} = \rho_2.V.A \tag{2}$$

Assuming ideal gas behavior, dry air mass rate can be written as a function of atmospheric pressure P_{atm} (kPa), saturated water vapour pressure P_{sat2} (kPa) at outlet dry-bulb temperature and relative humidity UR (%):

$$\dot{m}_{as} = \frac{(P_{atm} - UR_2.P_{sat2})}{0{,}000287.(273{,}15 + T_2)}.V.A \tag{3}$$

The water evaporated mass rate $\dot{m}_{ev}$ (kg/s) can be written as a function of mass flow rate of dry air, at absolute humidity ratio, ω_1 (kg/kg) and ω_2 (kg/kg), respectively at inlet and outlet:

$$\dot{m}_{ev} = m_{as}.(\omega_2 - \omega_1) \tag{4}$$

Applying perfect gas law, humidity ratio ω_1 (kg/kg), can be written as:

$$\omega_1 = \frac{0{,}62198.UR_1.P_{sat1}}{P_{atm} - UR_1.P_{sat1}} \tag{6}$$

Where P_{atm} (kPa) is the atmospheric pressure, P_{sat1} (kPa) is the saturated water vapour pressure at inlet dry-bulb temperature and UR_1 (%) is the relative humidity. The same form is obtained to the humidity ratio ω_2 (kg/kg), at outlet dry-bulb temperature:

$$\omega_2 = \frac{0{,}62198.UR_2.P_{sat2}}{P_{atm} - UR_2.P_{sat2}} \tag{7}$$

The cooling efficiency is given by:

$$\eta = (T_1 - T_2)/ (T_1 - T_{W1}) \tag{8}$$

Where T_{WT} (°C) is the wet-bulb temperature. The coefficient of mass transfer h_m (m/s) can be expressed function of water evaporated mass flow rate $\dot{m}_{ev}$ (kg/s), exchange effective area A_{ef} (m^2), and log mean mass density of water vapor $\Delta\rho_v$ (kg/m^3):

$$h_M = \dot{m}_{ev} / A_{ef}.\, \Delta\rho_v \tag{8}$$

$$\Delta\rho_v = (\rho_{v2} - \rho_{v1})/\ln[(\rho_{v2} - \rho_S)/(\rho_{v1} - \rho_S)] \tag{9}$$

Where ρ_{v1} (kg/m^3) and ρ_{v2} (kg/m^3) are the mass densities of water vapour in the air stream before and after evaporative exchanger, and ρ_S (kg/m^3) is the mass density of water vapour at wet-bulb temperature. Handling Equations (3), (4), (8) e (9) at coefficient of mass transfer given by (Liao, 2002):

$$h_M = \rho_a.V_o.A_o \frac{(\omega_2 - \omega_1)}{A_{ef}}.\frac{ln[(\rho_{v2} - \rho_S)/(\rho_{v1} - \rho_S)]}{(\rho_{v2} - \rho_{v1})} \tag{10}$$

In the same way, the sensible heat rate can be obtained as a function of heat transfer coefficient h_H (W/m^2.K), exchange effective surface A_{ef} (m^2) and log mean temperature difference ΔT (°C) for a constant water temperature, as following:

$$\dot{q} = h_H.A_H.\Delta T \tag{11}$$

$$\Delta T(\mathrm{K}) = (T_2 - T_1)/\ln[(T_2 - T_{W1})/(T_1 - T_{W1})] \tag{12}$$

The heat transfer coefficient could be written as function of the cooling efficiency, given by:

$$h_H = \frac{\dot{q}}{A_{ef}}.\frac{ln(1-\eta)}{(T_2 - T_1)} \tag{13}$$

Inlet and outlet temperatures, respectively T_1 (°C) and T_2 (°C), outlet velocity air V (m/s), inlet relative humidity UR_1 (%) and exit area A (m^2), are directly measurable on the prototype. Saturated pressure P_{sat1} (kPa), P_{sat2} (kPa), outlet relative humidity UR_2 (%) and wet-bulb temperature T_{WT} (°C) are numerically calculated using *EES-Engineering Equation Solver* software. The others interesting parameters are calculated by the equations shown above.

3. PROTOTYPES

Initially, the theory of flows through under a set of cylindrics tubes, presented in Incropera (2003), and was used to design the ceramic evaporative exchanger prototypes. After of mathematical method to be computationally implemented, it was possible estimate the airflow rate function of drop pressure, across the exchanger, needed to specify commercial fans. The prototype housing is made of welded polypropylene plates. Figure 1 shows the prototype chamber-test. The dimensional characteristics of three ceramic evaporative porous elements are shown in Fig. 2.

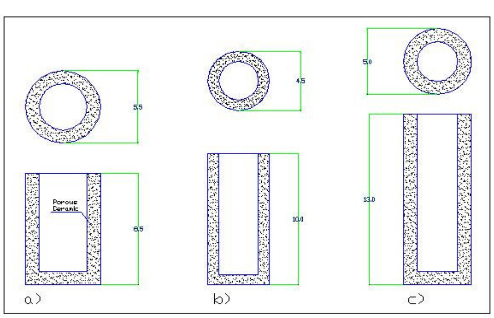

Figure 2. Porous ceramic evaporative elements.

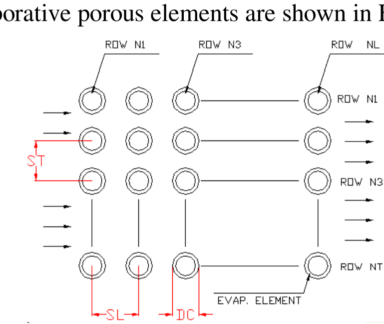

Figure 3. Matrix of evaporative elements.

Prototype 'A' has surface unit area, A_{el} = 17,1x10^{-3}m^2 and 29 evaporative elements with change effective area, A_{ef} = 0,49 m^2. Prototype 'B' has surface unit area, A_{el} = 15,73x10^{-3}m^2 and 47 evaporative elements and A_{ef} = 0,74 m^2. Prototype 'C' has surface unit area, A_{el} = 22,38x10^{-3}m^2 and 42 evaporative elements ante A_{ef} = 0,94m^2. The matrix of evaporative elements is arrangement as shown in Fig. 3.

4. DATA COLECTION

Experiments were performed on prototypes under a variety of environmental conditions and velocity to assess the heat and mass transfer. Under condition, the system is allowed to reach steady state before moving to the next step. Inlet and outlet air-dry bulb temperature were measured with 5 channels digital thermometer bi-metallic sensor. Air velocity was measured with digital helices sensor. For relative humidity was used a digital hygrometer bimetallic sensor. Their technical characteristics are presented on Tab. 1.

Table 1. Technical characteristics of measurement devices.

Dimension	Unit	Scale	Resolution	Precision	Sensor	Rereference
Temperature	°C	-50 – 105	0,1	+/-0,1%	Bi-metalic	Penta Full Cage
Velocity	m/s	0,3 – 45	0,1	+/-0,3%	Helice	TAD 500
Humidity	%	0 – 100	0,1	+/-3%	Bi-metalic	HT-260

Tables 2 and 3 provide the expanding uncertainties of variables due at systematic error associated to the measurement equipments.

Table 2. Relative uncertainties at the parameters directly measured.

Parameters	T_1 (°C)	T_2 (°C)	UR_1 (%)	V (m/s)	A (m^2)	A_{ef} (m^2)
Relative uncertainty (%)	0,5	0,5	3,3	5,3	0,2	0,5

Table 3. Relative uncertainties at the fulfillment parameters.

Parameters	η (%)	$\dot{q}$ (W)	h_M (m/s)	h_H (W/m^2.K)
Relative uncertainty	3,5	6,5	9,5	7,5

5. RESULTS AND DISCUSSION

Figures 4, 5 and 6 present the evaporative cooling efficiency (η) as a function of the dry air mass flow rate ($\dot{m}_{as}$) for three evaporative ceramics prototypes.

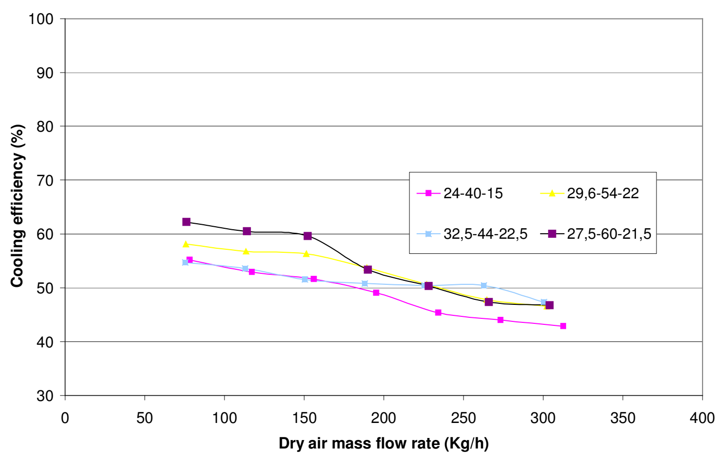

Figure 4. Evaporative cooling efficiency as a function of dry air flow rate: prototype A_{ef} = 0,49 m^2

Each line is associated with just one specific climatic condition represented on legend respectively, by T_1 (°C), UR (%) and T_{WT} (°C). Results indicate that evaporative cooling efficiency decrease as increase of mass flow rate and with the decrease of relative humidity for the same wet-bulb temperature condition. Results indicate also that cooling efficiency is dependent of the climatic condition in time of measurement.

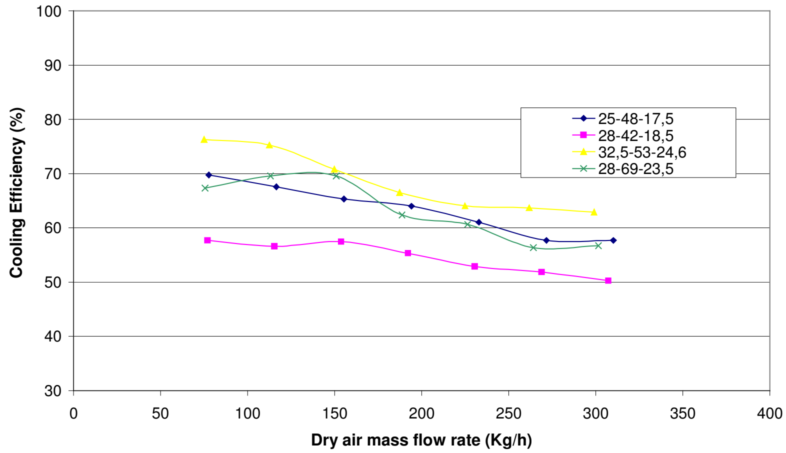

Figure 5. Response of evaporative cooling efficiency varied with dry airflow rate: prototype A_{ef} = 0,74 m^2.

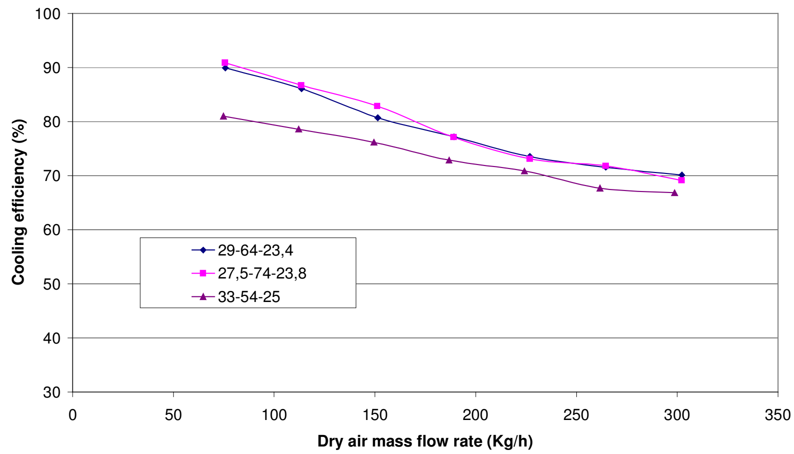

Figure 6. Evaporative cooling efficiency as a function of dry air flow rate: prototype A_{ef} = 0,94 m^2

Table 4 shows the best values of cooling efficiency achieved to the respective climatic condition. The ceramic evaporative prototype with change effective area A_{ef} = 0,49 m^2, obtained maximum cooling efficiency (η) of 62,2 %. The prototype with A_{ef} = 0,74 m^2, obtained η = 76,3%, and the prototype with A_{ef} = 0,94 m^2, attained η = 90,9 %. Johnson (2003) making use of micro porous hollow fiber membranes and Liao (2002) utilizing a pad made of PVC sponge mesh obtained cooling efficiency close to 80% of cooling efficiency.

Table 4. Test response of evaporative cooling efficiency: maximum values.

Ceramic evaporative Prototype	T_1 (°C)	*UR* (%)	η (%)
A_{ef}= 0,49 m^2	27,6	60,4	62,2
A_{ef} = 0,74 m^2	32,6	53,5	76,3
A_{ef} = 0,94 m^2	27,5	74,7	90,9

Figure 7 shows the arithmetic average of evaporative cooling efficiency for all climatic conditions measured in each prototype. Results indicate that cooling efficiency is directly proportional to the heat and mass effective area exchange.

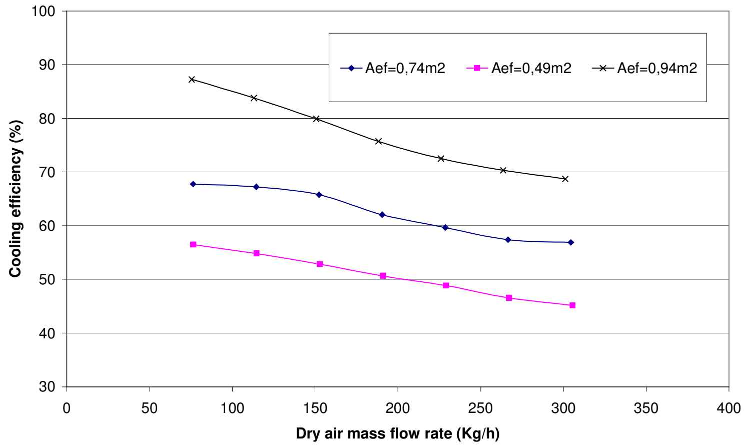

Figure 7. Average of evaporative cooling efficiency for prototypes as a function of dry air flow rate.

Table 5 are presented the temperature difference between the inlet and outlet air stream in evaporative cooler to the respective couple, dry-bulb temperature and relative humidity. The major difference is achieved for the low velocity air stream (V = 2 m/s).

Table 5. Temperature difference between the inlet and outlet air stream in evaporative prototype.

Ceramic Prototype	T_1(°C)	*UR*(%)	ΔT(°C): V=8m/s	ΔT(°C): V=2m/s
A_{ef} = 0,50m^2	32,5	44	4,6	5,4
A_{ef} = 0,74m^2	32,6	53	4,9	6,1
A_{ef} = 0,94m^2	33,5	54	5,2	6,7

For the same climatic condition results indicate that the cooling power increases almost linearly while increases the dry air stream, Figures 8 and 9. The prototype with effective area equal to 0,49 m^2 obtained maximum cooling power of 386 W, 780 W/m^2 of cooling power per unit of effective area, at dry air flow rate equal to 300 kg/h, for inlet air conditions of T_1 = 32,6 °C e *UR* = 44 %. Prototype with A_{ef} = 0,74 m^2 obtained $\dot{q}$ = 409 W, $\dot{q}/A_{ef}$ = 557 W/m^2, for T_1 = 32,6 °C e *UR* = 53%. The prototype A_{ef} = 0,94 m^2 achieved $\dot{q}$ = 434 W, $\dot{q}/A_{ef}$ = 462 W/m^2 for T_1 = 32,9 °C e *UR* = 55 %. Ibrahim (2003) utilizing a prototype made of ceramic material obtained 224 W/m^2 of cooling power per unit of effective area, for inlet air conditions for T_1 = 35,5 °C e *UR* = 25%.

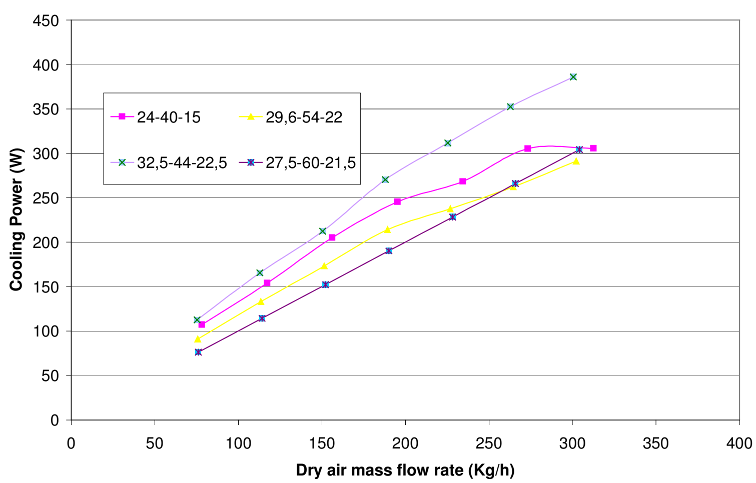

Figure 8. Cooling power as a function of dry air flow rate: prototype with A_{ef} = 0,49 m^2

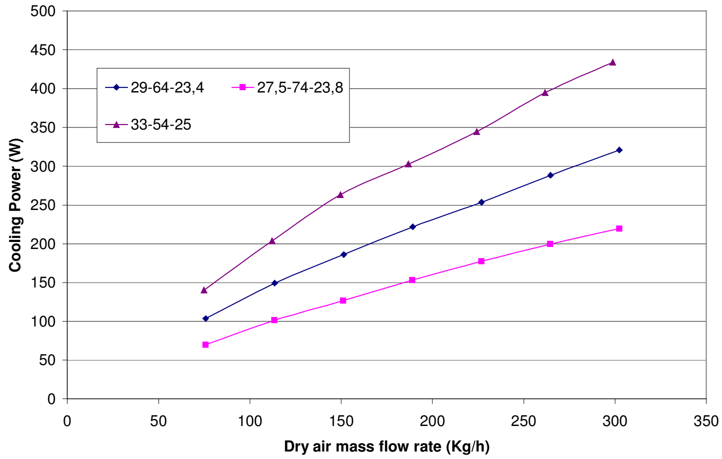

Figure 9. Cooling power as a function of dry airflow rate: prototype with $A_{ef} = 0,94\ m^2$.

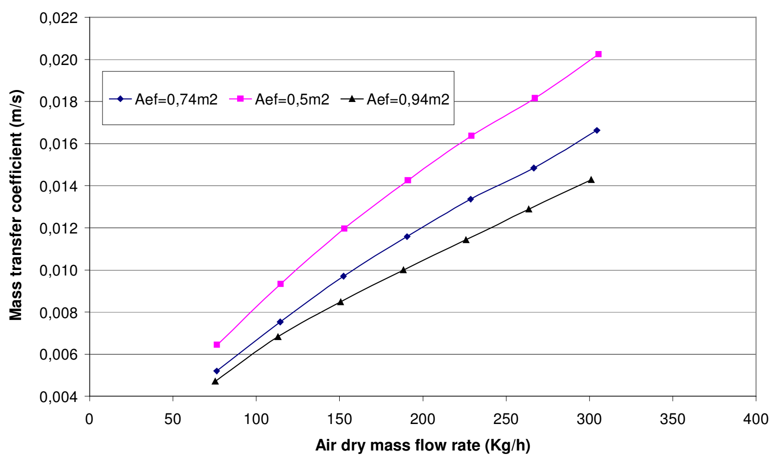

Figure 10. Mass transfer coefficient as a function of dry airflow rate for ceramics prototypes.

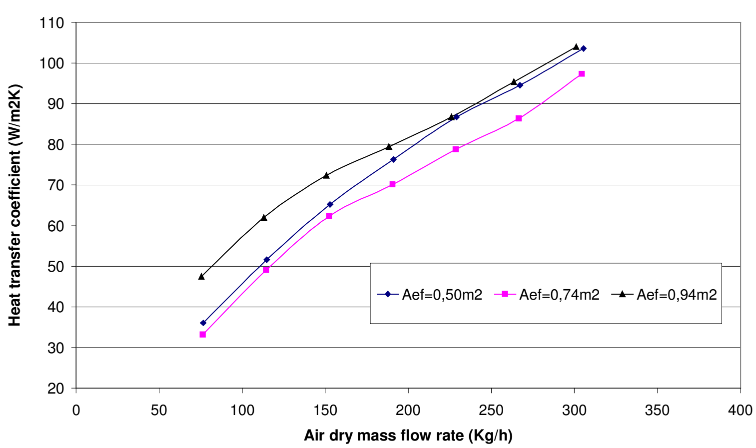

Figure 11. Heat transfer coefficient versus dry airflow rate for ceramic evaporative prototypes.

Figures 10 and 11 show the aritimetic average values of mass and heat tranfer coefficients respectively, versus dry air mass flow rate for all climatic conditions measured in each prototype. The results indicate that mass and heat coefficients are are proportional to air volumetric fluo rate for the same climatic condition.

On Table 5 are presented some results found in literature with results in this work. The author's data are at exit velocity of 5 m/s, to the respective dry air mass flow rate of 190 kg/h.

Table 5. Comparative results between some works published.

Prototype	T_1 (°C)	UR_1 (%)	H (%)	$\dot{q}$ (W/m^2)	h_M (m/s)	h_H (W/m^2.K)
Ceramic A_{ef} = 0,49 m^2	32,7	43	51	546	0,012	64,7
Ceramic A_{ef} = 0,74 m^2	32,5	53	66	382	0,011	66,5
Ceramic A_{ef} = 0,94 m^2	33,2	54	73	322	0,009	62,2
Ibrahim (2003)	35,5	25	-	224	-	-
Johnson (2003)	31	50	80	500	0,03	120
Liao (2002)	-	-	80	0,083	5,5x10^{-5}	0,15

6. CONCLUSIONS

Three models of wet-surface heat and mass exchangers were investigated. Porous ceramic materials have been tested as medium of mass and heat transfer in direct evaporative cooling process. The ceramic evaporative prototype with A_{ef} = 0,49 m^2, achieved maximum cooling efficiency of 62,2 %. The prototype with A_{ef} = 0,74 m^2, obtained η = 76,3%, and the prototype with A_{ef} = 0,94 m^2, attained η = 90,9 %. Results indicate that cooling efficiency is directly proportional to the effective area of heat and mass surface. All those results are dependent of environmental conditions. The prototype with A_{ef} = 0,49 m^2 achieved maximum cooling power of 386 W, 780 W/m^2 of cooling power per unit of effective area, at dry air flow rate equal 300 kg/h, for inlet air conditions: T_1 = 32,6 °C e UR = 44 %. The prototype with A_{ef} = 0,94 m^2 achieved $\dot{q}$ = 434 W, $\dot{q}$ /A$_{ef}$ = 462 W/m^2 for T_1 = 32,9 °C e UR = 55 %.

The present study demonstrated the viability to express parametric results based on measurements of air velocity; relative humidity, and inlet and outlet dry temperature. The porous ceramic elements used in this study shown great potential to be explored as evaporative cooler medium to promote thermal comfort in buildings.

7. REFERENCES

Oliveira, A.M., 2007 "Climatização por Resfriamento Evaporativo: Estudo Teórico e Experimental de Protótipos". Programa de Pós Graduação da Engenharia Mecânica da Universidade Federal de Minas Gerais, 46° tese.

Davis, W. W., 1989, "Spray for Humidification and Air Flow Reduction". Ashrae (3290), p. 351 a 356.

Givoni, B., 1992, "Comfort Climate Analysis and building design guidelines". Energy and Buildings, 18:11-23.

Hsu, s.t., Lavan, Z., Worek, W.M., 1989, "Optimization of Wet-Surface Heat Exchanger". Energy (14), p.757.

Johnson, D.W. et alli; 2003, "Analysis of Heat and Mass Transfer Phenomena in hollow fiber membranes used for evaporative cooling". Journal of Membrane Science (227), p.159.

Incropera, F.P., De Witt, D.P., 2003, "Transferência de Calor e Massa", Editora LTC, 5° Edição, 2003.

Kachhwaha, S.S.; Dhar, P.L., 1998, "Experimental studies and numerical simulation of evaporative cooling of air with a water spray". International Journal of Heat and Mass Transfer (41), p.447.

Kim, H.Y., at all., 2004, "Enhancement of Natural Convection and Pool Boiling Heat Transfer via Ultrasonic Vibration", International Journal of Heat and Mass Transfer (47), p.2831.

Kim, S.J., Vafai, K., 1989, "Analysis of a Natural Convection about a Vertical Plate embedded in a Porous Medium", International Journal of Heat and Mass Transfer 1989, p.665.

Ibrahim, E., at all., 2003, "Performance of Porous Ceramic Evaporators for Building Cooling Application", Energy and Buildings (35), p.941.

Liao, C.M., Chiu, K.H., 2002, "Wind Tunnel Modeling the system performance of a alternative evaporative cooling pads in Taiwan region", Building and Environment (37), p.177.

Lomas, K.J. at all., 2004, "Building Bioclimatic Charts for non-domestic buildings and passive downdraught evaporative cooling", Building and Environment (39), p.661.

Mathaudhu, S.S., Oct 200, "Evaporative Cooling in California"; Ashrae Journal, p.81.

Olesen, B.W., Aug. 2000, "Guidelines for Comfort"; Ashrae Journal, p.41.

Park, K.A., Bergles, A.E., 1988, "Ultrasonic Enhancement of Saturated and sub cooled Pool Boiling", International Journal of Heat and Mass Transfer, (31), p.664.

Paschold, H., at all, 2003, "Laboratory Study of the Impact of Evaporative Coolers on indoor PM Concentrations", Atmospheric Environment (37), p.1075.

Van Wylen, Sonntag, Borgnakke, 1995, "Fundamentos da Termodinâmica Clássica". Editora Edgard Blucher, 4° edição.

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