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THE DESIGN OF DEHUMIDIFIERS FOR USE IN DESICCANT COOLING AND DEHUMIDIFICATION SYSTEMS

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ABSTRACT

The use of rotary dehumidifiers in open cycle desiccant cooling systems is investigated by analyzing the performance of the rotary heat exchanger-rotary dehumidifier subsystem. For a given cooling load, the required regeneration heat supply can be minimized by choosing appropriate values for the regeneration air mass flow rate and the wheel rotation speed. A map is presented showing optimal values for rotational speed and regeneration flow rate as a function of the regeneration air inlet temperature and the process air inlet humidity ratio. This regeneration temperature is further optimized as a function of the process humidity ratio. In the analysis, the control strategy adjusts the process air mass flow rate to provide the required cooling load. Additional control options are considered and the sensitivity of the regeneration heat required to the wheel speed, regeneration air mass flow rate and inlet temperature is discussed. Experimental data reported in literature are compared with analytical results and indicate good agreement.

NOMENCLATURE

A_j	total heat transfer area of the dehumidifier matrix in period j [area]
COP	coefficient of performance
c_p	moist air specific heat [energy/(dry mass-temp)]
h	heat transfer coefficient [energy/(area-time-temp)]
h_w	mass transfer coefficient [mass/(area-time)]
i	moist air enthalpy [energy/dry mass]
i_{fg}	heat of vaporization [energy/mass]
i_{wv}	water vapor enthalpy [energy/mass]
I	desiccant matrix enthalpy [energy/dry mass]
L	axial flow length through the matrix [length]
Le	NTU_t/NTU_w, overall Lewis number [dimensionless]
$\dot{m}$	mass flow rate [mass/time]
M_d	mass of desiccant in the dehumidifier matrix [mass]
M_f	mass of air in the dehumidifier matrix [mass]
NTU_t	$hA/\dot{m}c_p$, overall number of transfer units for heat transfer [dimensionless]
NTU_w	$h_wA/\dot{m}$, overall number of transfer units for mass transfer [dimensionless]
Q	thermal energy supply [energy]
RPM	dehumidifier wheel revolution speed [$time^{-1}$]
t	temperature
T	time required for a complete rotation of the matrix [time]
w	moist air humidity ratio [dimensionless]
W	matrix water content [dimensionless]
x	axial coordinate measured from period entrance [dimensionless]
z	axial displacement through matrix measured from period entrance [length]
Γ_j	jth operating parameter of the rotary dehumidifier defined in Eq. (1) [dimensionless]
Δ	difference
ε_h	enthalpy effectiveness of the rotary dehumidifier [dimensionless]
ε_w	humidity ratio effectiveness [dimensionless]
θ	time
θ_j	duration of period j [time]
$\dagger$	time coordinate [dimensionless]
τ_{dj}	dwell time of a fluid particle in period j [time]

Subscripts

d	desiccant
f	evaluated at fluid state
id	ideal outlet state
j	period index
load	load
m	evaluated at, or in equilibrium with, the matrix state
min	minimum
opt	optimal
reg	regeneration
room	evaluated at room air state
sys	system
t	heat transfer or temperature

w mass transfer or moisture
wv water vapor

Superscripts
• rate
- average value for a period

INTRODUCTION

Desiccant air conditioning and industrial drying systems using a rotary dehumidifier have been proposed as an alternative to conventional vapor compression units. Various systems for commercial and residential applications have been studied with respect to energy consumption and system performance [1-12,14,15]. Prototype units have been built and tested by AiResearch Manufacturing Company [9,23], the Institute of Gas Technology [10], and Exxon. Currently, both DOE and the Gas Research Institute are supporting further development of these systems.

In desiccant air conditioning systems, air is dried by passing it over the desiccant and the heat of sorption is removed by sensible cooling. The air is further cooled by adiabatic humidification and is directed into the residence as cool dry air. An overview of the various proposed and tested cycles is given in [3]. Parameters influencing system performance are regeneration air inlet temperature, wheel revolution speed, desiccant mass and the ratio of regeneration air to process air mass flow rates. An optimal choice of these parameters will reduce the regeneration heat required for a given cooling load. Minimizing both air stream mass flow rates may also reduce the electrical fan power. Jurinak [3] analytically investigated the influence of regeneration air inlet temperature, wheel revolution speed and mass flow rate ratio on the performance of entire open cycle solid desiccant cooling systems. Values for the capacitance rate parameters Γ_1 and Γ_1/Γ_2 as defined by Eq. (1) of 0.15 and 0.60, respectively, for the recirculation mode, and values of 0.15 and 0.80 respectively for the ventilation cycle are recommended. The COP based on thermal energy input shows a maximum between 65°C and 85°C regeneration temperature, while the COP based on electrical energy input, which includes parasitic power, is maximal at 105°C. Ingram and Vliet [17] present performance charts for a solid desiccant rotary dryer. These charts show the process outlet state of the dehumidifier as a function of the inlet conditions for a set of optimal design parameters, including a dimensionless wheel period and number of transfer units. The optimization was performed only with respect to minimal outlet humidity ratio of the process air stream, and did not consider regeneration energy.

In this paper, the optimization of dehumidifier operation for minimum energy use is determined, and the sensitivity of performance to deviations from optimum operation is evaluated. Results of the optimization analysis are compared to available experimental data and are in good agreement.

ROTARY DEHUMIDIFIER MODEL

Analytical models for the flow of moist air through packed beds of desiccant material have been developed based on the governing equations for heat and mass. Comparisons have been made between analytical solutions and experimentally obtained breakthrough curves [16,24-26,29,30]. The agreement between theory and experiment is satisfactory. Rotary dehumidifiers have been modeled analytically [13,18,19], and numerically [9,13,24]. Model predictions are compared to experimental data by Ball et al. [17], Pla Barby [24], and Rousseau [23]. It is found that the various models are satisfactory and allow acceptable accuracy in making dehumidifier performance predictions.

The analogy theory, introduced by Banks et al. [18,13], relates the performance of a rotary heat and mass exchanger to a superposition of two analogous thermal regenerators. Breakthrough curves based on this theory are given by Close [29,30], and are compared to experimental data obtained by Close [29], and Bullock et al. [31]. The analogy method gives good agreement compared to tests on a silica gel bed which is subjected to step changes in the entering fluid state. Ball et al. [27] compared experimental data for a silica gel rotary dehumidifier to predictions by Nelson's model [28], which is also based on the analogy theory. Typical differences between experimental and model predicted values of the process exit temperature are less than 2 °C, and humidity ratio exit differences are less than 1 g/kg. Later, Maclaine-cross developed MOSHMX [13], a computer code that numerically solves the heat and mass equations for modeling rotary dehumidifiers. The code is superior in accuracy to the analogy-theory based models, but is obtained at high computational effort. Recently, Van den Bulck [20] developed an effectiveness approach following that for regenerators which has accuracy similar to MOSHMX, but involves far less computation. This model is used for the analysis presented in this paper, and its concepts are briefly outlined below.

The nomenclature and coordinate system for the rotary dehumidifier are illustrated in Fig. 1. The adsorbent matrix is arranged as a rotating cylindrical wheel of length L and has a total mass of dry desiccant M_d. Two air streams are blown in counterflow through the regenerator. The process air stream has a low temperature and high relative humidity while the regeneration air stream has a high temperature and low relative humidity. For each period, the axial coordinate, z, is defined as positive in the fluid flow direction, while the rotary position is indicated by the time coordinate θ.

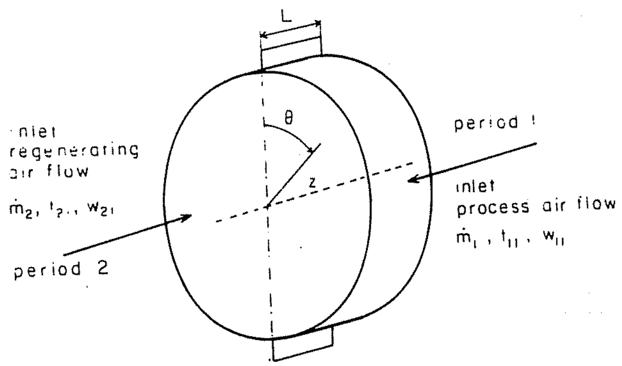

FIGURE 1 Nomenclature and coordinate system for the rotary dehumidifier.

The model which describes the exchange of heat and mass between the moist air and the desiccant matrix is based on the following conventional assumptions for this situation [3,13]:

1. The matrix is modeled as being of parallel passage form, consisting of a homogeneous solid with constant matrix characteristics and porosity, through which an air-water vapor mixture flows with constant velocity. Pressure drop effects through the bed are small with respect to absolute pressure [23], and are neglected.
2. The state properties of the air streams are spatially uniform at the inlet of each period. There is no mixing or carry-over of streams.
3. A one dimensional approach is applied. There is no radial variation of fluid or matrix states, and diffusion fluxes of heat and mass due to tangential gradients of matrix and air state properties are neglected.
4. The axial heat conduction and water vapor diffusion flux are negligible in both the matrix and the air streams.
5. The thermal and moisture capacities of the air entrained in the matrix is negligible compared to the matrix capacities.
6. Transport of water vapor occurs only through ordinary diffusion and transport of heat occurs only through ordinary heat conduction. Flux coupling is neglected.
7. The heat and mass transfer processes between the desiccant matrix and the air stream can be described by lumped transfer coefficients.
8. The steady state performance only of the dehumidifier is considered.

   The capacitance rate parameters, Γ_1 and Γ_2, are defined as the ratio of matrix to fluid mass capacity rate

$$\Gamma_j = \frac{M_{dj}\ \tau_{dj}}{\theta_j\ M_{fj}} = \frac{M_d}{T\ \dot{m}_j} \qquad j = 1,2 \tag{1}$$

and the following dimensionless coordinates are introduced

$$x = \frac{z}{L} \qquad ; \quad 0 \leq x \leq 1$$

$$\tau = \frac{\theta}{\tau_{dj}}\frac{M_{fj}}{M_{dj}} = \frac{\theta}{\theta_j}\frac{1}{\Gamma_j} \quad ; \quad 0 \leq \tau \leq \frac{1}{\Gamma_j} \tag{2}$$

Under the assumptions 1 to 8, the conservation and transfer rate equations for period j of the heat and mass regenerator have been written as [13]:

$$\frac{\partial w_f}{\partial x} + \frac{\partial W_m}{\partial \tau} = 0$$

$$\frac{\partial w_f}{\partial x} = NTU_{w,j}\ (w_m - w_f)$$

$$\frac{\partial i_f}{\partial x} + \frac{\partial I_m}{\partial \tau} \tag{3}$$

$$\frac{\partial i_f}{\partial x} = NTU_{t,j}\ \frac{\partial i_f}{\partial t_f}\ (t_m - t_f) + i_{wv}\ NTU_{w,j}\ (w_m - w_f)$$

Equations (3) are coupled through the thermodynamic property relationships for the desiccant-air-water vapor mixture. Property relations for silica gel are obtained from the literature [3].

$$w_m = w_m(W_m, t_m)$$
$$i_m = i_m(w_m, t_m) \tag{4}$$
$$I_m = I_m(W_m, t_m)$$

The initial conditions for this system of equations are:

$$w_f(x=0,\tau) = w_{j1}$$
$$0 \leq \tau \leq \frac{1}{\Gamma_j}\ ;\ j = 1,2 \tag{5}$$
$$i_f(x=0,\tau) = i_{j1}$$

The periodic equilibrium boundary conditions for the matrix state properties are:

$$\lim_{\tau_1 \to (1/\Gamma_1)^-} W_m(x,\tau_1) = \lim_{\tau_2 \to 0^+} W_m(1-x,\tau_2)$$
$$\lim_{\tau_1 \to (1/\Gamma_1)^-} I_m(x,\tau_1) = \lim_{\tau_2 \to 0^+} I_m(1-x,\tau_2)$$
$$0 \leq x \leq 1\ ;\quad j = 1,2 \tag{6}$$

$$\lim_{\tau_1 \to 0^+} W_m(x,\tau_1) = \lim_{\tau_2 \to (1/\Gamma_2)^-} W_m(1-x,\tau_2)$$
$$\lim_{\tau_1 \to 0^+} I_m(x,\tau_1) = \lim_{\tau_2 \to (1/\Gamma_2)^+} I_m(1-x,\tau_2)$$

Numerical solutions have been obtained for these equations by a number of authors [9,13,24]. There are not methods available for correlating these results in terms of nondimensional parameters. Such a method is presented in the next section.

The ε-NTU METHOD FOR ROTARY DEHUMIDIFIERS

Two state properties of the moist air are required to fully characterize the process outlet state of a dehumidifier. These properties may be obtained by using a conventional effectiveness approach, in which the dehumidifier is compared to a corresponding dehumidifier with infinite transfer coefficients. Two effectiveness factors for the process outlet state are needed. The effectiveness for humidity ratio is defined as

$$\varepsilon_w = \frac{w_{11} - \overline{w}_{12}}{w_{11} - (\overline{w}_{12})_{id}} \tag{7.a}$$

and for enthalpy

$$\varepsilon_h = \frac{\overline{i}_{12} - i_{11}}{(\overline{i}_{12})_{id} - i_{11}} \tag{7.b}$$

where the subscript id indicates the outlet state of a dehumidifier operating at the same inlet conditions, the same Γ_j parameters, and with infinite overall transfer coefficients for mass and heat. Equations (3), (5) and (6) show that the effectiveness are function of inlet temperature and humidity ratio, capacitance rate parameters Γ_j, transfer parameters $NTU_{t,j}$ and the Lewis number Le. Effectiveness expressions for a nominal silica gel rotary dehumidifier are presented in Ref. [20]. These expressions are obtained by combining the solutions for the ideal dehumidifier with values from a numerical analysis of a dehumidifier with finite transfer coefficients [13].

The Ideal Rotary Dehumidifier

In the ideal dehumidifier, the overall heat and mass transfer coefficients are infinite. Thus at all times, each differential desiccant-moist air subsystem is in complete thermodynamic equilibrium (i.e., thermal and vapor pressure equilibrium).

The conservation equations (3) may then be expressed as

$$\frac{\partial w_m}{\partial x} + \frac{\partial W_m}{\partial \tau} = 0$$
$$\frac{\partial i_m}{\partial x} + \frac{\partial I_m}{\partial \tau} = 0 \qquad (8)$$

Equations (8) combined with the property relationships (4) and the initial and boundary conditions (5), (6), form a system of two coupled conservation laws. Each is a hyperbolic partial differential equation, and is non-linear because of the non-linear property relationships. Solutions may be obtained by the method of characteristics and the shock wave method [21]. These methods provide a set of analytical equations that allow prediction of the performance of an ideal dehumidifier for the entire range of operating parameters, Γ_j, and for any inlet conditions. The functional form of the equations is presented in Ref. [20].

DESIGN PARAMETER OPTIMIZATION FOR DEHUMIDIFIERS IN DESICCANT COOLING SYSTEMS

The analysis will focus on the dehumidifier-regenerator sub-system that is common to both ventilation and recirculation cycles, and is shown schematically in Fig. 2. The independent variables are the dehumidifier inlet conditions, t_{j1}, w_{j1}, the mass flow rates in both periods, and the wheel rotational speed, RPM.

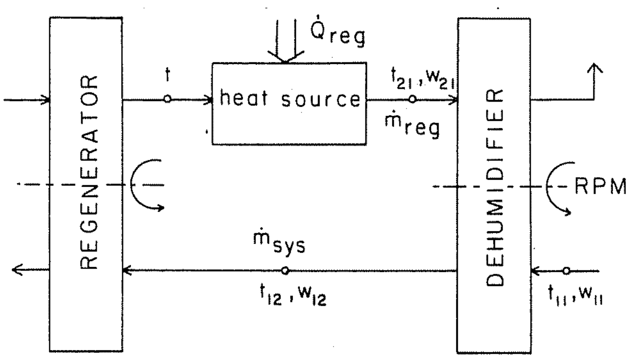

FIGURE 2 Dehumidifier-regenerator subsystem of a desiccant cooling system.

The regeneration heat supply rate, $\dot{Q}_{reg}$, is given by

$$\dot{Q}_{reg} = \dot{m}_{reg} c_p (t_{21} - t) \qquad (9)$$

Introducing Δt as the temperature difference between the air leaving and entering the regenerator, t may be expressed as

$$t = t_{12} - \Delta t \qquad (10)$$

In these cycles the variation of Δt with inlet conditions and mass flow rates through the heat exchanger is small compared to the temperature increase provided by the heat source. It is assumed that Δt is a constant equal to 4°C. This temperature difference corresponds to a heat exchanger effectiveness of 90%, which is typical of a high performance rotary regenerator [32].

Using the parameters Γ_1 and Γ_2, the regeneration flow rate may be expressed by

$$\dot{m}_{reg} = \frac{\Gamma_1}{\Gamma_2} \dot{m}_{sys} \qquad (11)$$

Applying an energy balance on the conditioned space yields the process flow rate required to meet the load as

$$\dot{m}_{sys} = \frac{\dot{Q}_{load}}{\Delta i} \qquad (12)$$

where Δi is the difference in enthalpy between the processed air entering the room and the room air. The temperature of the process air at the exhaust of the heat exchanger is assumed to be equal to the room temperature. Tests on experimental open cycle desiccant cooling systems have shown this to be a very good approximation for both the ventilation cycle [9], and the recirculation cycle [23]. Equation (12) may thus be written as

$$\dot{m}_{sys} = \frac{\dot{Q}_{load}}{i_{fg}(w_{room} - w_{12})} \qquad (13)$$

Substituting Eqs. (10), (11) and (13) into (9), the regeneration heat supply may be expressed as

$$\dot{Q}_{reg} = \frac{c_p}{i_{fg}} \frac{\Gamma_1}{\Gamma_2} \frac{(t_{21} - t_{12} + \Delta t)}{(w_{room} - w_{12})} \dot{Q}_{load} \qquad (14)$$

The dehumidifier transfer parameters are taken to be a $NTU_{t,1}$ of 15 which is typical of a high performance regenerator [32], and a Lewis number of unity. The regeneration period NTU is given by

$$NTU_{t,2} = NTU_{t,1} \frac{\Gamma_2}{\Gamma_1} \qquad (15)$$

These values of transfer coefficients imply that the resistances for heat and mass transfer between the air stream and the surface of the desiccant particles dominates the overall transfer process and that the flow of the air through the matrix is laminar.

The cooling load, $\dot{Q}_{load}$, the process air inlet state (t_{11}, w_{11}), the regeneration air inlet humidity ratio, w_{21}, and the room air state, w_{room}, are taken as specified. The regeneration heat required, given by Eq. (14), may then be minimized by choosing optimal values for regeneration temperature, t_{21}, and capacitance rate parameters, Γ_j. The cooling load is held constant and w_{room} is set equal to the ARI standard room air humidity ratio of 0.0111 kg/kg. The results of an analysis based on this model [20] show that the regeneration heat required is less sensitive to Γ_1 than to Γ_1/Γ_2. These results also show that the influence of t_{11} and w_{21} on the optimal value of Γ_1/Γ_2 is of second order compared to the influence of w_{11} and t_{21}. The optimal values of Γ_1 and Γ_1/Γ_2 may therefore be averaged with respect to t_{11} and w_{21} and correlated as a function only of w_{11} and t_{21}. These averaged optimal values are shown in Fig. 3 for specified process air inlet humidity ratio and regeneration air inlet temperature. This figure establishes the optimum operating conditions, Γ_1, Γ_2, for given process humidity and regeneration temperature.

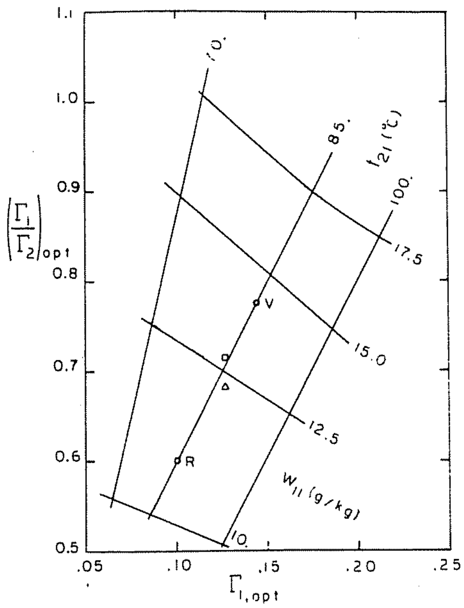

FIGURE 3 Regeneration temperature and process humidity in a $\Gamma_{1,opt}$ - $(\Gamma_1/\Gamma_2)_{opt}$ chart. Δ: Airesearch experimental [23]; □ : model predicted.

The results in Fig. 3 demonstrate that an increase in process humidity w_{11} for a given regeneration temperature causes both $\Gamma_{1,opt}$ and $(\Gamma_1/\Gamma_2)_{opt}$ to increase. To extract more moisture from the process air stream as w_{11} increases, the matrix should be operated at a higher regeneration mass flow rate, contain more desiccant material, and/or rotate at a higher speed. Increasing the regeneration temperature t_{21} for a given process humidity ratio causes $\Gamma_{1,opt}$ to increase but $(\Gamma_1/\Gamma_2)_{opt}$ decreases. The process outlet humidity ratio will decrease and hence also the process mass flow rate. Therefore, a lower regeneration flow rate is required. The optimal amount of adsorbent in the dehumidifier or its revolution speed might increase or decrease depending on the inlet conditions.

Points marked with V and R in Fig. 3 show typical states for the ventilation and recirculation cycles, respectively. The values for $(\Gamma_1/\Gamma_2)_{opt}$ for these points correspond to the values reported by Jurinak [3], which were obtained by a complete detailed system analysis.

If the dehumidifier is operated at the optimal values of Γ_1 and Γ_2 for the given inlet conditions, the outlet humidity ratio of the process air stream equals the minimum obtainable value for these inlet conditions [20]. Also, the regeneration mass flow rate is the minimum flow rate that can still regenerate the matrix without decreasing the cooling capacity of the system. Minimizing the regeneration heat supply for fixed inlet conditions corresponds to minimizing the regeneration mass flow rate. Hence, choosing the right values for Γ_1 and Γ_2 yields another advantage, it minimizes both mass flow rates and therefore the electrical fan power while producing maximum dehumidification.

For a "tuned" dehumidifier, which has optimal values for Γ_1 and Γ_2, the regeneration heat supply may be correlated versus t_{11}, w_{11}, t_{21} and w_{21}. Figure 4 gives $\dot{Q}_{reg}$ as a function of the regeneration temperature and process humidity ratio for given process temperature and regeneration humidity ratio. $\dot{Q}_{reg}$ is normalized with respect to the minimum regeneration heat required for a process humidity of 0.010 kg/kg and a regeneration temperature of 85°C. If the dehumidifier is used in the recirculation cycle mode, the value of w_{11} is in the range 0.010 - 0.013 kg/kg. Figure 4 shows that the regeneration heat supply is only a weak function of the regeneration temperature in this humidity range. Thus, the COP of the recirculation cycle mode based on thermal energy input is not sensitive to an optimal choice of the regeneration temperature for optimal choices for Γ_1 and Γ_2. However, if the dehumidifier is used in the ventilation cycle mode, higher process humidity ratios (0.015-0.0175) result and a high regeneration temperature is recommended. Previous analytical studies have shown the COP based on thermal energy input to be maximal for t_{21} between 65°C and 85°C, but these studies were performed either assuming balanced flow [16], or fixed values for Γ_1 and Γ_2 [3]. Also, these studies do not presume a fixed total cooling load, as is done here, but instead presume a fixed process mass flow rate.

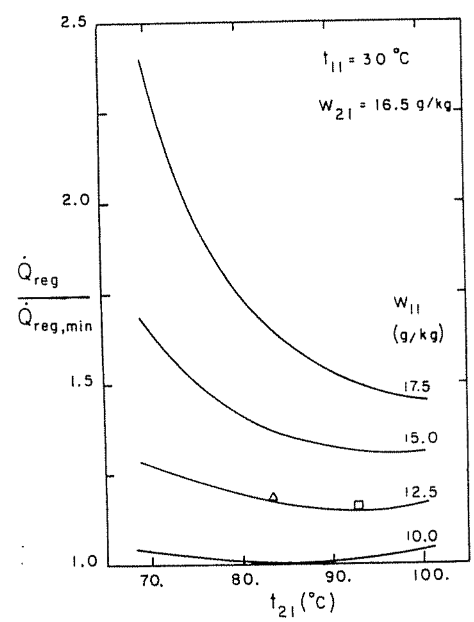

FIGURE 4 Normalized regeneration heat required as a function of regeneration temperature for a tuned dehumidifier. Δ: AiResearch experimental [23]; □ : model predicted.

Increasing the regeneration temperature will lower the required process mass flow rate because the minimum obtainable process outlet humidity ratio decreases. Increasing this temperature also allows the dehumidifier to be operated at a lower value for $(\Gamma_1/\Gamma_2)_{opt}$, which decreases the regeneration mass flow rate even further. The parasitic power of desiccant cooling systems will therefore decrease with increasing regeneration temperature [3,16]. Thus, to maximize system performance, the regeneration temperature should be high. All experimental prototype air conditioning systems using a silica gel dehumidifier operate at a regeneration temperature between 80°C and 100°C [7,9,10]. The selection of operation temperatures were arrived at experimentally, and are consistent with the analysis presented here.

SENSITIVITY ANALYSIS

The optimal values for the regeneration temperature, regeneration mass flow rate and wheel speed are presented in Figs. 3 and 4. If in operation the values for these parameters are different from their respective optimal values, the required regeneration heat supply will increase. This increase will be a function of the extent of the deviation, and also the strategy employed to control the dehumidifier system. Control options on the wheel speed might be to hold the speed constant, vary the speed linearly with the process mass flow rate or the regeneration mass flow rate. Control options on the regeneration mass flow rate might be to hold this flow rate constant, or vary the regeneration flow rate linearly with the process mass flow rate. In the control strategy, the regeneration temperature is kept constant unless mentioned otherwise.

A sensitivity analysis was carried out for two typical dehumidifier parameter sets, reflecting the use of the dehumidifier in a ventilation and recirculation cycle system [20]. The parameters listed in Table 1 are based on the ARI standard room and ambient states of (26.7°C, 0.0111 kg/kg) and (35.0°C, 0.0142 kg/kg) as specified by the Solar Energy Research Institute for testing desiccant air conditioners [22]. The results are summarized below.

TABLE 1 Dehumidifier Parameters for Sensitivity Analysis.

VENTILATION CYCLE	RECIRCULATION CYCLE
t_{11} = 35.0 °C	t_{11} = 26.7 °C
w_{11} = 0.0142 kg/kg	w_{11} = 0.0111 kg/kg
t_{21} = 85.0 °C	t_{21} = 85.0 °C
w_{21} = 0.0140 kg/kg	w_{21} = 0.0190 kg/kg
$\Gamma_{1,opt}$ = 0.170	$\Gamma_{1,opt}$ = 0.093
$\left(\frac{\Gamma_1}{\Gamma_2}\right)_{opt}$ = 0.823	$\left(\frac{\Gamma_1}{\Gamma_2}\right)_{opt}$ = 0.600

1. Effect of Regeneration Air Mass Flow Rate

If the regeneration mass flow rate is increased above the minimal value required to provide the given cooling load, the regeneration heat supply has to increase. Figure 5 shows the relative increase of $\dot{Q}_{reg}$ as a function of the relative increase of the regeneration flow rate for the recirculation and ventilation cycle, and for various control strategies.

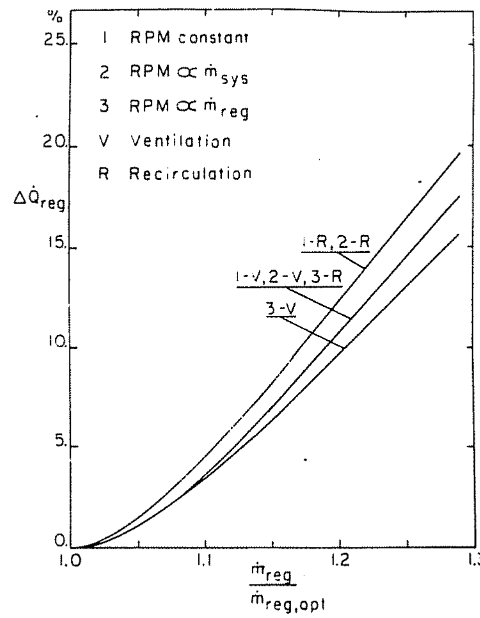

FIGURE 5 Influence of regeneration flow rate on regeneration heat required.

The results in Fig. 5 show only minor differences between the ventilation and recirculation cycle systems. The influence of the different control strategy options is small. Increasing the regeneration flow rate to 20% above the minimal value required causes an increase of about 10% in required regeneration heat supply for the same cooling load. Hence, considerable energy savings may be accomplished by tuning the regeneration mass flow rate to its proper value. This is the minimal value required for the given cooling load and inlet conditions.

2. Wheel Revolution Speed

The sensitivity of the dehumidifier performance to deviations of the wheel rotation speed from its optimal value is presented in Fig. 6. Various control strategies for the regeneration mass flow rates are examined for constant regeneration temperature. The results show that by keeping the regeneration mass flow rate proportional to the process mass flow rate (2-V and 2-R), the regeneration heat supply is the least influenced by wheel speed perturbations. In this case, the wheel speed may vary from -20% to +30% without significantly affecting $\dot{Q}_{reg}$. Positive wheel speed deviations are less influential than negative deviations. The dehumidifier performance is, in general, slightly more sensitive to the wheel revolution speed for the ventilation cycle than for the recirculation cycle.

3. Effect of Regeneration Air Inlet Temperature

In Fig. 3 it is shown that $\Gamma_{1,opt}$ and $\Gamma_{2,opt}$ are primarily a function of the regeneration temperature for a given process humidity ratio. Moreover it is shown in Fig. 4 that if Γ_1 and Γ_2 are controlled in response to the regeneration temperature, the required regeneration heat supply varies only slightly with the regeneration temperature provided

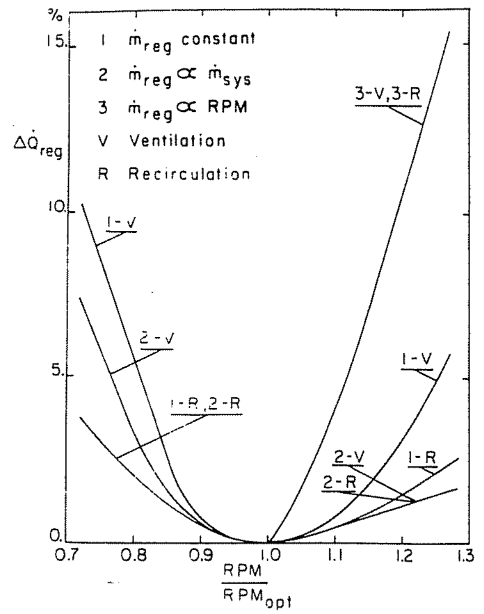

FIGURE 6 Influence of wheel revolution speed on regeneration heat required.

that this temperature is chosen properly. However, this assumption requires a rather complex control strategy in which the regeneration mass flow rate and wheel speed are tied to both the regeneration temperature and process mass flow rate. Figure 7 shows the relative variation of the required regeneration heat supply with regeneration temperature for four other simple control options for the recirculation cycle. Curves for the ventilation cycle show the same trends. The results show that the regeneration heat supply is the least affected by the regeneration temperature for control option 2 in which the wheel speed is contant and the regeneration mass flow rate is varied in proportion to the process mass flow rate. In this case the regeneration temperature may vary from 75°C to 100°C without significantly increasing Q_{reg}. Options in which the regeneration mass flow rate is kept constant are to be avoided, while it is preferable to keep the matrix rotating at a constant speed.

COMPARISON WITH EXPERIMENTAL RESULTS

Desiccant cooling systems are under development and only limited experimental data are available. AiResearch Manufacturing Company has developed a prototype unit of an open cycle solid desiccant air conditioner [9,23]. Their data are described here in the context of the present analysis. The dehumidifier and the regenerator are arranged as two coaxial cylinders rotating around parallel axes. The heater is mounted between the dryer and the heat exchanger. The desiccant is silica gel.

AiResearch set up a system with the prototype unit and two humidifiers arranged in a recirculation type cycle [23]. They conducted experiments on the system to determine values for the design parameters that maximize the COP, defined as the ratio of cooling capacity to thermal heat input. The sensitivity of the COP to process and regeneration air mass flow rate and wheel speed were also investigated. The cooling capacity of the system at design conditions was rated at 4.7 kW. Typical test data of the system are listed in Table 2.

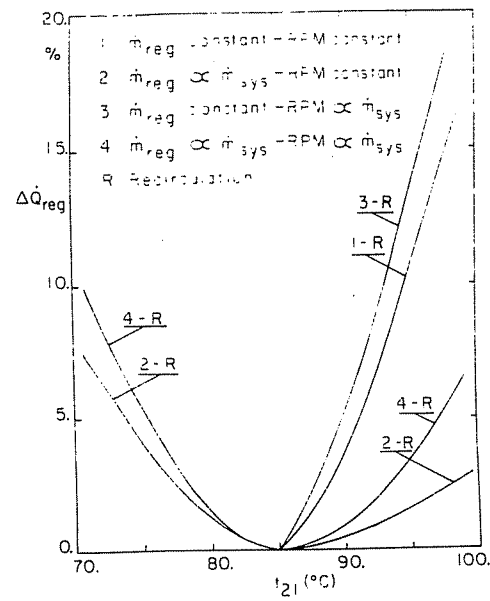

FIGURE 7 Influence of regeneration temperature on regeneration heat required.

TABLE 2 Experimental Data for the AiResearch Test Facility [23].

$\bar{t}_{11}$ = 28.6°C	$\dot{m}_{reg}$ = 0.34 kg/s
$\bar{w}_{11}$ = 12.8 g/kg	$NTU^a_{t,1}$ = 10
$\bar{t}_{21}$ = 83.3°C	M_d = 45.4 kg
$\bar{w}_{21}$ = 19.0 g/kg	T = 720 s

[a] Estimated value

The optimal process air mass flow rate (i.e., the flow rate which yielded the maximum COP) was determined to be 0.50 kg/s. Based on these measurements, the optimal values for the Γ_j parameters may be calculated from Eq. (1). The experimental values for these parameters are compared with values obtained from the model (for the inlet conditions listed in Table 2) in Fig. 3, and indicate good agreement. At these inlet conditions, the COP of the system is not sensitive to the regeneration temperature, as shown in Fig. 4.

The sensitivity of the regeneration heat supply to wheel speed was experimentally determined by AiResearch [23]. Process and regeneration air mass flow rate were kept constant. Measurements were taken for runs at wheel speed differing +10% and -10% from the optimal value. The reported results indicate no measurable effect on cooling capacity and COP of the system. This result is in agreement with the analytical results shown in Fig. 6, which shows sensitivity to wheel speed about the optimum.

The sensitivity of the COP to process and regeneration air mass flow rate was investigated experimentally. The cooling capacity of the system varied with flow rates, therefore no comparison between the reported data and the model can be made. However the experiments indicate the same effect on the COP as shown in Fig. 5. These results substantiate the analysis presented here.

CONTROL STRATEGIES FOR DESICCANT DEHUMIDIFIER COOLING SYSTEMS

The optimal strategy for controlling a dehumidifier cooling system will provide the given cooling load with the least possible regeneration heat supply. This control strategy will not only maximize the COP based on thermal energy input but also minimize electrical energy input. An ideal controller would therefore sense room state and ambient conditions and adjust the process and regeneration mass flow rate, the wheel revolution speed and the regeneration temperature to optimal values. Such tasks can only be accomplished by a direct digital control system. Since this might not pay off for small scale applications, simpler control strategies are needed.

The sensitivity analysis allows one to examine the influence of different control strategy options. If it is assumed that the process mass flow rate is adjusted to provide the given cooling load, the following control options are recommended:

1. <u>The regeneration mass flow rate should vary in proportion to the process mass flow rate.</u>

At the optimal point, the outlet humidity ratio of the process air stream is minimal. Hence, any deviation from this optimal point will increase the process mass flow rate. If, however, the regeneration mass flow is increased, the drying capacity of the matrix increases. This will in turn lower the process outlet humidity ratio, and hence, the process mass flow rate will drop. Thus, this control strategy is inherently stable, and will always provide the least possible flow rates for the following set of given parameters: ratio of regeneration mass flow rate to process mass flow rate, wheel revolution speed, regeneration temperature, room state, cooling load and ambient conditions. It is important to choose and maintain a proper value for the ratio of regeneration mass flow rate to process mass flow rate.

2. <u>The wheel speed may be kept constant.</u>

As has been shown, the influence of wheel speed control on the regeneration energy demand is of second order. Since the dehumidifier wheel rotates at a very low speed, it is important that this speed may be set constant. The choice of the speed is not critical. Deviations of -20% to +30% of the optimal speed do not significantly decrease the COP of the system, provided that condition 1 has been met.

3. <u>The regeneration heat supply is minimized by high regeneration temperatures.</u>

To minimize the electrical energy input for fans, the regeneration temperature should be set to 85°C to 100°C. The COP based on thermal energy input is only weakly influenced by the regeneration temperature in this temperature range. Once this temperature has been chosen, the wheel speed and the ratio of regeneration to process mass flow rate can be determined according to Fig. 3.

CONCLUSIONS

The performance of solid desiccant cooling and drying systems depends on the operating parameters of the dehumidifier. For a given cooling load, the required regeneration heat supply may be minimized by choosing proper values for these parameters, while also the dehumidifier control strategies influence system performance. The analysis presents a method for choosing optimal values, and is substantiated by available experimental results.

Substantial energy savings can be achieved by reducing the regeneration mass flow rate from balanced flow to between 60% and 80% of the process mass flow rate, depending on the regeneration temperature and process humidity ratio. The ratio of matrix mass capacity rate to process air mass flow rate should vary between 0.10 and 0.18. To reduce required mass flow rates, high regeneration temperatures of 85°C to 100°C are recommended.

The optimal control strategy will let the regeneration mass flow rate vary in proportion to the process mass flow rate. The wheel speed may be kept constant, but fine tuning of this speed is not important.

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