THEORETICAL ANALYSIS OF AN EVAPORTAIVE DESICCSANT COOLING PROCESS

Thomaz Smith Reis, thomasmithr@yahoo.com.br
Paulo Eduardo Lopes Barbieri, barbieri@des.cefetmg.br
José Henrique Martins Neto, henrique@des.cefetmg.br
Centro Federal de Educação Tecnológica de Minas Gerais - Departamento de Engenhraira Mecânica, Av. Amazonas, 7675, Nova Gameleira, Belo Horizonte, MG- CEP-30510-000.

Abstract. *This paper presents results and analyses of simulations performed for a desiccant evaporative air conditioning system operating in two different modes: ventilating and recirculating. The analyses take in account the first and second thermodynamic laws. The simulations were performed for different exterior air conditions. The coefficient of performance (COP) shows to decrease with the increase of the absolute humidity at fixed external temperature for both operating modes: 80% for ventilating and 33% for recirculating when the absolute humidity changes of 4 to 20* g_{vap}/kg_{dry_air} *. They show yet a decrease of 2% for the first and 8% for the second with the increase of the exterior air dry bulb temperature, 27 to 35°C, at fixed humidity ratio. The second law analysis shows that both the reversible and the Carnot cycles follow the same behavior of the COP found for the real cycles. A comparison analyses for the two different operating modes was performed and the results show that strategies using combined modes can offer better opportunity to increase the COP.*

Keywords*: Desiccant, evaporative cooling, air conditioning, heat driven cycle*

1. INTRODUCTION

Over the last fifteen years, active desiccant systems have become a common component of HVAC systems. Desiccant cooling systems are heat-driven cooling units and they can be used as an alternative to conventional vapor compression and absorption cooling systems. The desiccants are natural or synthetic substances capable of absorbing or adsorbing water vapor due the difference of water vapor pressure between the surrounding air and desiccant surface. Prevalently used desiccant materials include lithium chloride, triethylene glycol, silica gels, aluminium silicates, aluminium oxides, lithium bromide solution and lithium chloride solution with water.

According Harriman III *et al.* (1999) its operation is based on the use of a rotary dehumidifier (desiccant wheel, DW) where air is dehumidified. The amount of moisture removed by desiccant wheel depends on a number of variables including the entering air temperature and moisture, the type and quantity of desiccant, the depth of the wheel, the surface area of the honey-comb, the velocity of air moving through the wheel and the wheel rotation speed. However, the most common variable used by commercial manufactures to remove the moisture of the wheel is the air reactivation. Commercial desiccant wheel are usually reactivated with air temperatures varying between 82°C and 107°C.

Note that the temperature rise of the supply dry air is higher when more moisture is removed from it. The temperature rise in the supply air comes from the conversion of latent heat to sensible heat because moisture is removed from the air.

A desiccant cooling system comprises principally three components, namely the regeneration heat source, the dehumidifier (DW), and the cooling unit. The cooling unit can be the evaporator of a traditional air conditioner, an evaporative cooler or a cold coil. The role of the cooling unit is to handle the sensible load while the desiccant is responsible to remove the latent load. The regeneration heat source supplies the thermal energy necessary for driving out the moisture that was absorbed by the desiccant element during the sorption phase. Renewable energies such as solar, geothermal as well as, waste heat from conventional fossil-flue systems may be used to regenerate the desiccant wheel (Daou *et al*. (2006)) .

A typical system combines a desiccant cooling system, with direct and indirect evaporative systems, allowing a filtered and cooling air supply under controlled temperature, humidity and speed conditions so as to propitiate environmental thermal comfort, even in equatorial and tropical climates. These typical systems can be operated in a recirculation mode, in a ventilation mode and in a combined mode (ventilation-recirculation modes). A schematic of the desiccant evaporative cooling system operating in ventilation, in recirculation and combined modes are shown in Fig. 1.

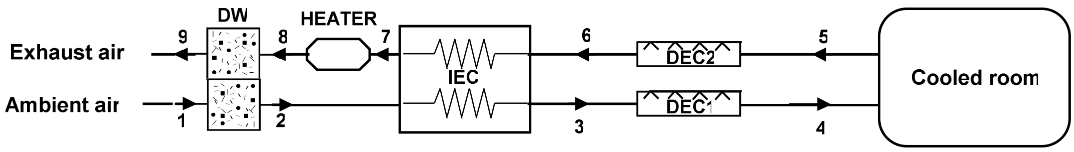
A number of studies are dedicated to the design, development, and analysis of desiccant evaporative cooling systems. According Kanoglu *et al.* (2004) some theoretical studies concentrate on analysis and optimization of operating variables for the entire systems and some are aimed only to the desiccant wheel. There are also some studies that concentrate on the potential use of desiccant evaporative cooling systems in various locations in USA, Europe e South America.

A good number of theoretical and experimental studies have been done to study specific aspects of desiccant evaporative cooling systems; however, not many studies have been performed on the second law analysis. Lavan *et al.*

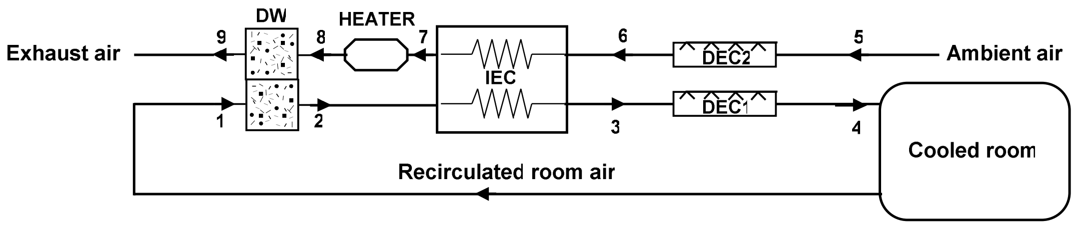
(1982) apud Kanoglu *et al.* (2007) present a general second law analysis of these systems and introduce the concept of the equivalent Carnot temperatures for evaluating reversible COP, which depends on the operating parameters.

Camargo (2003) presented an analysis of the combined mode, involving second law analysis, which can be used to establish potentials and limitations of air conditioning systems by evaporative and desiccant cooling systems when used for human thermal comfort and lower power consumption.

Kanoglu *et al.* (2004) presents a procedure for energy and exergy analysis of desiccant evaporative cooling systems in ventilation mode applied to an experimental working unit. The analysis of Kanoglu *et al.* (2004) showed that an exergy analysis can provide some useful information respect to the theoretical upper limit of the system performance, which cannot be obtained from an energy analysis alone.



(a) Ventilation mode



(b) Recirculation mode.

Figure 1. Schematic desiccant evaporative cooling mode.

Kanoglu *et al.* (2007) presented a model of desiccant evaporative cooling systems in ventilation mode and recirculation mode. Parametric studies were performed to investigate the effects of ambient temperature and relative humidity over the COP (coefficient of performance) and cooling load profiles. As part of this analysis, variations of the thermal COP, the cooling load, the reversible COP and the Carnot COP were obtained due to the change of the ambient temperature and relative humidity for both operation modes. According, Kanoglu *et al.* (2007) the thermal COP and reversible COP have similar trends. The thermal COP may be considered as a practical limit and possible to obtain with the current technology, while the reversible COP is the theoretical upper limit for desiccant evaporative cooling systems.

In this paper, a procedure for the energy and exergy analyses based on Kanoglu *et al.* (2007) model is applied to a desiccant evaporative cooling systems operating in ventilation mode and recirculation mode (Fig. 1a,b). This model assumes the effectiveness for the direct and indirect evaporative cooling equal to 100%. Also, the desiccant cooling process is considered ideal.

2 MODEL DESCRIPITON

A procedure is developed to analyze the desiccant evaporative cooling system operating in both ventilation and recirculation modes, from the point of view of the first and second thermodynamics laws. A schematic of the desiccant evaporative cooling system, operating in ventilation mode, is shown in Fig. 1(a). The model proposed for the desiccant cooling process assumes the desiccant wheel (DW) ideal, which means that the air is completely dehumidified at outlet of wheel, having an absolute humidity, w, equal to zero.

$$w_2 = 0 \tag{1}$$

When the moisture air flows through the desiccant element of the DW the latent heat of the water vapor is converted to sensible heat, promoting the increase of the dry air temperature, named the “air of the process”. An energy balance on the desiccant wheel gives,

$$h_2 = h_1 + (w_1 - w_2)\,h_w \tag{2}$$

where h is the enthalpy of the moisture air at the respective points shown in Fig. 1(a), and h_w is the enthalpy of liquid water. The contribution of the liquid water energy is small and can be neglected because both the absolute humidity and liquid water enthalpy are small. Then,

$$h_2 = h_1 \tag{3}$$

An energy balance is performed in the indirect evaporative cooling (IEC) shown in Fig. 1(a), carrying out the following equation:

$$h_2 - h_3 = h_7 - h_8 \tag{4}$$

This model considers the effectiveness for the indirect evaporative cooling heat exchanger equal to 100%, so,

$$T_3 = T_6 \tag{5}$$

where T is the air dry-bulb temperature at the, respective to the points shown in (Fig. 1a). For the ventilating mode the air mass flow rates are the same in both the process and in the regeneration lines. Also, the absolute humidity remains constant through this heat exchanger, that is,

$$w_3 = w_2 \quad \text{and} \quad w_7 = w_6 \tag{6}$$

Before entering the room, the dry air is cooled by an evaporative cooler heat exchanger, which is considered to follow a saturation adiabatic process (3-4). In that case, liquid water is sprayed in the air stream promoting the reduction of the air temperature. The air temperature drop occurs due to the necessity of latent heat to evaporate the liquid water that comes from the reduction of the sensible heat of the air. Considering as limits of border of the control volume the process air at the entrance and exit of the direct evaporative cooler (DEC1), and the liquid water of the sprinkler, the energy balance gives,

$$h_3 + (w_4 - w_3) h_w = h_4 \tag{7}$$

As mentioned before, the liquid water enthalpy can be neglected, because both the difference of absolute humidity and the enthalpy of liquid water are very small, then,

$$h_3 = h_4 \tag{8}$$

Considering the effectiveness of the direct evaporative heat exchanger equal to 100%, then the relative humidity at the outlet of the equipment will be,

$$\phi_4 = 1 \tag{9}$$

Similarly, for the ideal direct evaporative cooling process installed in the regeneration lines (DEC2), we have,

$$h_6 = h_5 \tag{10}$$

$$\phi_6 = 1 \tag{11}$$

The state 5 corresponds to the room condition which is the room air return temperature, being a fixed value called (T_5, ϕ_5). The enthalpy difference between points 5 and 4 corresponds to the specific room air conditioning load, which has the contribution of both sensible and latent loads.

Pursuing its path in the regeneration line the air passes through another direct and indirect evaporative cooling process before entering the heater. The external heat supplied to the regeneration air is given by,

$$q_{reg} = h_8 - h_7 \tag{12}$$

The model considers the process through the DW ideal, process in which means that the regeneration heat supplied by the heater must be no more than the necessary latent heat for the complete dehumidification of the ambient air entering the DW. Then, the regeneration heat also may be calculated from,

$$q_{reg} = (w_1 - w_2) h_{fg} \tag{13}$$

where h_{fg} is the latent heat of vaporization of the water.

In that manner performing balance of mass in the DW, we obtain,

$$\left(w_9 - w_8\right) = \left(w_1 - w_2\right) \tag{14}$$

Similarly to process 1-2 a energy balance in the process 8-9 gives,

$$h_8 + (w_9 - w_8)\,h_w = h_9 \tag{15}$$

Once more the liquid water enthalpy is negligible because both the difference of absolute humidity and enthalpy of liquid water are very small, then,

$$h_8 = h_9 \tag{16}$$

The room load is given by,

$$q_{cool} = h_5 - h_4 \tag{17}$$

The coefficient of performance of the system, COP, is defined as the ratio of the room load to the regeneration heat, conforms is presented bellow.

$$COP = \frac{q_{cool}}{q_{reg}} \tag{18}$$

The Eqs. (1) to (18) can be solved simultaneously considering as input variables the ambient air state, 1, and the room state, 5.In the present work the proprieties of the moist air and the solution of the equations are evaluated using the commercial equation solver of the Klein and Alvarado (2008).

The cycle described above can also be applied for a desiccant evaporative cooling system operating in the recirculation mode, as illustrated in Fig. 1(b). In the recirculation mode, the room air is recirculated to the process line while the outside ambient air is drawn into of the regeneration line. Note that, for this mode, the states 1 and 5 represent the room and external ambient air states, respectively.

A second law analysis was also performed in this work. The reversible coefficient of performance, COP_{rev}, and the Carnot coefficient of performance, COP_C, were determined for both the ventilation and recirculation modes using the procedure based on Kanoglu *et al.* (2007) and Lavan *et al.* (1982). The COP_C is calculated assuming that the entire cycle is totally reversible. It means that for a heat-driven cooling system the heat from the heat source comes from a heat engine working at the Carnot cycle, while the work delivered by this engine feeds another Carnot refrigeration cycle, in order to remove the heat generated in the room. According Kanoglu *et al.* (2007) the expression for COP_C is,

$$COP_c = \left(1 - \frac{T_{ambient}}{T_{heat\,source}}\right)\left(\frac{T_{cooled\ room}}{T_{ambient} - T_{cooled\ room}}\right) \tag{19}$$

where $T_{ambient}$, $T_{heat\ source}$ and $T_{cooled\ room}$ are the temperatures of the air ambient, heat source and cooled room, respectively. In the systems presented in Figs. 1(a,b) the temperatures of air ambient and the cooled room are, respectively, T_1 and T_5 for the ventilation mode, and T_5 and T_1 for the recirculation mode. The temperature of the heat source is considered equal to the regeneration temperature, T_8.

The desiccant evaporative cooling systems involve components that are inherently irreversible, for example, adiabatic humidification and desiccant processes (Kanoglu *et al.* (2007)). Lavan *et al.* (1982) investigated the desiccant evaporative cooling systems using different approaches, which were based on equivalent Carnot temperatures for the evaporator, condenser and heat source. In that way, the COP_{rev} is defined as,

$$COP_{rev} = \left(1 - \frac{T_{ambient}}{T_{e,hs}}\right)\left(\frac{T_{e,evap}}{T_{e,cond} - T_{e,evap}}\right) \tag{20}$$

where $T_{e,hs}$, $T_{e,cond}$ and $T_{e,cond}$ are the equivalent temperatures for heat source, evaporator and condenser, respectively. Lavan *et al.* (1982) also defined the equivalent Carnot temperature as,

$$T_e = \frac{\sum \dot{m}_i h_i - \sum \dot{m}_o h_o}{\sum \dot{m}_i s_i - \sum \dot{m}_o s_o} \tag{21}$$

where T_e is equivalent temperature, s is the entropy for the fluid, and the subscripts i and o stand for the inlet and outlet states. Kanoglu *et al.* (2007) applied Eq. 21 in their model for the ventilation mode, and obtained the following expressions for the equivalent temperatures,

$$T_{e,hs} = \frac{h_7 - h_8}{s_7 - s_8} \tag{22}$$

$$T_{e,evap} = \frac{h_4 - h_5 + \Delta w_{\text{cooled room}} h_w}{s_4 - s_5 + \Delta w_{\text{cooled room}} s_w} \tag{23}$$

$$T_{e,cond} = \frac{h_9 - h_1 - \left(\Delta w_{DEC1} + \Delta w_{DEC2} + \Delta w_{\text{cooled room}}\right) h_w}{s_9 - s_1 - \left(\Delta w_{DEC1} + \Delta w_{DEC2} + \Delta w_{\text{cooled room}}\right) s_w} \tag{24}$$

In the equations presented above Δw_{DEC1} and Δw_{DEC2} refer to the moisture added per kg of dry air in the DECs installed in the process and regeneration lines, respectively, while the $\Delta w_{\text{cooled room}}$ is the moisture added per kg of dry air in the cooled room due to the latent load and s_w is the entropy of liquid water at the ambient air state. In the recirculation mode Kanoglu *et al.* (2007) defined the equivalent temperatures as,

$$T_{e,evap} = \frac{h_4 - h_1 + \Delta w_{\text{cooled room}} h_w}{s_4 - s_1 + \Delta w_{\text{cooled room}} s_w} \tag{25}$$

$$T_{e,cond} = \frac{h_9 - h_5 - \left(\Delta w_{DEC2} + \Delta w_{DW}\right) h_w}{s_9 - s_5 - \left(\Delta w_{DEC2} + \Delta w_{DW}\right) s_w} \tag{26}$$

In the equations above Δw_{DW} is the absolute humidity exchange in the desiccant wheel.
Therefore, applying Eqs. (20) to (26) we can determine the COP_{rev} for desiccant evaporative cooling systems.

3 RESULTS' AND DISCUSSIONS

The model presented above was applied to a desiccant evaporative cooling system for both the ventilating (Fig. 1(a)) and recirculation modes (Fig. 1(b)). The dry air bulb temperature and relative humidity of the air inside the room were maintained constant and equal to 25°C and 50%, respectively. A psychrometric diagram of the systems is showed below (Fig. 2). In these, for the external air, the dry bulb temperature is 30°C and humidity ratio $14 g_{vap}/kg_{dry\_air}$.

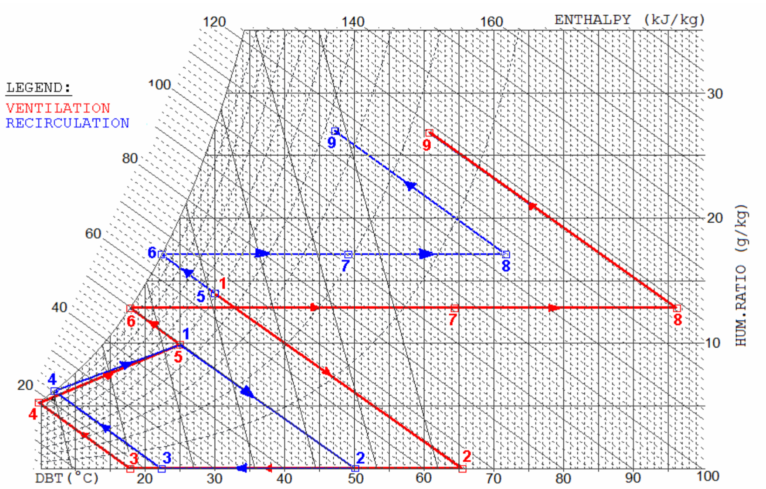
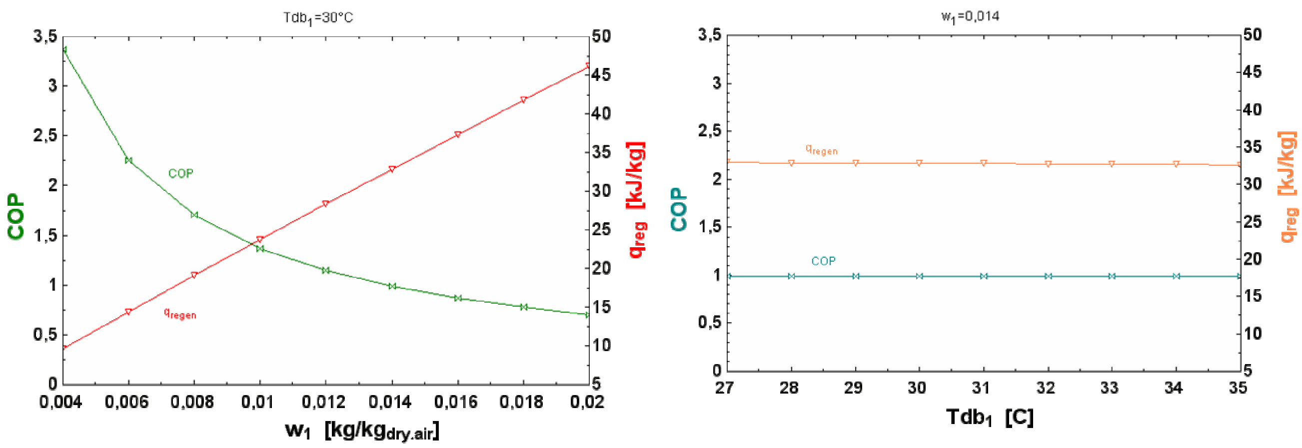


Figure 2. Psychrometric diagram of systems.

It was performed parametric analyses to see the behavior of COP under the influence of the external air conditions, specifically the dry bulb temperature an absolute humidity.

The behavior of the COP and q_{reg} for the ventilating mode is presented in Fig 3(a) and Fig. 3(b). The influence of the absolute humidity of the air exterior is presented in Fig. 3(a), for an external air temperature of 30°C, while the influence of the dry bulb temperature of the air exterior is presented in Fig. 3(b), for an absolute humidity of 14 grams of vapor per kg of dry air.



(a) Effect of the absolute humidity.

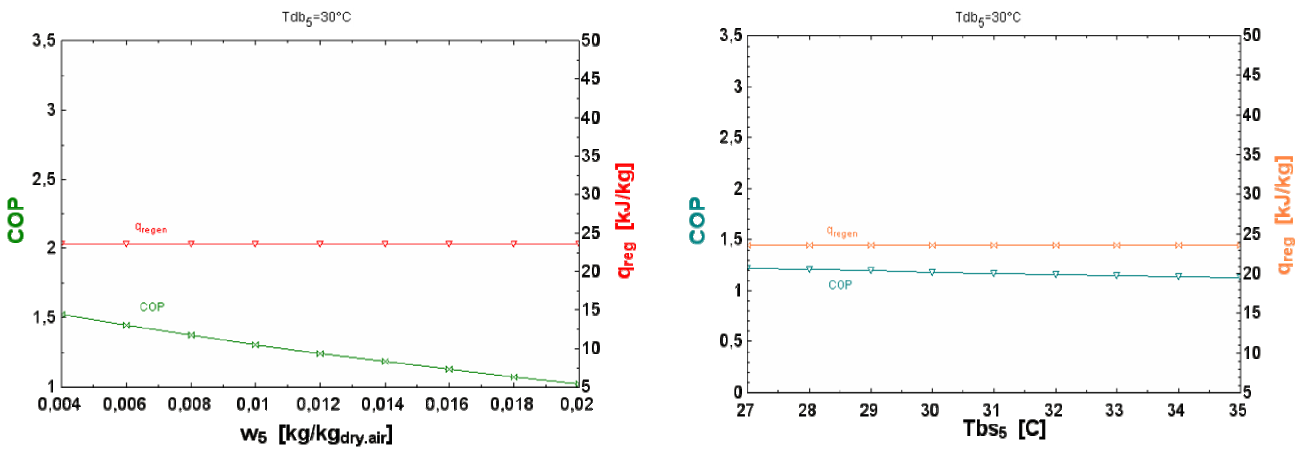
(b) Effect of the dry bulb temperature.

Figure 3. Behavior of the COP and q_{reg} as a function of the air exterior condition for the ventilating mode.

It can be observed from Fig 3(a) that the COP decreases and the regeneration heat increases as the absolute humidity increases. From Eq. (13) it can be seen that the regeneration heat depends mainly on the amount of moisture to be removed from the wheel, therefore, the increase of the absolute humidity at the inlet of the wheel will promote an increase of the difference of absolute humidity through the wheel (w_2 = 0), and consequently increases of the regeneration heat. As the cooling load was maintained constant then COP will drop.

It can be observed from Fig. 3(b) that the COP increases slightly and the regeneration heat decrease slightly as the dry bulb of the exterior air temperature increases. From Eq. (13) it can be seen that the regeneration heat also depends on the latent heat of the water. For our model we have taken the latent heat at the temperature of the air leaving the well (point 2). Therefore, as the exterior air temperature increases the temperature leaving the wheel (point 2) also increases because w_2 = 0, consequently the latent heat decreases and the regeneration decreases. As result the COP increases because the room load was maintained constant.

The behavior of the COP and q_{reg} for the recirculating mode is presented in Fig 4(a) and Fig. 4(b). The influence of the absolute humidity of the air exterior is presented in Fig. 4(a), for an external air temperature of 30°C, while the influence of the dry bulb temperature of the air exterior is presented in Fig. 4(b), for an absolute humidity of 14 grams of vapor per kg of dry air.



(a) Effect of the absolute humidity.

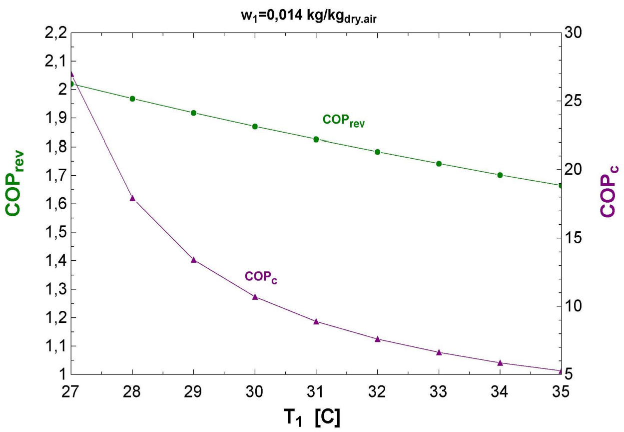
(b) Effect of the dry bulb temperature.

Figure 4. Behavior of the COP and q_{reg} as a function of the air exterior condition for the recirculating mode.

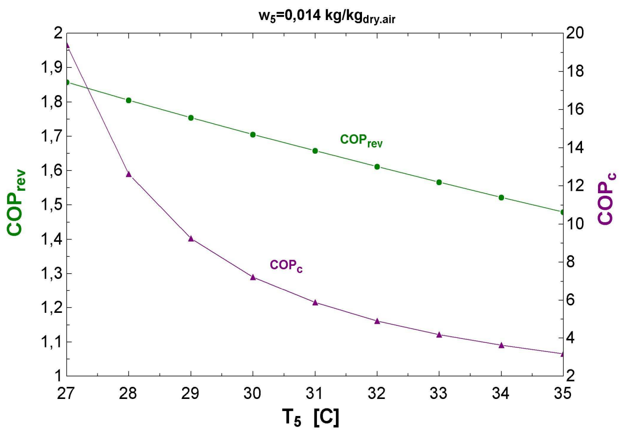
It can be observed from Fig. 4(a) that the COP decreases and the regeneration heat maintain constant as the absolute humidity increases. The temperature of air at point 6 increases as the absolute humidity of the exterior air (point 5) increases due to the isenthalpic process which occurs in the direct evaporative cooler. As this model assumes the temperature of the air at point 3 equal to that of point 6, then the temperature of point 3 increases, and consequently, the temperature at point 4 also increases (isenthalpic line). The temperature difference of points 5 and 4 decreases because the state of point 5 is fixed (room air temperature). Consequently, the enthalpy difference of these same points decreases and the room load decreases. On the other hand, the regeneration heat remains fixed because the state of the room air was maintained fixed and $w_2 = 0$. Therefore, COP decreases as the absolute humidity of the exterior air increases.

It can be observed from Fig. 4(b) that the COP decreases and the regeneration heat maintain constant as the dry bulb of the exterior air temperature increases. Like the previous explanation, the increasing of the absolute humidity of the exterior air causes increasing of the temperature of point 6 which promotes the increase of the temperature of point 3, and consequently, increase of the temperature at point 4 (isenthalpic line). For the same reason shown previously the room load decreases. The regeneration heat again remains fixed because the state of the room air was maintained fixed and $w_2 = 0$. Therefore, the COP also decreases as the absolute humidity of the exterior air increases.

At this point it is presented the influence of the variations of temperature and absolute humidity of the external air in the behavior of COP_C and COP_{rev} using a thermodynamic second law analysis. The effect of the dry bulb temperature of the exterior air is shown in Fig. 5 considering the absolute humidity constant and equal to 14 grams of vapor per kg of dry air for the ventilating mode.



(a) Ventilating mode



(b) Recircularting mode.

Figure 5. Behavior of COP_{rev} and COP_c for dry bulb temperature changes of the exterior air

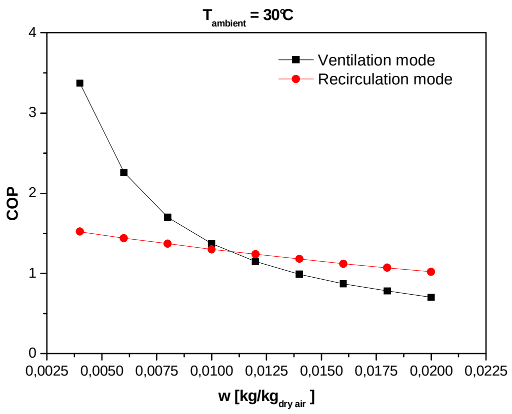
The behavior of COP_{rev}, shown in Fig. 5(a) is similar to the COP behavior presented previously in Fig. 3. According Lavan *et al.* (1982) the increase of the dry bulb temperature of the exterior air causes an increase in both the heat source and the condenser temperatures, and consequently, the COP_{rev} tends to decrease. The COP_c also decreases with the increasing of the exterior air dry bulb temperature, however, it can be seen from Fig. 5(a) that this reduction is much more accentuated because this parameters depends only on the temperatures.

The effect of the dry bulb temperature of the exterior air is shown in Fig. 5(b) considering the absolute humidity constant and equal to 14 grams of vapor per kg of dry air for the recirculating mode.

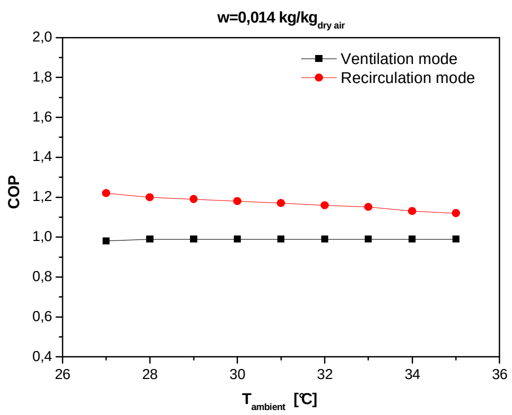
From Fig. 5(b) it can be observed that the behavior of the COP_{rev} and COP_c with the variation of exterior air the dry bulb temperature is similar to that of the ventilating mode.

From the parametric analyses shown in Figs. 5(a) and 5(b) we can say that for both operating modes COP_C represents the limit superior maximum for the coefficient of performance. On the other hand, COP_{rev} represents the limit superior, although this value is not the maximum value, depending of the operating system conditions. According Kanoglu *et al.* (2007) the COP_{rev} is the best parameter to measure the performance of the desiccant evaporative cooling systems.

An additional analysis to compare the ventilating and recirculating modes is presented in Figs. 6(a) and 6(b), using the COP values obtained from Figs. 3 and 4, which consider, respectively, the variations of the absolute humidity and the dry bulb temperature of the exterior air. Fig. 6(a) shows that for low values of the absolute humidity the ventilating mode shows higher values of COP; however, as the absolute humidity increases the COP of the ventilating mode decreases rapidly and at certain point a common value of COP is found for both modes. Beyond this point it can be observed that the COP for the recirculating mode is a little superior than the COP of the ventilating mode. From Fig 6(b) it can be noticed that the COP values for the recirculating mode is always superior to those met for the ventilating the mode. From these two figures it can be recommend control strategies that promote mixture of the return and outside air in order to obtain better values of COP.



(a) Effect of the absolute humidity with fixed external air dry bulb temperature.



(b) Effect of the external dry bulb temperature with fixed absolute humidity.

Figure 6. Comparison of the COP for the ventilating and recirculated modes.

4 CONCLUSIONS

This work presents a model to analyze desiccant evaporative cooling systems operating in both ventilating and recirculating modes, through the point of view of the first and second thermodynamic laws.

The model assumes the efficiencies of the direct evaporative coolers, the indirect heat exchanger and the desiccant wheel equal to 100%.

It has been analyzed the influence of the exterior air conditions in the coefficient of performance of the cooling system. The results show a considerable COP decreases with the increase of the absolute humidity for both modes. Regarding the variation of the exterior air temperature, the COP shows insensitive for the ventilating mode, although it decreases as the temperature increases for the recirculating mode.

The COP_{rev} and COP_C follow the same behavior of the real COP for the variation of the exterior air day bulb temperature. The COP_{rev} can be a good parameter to measure the highest efficiency of a real cooling system.

It can be recommended a strategy which provides the mixture of the exterior with the return air in order to improve the efficiency of the cooling system, being necessary, therefore, to simulate in the future the combined model.

5. ACKNOWLEDGEMENTS

The authors would like to thank the sponsors CNPq and FAPEMIG.

6 REFERENCES

Camargo, J., R. 2003. "Sistemas de resfriamento evaporativo e evaporativo-adsortivo aplicados ao condicionamento de ar," Faculdade de Engenharia - Universidade Estadual Paulista - UNESP, Guaratinguetá - SP, Brasil.

Daou, K., Wang, R. Z., and Xia, Z. Z. 2006. "Desiccant cooling air conditioning: a review." Renewable and Sustainable Energy Reviews, 10, 55-77.

Harriman III, L. G., Witte, M. J., Czachorski, M., and Kosar, D. R. 1999. "Evaluating active desiccant systems for ventilating commercial buildings." ASHRAE Journal, 28-34.

Kanoglu, M., Bolattürk, A., and Altuntop, N. 2007. "Effect of ambient conditions on the frist and second law performance of an desiccant cooling process." Renewable Energy, 32, 931-946.

Kanoglu, M., Çarpinlioglu, M. Ö., and Yildirim, M. 2004. "Energy and exergy analyses of an experimental open-cycle desiccant cooling system." Applied Thermal Engineering, 24, 919-932.

Klein, S. A., and Alvarado, F. L. (2008). "Engineering equation solver (EES)." <www.fchart.com> (jun, 2008).

Lavan, Z., Monnier, J. B., and Worek, W. M. 1982. "Second law analysis of desiccant cooling systems." ASME - journal Solar Energy Engineering, 104, 229-236.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.