EVAPORATIVE COOLING: AN EXPERIMENTAL INVESTIGATION

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***Abstract**: The air conditioning by evaporative cooling method is considered energetically efficient. This means that it does not need the fluid compression, as normally occurs on the conventional refrigeration methods. Also, it uses water as a work refrigerant, which does not cause any environmental damage. Of simple construction, and allied to the points previously mentioned, it is, then, a much more economical option compared to the conventional systems. We present in this work the functioning principals, the theoretical foundation and the description of an experimental developed apparatus. We present as results, the temperatures and relative humidity measured in the exit of the apparatus and these results are discussed. Results from a theoretical prediction are also presented, and comparisons are done between these and the experiments. As a conclusion, such results showed that the method is able to develop significant refrigeration. The undesirable effect was the relative moisture increase of the air conditioner. Another factor is that the method keeps efficient only for relative humidity of up to 55 % of the air to be conditioned.*

***Keywords**: Resfriamento evaporativo, conforto térmico, condicionamento de ar.*

1. INTRODUCTION

Evaporative cooling is a simple and practical method of air-conditioning. (Fig. 1) It is an endothermic process, removing heat from an environment, and, for example, it removes approximately 580 kcal to evaporate a liter of water at room temperature. With reference to Fig. 1, the evaporative panel showed there, is made of layers of high quality paper *Kraft*, corrugated and porous. This material is permeated with a resin, which gives it a great rigidity and durability

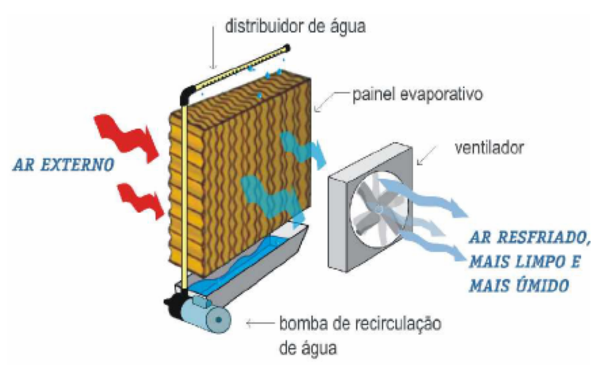

Figure 1. Evaporative cooling.

Temperature reduction between air-conditioner inlet an outlet depends mainly on external air relative humidity: lower humidity gives higher temperature reductions. Temperature reduction depends also on inlet temperature and barometric pressure, which depends on local altitude. Outlet air, leaving the evaporative cooler, does not pass again into the equipment. In this paper the evaporative cooler system was mounted in a laboratory, taking always air from outside. It produced a positive pressure into the room and made easier internal air renovation. Electrical energy consumption was only from a fan and a small water pump. An air conditioning conventional system uses 10 to 15 more electrical energy.

Camargo e Ebinuma (2002) presented a mathematical model of direct an indirect evaporative cooling process, under certain simplified assumptions, based in heat and mass transfer theory. So, they obtained relations between the mainly parameters involved in evaporative cooling process. Those parameters can be used in simulation, design and design optimization.

Castro e Pimenta presented a theoretical approach on an evaporative cooling system made of panels in direct contact with air. They presented a mathematical model of heat and mass transfer processes in those panels and model to calculate pressure drop in a cellulose commercial panel. Figure 2 shows a schematic of the evaporative cooler developed. Such a system consists basically of a panel (cellulose impregnated with resin, which acts as a filter, retaining particles greater than 3μm), water pump, container sprinkler.

Figure 3 presents a photo of the panel used in the present work. The cellulose panel has a big wet surface area in contact with air stream and a small pressure drop.

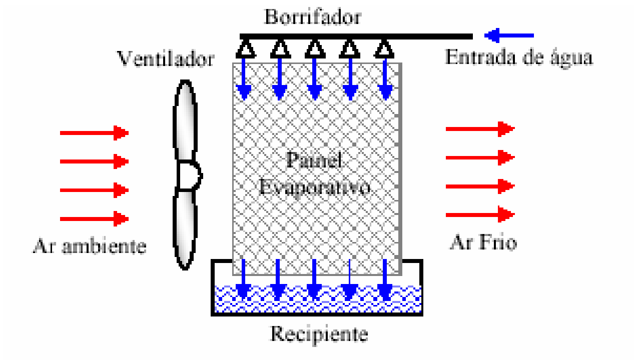

Figure 2. Evaporative cooling system

Figure 3. Evaporative panel used in this work.

2. EXPERIMENTAL APPARATUS

The experimental apparatus is showed in the photo of Fig. 4 and is better understood with help from Fig. 5. It consists mainly of: a) radial fan that promotes air admission and blows air over the panel, b) diffuser which couples the fan and the box containing the panel, c) insufflating nozzle to the air conditioning ambient (after passing the panel). The apparatus has also: d) an air flow control valve placed at external air inlet, e) electrical resistance heater placed into the duct, near to the radial fan, with the purpose of reducing relative humidity, “drying” the air inlet. In this way ,it is possible to obtain several measurements , in a relative humidity range of 25% to 60%. Further, the apparatus has: f) electrical power supply (Variac) to feed the heater with several powers., g) water reservoir placed under the panel box, h)small pump to recirculate water (from water reservoir to panel), i) a small duct, placed above the panel, which drips water on it, j) a glass cover, necessary to panel visualization. The instrumentation utilized was: l) a probe (thermo-higrometer), which measured relative humidity and dry bulb temperature (measurements were done in orifices at the diffuser inlet and nozzle outlet, at symmetrical positions), m) ten thermocouples to measure temperatures of the control surface formed by diffuser – box-nozzle (such measurements were necessary in order to evaluate the heat transfer rate to the ambient, whose temperature was measured with glass thermometers), n) two inclined-tube manometers to measure the pressure drop and to make possible compute the mass rate flow.

Figure 4. Experimental apparatus

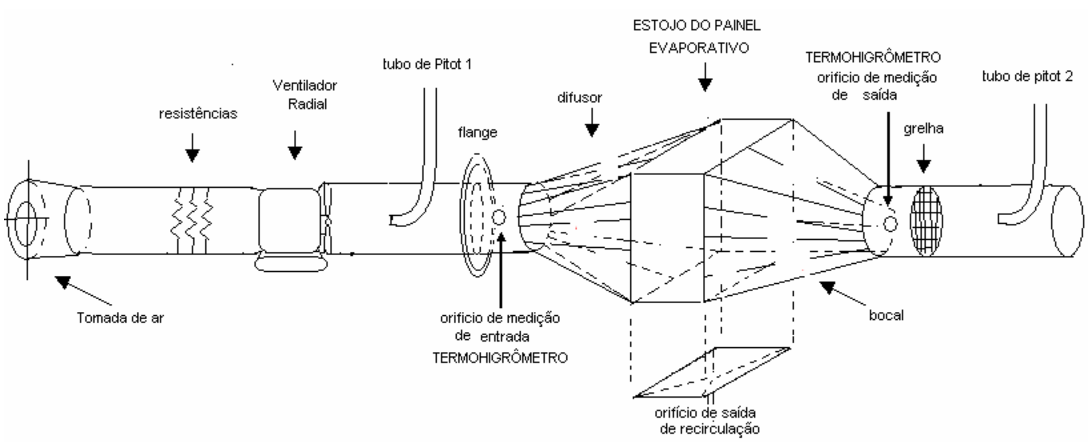

Figure 5. Experimental apparatus schematic.

3. THEORETICAL MODEL

Consideration will be given now to the theoretical formulation necessary to understand the results. An energy balance in the studied control surface, coupled with a psychrometric analysis, will be established. Also, it will be derived a relation giving the heat transfer rate loss from that surface to the environment. This relation is necessary to the mentioned energy balance. Also, in this energy balance, it is necessary to compute the flow mass velocity. The procedure to obtain this parameter will be presented. Also, the calculations of apparatus pressure drop, efficiency parameters, the deviation and humidity increment resulting from evaporative cooling process will be presented.

Saturated air is a mixture of dry air and saturated steam. The steam amount contained in dry air ranges from zero to a maximum, depending from pressure and temperature, when the mixture is in this saturated state. In typical evaporative cooling process, in an ideal formulation, the air reaches till 80% of saturation state and enthalpy remains constant, as it is presented in Fig. 6. In a real process the enthalpy suffers a small variation (Stocker and Jones, 1985).

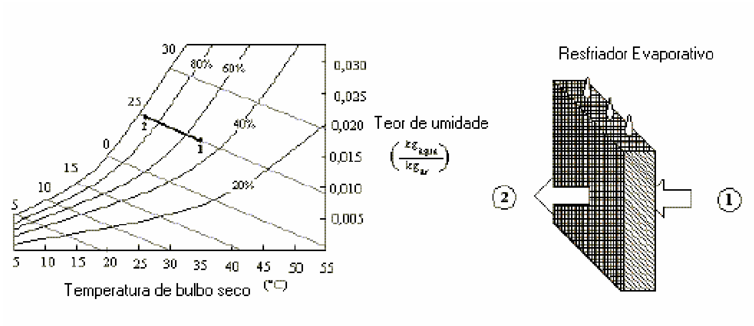

Figure 6. Evaporative cooling process – Psychrometric chart

3.1. Energy Balance

Next, it will be presented the process modeling, based in the *First Law of Thermodynamics.* The main objective is to quantify some evaporative cooling process parameters. Heat and mass transfer balances were applied to air stream that crosses the panel. The process was considered non-adiabatic.

Figure 7 helps the understanding of those balances. (Camargo e Edinuma, 2002):

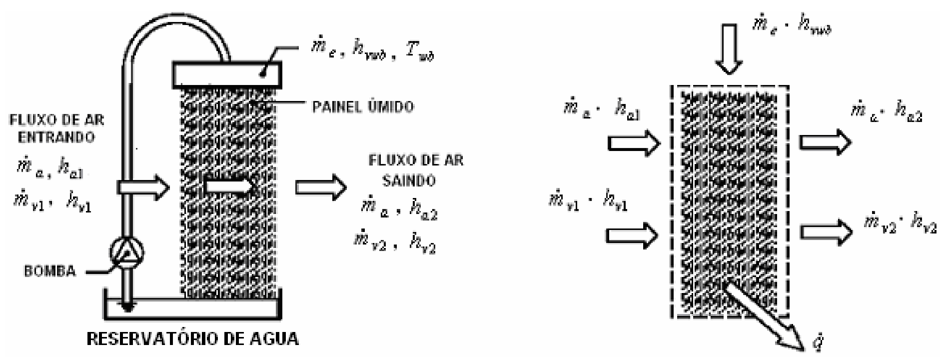

Figure 7. Mass and energy balance schematic in the evaporative cooler

Taking into account the considerations made above and applying the *Energy Equation in Steady Flow,* results

$$\dot{m}_a h_{a1} + \dot{m}_{v1}\ h_{v1} + \dot{m}_e\ h_{vwb} = \dot{m}_a\ h_{a2} + \dot{m}_{v2} h_{v2} + \dot{q} \tag{1}$$

where $\dot{q}$ is the heat transfer rate *[W];* $\dot{m}_a$ is the dry air mass flow rate *[kg/s];* $\dot{m}_{v1}$ *and* $\dot{m}_{v2}$ are the steam mass flow rate at inlet and outlet, respectively *[kg/s];* $\dot{m}_e$ is the evaporated water flow rate *[kg/s];* h_{a1} *and* h_{a2} are the dry air enthalpies at inlet and outlet, respectively *[J/k $_{dry\ air}$];* h_{vwb} is the steam enthalpy at wet bulb temperature *[J/kg $_{dry\ air}$];* h_{v1} *e* h_{v2} are the steam enthalpies at inlet and outlet, respectively *[J/kg $_{dry\ air}$];*. Mass balance at the mixture stream gives:

$$\dot{m}_e = \dot{m}_{v2} - \dot{m}_{v1} \tag{2}$$

Dividing Eq. (2) by $\dot{m}_a$, one obtains:

$$\frac{\dot{m}_e}{\dot{m}_a} = \left(\frac{\dot{m}_{V2}}{\dot{m}_a} - \frac{\dot{m}_{V1}}{\dot{m}_a} \right) = (w_2 - w_1) \tag{3}$$

where w_1 is the specific (absolute) humidity at panel's inlet and w_2 is the specific (absolute) humidity at panel's outlet. From the specific heat at constant pressure definition, cp_a (Wan Wylen et al.,1998), and denoting T_1 e T_2 *[K]* as panel's inlet an outlet temperatures , respectively, Eq. (1) gives T_2, that is, jointly with outlet's humidity, the result of greatest interest in this work.:

$$T_2 = \frac{\left(\dot{m}_a \left[cp_{ar}.T_1 + w_1.h_{v1} - w_2.h_{v2} + (w_2 - w_1)h_{l\,água}\right] - q\right)}{\dot{m}_a.cp_{ar}} \tag{4}$$

Relative humidity is easily found with help of a psychrometric chart, for example.

3.2. Heat transfer and mass flow rates

The convection heat transfer rate exchanged in the control surface needed in Eq. (4) for T_2 computation is presented now. It is assumed a totally free convection. Convection heat transfer coefficients may be found in Incropera and DeWitt, 1998, for example. First, it is necessary to obtain the *Rayleigh* number *(Ra)*, defined by Eq. (5). The *Nusselt* num, is then computed using Eq. (6). The mean convection heat transfer coefficient is obtained using Eq. (7). Thus, the heat transfer rate $\dot{q}$ defined by Eq. (8) is a function of thermo-physical properties and the surface temperature, T_{sup} and the non-disturbed medium temperature, T_{sup}.

$$R_a = \frac{g\beta(T_{sup} - T_\infty)L^3}{\nu\alpha} \tag{5}$$

$$N_u = 0{,}27\ R_a^{1/4} \tag{6}$$

$$\bar{h} = \frac{N_u.k}{L} \tag{7}$$

$$\dot{q} = \bar{h}A_s(T_{SUP} - T_\infty) \tag{8}$$

The mass flow rates, also needed in Eq. (4), were obtained applying the continuity equation in an incompressible duct flow. First, the fluid velocity is computed using the data obtained in the manometers (dynamical and statatic pressures). Air density is evaluated using the perfect gas law. Once obtained the fluid mean velocity, C_{MED} , the air ,steam and mixture mass flow rates are calculated using incompressible flow continuity equation in the form of Eq. (9), Eq. (10) and Eq. (11). In these equations *A* is the duct cross section area.

$$\dot{m}_{ar} = \rho_{ar}.C_{MED}.A \tag{9}$$

$$\dot{m}_{vap} = \rho_{vap}.C_{MED}.A \tag{10}$$

$$\dot{m}_t = \dot{m}_{ar} + \dot{m}_{vap} \tag{11}$$

4. RESULTS

The cooling efficiency in this work is defined as the temperature drop percentage obtained from cooling. To calculate the cooling efficiency (η_{REF}), Eq. (12), this is based on the measured experimental values from thermo-hygrometer, through of the temperature inputs and outputs, T_e *and* T_s., respectively.

$$\eta_{REF} = \left[\frac{T_e - T_s}{T_e} \right] .100\ \% \tag{12}$$

Figure 8 presents η_{REF} (%), as function of the mixture mass flow rate. It can be noted that this varies from 17 to 30 %, for all range of the mass flow rate. Looking this figure, it appears to have a optimum relationship between $\eta_{RESF.} / \dot{m}_t$, in which would be implicitly linked to the area of exchange of the evaporative panel. From this optimum point (30%) to ahead, with the increasing of the size of the evaporative panel (or area of exchange), this situation would be insufficient to maintain the same efficiency.

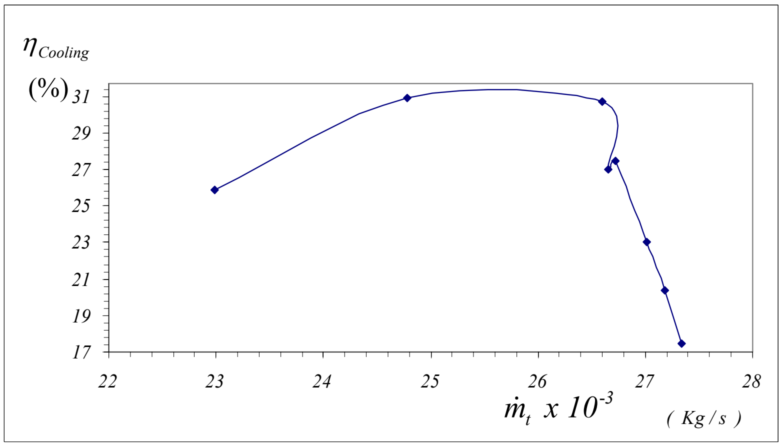

Figure 8. Cooling efficiency versus mixture mass flow rate.

The *deviation* in this work is defined as a measurement of the difference between the experimental and theoretical values. This is based on the values of output temperature provided by cooling. Thus, based on the output temperatures, theoretically calculated and experimentally measured, is formulated the Eq. (13).

$$\psi = \left(\frac{T_{S\ MEDIDA} - T_{S\ TEÒRICA}}{T_{S\ MEDIDA}} \right) 100\ \% \tag{13}$$

Figure 9 presents this deviation (%). There is a maximum value of 1%, in which this gives confidence in the developed experimental and theoretical models.

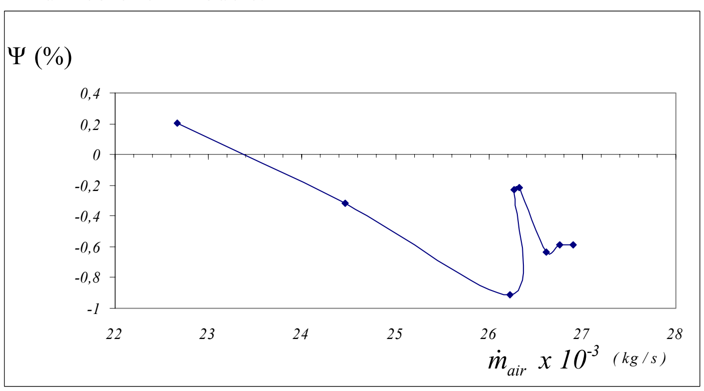

Figure 9. Deviation versus dry air mass flow rate.

Figure 10 presents the relative humidity increment (%), in which is defined by relationship between the experimental relative humidity measurements between the input and output of the device. An increment of the relative humidity is consequence of this method of cooling (Stoeckers e Jones, 1985). In this is verified that the highest increasing in the relative humidity occurs around $25x10^{-3}$ *kg/s*, which is a point close to the highest cooling efficiency point, as can be observed in the Fig. 8.

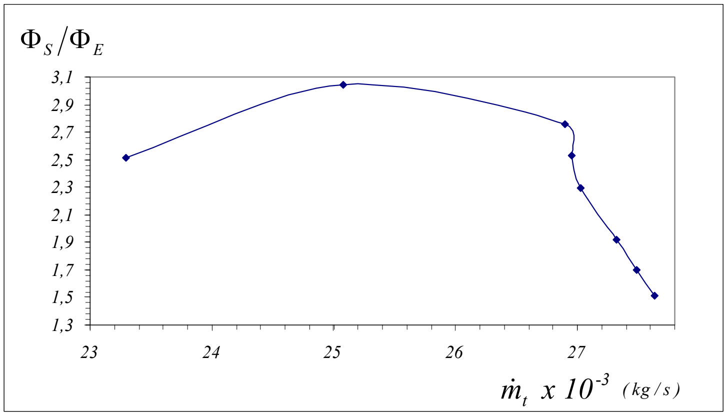

Figure 10. Increasing of the relative humidity versus mixture mass flow rate.

Figure 11 presents the pressure loss, which is consequence of the use of the evaporative panel. The pressure loss is given by the static pressure difference between the input and output of the device. The objective of this study is to provide the scaling parameters to specify a given fan for an evaporative cooler.

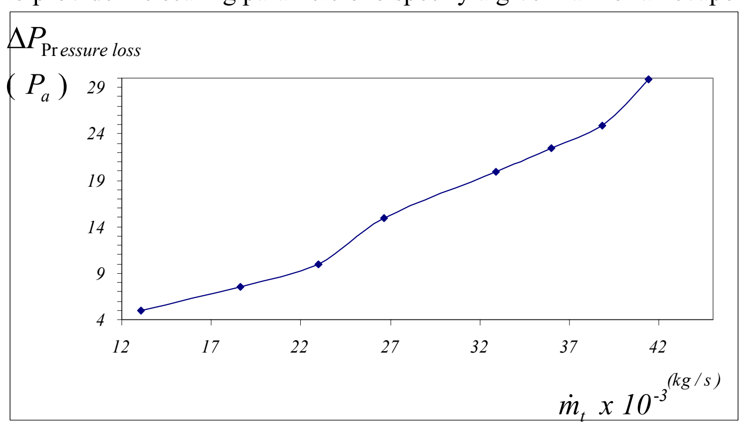

Figure 11. Pressure loss versus mixture mass flow rate.

Figure 12 presents the cooling capacity per mass unit *(kJ/kg)* through of the use of the developed apparatus. Note that the cooling capacity remains with certain uniformity between 23 to 26,5 (x 10^{-3} *kg/s)*.

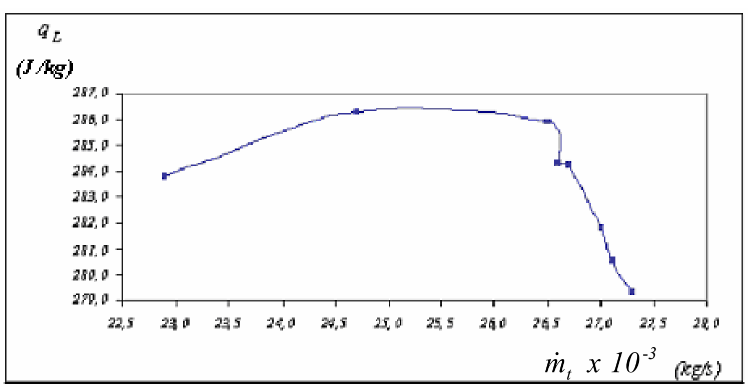

Figure 12. Cooling capacity per mass unit versus flow mass rate.

5. CONCLUSIONS

The main objective in this study was to develop a system to obtain a temperature drop from of outside air to be blown to internal environments, in order to improve the thermal comfort of these environments. It was developed an experimental device associated with a good measurement system. This system is composed of temperature, humidity and pressure sensors in which were associated with appropriate measurement devices. This allowed providing physical parameters with high degree of confidence and low level of uncertainty. The developed theoretical model, based on the *First Law of Thermodynamics for Volume Control for Steady–State*, proved itself to be quite appropriate; it has followed the approach in which can be met in the mentioned literature. The model has accounted the balance of enthalpy; in additional, it has considerated the heat exchange on the surface of control volume (non-adiabatic), in which is normally neglected. This model enabled to calculate the desirable temperature, or output temperature, which is the greatest interesting parameter. The heat transfer rates have been based on experimental data and various correlations. For this objective, various temperatures have been measured at various positions of the surface of the control volume and at environmental; these enabled to calculate the local heat convection coefficients for each part of the control surface. Such coefficients were obtained from correlations, based on the Numbers of *Rayleigh* and *Nusselt*, and they were very similar with those presented in the literature, for free convection cases. Thus, this work avoided to employ the usual treatment in which is to adopt values for the convection coefficient. The output temperature was obtained by experimental and theoretical models. Comparisons between both models have been implemented and the difference calculated. It has been found low deviations, less than 01 °C. Thus, this result confirms that high performance models have been developed, theoretical and experimental, and that the results for each one validate another, reciprocally. Moreover, the low deviation confirms the low uncertainty with high degree of confidence in both models.

The designed apparatus is of easy construction, low cost and simple operation. Basically, it is composed of galvanized steel sheets, a paper element (evaporative panel), a small water pump and a fan. It has low energy consumption because it dispenses the use of compressor, in which also has high acquisition cost.

It has been obtained a high temperature drop, up to 10 ° C. Moreover, the system allows a humidification of the environment, in which depending of the situation can be very desirable. For example in Brazil during the period of July to October, it is usually to be very dry. A humidification would avoid breathing problems in which normally occurs in this time of year.

The output relative humidity remains virtually constant, around 80%, for a wide input relative humidity range, 25 to 55%. Thus, the apparatus develops a relative humidity more or less constant to the environment. Even being above of the typically desirable (60%), in same situations, for example in certain storing, the higher humidity is very desirable to avoid the drying.

6. REFERENCES

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