ANALYSIS OF HEAT AND MASS TRANSFER REGENERATOR EFFECTIVENESS ON THE PERFORMANCE OF A VENTILATION DESICCANT CYCLE FOR HOT AND HUMID CLIMATES

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Abstract. *A simple algorithm for simulating a ventilation-type desiccant cooling cycle has been proposed. This desiccant cooling cycles differ from the conventional refrigeration cycles, since they use no refrigerant fluids, thereby being exempt from the environmental problem brought by CFCs gases. The proposed algorithm was computationally implemented and simulation results of the ventilation cycle in hot and humid climates were discussed. Simulation results were calculated varying the heat and mass transfer effectiveness of the desiccant wheel and heat exchanger, and by linking the algorithm to a robust desiccant wheel simulation code previously developed. An analysis of cycle performance (COP) for different Brazilian cities with hot and humid climates was carried-out, showing that the cycle has an improved performance for warmer and moister outdoor conditions.*

Keywords: *Desiccant Cycle, Evaporative Cooling, Computer Simulation, Adsorption*

Nomenclature

c, c_p	specific heats
C	sensible heat capacity rate
C_r^*, C^*	sensible heat capacity ratios
h	convective transfer coefficient
i	specific enthalpy
$i_{v,\Delta T}$	interface heat of sorbate transfer
m, $\dot{m}$	mass and mass flow rate
N_tu	number of transfer units
T	temperature
V	volumetric capacity rate
V_r^*, V^*	volumetric capacity ratios
W	concentration in adsorbed phase of desiccant
Y	water vapor concentration in dry basis
Greek symbols	
α	thermal diffusivity
ϵ_f	porosity of sorbent matrix
ε	effectiveness
ρ	specific mass
τ_I, τ_II	duration of periods I and II in desiccant wheel
τ_{dw}	dwell time
Subscripts & Superscripts	
a	dry air
dw	desiccant wheel
f	porous adsorbent in desiccant wheel
hs	heat source
he	heat exchanger
l	liquid water
v	water vapor
in, out	inlet and outlet
m	mass transfer
i	enthalpy transfer

h sensible heat transfer
$*$ dimensionless quantity
$\star$ dry reference quantity

1. INTRODUCTION

It is known that conventional vapor-compression cooling systems are somewhat inefficient in hot and humid climates. This happens, due to the high latent load encountered in these climates. Hence, in order to handle the associated latent load and meet minimum comfort requirements, conventional systems must cool the process air to temperatures much lower than required by the sensible load, thereby reducing cycle performance. In addition, this overcooling, also requires a reheat system to meet comfort conditions, which again reduces overall cycle efficiency. Another drawback associated with vapor-compression cycles is the fact that they employ CFC and HCFCs gases, which may be harmful to the environment, by contributing to the depletion of the ozone layer as well as to global warming.

Because of the disadvantages involved in vapor-compression cycles, researchers have focused their attention and efforts to develop alternative cooling systems. Over the last years, desiccant cooling systems have been gradually used as substitutes for conventional systems. Unlike conventional cycles, which require high grade energy and use CFCs gases, desiccant cycles employ simple technology and are benign to the environment. The principle of operation of these cycles consists of using a desiccant dehumidifier (usually in the form of a desiccant wheel) to dry the process air, followed by an evaporative chiller, which cools it down. Desiccant cycles receive this name because of the desiccant materials used in the dehumidifier. These materials are adsorbents with a high affinity for water, having low vapor pressure at their surface, thereby uptaking water when cold and dry and releasing water when hot and wet. Developments in desiccant cooling cycles can be seen in the studies of Jain, Dhar et al. (1995), Belding and Delmas (1997), Vineyard, Sand et al. (2000), Jalalzadeh-Azar, Steele et al. (2000), Camargo, Godoy Jr. et al. (2005), Mazzei, Minichiello et al. (2002), among others. Other desiccant-based cycles are hybrid vapor-compression cycles, which have been widely studied. These cycles use a desiccant wheel to work as a pre-dehumidifier and as a reheat system, improving overall cycle performance, as can be seen in (Worek and Moon, 1988).

In spite of all advantages associated with desiccant cycles, their current efficiency (measured by the coefficient of performance – COP) are notably lower than the ones encountered in conventional cycles. Because of this, there is a natural demand for improving the performance of desiccant cycles. Hence, computational simulations have been utilized as an analysis tool for optimizing such cycles, which requires appropriate formulations of these systems, and consequently a good understanding of the involved transport mechanisms. Looking into the heat and mass transfer involved in these cycles, the main difficulty lies in the simulation of the regenerative exchangers present in these systems. Recently, a unified formulation (Sphaier and Worek, 2004; Sphaier, 2007) and a robust computational implementation (Sphaier and Worek, 2008) have been developed for the simulation of desiccant and enthalpy wheels.

Although desiccant cooling cycles can provide comfortable room conditions at low energy costs, using environment friendly technologies, and being especially effective at warmer and humid weather, its employment is rarely seen throughout Brazil. This papers presents an analysis of the operation of a desiccant cooling cycle for various outdoor conditions encountered in different Brazilian cities. Despite the existence of different types os desiccant cooling cycles, a ventilation cycle was the main focus of this study. Hence, heat and mass transport through the components of a ventilation cycle were mathematically modeled and computationally solved. With the simulation results, an investigation of the effects of varying parameters related to the operation of a ventilation cycle was carried out.

2. DESCRIPTION OF THE VENTILATION CYCLE

The ventilation cycle is probably the simplest desiccant cycle available and the most reported in studies; it is constituted of a desiccant wheel, a sensible heat exchanger, a heat source and an evaporative cooler, as can be seen in Fig. 1. It is important to observe that this cycle is designed to use 100% of fresh air.

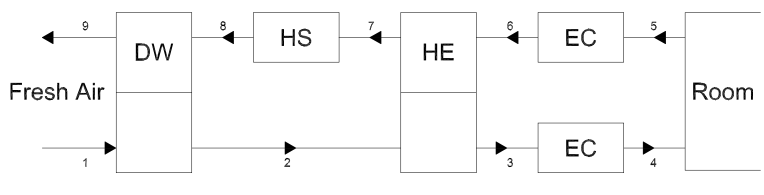

Figure 1. Ventilation cycle schematics.

Figure 2 displays an example of the ventilation cycle in a psychrometric chart. As can be seen, the outdoor air is first dehumidified by the desiccant wheel, becoming hot and dry. It is then cooled by the heat exchanger and evaporative cooler. On the other branch, the return air is first cooled in another evaporative cooler, and then is heated in heat exchanger. Finally, to archive the regeneration temperature, the return air heated again in the heat source.

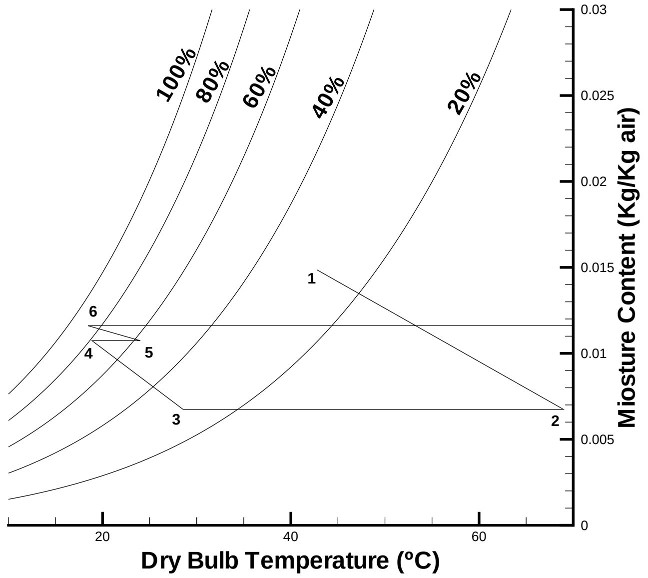

Figure 2. Ventilation cycle on a psychrometric chart.

3. MATHEMATICAL MODELING OF THE VENTILATION CYCLE COMPONENTS

3.1 Desiccant wheel

The desiccant wheel is a type of heat and mass transfer regenerator, employing a high fraction of desiccant material in its matrix and operating at relatively slow rotational speeds. In order to simulate the behavior of the desiccant wheel encountered in the ventilation cycle, the simplified one-dimensional version of the formulation developed in (Sphaier and Worek, 2004; Sphaier, 2007), as discussed in (Sphaier and Worek, 2006), is employed. The normalized governing equations can be written in the following form:

$$\Theta\,\mathrm{V}_\mathrm{r}^*\left((1-\epsilon_f)\,\Omega\frac{\partial W_f^*}{\partial t^*}+\epsilon_f\,\frac{\partial Y_f^*}{\partial t^*}\right)=-\,\mathrm{N}_\mathrm{tu}^\mathsf{m}\,(Y_f^*-Y^*), \tag{1}$$

$$\Theta\,\mathrm{C}_\mathrm{r}^*\,\chi_f\,\frac{\partial T_f^*}{\partial t^*}=-\,\mathrm{N}_\mathrm{tu}^\mathsf{h}\,(T_f^*-T^*)-\mathrm{N}_\mathrm{tu}^\mathsf{h}\,(T_f^*-T^*)\,\frac{\mathrm{C}_\mathrm{r}^*}{\mathrm{V}_\mathrm{r}^*}\,i_{v,\Delta T}^*+\Theta\,\mathrm{C}_\mathrm{r}^*\,\Omega\,(1-\epsilon_f)\,\frac{\partial W_f^*}{\partial t^*}\,i_{sor}^*, \tag{2}$$

$$\tau_{dw}^*\,\frac{\partial Y^*}{\partial t^*}+\frac{\partial Y^*}{\partial x^*}=\mathrm{N}_\mathrm{tu}^\mathsf{m}\,(Y_f^*-Y^*), \tag{3}$$

$$\chi\left(\tau_{dw}^*\,\frac{\partial T^*}{\partial t^*}+\frac{\partial T^*}{\partial x^*}\right)=\mathrm{N}_\mathrm{tu}^\mathsf{h}\,(T_f^*-T^*), \tag{4}$$

where some of the dimensionless parameters can assume different values for each process stream that crosses the desiccant wheel (denoted I and II). Their definitions are given by:

$$\mathrm{N}_\mathrm{tu,I}^\mathsf{h}=\frac{(h^\mathsf{h}A_s)|_\mathrm{I}}{\mathrm{C_I}},\qquad \mathrm{N}_\mathrm{tu,II}^\mathsf{h}=\frac{(h^\mathsf{h}A_s)|_\mathrm{II}}{\mathrm{C_{II}}},\qquad \mathrm{N}_\mathrm{tu,I}^\mathsf{m}=\frac{(h^\mathsf{m}A_s)|_\mathrm{I}}{\mathrm{V_I}},\qquad \mathrm{N}_\mathrm{tu,II}^\mathsf{m}=\frac{(h^\mathsf{m}A_s)|_\mathrm{II}}{\mathrm{V_{II}}}, \tag{5}$$

$$\mathrm{C}^*=\frac{\mathrm{C_{min}}}{\mathrm{C_{max}}}=\frac{\mathrm{V_{min}}}{\mathrm{V_{max}}}=\mathrm{V}^*,\qquad \mathrm{C}_\mathrm{r}^*=\frac{\mathrm{C}_{\mathrm{r},f}}{\mathrm{C_{min}}},\qquad \mathrm{V}_\mathrm{r}^*=\frac{\mathrm{V}_{\mathrm{r},f}}{\mathrm{V_{min}}}, \tag{6}$$

$$\tau_{dw}^*=\frac{\tau_{dw}}{\tau}=\frac{L}{u\,\tau},\qquad \Theta_\mathrm{I}=\frac{\mathrm{C_{min}}}{\mathrm{C_I}}=\frac{\mathrm{V_{min}}}{\mathrm{V_I}},\qquad \Theta_\mathrm{II}=\frac{\mathrm{C_{min}}}{\mathrm{C_{II}}}=\frac{\mathrm{V_{min}}}{\mathrm{V_{II}}}, \tag{7}$$

where,

$$\mathrm{C_I} = (\dot{m}^\star c_p^\star)|_\mathrm{I}, \qquad \mathrm{C_{II}} = (\dot{m}^\star c_p^\star)|_\mathrm{II}, \qquad \mathrm{C_r} = \frac{(m_f^\star c_f^\star)}{\tau_\mathrm{I} + \tau_\mathrm{II}}, \tag{8}$$

$$\mathrm{V_I} = \frac{\mathrm{C_I}}{\rho^\star c_p^\star}, \qquad \mathrm{V_{II}} = \frac{\mathrm{C_{II}}}{\rho^\star c_p^\star}, \qquad \mathrm{V_r} = \frac{\mathrm{C}_{\mathrm{r},f}}{\rho_f^\star c_f^\star}. \tag{9}$$

The effectiveness of the desiccant wheel is defined in terms of its inlet and outlet enthalpies and water vapor concentrations of both airstreams:

$$\varepsilon_\mathsf{m}^{dw} = \frac{\mathrm{C_I}}{\mathrm{C_{min}}} \frac{Y_{\mathrm{I},out} - Y_{\mathrm{I},in}}{Y_{\mathrm{II},in} - Y_{\mathrm{I},in}}, \qquad \varepsilon_\mathsf{i}^{dw} = \frac{\mathrm{C_I}}{\mathrm{C_{min}}} \frac{i_{\mathrm{I},out} - i_{\mathrm{I},in}}{i_{\mathrm{II},in} - i_{\mathrm{I},in}}, \tag{10}$$

3.2 Sensible heat exchanger

The sensible heat transfer regenerator works similarly to the desiccant wheel; however, there is no adsorbent material within the rotary matrix, such that the exchanger can only provide sensible heat transfer between the process streams. Hence, the operation of this device can be simulated as a simplified case of the desiccant wheel. The sensible heat regenerator effectiveness is defined as:

$$\varepsilon^{he} = \frac{\mathrm{C_I}}{\mathrm{C_{min}}} \frac{T_{\mathrm{I},out} - T_{\mathrm{I},in}}{T_{\mathrm{II},in} - T_{\mathrm{I},in}}, \tag{11}$$

3.3 Evaporative cooler

Evaporative coolers use the adiabatic saturation process to cool the airstream. Because of that, energy consumption is low when compared to other air coolers, but the water consumption is higher. Lately, many studies have been conducted to analyze evaporative coolers (Belding and Delmas, 1997; Camargo, D.Ebinuma et al., 2003; Castro and Pimenta, 2004). A simple formulation for the operation of the evaporative cooler within the considered cycle was obtained by applying global energy and mass balances to the evaporative cooler. Figure 3 displays a control volume for this component.

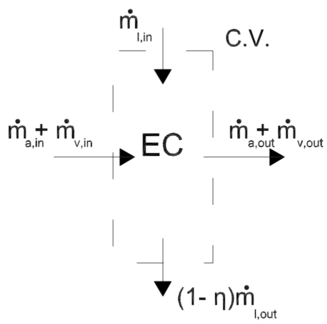

Figure 3. Control volume for evaporative cooler.

The mass and enthalpy conservation balances, yields:

$$\dot{m}_{a,in} = \dot{m}_{a,out} = \dot{m}_a \tag{12}$$

$$\dot{m}_{v,in} + \dot{m}_l = (1-\eta)\dot{m}_l + \dot{m}_{v,out} \tag{13}$$

$$\dot{m}_a\, i_{a,in} + \dot{m}_{v,in}\, i_{v,in} + \dot{m}_l\, i_{l,in} = \dot{m}_a\, i_{a,out} + \dot{m}_{v,out}\, i_{v,out} + (1-\eta)\dot{m}_l\, i_{l,out} \tag{14}$$

where,

$$\dot{m}_v = Y\,\dot{m}_a \tag{15}$$

Approximating the water vapor and dry air as perfect gases with constant proprieties, their specific enthalpies can be written as:

$$i_a = i_{a,\mathrm{ref}} + c_{p,a}(T - T_\mathrm{ref}) \tag{16}$$

$$i_v = i_{v,\mathrm{ref}} + c_{p,v}(T - T_\mathrm{ref}) \tag{17}$$

$$i_l = i_{l,\text{ref}} + c_{p,l}(T - T_{\text{ref}}) \tag{18}$$

Using equations (16,17,18,15) together with the balance equations (12,13,14), yields a relation between the inlet and outlet temperatures and vapor concentrations:

$$Y_{out} \; = \; Y_{in} + \eta \frac{\dot{m}_l}{\dot{m}_a} \tag{19}$$

$$T_{out} \; = \; \frac{T_{in}\left(\dot{m}_a\left(c_{p,a} \; + c_{p,v}\, Y_{in}\right) + \dot{m}_l\, c_{p,l}\right) \; + \; \dot{m}_l\left(i_{l,\text{ref}} - i_{v,\text{ref}} + \left(c_{p,v} - c_{p,l}\right) T_{\text{ref}}\right)\eta}{\dot{m}_a\left(c_{p,a} \; + c_{p,v}\, Y_{in}\right) + \dot{m}_l\left(\left(1-\eta\right) c_{p,l} + c_{p,v}\,\eta\right)} \tag{20}$$

3.4 Heat source

The heat source here is simply modeled as device to warm up the process air. A simple control volume for this device is displayed in Fig. 4.

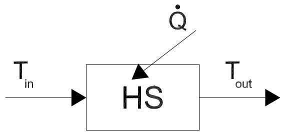

Figure 4. Heat source control volume.

Applying mass and enthalpy conservation balances to the control volume, one obtains:

$$Y_{in} \; = \; Y_{out} \; = \; Y \tag{21}$$

$$\dot{m}_a\, Y_{out}\, i_{v,out} \; + \; \dot{m}_a\, i_{a,out} \; - \; \dot{m}_a\, Y_{in}\, i_{v,in} \; - \; \dot{m}_a\, i_{a,in} \; = \; \dot{Q}_{hs} \tag{22}$$

Combining equations (21) and (22) and substituting the enthalpies yields:

$$T_{out} = \frac{\dot{Q}_{hs}}{\dot{m}_a\left(c_{p,a} \; + \; c_{p,v} Y\right)} - T_{in} \tag{23}$$

4. RESULTS AND DISCUSSION

The overall performance of the cycle is assessed by the coefficient of performance (COP). The COP is a performance rating that can be calculated by dividing the the room and supply air enthalpy difference by the heat source rate per unit of mass flow rate. Hence, according to Fig. 1, the cycle COP can be calculated by:

$$\text{COP} \; = \; \frac{i_5 \; - \; i_4}{i_8 \; - \; i_7} \tag{24}$$

Simulation results of the performance the desiccant ventilation cooling cycle were carried out using outdoor data for five different Brazilian cities, as reported in Tab. 1, obtained from the NBR6401 norm.

Table 1. Dry and wet bulb temperatures for different Brazilian cities.

City	DBT (°C)	WBT(°C)
Manaus (AM)	35.0	29.0
Rio de Janeiro (RJ)	35.0	26.5
São Paulo (SP)	31.0	24.0
Terezinha (PE)	38.0	28.0
Petrolina (PE)	36.0	25.5

The influence of varying effectiveness values for the desiccant wheel and sensible heat exchanger on overall cycle performance (COP) were carried out in two steps. Initially, the effectiveness of the sensible heat exchanger was held at fixed value, and the following input data were considered:

1. mass flow = 1 kg/s.
2. Evaporative cooler effectiveness = 0.6.

3. Regeneration temperature = 130.

4. Room load = 15 kW.

5. Sensible heat exchanger effectiveness = 0.6.

In order to analyze the influence the desiccant wheel's mass and enthalpy transfer effectiveness separately, a constant value for the mass transfer effectiveness ($\varepsilon_{\mathrm{m}} = -0.3$) was chosen and the enthalpy effectiveness was varied. Figure 5 displays the simulation results for this case. As one can observe, the COP increases with increasing desiccant wheel enthalpy effectiveness. The results also demonstrate that higher cycle performance values are obtained for warmer climates. In addition, it can be observed that at the same DBT, the COP values increase with the WBT. This shows how this type of the cycle can perform better at humid climates. Finally, one should mention that the behavior of the COP values followed a trend similar to the one reported by Jain, Dhar et al. (1995).

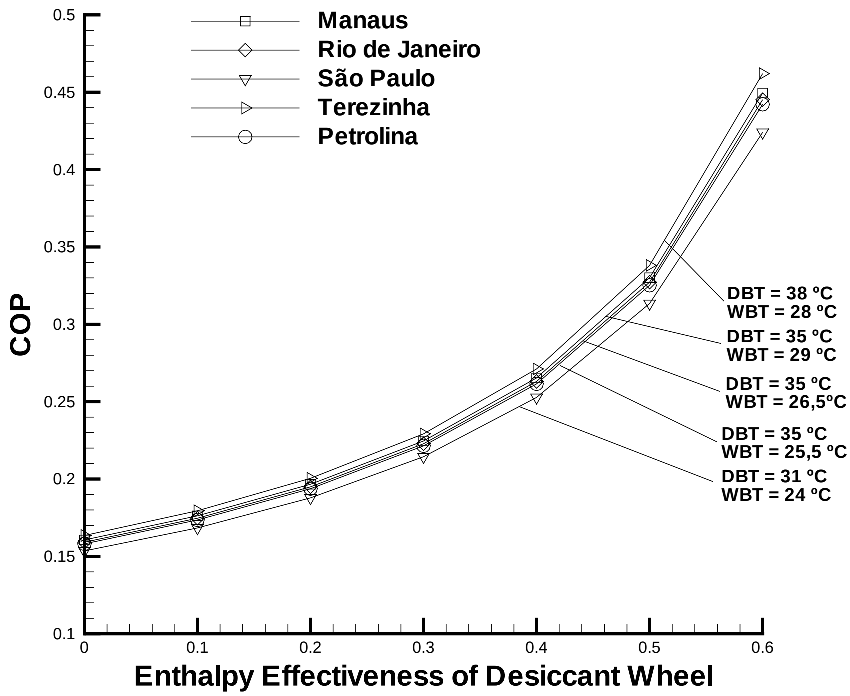

Figure 5. Effect of varying the enthalpy effectiveness of the desiccant wheel.

Then, for investigating the effects of desiccant wheel mass transfer effectiveness on cycle performance, the enthalpy effectiveness was fixed at 0.15, and the all previous considerations were maintained. Figure 6 displays the results for varying the desiccant wheel mass transfer effectiveness from -0.8 to 0.0. Is should be mentioned that mass transfer effectiveness for the desiccant wheel are negative because it is the airstream with higher water content (regeneration stream) that removes moisture from the the lower water content stream. This happens because the regeneration stream is at a sufficiently higher temperature. As one can also observe in Fig. 6, again, the COP increases with outdoor temperature and humidity.

As a second step, the unified formulation for the heat and mass transfer regenerator (1–4) was used to simulate the desiccant wheel and the effectiveness of the sensible heat exchanger was manually varied. Figure 7 displays these results, showing that the cycle COP increases with the component's effectiveness, as expected. In addition, the same behavior of increasing COP with increasing outdoor humidity (WBT) is generally seen. However, it is interesting to note that for the cases with higher DBT and WBT, this behaviors begins to change for higher values of heat exchanger effectiveness.

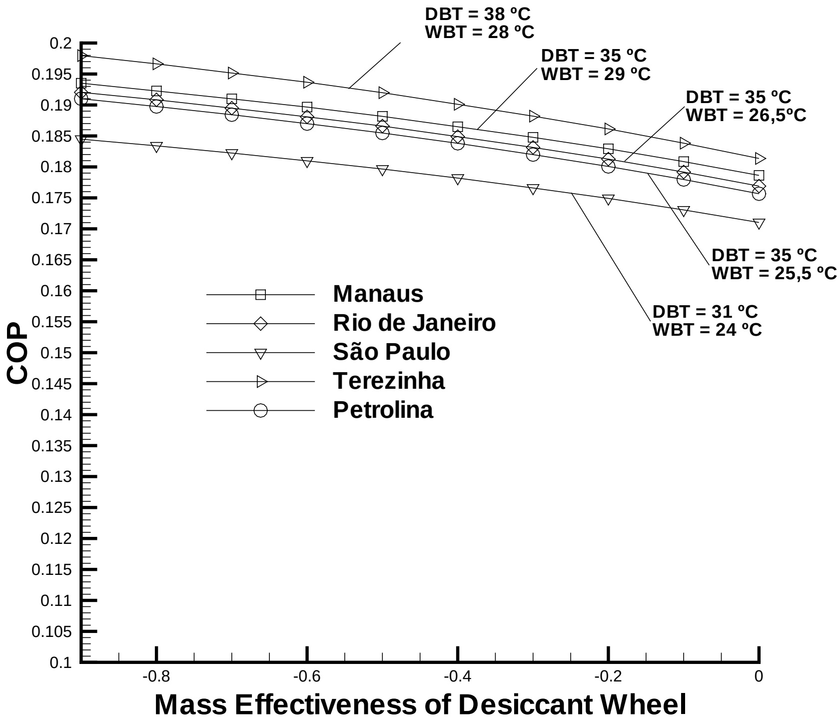

Figure 6. Effect of varying the mass effectiveness of the desiccant wheel

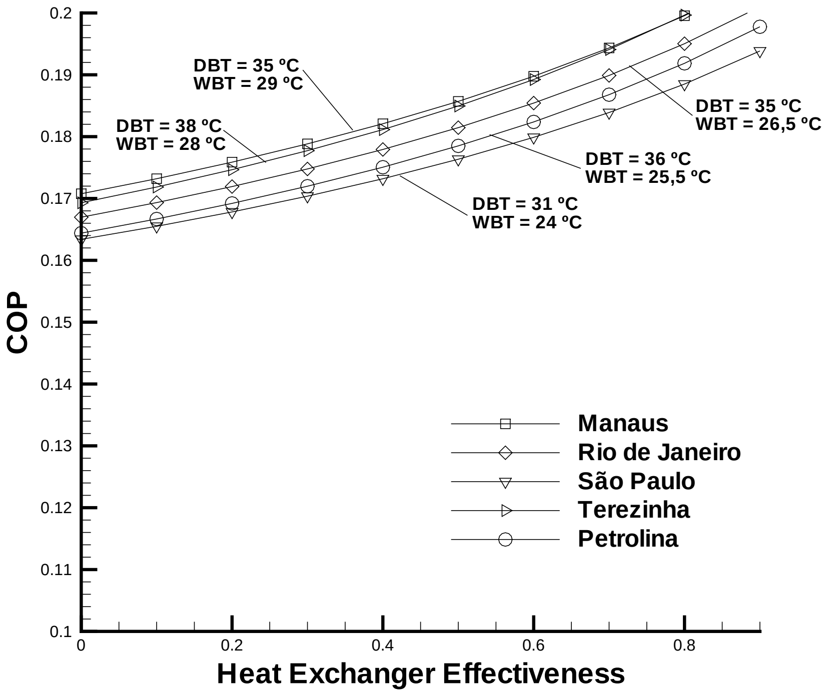

Figure 7. Effect of varying the sensible heat transfer regenerator effectiveness

5. SUMMARY AND CONCLUSIONS

This paper presented a simple method for predicting the performance of a ventilation desiccant cycle. Despite its simplicity, the algorithm invokes the robust numerical solution scheme devised in (Sphaier and Worek, 2008). The methodology was computationally implemented and a series of simulation results were calculated. The effects of varying the performance of the desiccant wheel and heat exchanger (cycle components) was investigated using weather data for five different Brazilian cities. The results show that the overall cycle performance (COP) increases with increasing magnitude of the effectiveness of the components, as expected. It was pointed out that the desiccant wheel mass transfer effectiveness presented negative values because it provided mass transfer from a lower concentration stream to a higher concentration one. Moreover, the results demonstrated that desiccant cycles are more efficient at warmer and moister climates. Nevertheless the obtained values for cycle performance are notably lower than the ones encountered for traditional vapor-compression cycles, thereby indicating a need for future efforts towards enhancing the performance of desiccant-based cycles.

6. ACKNOWLEDGMENTS

The authors would like to acknowledge the financial support provided by, CNPq, FAPERJ, and Universidade Federal Fluminense.

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