ANALYSIS OF DESICCANT VENTILATION AND RECIRCULATION CYCLES UNDER DIFFERENT ENVIRONMENTAL CONDITIONS

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Abstract. *Desiccant cooling systems have been increasingly investigated due to their environmental friendliness. A known characteristic of these type of cycles, is that they can be significantly influenced by outdoor conditions. Hence, the purpose of this study is to investigate, the operation of a desiccant cooling cycle is for various environmental conditions, and to show how these cycles can be adjusted to meet thermal comfort requirements. The analysis is based on a simple numerical procedure for desiccant cooling simulation in which the overall system operation is calculated from individual cycle components' characteristics. With the employed methodology, the conditioned space state is calculated for different environmental conditions and compared to a standard comfort-zone. The results show how the effectiveness of evaporative coolers influence the conditioned space for the ventilation and recirculation cycles.*

Keywords: *dehumidification, evaporative cooling, desiccant wheel, cooling cycle, recirculation*

1. NOMENCLATURE

T	air temperature, °C
Y	air absolute humidity, kg/kg
i	air enthalpy, kJ/(kg °C)
i_{lv}	latent heat of vaporization, kJ/kg
c	specific heat, J/(kg °C)
SHR	sensible heat ratio

Greek Symbols

ϵ	effectiveness
η	parameter of desiccant wheel
Ω	adimensional of refrigerated room

Subscripts

dw	desiccant wheel
hw	heat wheel
$ec1$	supplied air evaporative cooler
$ec2$	exhaust air evaporative cooler
reg	regeneration

2. INTRODUCTION

Desiccant cooling systems Bourdoukan *et al.* (2010) are particularly suitable regarding the environmental impact, due to the absence of refrigerants with ozone depleting properties. Moreover, the use of low-grade waste heat as the primary energy source also characterizes a low global warming potential, when compared to vapor compression systems. Due to the environmental friendliness of these cycles, a number of studies have addressed the dynamic simulation of desiccant cycles for better understanding their operation and improving their design. Panaras *et al.* (2007) proposed a methodology for the definition of the system's achievable working range under specific set of space comfort requirements, simulating a solar desiccant air conditioning system in a typical residential building. Kanoğlu *et al.* (2007) simulated an open desiccant cooling process applied to ventilation and recirculation operation modes, calculating First and Second-Law COPs. Camargo *et al.* (2003) performed a thermo-economic analysis of a desiccant cooling system, and simulations with direct and indirect evaporative coolers were presented. Zhang and Niu (2003) proposed a decoupled cooling system, in which the latent load is handled by a desiccant wheel while a chilled ceiling panel provides the sensible cooling. Panaras *et al.* (2010) proposed a methodology for dimensioning desiccant cooling systems using a simple steady state model. In spite of the relevance of previous studies, some works rely on unrealistic figures for some components, such as both evaporative coolers operating on near 100% efficiencies [3,4] or a room thermal load with no latent heat component [2] (SHR = 0). In a recent study Sphaier and Nóbrega (2012) a simple numerical algorithm for simulating the steady-state operation of desiccant cooling cycles was developed, showing how cycle components' characteristics can affect overall system operation. Both ventilation and recirculation modes were investigated, but the results were focused on a specific environmental condition, and the conditioned space temperature was also fixed at a standard 25°C value.

3. PRESENTATION OF THE DESICCANT CYCLES

Desiccant cooling cycles are usually studied as the Ventilation Cycle also, known as the Pennington Cycle, and the Recirculation Cycle. In this study those cycles are modeled and the components of which include a desiccant wheel (dw), a sensible heat wheel (hw), a thermal source (hs) and two evaporative coolers (ec1, ec2). Fig. 1 displays the working schematic of the ventilation cycles and the Fig. 2 shows the same for the recirculation cycles. There are nine air states to be determined, which amounts to eighteen variables, given in terms of temperature (T) and absolute humidities (Y). The desiccant cooling system performance depends on a number of design conditions, such as the adsorptive material, the evaporative coolers and the heat wheel effectiveness. How will be shown in the results, the outside air conditions are also of importance and it are able to modify the room comfort conditions.

4. MODEL EQUATIONS

The model equations are based on relations defining performance characteristics of each cycle component, together with mass and energy balances. The heat wheel effectiveness and the evaporative cooler's effectiveness are given by:

$$\epsilon_{hw} = \frac{T_2 - T_3}{T_2 - T_6} = \frac{T_6 - T_7}{T_2 - T_6}, \qquad \epsilon_{ec1} = \frac{T_3 - T_4}{T_3 - T_{3,sat}}, \qquad \epsilon_{ec2} = \frac{T_5 - T_6}{T_5 - T_{5,sat}}, \tag{1}$$

in which $T_{3,sat}$ and $T_{5,sat}$ correspond to the adiabatic saturation temperatures associated with states 3 and 5, respectively.

Desiccant wheel simulations can be calculated by solving coupled heat and mass transfer equations Nóbrega and Brum (2011); Sphaier and Worek (2009) However, for prompt cooling cycle simulations, this alternative becomes limited due to the effort associated with the numerical solution of the transport equations involved. An alternative approach is to employ algebraic correlations for determining the outlet states. One such methodology stem from the analogy method developed by Maclaine-Cross and Banks (1972):

$$\eta_1 = \frac{F_1(T_2, Y_2) - F_1(T_1, Y_1)}{F_1(T_8, Y_8) - F_1(T_1, Y_1)}, \qquad \eta_2 = \frac{F_2(T_2, Y_2) - F_2(T_1, Y_1)}{F_2(T_8, Y_8) - F_2(T_1, Y_1)}, \tag{2}$$

where η_1 and η_2 depend on the type of desiccant wheel and the functions F_1 and F_2 are defined in terms of temperature and absolute humidity as:

$$F_1(T, Y) = -\frac{2865}{(T + 273.15)^{1.49}} + 4.344\, Y^{0.8624}, \qquad F_2(T, Y) = \frac{(T + 273.15)^{1.49}}{6360} - 1.127\, Y^{0.07969}. \tag{3}$$

These equations, together with the heat and mass balances across the desiccant wheel, allow the determination of the outlet states once the inlet states are known. Typical values for the η's as employed in Sheridan and Mitchell (1985), are

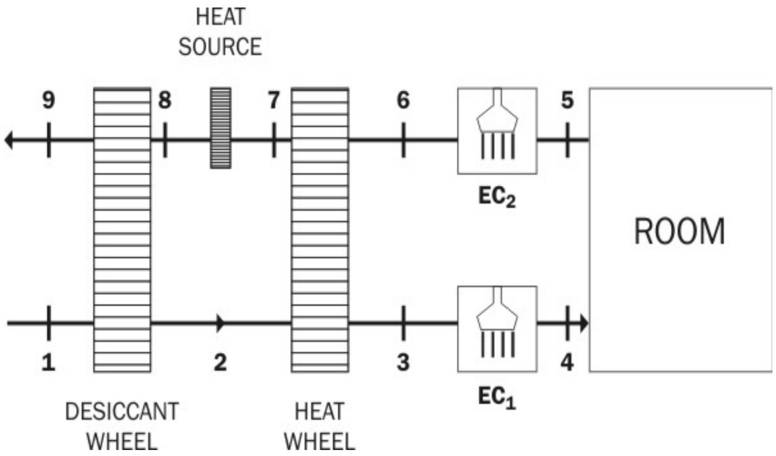

Figure 1. Ventilation cycle.

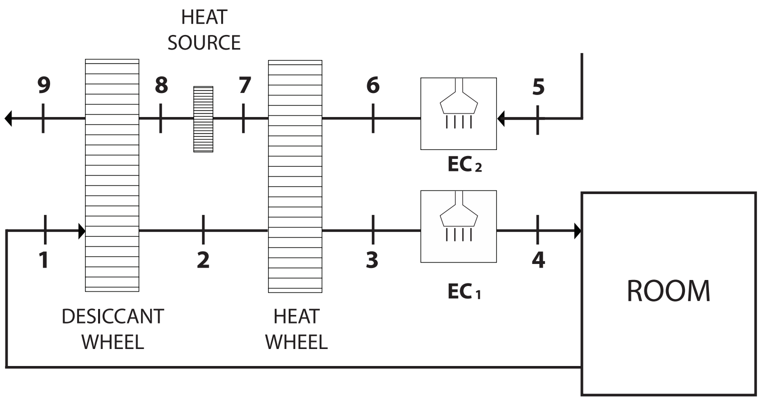

Figure 2. Recirculation cycle.

given below:

- High Performance Dehumidifier (HPD): $\eta_1 = 0.05$, $\eta_2 = 0.95$;
- Low Performance Dehumidifier (LPD): $\eta_1 = 0.07$, $\eta_2 = 0.80$;

The remaining model equations represent mass and energy balances across each component. The first two equations represent energy balances across the evaporative coolers:

$$i_3 = i_4, \qquad i_6 = i_5, \tag{4a}$$

whereas the mass and energy balances applied to the desiccant wheel are given by:

$$i_2 - i_1 = i_8 - i_9, \qquad Y_2 - Y_1 = Y_8 - Y_9. \tag{4b}$$

The mass balances and energy balances across the heat wheel are written as

$$Y_3 = Y_2, \qquad Y_6 = Y_7, \qquad T_6 - T_7 = T_3 - T_2, \tag{4c}$$

and the mass balance across the thermal source yields

$$Y_7 = Y_8. \tag{4d}$$

The last equations represent the process which the air undergoes as it crosses the conditioned space. Since the points are numbered differently for each cycle, two expressions need be used:

$$\mathrm{SHR} = 1 - \frac{i_{lv}\,(Y_5 - Y_4)}{i_5 - i_4}, \qquad (c_{p,a} + Y_4\, c_{p,v})\,(T_5 - T_4) = \Omega\,\frac{R\,T_4}{p_{atm}}\,(T_5 - T_1), \tag{4e}$$

in which SHR is the *sensible heat ratio* and $\Omega = UA/\mathrm{VFR}$ is a parameter that relates the room heat transfer conductance UA with the supply air volumetric flow rate.

The previously described non-linear system of equations is numerically solved using an iterative procedure based on Newton-Raphson's method, with a verification step to check wether the root search is maintained within physically acceptable limits. The solution procedure is explained in details in Sphaier and Nóbrega (2012).

5. RESULTS AND DISCUSSION

The simulation results of the current model are now presented. The calculated data is displayed in terms of isolines of outdoor temperate (green curves) and outdoor relative humidity (orange curves), so that the influence of the outdoor condition on the room condition can be easily observed. Figure 3 presents for the ventilation cycle, the calculated room condition for different inlet conditions, and shows the behavior of varying the exhaust and supply air evaporative cooler. As can be observed, increasing the supply air evaporative cooler effectiveness results in an overall reduction of room temperature followed by an increase in relative humidity. Hence, the value of ϵ_{ec1} can be use as a control parameter to reduce T_5 while increasing ϕ_5. When looking into the effect of varying ϵ_{ec2}, one notices that an increase in its value also results in an overall temperature reduction; nevertheless, the experienced sensible cooling effect is less than that achieve by increasing ϵ_{ec1}. On the other hand, when inspecting the exhaust air evaporative cooler effect on ϕ_5 one notices that, rather than increasing the relative humidity of the condoned space, increasing ϵ_{ec2} apparently has no effect on ϕ_5.

Next, figure 4 presents the same graphics structure and also keeping the parameters variation, however the recirculation cycles are studied at this time. The aim is evaluate as the air recirculation could affect the room condition for different inlet conditions. Firstly, it can be observed that the area of the polygon generated by varying conditions of inlet air is lower than in the ventilation cycle. This was expected, once the external air makes contact the room's air only in the heat wheel. Therewith, the outdoor conditions have a lower influence in the overall cycle. As can be observed again, increasing the supplied air evaporative cooler effectiveness results in an strong overall reduction of room temperature followed by an increase in relative humidity. The low temperatures that were achieved is due to the complete air that entering in the desiccant wheel comes from the room. The value of ϵ_{ec1} still can be use as a control parameter to reduce T_5 while increasing ϕ_5, further, the sensible cooling effect of ϵ_{ec2} is less than that achieve by increasing ϵ_{ec1} too.

6. CONCLUSIONS

This paper presented simulation results of a desiccant cooling cycle operating under various environmental conditions. The results showed how one can control the conditioned space temperature and humidity levels by varying the evaporative coolers' effectiveness and the regeneration temperature. As can be seen, the recirculation cycle produces overall temperatures lower, as well as, the value of ϵ_{ec1} is able to use as a control parameter to modify the temperature of the room. Although, the variation of the value of ϵ_{ec2} didn't have an important effect in the two cycles that this study have presenting. As was proposed in the early, the desiccant cooling cycles studied are significantly influenced by outdoor conditions.

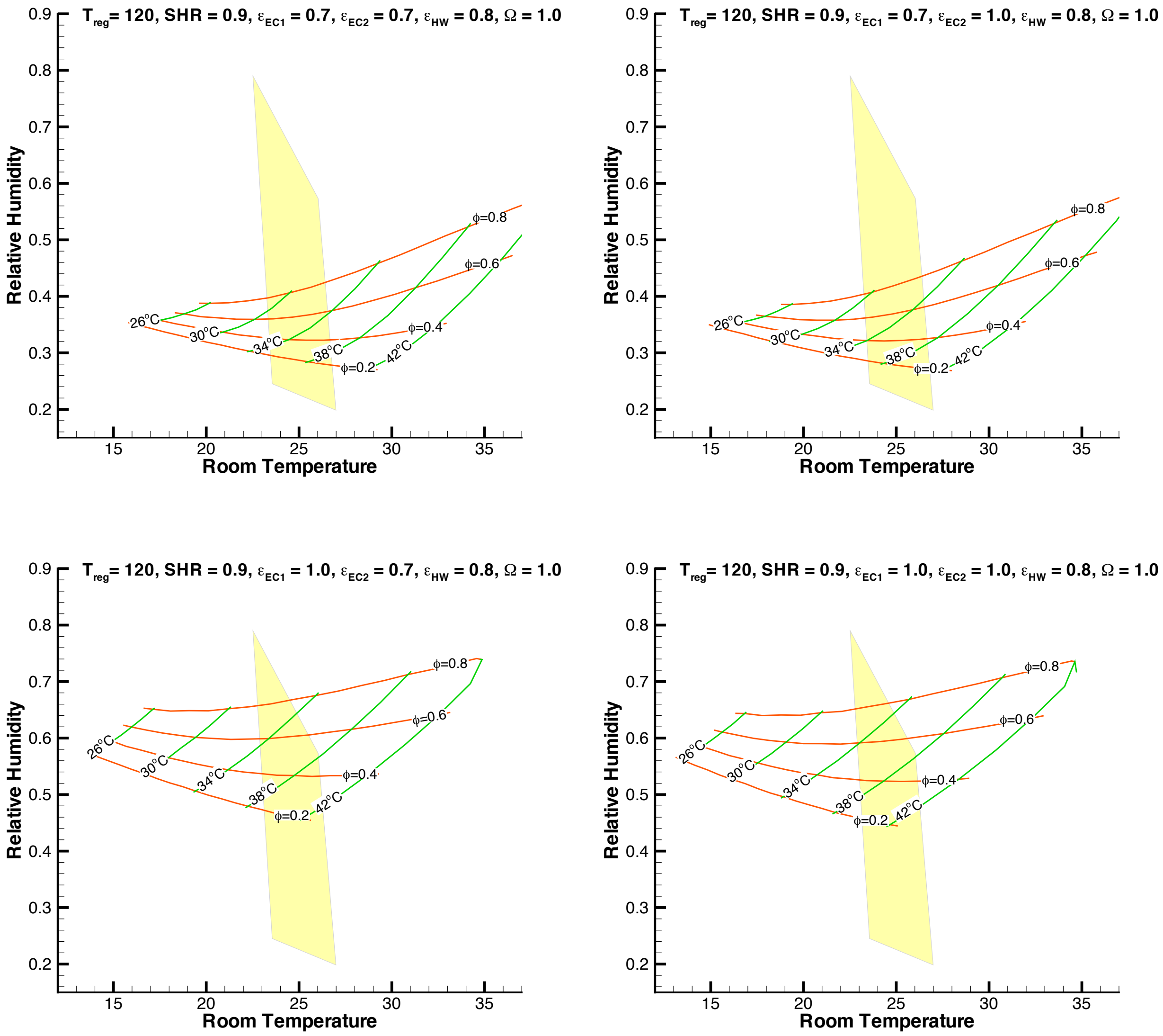

Figure 3. Calculated room condition for different environment conditions in Ventilation Cycles: effect of varying the evaporative coolers' effectiveness.

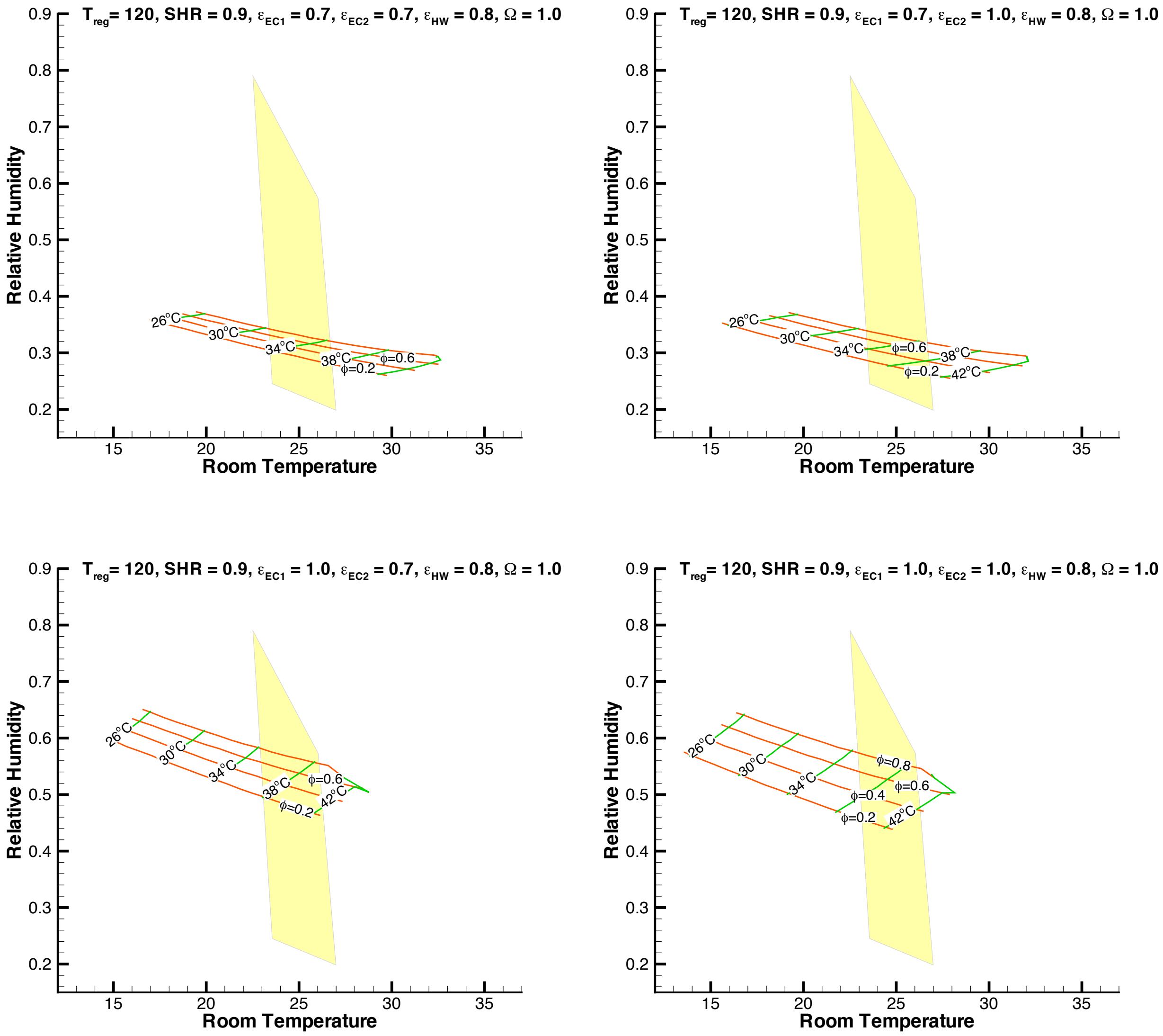

Figure 4. Calculated room condition for different environment conditions in Ventilation Cycles: effect of varying the evaporative coolers' effectiveness.

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