The Use of Dehumidifiers in Desiccant Cooling and Dehumidification Systems

E. Van den Bulck

J. W. Mitchell
Mem. ASME

S. A. Klein
Assoc. Mem. ASME

University of Wisconsin
Solar Energy Laboratory,
Madison, WI 53706

The use of rotary dehumidifiers in gas-fired 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 functions 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 the literature are compared with the analytical results and indicate good agreement.

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. The component configurations and psychrometric processes of the ventilation and recirculation cycles are illustrated in Figs. 1 and 2. An overview of the various proposed and tested cycles is given in [1, 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] numerically 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 equation (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.

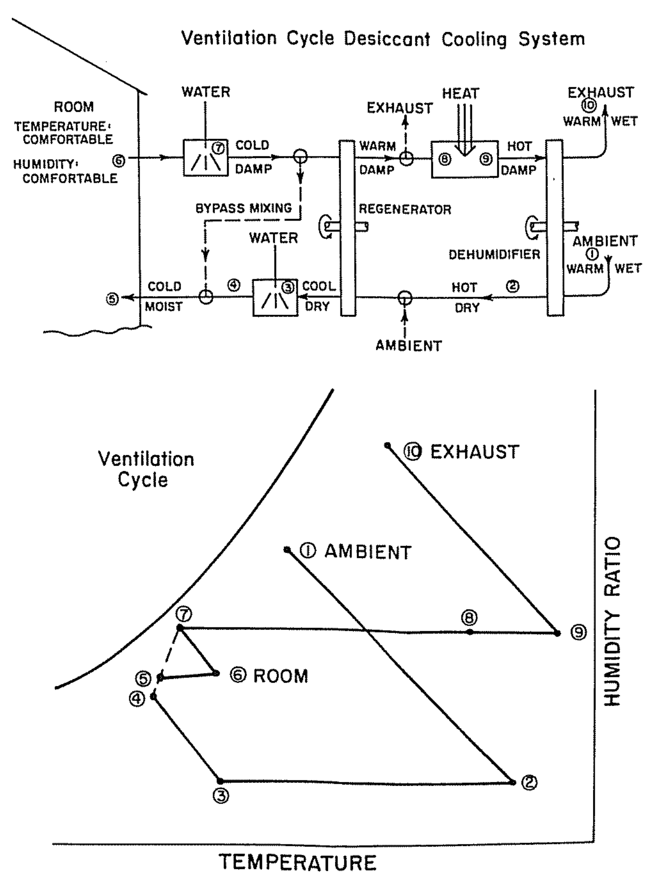

Fig. 1 Schematic and psychrometric diagram of a ventilation cycle desiccant cooling system (from [3])

Contributed by the Heat Transfer Division for publication in the JOURNAL OF HEAT TRANSFER. Manuscript received by the Heat Transfer Division August 30, 1985. Paper No. 84-HT-32.

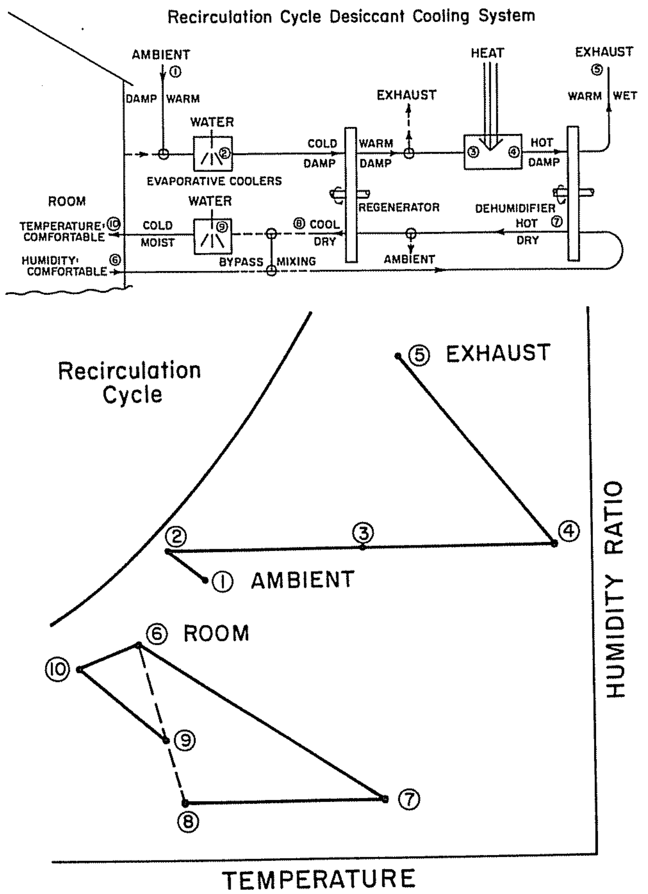

Fig. 2 Schematic and psychrometric diagram of a recirculation cycle desiccant cooling system (from [3])

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, 20], and numerically [9, 13, 24]. Model predictions are compared to experimental data by Ball et al. [27], 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. [13, 18, 19], 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 0.001 kg/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 et al. [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

Nomenclature

A_j = total heat transfer area of the dehumidifier matrix in period j, m^2
COP = coefficient of performance
c_p = moist air specific heat, J/kg dry air•K
h = heat transfer coefficient, W/m^{2}K
h_w = mass transfer coefficient, kg dry air/m^{2}s
i = moist air enthalpy, J/kg dry air
i_{fg} = heat of vaporization, J/kg
i_{wv} = water vapor enthalpy, J/kg
I = desiccant matrix enthalpy, J/kg dry desiccant
L = axial flow length through the matrix, m
Le = NTU$_t$/NTU$_w$, overall Lewis number
$\dot{m}$ = moist air mass flow rate, kg dry air/s
M_d = mass of desiccant in the dehumidifier matrix, kg dry desiccant
M_f = mass of air in the dehumidifier matrix, kg dry air
NTU$_t$ = $hA/\dot{m}c_p$, overall number of transfer units for heat transfer
NTU$_w$ = $h_wA/\dot{m}$, overall number of transfer units for mass transfer
$\dot{Q}$ = thermal energy supply rate, W
RPM = dehumidifier wheel revolution speed, s^{-1}
t = temperature, °C
T = time required for a complete rotation of the matrix, s
w = moist air humidity ratio, kg/kg dry air
W = matrix water content, kg/kg dry desiccant
x = axial coordinate measured from period entrance
z = axial displacement through matrix measured from period entrance, m
Γ_j = jth operating parameter of the rotary dehumidifier defined in equation (1)
Δ = difference
ϵ_h = enthalpy effectiveness of the rotary dehumidifier
ϵ_w = humidity ratio effectiveness
θ = time, s
θ_j = duration of period j, s
τ = time coordinate
τ_{dj} = dwell time of a fluid particle in period j, s

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
11 = process air inlet state of the dehumidifier
12 = process air outlet state of the dehumidifier
21 = regeneration air inlet state of the dehumidifier
22 = regeneration air outlet state of the dehumidifier

Superscripts

$\dot{}$ = rate
$\bar{}$ = average value for a period

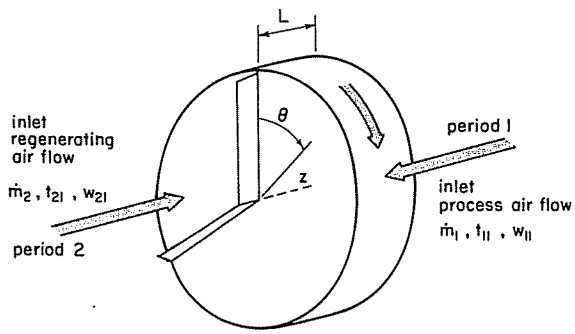

Fig. 3 Nomenclature and coordinate system for the rotary dehumidifier

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. 3. 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 passed 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.

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.

3 The thermal and moisture capacities of the air entrained in the matrix are negligible compared to the matrix capacities.

4 The mixing or carryover of process and regeneration air streams is neglected. Banks [33] has numerically investigated the effect of fluid carryover on the performance of rotary heat exchangers and showed that the regenerator effectiveness increases linearly with the ratio of fluid dwell time τ_{dj} to period duration θ_j. The proportionality constant was defined as the carryover effect and is a function of the overall number of transfer units and the heat capacity rate ratio. For well-designed rotary dehumidifiers, the fluid dwell time is ~ 0.2 s, the period duration is ~ 360 s, the overall number of transfer units is ~ 10, and the equivalent heat capacity rate ratio ranges from 2 to 5. Following Banks [33], the carryover effect is 0.05. The increase in dehumidifier effectiveness is then $(0.05)(0.2)/(360) = 3\times10^{-5}$, which shows that the effect of carryover on dehumidifier performance may be neglected.

5 A transient 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.

6 The axial heat conduction and water vapor diffusion flux are negligible in both the matrix and the air streams.

7 Transport of water vapor within the matrix occurs only through ordinary diffusion and transport of heat occurs only through ordinary heat conduction. Flux coupling is neglected.

8 The heat and mass transfer processes between the desiccant matrix and the air stream can be described by lumped transfer coefficients.

9 The periodic steady-state performance 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

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

The regeneration air flow rate will in general be less than the process air flow rate for well-designed dehumidifiers. To account for unbalanced flow, the dehumidifier may be designed with an unequal partition of the wheel face area (Fig. 3). Flow unbalance and unequal area split are described by the ratio of the capacitance rate parameters. The following dimensionless coordinates are introduced

$$\begin{aligned} x &= \frac{z}{L}; && 0\le x\le 1 \\ \tau &= \frac{\theta}{\tau_{dj}}\frac{M_{fj}}{M_{dj}} = \frac{\theta}{\theta_j}\frac{1}{\Gamma_j}; && 0\le\tau\le\frac{1}{\Gamma_j} \end{aligned} \tag{2}$$

Under the assumptions 1 to 9, the conservation and transfer rate equations for period j of the heat and mass regenerator have been written as [13]

$$\begin{aligned} &\frac{\partial w_f}{\partial x} + \frac{\partial W_m}{\partial \tau} = 0 \\ &\frac{\partial w_f}{\partial x} = \mathrm{NTU}_{w,j}(w_m - w_f) \\ &\frac{\partial i_f}{\partial x} + \frac{\partial I_m}{\partial \tau} = 0 \\ &\frac{\partial i_f}{\partial x} = \mathrm{NTU}_{t,j}\frac{\partial i_f}{\partial t_f}(t_m - t_f) + i_{wv}\mathrm{NTU}_{w,j}(w_m - w_f) \end{aligned} \tag{3}$$

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]

$$\begin{aligned} w_m &= w_m(W_m, t_m) \\ i_m &= i_m(w_m, t_m) \\ I_m &= I_m(W_m, t_m) \end{aligned} \tag{4}$$

The initial conditions for this system of equations are

$$\begin{aligned} w_f(x=0,\ \tau) &= w_{j1} \\ i_f(x=0,\ \tau) &= i_{j1} \end{aligned} \qquad 0\le\tau\le\frac{1}{\Gamma_j};\ j=1,\ 2 \tag{5}$$

The periodic equilibrium boundary conditions for the matrix state properties are

for $0 \le x \le 1$:

$$\begin{aligned} &\lim_{\tau_1\to(1/\Gamma_1)^-} W_m(x,\tau_1) = \lim_{\tau_2\to 0^+} W_m(1-x,\tau_2) \\ &\lim_{\tau_1\to(1/\Gamma_1)^-} I_m(x,\tau_1) = \lim_{\tau_2\to 0^+} I_m(1-x,\tau_2) \\ &\lim_{\tau_1\to 0^+} W_m(x,\tau_1) = \lim_{\tau_2\to(1/\Gamma_2)^-} W_m(1-x,\tau_2) \\ &\lim_{\tau_1\to 0^+} I_m(x,\tau_1) = \lim_{\tau_2\to(1/\Gamma_2)^-} I_m(1-x,\tau_2) \end{aligned} \tag{6}$$

Numerical solutions have been obtained for these equations by a number of authors [9, 13, 24]. There are no 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

$$\epsilon_w = \frac{w_{11} - \bar{w}_{12}}{w_{11} - (\bar{w}_{12})_{id}} \tag{7a}$$

and for enthalpy

$$\epsilon_h = \frac{\bar{i}_{12} - i_{11}}{(\bar{i}_{12})_{id} - i_{11}} \tag{7b}$$

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 effectivenesses are functions of inlet temperature and humidity ratio, capacitance rate parameters Γ_j, transfer parameters $\mathrm{NTU}_{t,j}$, and the Lewis number Le. Effectiveness expressions for a nominal silica gel rotary dehumidifer are presented in [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

$$\begin{aligned} \frac{\partial w_m}{\partial x} + \frac{\partial W_m}{\partial \tau} &= 0 \\ \frac{\partial i_m}{\partial x} + \frac{\partial I_m}{\partial \tau} &= 0 \end{aligned} \tag{8}$$

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 nonlinear because of the nonlinear 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 [20].

Design Parameter Optimization for Dehumidifiers in Desiccant Cooling Systems

The analysis will focus on the dehumidifier–regenerator subsystem that is common to both ventilation and recirculation cycles, and is shown schematically in Fig. 4. 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. The desiccant is silica gel.

The heat source is a gas furnace with adjustable thermal energy output $\dot{Q}_{\mathrm{reg}}$ to provide the specified regeneration temperature t_{21}. For solar-fired systems, the energy output from the collectors decreases with increasing regeneration temperature and the analysis should take the performance characteristics of the collectors into account. The following analysis considers gas-fired systems only. The regeneration heat supply rate is obtained from an energy balance on the heat source

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

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

$$t = t_{12} - \Delta t \tag{10}$$

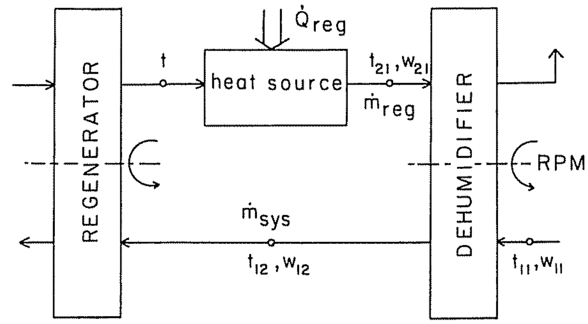

Fig. 4 Dehumidifer-regenerator subsystem of a desiccant cooling system

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 the temperature difference Δt is a constant equal to 4°C, which corresponds to a heat exchanger effectiveness of 90 percent for typical operating conditions. This effectiveness is typical of high-performance rotary regenerators which have large matrix to fluid heat capacity rate ratio [32]. The process air stream entering the heat exchanger has a nonuniform angular distribution of temperature (and humidity ratio). It has been shown by Brandemuehl and Banks [34] that nonuniformities in inlet fluid temperatures have little effect on the periodic steady-state performance of high-effectiveness regenerators at large matrix to fluid heat capacity rate ratio.

Using the parameters Γ_1 and Γ_2, the regeneration flow rate can be expressed in terms of the process mass flow rate

$$\dot{m}_{\mathrm{reg}} = \frac{\Gamma_1}{\Gamma_2} \dot{m}_{\mathrm{sys}} \tag{11}$$

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

$$\dot{m}_{\mathrm{sys}} = \frac{\dot{Q}_{\mathrm{load}}}{\Delta i} \tag{12}$$

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 (see Figs. 1 and 2). 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

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

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

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

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

$$\mathrm{NTU}_{t,2} = \mathrm{NTU}_{t,1} \frac{\Gamma_2}{\Gamma_1} \tag{15}$$

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 dominate the overall transfer process and that the flow of the air through the matrix is laminar. The actual Lewis number for packed bed silica gel dehumidifiers is of the order of 2 [35]. It has been shown by Van den Bulck et al. [20], that the effect of the Lewis number on the performance of regenerative dehumidifiers is small for high overall NTU and Lewis numbers less than 2.

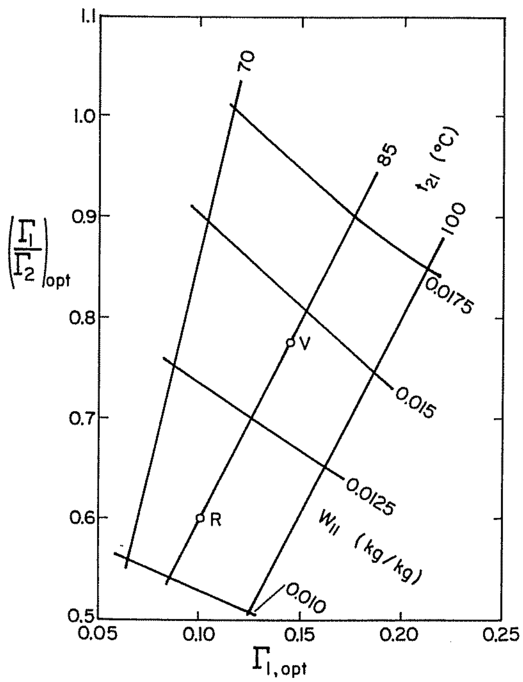

Fig. 5 Regeneration temperature and process humidity in a $\Gamma_{1,opt}$ – $(\Gamma_1/\Gamma_2)_{opt}$ chart

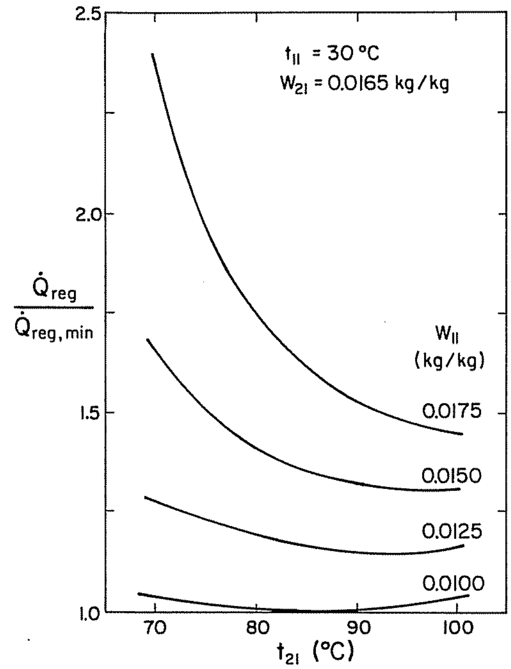

Fig. 6 Normalized regeneration heat required as a function of regeneration temperature for a tuned dehumdifier

Equation (14) gives the regeneration heat supply rate as a function of the cooling load, the room air humidity, the ratio of the capacitance rate parameters, the regeneration temperature, and the process air outlet state. In this equation, the process outlet state is a function of both process and regeneration air inlet states and capacitance rate parameters. Optimum system performance is achieved when the regeneration energy is minimum for a specified cooling load. In this analysis, the cooling load is held constant and w_{room} is set equal to the ARI standard room air humidity of 0.0111 kg/kg. Parameters for the optimization study are the process air temperature and humidity and the regeneration air humidity. The variables which can then be used to minimize $\dot{Q}_{reg}$ are the regeneration temperature and capacitance rate parameters Γ_1 and Γ_2. In the subsequent analysis, the optimization is carried out in two steps. First, values for Γ_1 and Γ_2 which minimize $\dot{Q}_{reg}$ are determined as functions of process and regeneration air inlet states. In a second step, the regeneration temperature that minimizes $\dot{Q}_{reg}$ for a "tuned" dehumidifier (i.e., optimal Γ_1 and Γ_2) is determined as a function of process temperature and humidity and regeneration humidity.

The results of an analysis based on this model [36] show that the regeneration heat required is less sensitive to Γ_1 than to Γ_1/Γ_2. These results also show that the effect of process temperature t_{11} and regeneration humidity w_{21} on the optimal value of Γ_1/Γ_2 is of second order compared to the effect of process humidity w_{11} and regeneration temperature 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. 5 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.

The results in Fig. 5 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 rotational speed might increase or decrease depending on the inlet conditions.

Points marked with V and R in Fig. 5 show typical states for the ventilation and recirculation cycles, respectively. The values for $(\Gamma_1/\Gamma_2)_{opt}$ for these points agree with the values reported by Jurinak [3], which were obtained by a complete detailed analysis of the complete system, not just the rotary dehumidifier–regenerator subsystem.

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 [36]. 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 is a function of process and regeneration air inlet states. Figure 6 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 6 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

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
$\Gamma_{1,opt}=0.170$	$\Gamma_{1,opt}=0.093$
$\frac{\Gamma_1}{\Gamma_2}\_{opt}=0.823$	$\frac{\Gamma_1}{\Gamma_2}\_{opt}=0.600$

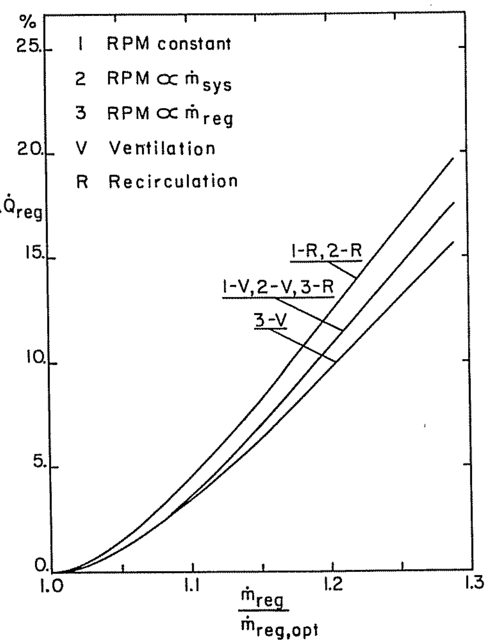

Fig. 7 Effect of regeneration flow rate on regeneration energy required

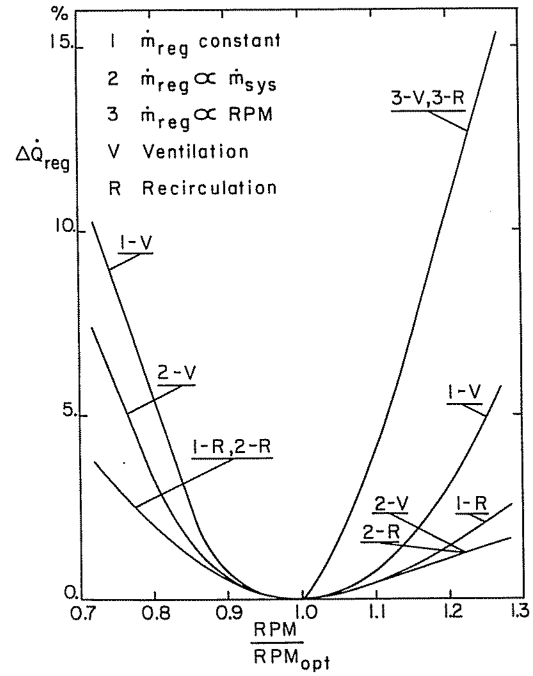

Fig. 8 Effect of wheel rotational speed on regeneration energy required

energy input to be maximal for t_{21} between 65°C and 85°C, but these studies were performed assuming either 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.

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 of $(\Gamma_1/\Gamma_2)_{opt}$, which decreases the regeneration mass flow rate even further. The parasitic power of desiccant cooling systems therefore decreases 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 regeneration temperatures between 80°C and 100°C [7, 9, 10]. The selected 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. 5 and 6. 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, or 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 [36]. 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 in Table 1.

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 7 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.

The results in Fig. 7 show only minor differences between the ventilation and recirculation cycle systems. The effect of the different control strategy options is small. Increasing the regeneration flow rate to 20 percent above the minimal value required causes an increase of about 10 percent 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 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. 8. 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 percent to +30 percent without significantly affecting

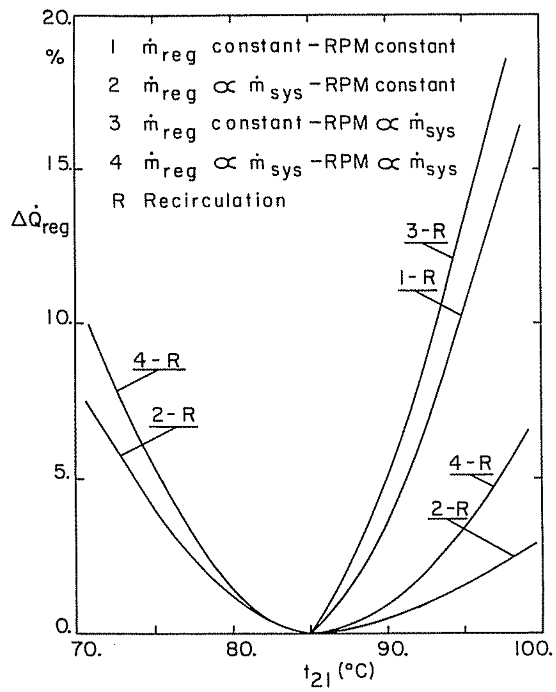

Fig. 9 Effect of regeneration temperature on regeneration energy required

$\dot{Q}_{reg}$. 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. 5 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. 6 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 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 9 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 constant 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 $\dot{Q}_{reg}$. 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 packed beds and 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.

Table 2 Experimental data for the AiResearch test facility [23]

$\bar{t}_{11} = 28.6°C$	$\dot{m}_{reg} = 0.34$ kg/s
$\bar{w}_{11} = 0.0128$ kg/kg	$NTU_{t,1}^{a} = 10$
$\bar{t}_{21} = 83.3°C$	$M_d^a = 45.4$ kg
$\bar{w}_{21} = 0.0190$ kg/kg	$T^a = 720$ s

[a]Estimated value

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.

Representative test data of the system are listed in Table 2. For these specified data, the variation of system COP with process air mass flow rate was determined experimentally and it was found that a flow rate of 0.50 kg/s yielded the maximum COP. From these measurements, values of the capacitance rate parameters that provided the optimum system performance can be calculated using equation (1)

$$\Gamma_1 = \frac{(45.4)}{(720)(0.50)} = 0.126 \tag{16a}$$

$$\frac{\Gamma_1}{\Gamma_2} = \frac{[(45.4)/(720)(0.50)]}{[(45.4)/(720)(0.34)]} = 0.68 \tag{16b}$$

These experimental values can be compared with values that are predicted by the model for this specific case as follows. Figure 5 shows analytically determined optimal values for the capacitance rate parameters as functions of regeneration temperature and process humidity. With the specified inlet conditions of Table 2 as entries in Fig. 5, the following values are obtained

$$\Gamma_{1,opt} = 0.125 \tag{17a}$$

$$\left(\frac{\Gamma_1}{\Gamma_2}\right)_{opt} = 0.71 \tag{17b}$$

This comparison establishes that the experimentally determined optimal values of the capacitance rate parameters agree with this analysis. The good agreement supports the approximation made with respect to the heat exchanger-dehumidifier subsystem.

AiResearch [23] conducted its experiments for a regeneration temperature of 83°C. The optimal regeneration temperature for the inlet process humidity as specified in Table 2 can be determined from Fig. 6 and is predicted to be ~ 92°C. Figure 6 also shows, however, that the optimum regeneration temperature is not well defined for low process humidities which are encountered in recirculation type systems.

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 percent and −10 percent 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 illustrated in Fig. 8, 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. Since the cooling capacity of the system varied with flow rates, 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. 7. 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 rotational 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 The regeneration mass flow rate should vary in proportion to the process mass flow rate. 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 The wheel speed may be kept constant. 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 practical that this speed may be set constant. The choice of the speed is not critical. Deviations of -20 percent to $+30$ percent of the optimal speed do not significantly decrease the COP of the system, provided that condition 1 has been met.

3 The regeneration heat supply is minimized by high regeneration temperatures. For gas-fired desiccant cooling systems at specified cooling load, the thermal regeneration energy input and electric power required to drive the air fans are minimized by regeneration temperatures ranging from 85 to 100°C. The COP based on thermal energy input is only weakly affected by the regeneration temperature in this range. Once the regeneration temperature has been chosen, the wheel speed and the ratio of regeneration to process air mass flow rate can be determined according to Fig. 5.

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. This analysis presents a method for choosing optimal values for gas-fired systems, 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 percent and 80 percent 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 rotational speed may be kept constant and fine tuning of this speed is not important.

Acknowledgments

This project was supported by the Solar Heating and Cooling Research and Development Branch, Office of Conservation and Solar Applications, U.S. Department of Energy.

References

1 Jurinak, J. J., Mitchell, J. W., and Beckman, W. A., "Open Cycle Desiccant Air Conditioning as an Alternative to Vapor Compression Cooling in Residential Applications," ASME *Journal of Solar Energy Engineering*, Vol. 106, 1984, pp. 252–260.

2 Dunkle, R. V., "A Method of Solar Air Conditioning," *Mech. Chem. Engng. Trans. Inst. Engrs. Aust.*, Vol. 1, 1965, pp. 73–78.

3 Jurinak, J., "Open Cycle Solid Desiccant Cooling—Component Models and Systems Simulations," PhD thesis, University of Wisconsin—Madison, 1982.

4 *Proceedings of the 1981 Annual DOE Active Heating and Cooling Contractor's Review Meeting*, U.S. Department of Energy, Washington, D.C., 1981.

5 Sheridan, J. C., and Mitchell, J. W., "Hybrid Solar Desiccant Cooling Systems," *Progr. Sol. Energy*, Vol. 5, 1982, pp. 579–584.

6 Macdonald, N. J., "Utilization of Condensor Heat for Desiccant Dehumidifiers in Supermarket Applications," *Trans. Am. Soc. Heat. Refrig. Air-Condit. Engrs.*, Vol. 89, Part 2A, 1983, pp. 225–235.

7 Monnier, J. B., Worek, W. M., and Lavan, Z., "Performance of a Cross-Cooled Solar-Powered Desiccant Cooling System," Rep. No. DES-81-1, Illinois Institute of Technology, Chicago, 1982.

8 Shultz, K. J., and Mitchell, J. W., "The Performance of Solar-Desiccant Systems Using Cooled Desiccant Beds," *Progr. Sol. Energy*, Vol. 5, 1982, pp. 585–590.

9 Rousseau, J., "Development of a Solar Desiccant Dehumidifier: Phase II Technical Progress Report," Rep. No. 81-18436, AiResearch Manufacturing Company, Los Angeles, 1981.

10 Wurm, J., et al., "SOLAR-MEC Development Program: Semiannual Progress Report," Rep. No. C00-4495-23, Institute of Gas Technology, Chicago, 1979.

11 Collier, R. K., Barlow, R. S., and Arnold, F. H., "An Overview of Open-Cycle Desiccant-Cooling Systems and Materials," ASME *Journal of Solar Energy Engineering*, Vol. 104, 1982, pp. 28–34.

12 Jurinak, J. J., and Mitchell, J. W., "Recirculation of Purged Flow in an Adiabatic Counterflow Rotary Dehumidifier," ASME JOURNAL OF HEAT TRANSFER, Vol. 106, 1984, pp. 369–375.

13 Maclaine-cross, I. L., "A Theory of Combined Heat and Mass Transfer in Regenerators," PhD thesis, Monash University, Australia, 1974.

14 Lunde, P., "Solar Desiccant Air Conditioning With Silica Gel," *Proc. 2nd Workshop on the Use of Solar Energy for Cooling of Buildings*, SAN/1122-76/2, 1976, p. 280.

15 Burns, P. R., Mitchell, J. W., and Beckman, W. A., "Hybrid Desiccant Cooling Systems in Supermarket Applications," *Trans. Am. Soc. Heat. Refrig. Air-Condit. Engrs.*, Vol. 91, Part 1, 1985.

16 Schlepp, D., "A High Performance Dehumidifier for Solar Desiccant Cooling Systems," Rep. No. SERI/TP-252-1979, Solar Energy Research Institute, Denver, 1983.

17 Ingram, J. B., and Vliet, G. C., "Solid Desiccant Rotary Dryer Performance Charts," *Progr. Sol. Energy*, Vol. 5, 1982, pp. 603–608.

18 Maclaine-cross, I. L., and Banks, P. J., "Coupled Heat and Mass Transfer in Regenerators—Prediction Using an Analogy With Heat Transfer," *Int. J. Heat Mass Transfer*, Vol. 15, 1972, pp. 1225–1242.

19 Banks, P. J., "Prediction of Heat and Mass Regenerator Performance Using Nonlinear Analogy Method," ASME JOURNAL OF HEAT TRANSFER, Vol. 107, 1985, pp. 222–238.

20 Van den Bulck, E., Mitchell, J. W., and Klein, S. A., "Design Theory for Rotary Heat and Mass Exchangers," *Int. J. Heat Mass Transfer*, Vol. 28, 1985, pp. 1575–1595.

21 Lax, P., *Hyperbolic Systems of Conservation Laws and the Mathematical Theory of Shockwaves*, SIAM, Philadelphia, 1973.

22 Schlepp, D., "SERI Test Standard: Methods of Testing for Rating Solar Assisted Desiccant Cooling Systems," Solar Energy Research Institute, Denver, 1979.

23 Rousseau, J., "Development of a Solar Desiccant Dehumidifier: Phase I Final Summary Report," Rep. No. 80-17481, AiResearch Manufacturing Company, Los Angeles, 1980.

24 Pla Barby, F., "Solid Desiccant Air Conditioning With Silica Gel Using Solar Energy," PhD thesis, The University of Texas, Austin, 1978.
25 Pesaran, A. A., "Moisture Transport in Silica Gel Particle Beds," PhD thesis, University of California, Los Angeles, 1983.
26 Yoshida, H., and Ruthven, D. M., "Dynamic Behaviour of an Adiabatic Adsorption Column I," *Chem. Engng. Sci.*, Vol. 38, 1983, pp. 877-884.
27 Ball, H. D., Anath, G. P., and Singer, R. J., "The Dehumidification of Air Using Solar Regenerated Beds of Silica Gel," ASME Paper No. 83-WA-SOL-2.
28 Nelson, J. S., "An Investigation of Solar Powered Open Cycle Air Conditioners," MS thesis, University of Wisconsin—Madison, 1976.
29 Close, D. J., "Combined Heat and Mass Transfer in Porous Media—The Further Development of an Analogy With Heat Transfer," PhD thesis, Monash University, Australia, 1974.
30 Close, D. J., and Banks, P. J., "Coupled Equilibrium Heat and Single Adsorbate Transfer in Fluid Flow Through Porous Medium—II," *Chem. Engng. Sci.*, Vol. 27, 1972, pp. 1157-1168.
31 Bullock, C. E., and Threlkeld, J. L., "Dehumidification of Moist Air by Adiabatic Adsorption," *Trans. Am. Soc. Heat Refrig. Air-Condit. Engrs.*, Vol. 72, Part 1, 1966, pp. 301-313.
32 Shah, R. K., "Thermal Design Theory for Regenerators," in: *Heat Exchangers: Thermal Hydraulic Fundamentals and Design*, S. Kakao, A. E. Bergles, and F. Mayinger, eds., Hemisphere Publishing Corp., New York, 1981, pp. 721-763.
33 Banks, P. J., "Effect of Fluid Carry Over on Regenerator Performance," ASME Journal of Heat Transfer, Vol. 104, 1982, pp. 215-217.
34 Brandemuehl, M. J., and Banks, P. J., "Rotary Heat Exchangers With Time Varying or Nonuniform Inlet Temperatures," ASME Journal of Heat Transfer, Vol. 106, 1984, pp. 750-758.
35 Allander, C. G., "Untersuchung des Adsorptionsvorganges in Adsorbentenschichten mit linearer Adsorptionsisotherme," *Kungl. Tekniska Hoegskolans handlingar*, Vol. 70, 1953, pp. 1-160.
36 Van den Bulck, E., "Analysis of Solid Desiccant Rotary Dehumidifiers," MS thesis, University of Wisconsin—Madison, 1983.