An Evaporative and Desiccant Cooling System for Air Conditioning in Humid Climates

J. R. Camargo
and E. Godoy Jr.
Universidade de Taubaté - UNITAU
Departamento de Engenharia Mecânica
Rua Daniel Danelli, s/n – Jardim Morumbi
12060-440 Taubaté, SP. Brazil
rui@mec.unitau.br
godoyjr@unitau.br

C. D. Ebinuma
Universidade do Estado de São Paulo – UNESP
Faculdade de Engenharia de Guaratinguetá–FEG
Departamento de Energia
Rua Ariberto Pereira da Cunha, 333
12500-000 Guaratinguetá, SP. Brazil
ebinuma@feg.unesp.br

Evaporative cooling operates using water and air as working fluids. It consists in water evaporation, through the passage of an air flow, thus decreasing the air temperature. This system has a great potential to provide thermal comfort in places where air humidity is low, being, however, less efficient where air humidity is high. A way to solve this problem is to use dehumidifiers to pre-conditioning the process air. This paper presents a system that can be used in humid climates coupling desiccant dehumidification equipment to evaporative coolers. The paper shows, initially, the main characteristics of the evaporative cooling and of the adsorption dehumidification systems. Later on the coupled systems, in which occurs a dehumidification by adsorption in a counter flow rotary heat exchanger following the evaporate cooling of the air in evaporative coolers, are analyzed. The thermodynamic equations of state are also presented. Following, this paper analyzes some operation parameters such as: reactivation temperature, R/P relationship (reactivation air flow/ process air flow) and the thermodynamic conditions of the entering air flow. The paper shows the conditions for the best operation point, with regard to thermal comfort conditions and to the energy used in the process. In addition this paper presents an application of the system in different climate characteristics of several tropical and equatorial cities.
Keywords*: Evaporative cooling, desiccant system, thermal comfort*

Introduction

In the last 10 years, evaporative and desiccant cooling technology for air conditioning systems has increased as an alternative to the conventional vapor compression systems. A typical system combines a dehumidification system that uses a rotary desiccant wheel, with direct or indirect evaporative systems, allowing a filtered and cooled air supplying temperature, humidity and speed conditions that propitiate environmental thermal comfort, even in equatorial and tropical climates. These systems cause a lot of electrical power saving, mainly in places where thermal energy sources are easily found, where the price of electrical energy is high, where the latent heat percentage is high or where the needed air dew point is low. So, evaporative cooling systems that use adsorption pre-dehumidification present a great perspective in thermal comfort. It can be used in co-generation systems, where the needed heat to regeneration can be gotten from gas turbines exhaust gases or from internal combustion engines or, still, from steam in plants that use steam turbines.

This paper aims to analyze the influence of some operation parameters, such as the reactivation temperature of the adsorbent, and the relationship between the reactivation air flow and the process air flow (R/P) on the performance of the system. In addition, this paper presents an application of a proposed system in different climate characteristics of several tropical and equatorial cities.

New technologies using desiccant dehumidification applied to evaporative cooling systems for human thermal comfort have been developed, such as presented by Shen and Worek (1996), Belding and Delmas (1997), Jalalzadeh-Azar (2000), Jalalzadeh et al (2000), Vineyard et al (2000), Jain et al (2000) and Zhenqian et al (2000).

Nomenclature

c_p *=specific heat of moist air at constant pressure, 1.013kJ/kg°C*
DBT = dry bulb temperature, °C
H = altitude above sea level ,m
L = latent heat of vaporization, kJ/kg
$\dot{m}_P$ *= process mass flow, kg/s*
P = atmospheric pressure, kPa
P_v *= vapor pressure, kPa*
P_{vs} *= saturation vapor pressure at DBT, kPa*
P_{swb} *= saturation vapor pressure at WBT, kPa*
Q = heater power, kW
RH = relative humidity, %
R/P = reactivation air flow/process air flow, dimensionless
T = temperature, °C
x = percent of outdoor air to process, dimensionless
w= specific humidity, g_w */ kg* $_{dry\ air}$
WBT = wet bulb temperature, °C

Greek Symbols

γ auxiliary coefficient defined by Eq. (6), kPa/°C
ε_d DEC effectiveness, dimensionless
ε_{cw} ECW effectiveness, dimensionless

Subscripts

numbers correspond to points in Fig. (2)
w corresponds to WBT
react reactivation

Paper accepted June, 2005. Technical Editor: Clóvis R. Maliska.

Evaporative Cooling Systems

Evaporative cooling consists in the use of the water evaporation or other fluid in the presence of a draught, with a consequent cooling of the air. Although it is not common in Brazil, evaporative cooling systems have a great potential to provide thermal comfort in places where the wet bulb temperature is low. Evaporative cooling equipment can be of the direct evaporative cooling (DEC) type or indirect evaporative cooling (IEC) type. Direct evaporative cooling equipment cool the air by direct contact with a liquid surface or with a wet solid surface, or even with sprays (Camargo, 2000). So, in a DEC, water is vaporized in the airstreams and the heat and mass exchanged between air and water decrease the air dry bulb temperature (DBT) and increase its humidity, keeping constant the enthalpy, the minimum temperature that can be reached is the wet bulb temperature (WBT) of the inlet air.

The effectiveness of an evaporative cooler is the rate between the real decreasing of the dry bulb temperature and the maximum theoretical decrease that the dry bulb temperature could be if the cooler was 100% efficient and the outlet air was saturated. In this case, the outlet dry bulb temperature would be equal to the wet bulb temperature of the inlet air. To an ideal evaporative cooler the dry bulb temperature and the dew point might be equal to the wet bulb temperature.

Adsorption Dehumidification System and Energy Conservation Wheel

The adsorption dehumidification is a physical process that fixes molecules of an adsorbate (water, in this case) on the adsorbent surface, usually porous and granulated. Desiccants attract moisture from the air by creating an area of low vapor pressure at the surface of the desiccant. The partial pressure of the water in the air is high, so the water molecules move from the air to the desiccant and the air is dehumidified (Harriman III, 1990). Thus, the essential characteristic of desiccant is their low surface vapor pressure. If the desiccant is cool and dry, its surface vapor pressure is low and it can attract moisture from the air, which has a high vapor pressure when it is moist. Actually, the adsorbents more used are SiO_2 (Silica Gel), ClLi (Lithium Chloride), Al_2O_3 (Activated Alumina), LiBr (Lithium Bromide) and Zeolithe. These substances are usually deposited in a support structure of fiberglass or aluminum. The aspect seams to a fine honeycomb. This process is regenerative because the adsorbent material, after saturated by the humidity, sets the water free, when submitted to a heat source (desorption). The thermal energy to the regeneration can be obtained by electric power, water vapor or hot air.

A typical configuration uses a rotary desiccant wheel that moves slowly and continuously between two crosswise air fluxes, the process and reactivation airstreams. The process air flow through the flutes formed by the corrugations, and the desiccant in the structure adsorbs the moisture from the air. As the desiccant picks up moisture it becomes saturated and its surface vapor pressure rises. Then as the wheel rotates into the reactivation air stream, the desiccant is heated by the hot reactivation air, and the surface vapor pressure rises, allowing the desiccant to release its moisture into the reactivation air. Following reactivation, the hot desiccant rotates back into the process air, where a small portion of the process air cools the desiccant so it can adsorb more moisture from the balance of the process air stream.

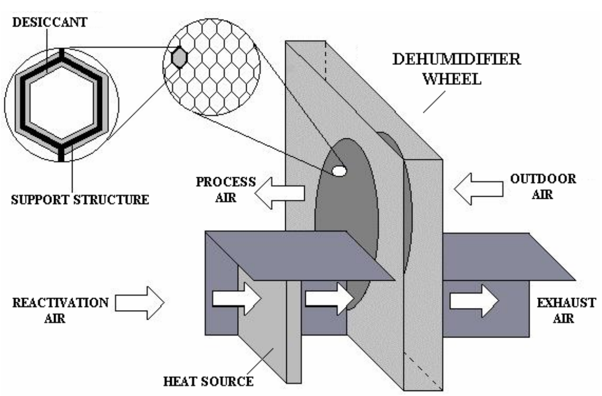

Figure 1. Rotary dehumidifier wheel.

In typical applications, 75% of the air passage is used by the process air and 25% by the reactivation air. Desiccant rotary wheels do not reduce the air energetic load; they only change latent heat (humidity) by sensitive heat (temperature). Figure 1 shows a dehumidifier wheel type honeycomb.

The addiction of a desiccant dehumidifier to an evaporative cooling air conditioning system provides a humidity control separate of the temperature control. It is especially good in applications where the latent thermal load is high comparing to the sensitive load, or when they get the maximum in different time. Usual applications are supermarkets, shopping centers, theaters, hospitals, hotels, motels and officers buildings.

The energy conservation wheel (ECW) is a rotary counter flow air-to-air exchanger used to transfer both sensible and latent heat between supply and exhaust air streams.

Physical Diagram of the System

Figure 2 shows the schematic diagram of the system utilized in this research. In this configuration the outdoor air is first mixed to the return air and flows through the dehumidifier changing latent heat into sensible heat. After, it is cooled in an ECW and in a DEC, providing to the room air conditioned in thermal conditions to human thermal comfort (process 0-1-2-3-4).

The reactivation air stream is also composed by a mixed of outdoor and return air stream that flows first through a DEC and, after, through an ECW. Immediately it is heated by a heat source that can be electrical, vapor or direct burn of a fuel and flow through the dehumidifier, removing the moisture of the adsorbent (reactivation process 5-6-7-8-9).

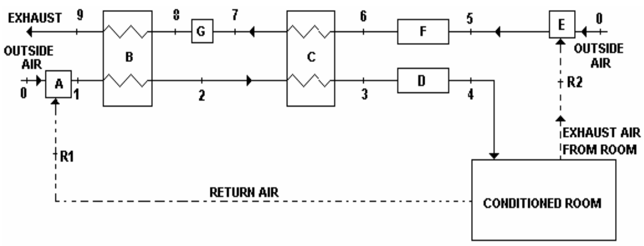

Figure 2. Schematic diagram of the system (unit A: mixer and fan of the process air; unit B: desiccant dehumidifier type rotary wheel; unit C: energy conservation wheel; D unit: direct evaporative cooler; unit E: mixer and fan of the reactivation air; unit F: direct evaporative cooler; unit G: source of reactivation energy).

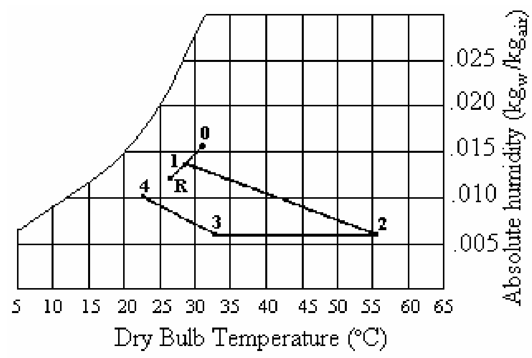

Figure 3. Psychrometric processes.

Figure 3 shows the psychrometric processes for the supply air according to the points that appears in Fig. 2. Point 0 corresponds to the external design conditions for each city, point R corresponds to return air, point 1 corresponds to mixed outdoor and return air condition, point 2 is the dehumidifier exit, point 3 is the ECW exit and point 4 is the DEC exit and it is the final condition of the process air.

Methodology

The analyzed operational parameters that have influence in the system performance are: temperature and moisture of inlet air (that is an outdoor and return mixed air), the reactivation temperature of the adsorbent and the relationship between the reactivation air flow and the process air flow (R/P).

To outdoor conditions it was used values according to ASHRAE 1% summer design conditions (ASHRAE, 1993) and to return air it was used conditions according to ARI (American Institute of Refrigeration).

To determinate the temperatures and moisture in the points presented in Fig. 2, the methodology described bellow was used:

a) The DEC and ECW effectiveness was estimated as 90% and 80%, respectively (Munters, 1999).

b) The used process air flow is constant and equal to $1.67\ m^3/s$. The return air and exhaust air from room flow rates are 50% to each side.

d) The adsorbent utilized is silica gel that has a low reactivation temperature (between 70ºC and 120 ºC).

e) The air condition at point 1 is determined by mixing return air with outdoor air, according to equations (1) and (2). Considering c_p independent of air specific humidity,

$$T_1 = x\,T_0 + (1-x)\,T_{R1} \tag{1}$$

where T_1 is the dry bulb temperature and x is the percent of outdoor air to process, according to:

$$x = \frac{outdoor\ air\ \ flow}{total\ \ process\ air\ \ flow} \tag{2}$$

T_{R1} is the dry bulb temperature at point R1 and the wet bulb temperature at point 1 (T_{1w}) is found by the same way. The conditions at point 5 are also found using equations (1) and (2) by replacing the process air flow by the reactivation air flow.

f) If the dry and wet bulb temperatures are known all thermodynamic properties can be found using the following equations.

The atmospheric pressure at H (m) above the sea level and the saturation vapor pressure at DBT is given by Moreira (1999) as:

$$P = 101.325(\,1 - 2.25577\,x10^{-5}\,H\,)^{5.2559} \tag{3}$$

$$\begin{aligned} ln(\frac{P_{vs}}{22{,}087.87}) &= \\ \frac{0.01}{T+273.15}&(\,374.136 - T\,)\sum_{i=1}^{8} F_i\,(\,0.65 - 0.01T\,)^{i-1} \end{aligned} \tag{4}$$

where F_1 = -741.9242; F_2 = -29.7210; F_3 = -11.55286; F_4 = -0.8685635; F_5 = 0.1094098; F_6 = 0.439993; F_7 = 0.2520658 and F_8 = 0.05218684.

The vapor pressure is given by Jensen et al (1990) as:

$$P_v = P_{swb} - \gamma\,(\,DBT - WBT\,) \tag{5}$$

where γ is calculated as $\gamma = \dfrac{c_p\,P}{0.62198\ L}$ (6)

$$L \cong (\,2.501 - 0.002361\ DBT\,)x10^3 \tag{7}$$

$$RH = \frac{P_v}{P_{vs}}\ x\,100 \tag{8}$$

$$w = 0.622\left(\frac{P_v}{P - P_v}\right) \tag{9}$$

g) Moisture removal in the dehumidifier varies for different reactivation configurations, reactivation energy, process entering conditions and wheel rotational speed. Using the performance graphs or performance data of the wheel’s manufactures were determinate the process outlet moisture and the process outlet temperature after passing through the desiccant wheel. To determinate the thermodynamic state at all the other points it used the following equations, where the numerical index corresponds to the points at Fig. 2.

$$T_3 = T_2 - \varepsilon_{cw}\,(\,T_2 - T_6\,) \tag{10}$$

$$w_3 = w_2 \tag{11}$$

$$T_4 = T_3 - \varepsilon_d\,(T_3 - T_{3w}) \tag{12}$$

$$T_{4w} = T_{3w} \tag{13}$$

$$T_6 = T_5 - \varepsilon_d\,(T_5 - T_{5w}) \tag{14}$$

$$T_{6w} = T_{5w} \tag{15}$$

$$T_7 = T_6 + \frac{(T_2 - T_3)}{(R/P)} \tag{16}$$

$$w_7 = w_6 \tag{17}$$

$$T_9 = T_8 - \left(\frac{T_2 - T_1}{R/P}\right) \tag{18}$$

$$w_9 = w_8 - \left(\frac{w_2 - w_1}{R/P}\right) \tag{19}$$

Normally, the reactivation temperature (T_8) is known but if the heater power (Q_R) is known, the reactivation temperature can be determined as:

$$T_8 = T_7 + \frac{Q_R}{\dot{m}_P\,c_P} \tag{20}$$

Based on Eqs. (1) to (20) the authors developed a software, named **SISREAD** (in Portuguese: **SIS**TEMA DE **R**ESFRIAMENTO **E**VAPORATIVO-**AD**SORTIVO) that allows to determine the supply air conditions. Figure 4 shows the SISREAD flow chart.

Results

This paper analyzes some operational parameters such as: reactivation temperature, R/P relationship and the thermodynamic conditions of the entering air flow.

Figure 5 shows the influence of the reactivation temperature in the process supply air. To obtain it the inlet data was the following: to point 1, Fig 2, TDB = 28.85 ^{o}C; WBT = 21.83 ^{o}C e w = 0.0135 kg_w/kg_{air} , that is the mixer conditions of 50% outdoor air and 50% return air to São Paulo (Brazil). To point 7, DBT = 48.2 ^{o}C; WBT = 28.35 ^{o}C e w = 0.0162 kg_w/kg_{air}, that is the outlet conditions of the ECW, calculated from Eqs. (16) and (17). The performance of the dehumidifier is obtained from two softwares available by manufacturers of desiccant wheels: *Novelaire Technologies - Desiccant Wheel Selection Program – Version 1.0.5* and *Munters Cargocaire DH Selection Program – Version 9.5a.*

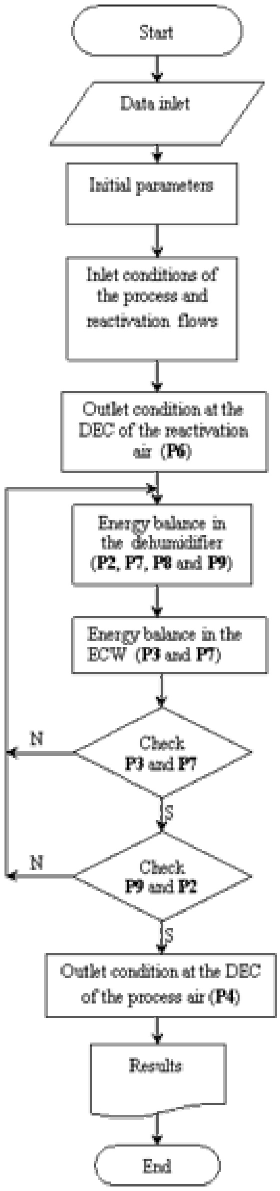

Figure 4. Flow chart of the SISREAD software.

Fig. 5(a) shows the influence of the reactivation temperature on the temperature of the supply air leaving the dehumidifier (point 2, Fig. 2) and Fig. 5(b) shows the influence of the reactivation temperature on absolute humidity.

Figure 6(a) shows the influence of mass flows of the reactivation and process air on the temperature and Fig. 6(b) shows the influence of mass flows of the reactivation and process air on the absolute humidity of the supply air leaving the dehumidifier. In this simulation the relationship R/P changes and the reactivation temperature is keeping constant and equal to 115^{o}C.

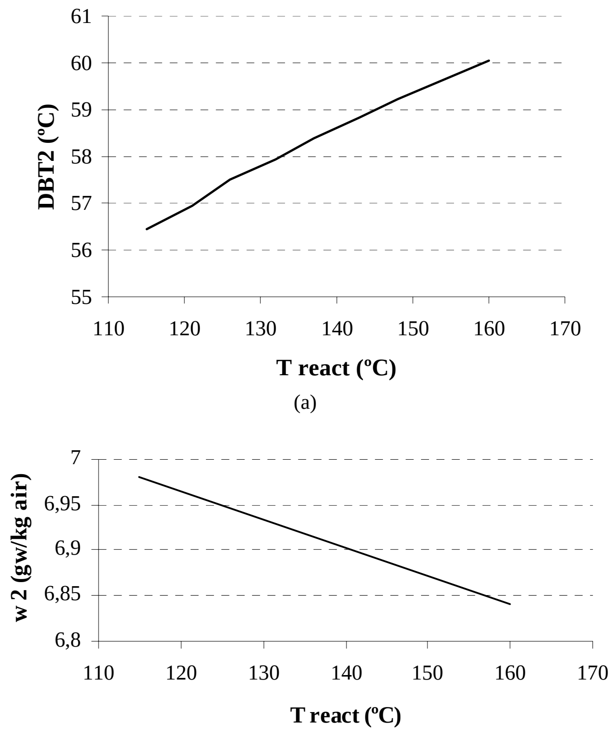

Figure 5. Influence of the reactivation temperature.

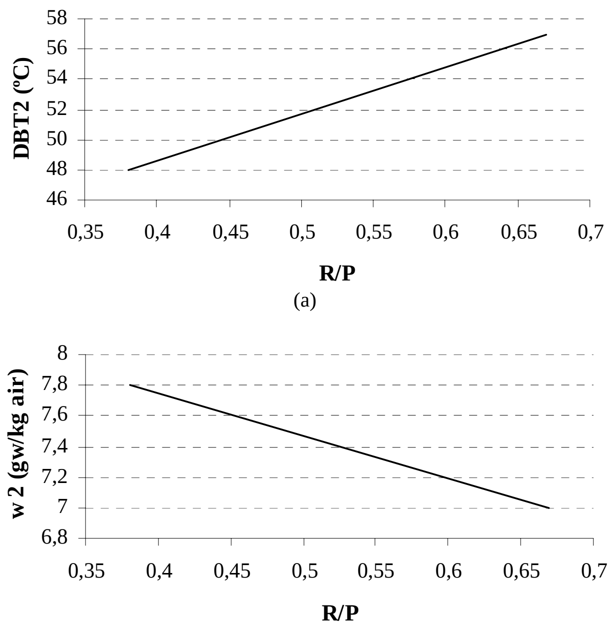

Figure 6. Influence of mass flows of the reactivation and process air.

Table 1 presents the values of the outdoor air (point 1, Fig 2) and supply air (point 4, Fig. 2) for several cities, characterized by humid climates.

Table 1. Outdoor and supply conditions for several cities.

CITY	OUTDOOR AIR		SUPPLY AIR	
	DBT (^oC)	WBT (^oC)	DBT (^oC)	WBT (^oC)
Belem (Brazil)	32	27	21.0	19.7
Brasília (Brazil)	30	22	18.2	16.8
Da Nang (Vietnam)	36	30	22.5	21.2
Hanoi (Vietnam)	37	29	21.9	20.5
Kuala Lumpur (Malaysia)	34	28	21.4	20.1
Madras (India)	40	29	21.9	20.4
Manaus (Brazil	35	29	22.7	21.7
Monrovia (Liberia)	32	28	21.5	20.2
Rio de Janeiro (Brazil)	34	27	20.8	19.5
São Luis (Brazil)	33	28	22.6	21.2
São Paulo (Brazil)	32	23	18.7	17.3
Tainam (Taiwan)	33	29	22.0	20.7

Discussion

In this paper it is analyzed the influence of the outdoor air condition on the air conditioning system performance for several cities in low latitudes as Belém, Monrovia, São Luis and Manaus, that are typical equatorial climates (hot and humid) and medium latitudes cities as Hanoi, Rio de Janeiro, São Paulo and Tainan, representing tropical climate. The city that presents the drier climate is Brasilia and the one that presents the wetter is Tainan. Table 1 shows that the minimum supply DBT is obtained for Brasilia (18.2^oC) and the maximum one for Manaus (22.7^oC). In despite of these cities present very different climate characteristics, the difference in the supply air temperature is only 4.5^oC. The values obtained for the supply air in all studied cities allow to arrive the thermal comfort zone in psychrometric chart.

In despite of the influence of the adsorbent reactivation temperature, it can be seen by Fig. 5 that an increasing in this temperature takes to an increasing in the supply air leaving the humidifier, and to a decreasing in its absolute humidity. It can still be observed that the temperature gradient is higher than the dehumidification gradient. The change in the absolute humidity is 2% and the change in the temperature is 6%. So, to evaporative cooling application it is better to work with the minimum reactivation temperature.

Figure 6 shows that a lesser R/P relationship takes to a lesser DBT2 temperature, what is desired to the supply air temperature and to the heat power used for reactivation. To the studied conditions, a variation of R/P from 0.385 to 0.670 takes to a variation in the power consumption from 51.7 kW to 90.1 kW, i.e. about 80.5 kJ/m^3 of air.

Conclusions

This paper presents an air conditioning system that couples a desiccant dehumidification equipment to indirect and direct evaporative coolers. In this system occurs a dehumidification by adsorption in a counter flow rotary heat exchanger following the evaporate cooling of the air using direct and indirect evaporative coolers.

This paper analyzes some operation parameters such as: reactivation temperature, R/P relationship (reactivation air flow/ process air flow) and the thermodynamic conditions of the entering air flow. The paper shows still the conditions for the best operation point with regard to the thermal comfort conditions and to the energy used in the process. In addition this paper presents an application of the system in different climate characteristics of several tropical and equatorial cities.

An analysis of the results shows that the lesser R/P relationship and lesser reactivation temperature take to the best operation point. The analyzes of the supply air condition shows that the system is able to provide human thermal comfort in humid climates, and it can be an alternative for the conventional air conditioning systems.

References

ASHRAE Fundamentals Handbook, 1993, “Weather Data”, chap. 24.

Belding W.A. and Delmas, M.P.F., 1997, “Novel desiccant cooling system using indirect evaporative cooler”, ASHRAE Transactions, vol. 103, part 1, pp. 841-847.

Camargo, J. R., 2000, “Análise de métodos para avaliar a viabilidade técnica de sistemas de resfriamento evaporativo aplicados ao condicionamento de ar para conforto”, Departamento de Engenharia Mecânica, Universidade de Taubaté, Dissertação de Mestrado, Taubaté, SP, 106p (in portuguese).

Harriman III, L.G., 1990, “The dehumidification handbook”, Munters Cargocaire, Amesbury, MA, USA, 194p.

Jain, S., Dhar, P.L. and Kaushik, S.C., 2000, “Experimental studies on the dehumidifier and regenerator of a liquid desiccant cooling system”, Applied Thermal Engineering 20, pp.253-267.

Jain, S., Dhar, P.L. and Kaushik, S.C., 2000, “Optimal design of liquid desiccant cooling systems”, ASHRAE Transactions: Research, pp. 79-86.

Jalalzadeh-Azar, A.A., 2000, “Consideration of transient response and energy cost in performance evaluation of a desiccant dehumidification system”, ASHRAE Transactions: Research, pp. 210- 216.

Jalalzadeh-Azar, A.A., Steele, W.G. and Hodge B.K., 2000, “Performance characteristics of a commercially available gás-fired desiccant system”, ASHRAE Transactions: Research, 106 (1), pp. 95-104.

Jensen, M.E., Burman, R.D. and Allen, R.G., 1990, “Evapotranspiration and irrigation water requirements”, ASCE Manuals and Reports on Engineering Practice, No. 70.

Moreira, J. R. S., 1999, “Fundamentos e aplicações da psicrometria”, RPA Editorial Ltda, São Paulo, 194p.

Munters, 1999, “Sistema de ventilação com resfriamento do ar através do processo natural de evaporação da água”, Lecture Notes, Curitiba (in portuguese).

Shen, C.M. and Worek, W.M., 1996, “The second–law analysis of a recirculation cycle desiccant cooling system: cosorption of water vapor and carbon dioxide”, Atmosferic Enviroment, vol. 30, n.9, pp. 1429-1435.

Vineyard, E.A., Sand, J.R. and Durfee, D.J., 2000, “Parametric analysis of variables that affect the performance of a desiccant dehumidification system”, ASHRAE Transactions: Research, 106 (1), pp. 87-94.

Zhenqian, C. and Mincheng, S., 2000, “Indirect evaporative cooling and desiccant dehumidifying using advanced heat pipe heat exchangers”, Air Conditioning in High Buildings’2000, IIF/CAR, Shangai, pp. 318-321.