Experimental Study of the Heat and Mass Transfer in a Packed Bed Liquid Desiccant Air Dehumidifier

V. Öberg

D. Y. Goswami[1]

Solar Energy and Energy Conversion Laboratory
Department of Mechanical Engineering
University of Florida
Gainesville, Florida 32611-6300

Desiccant cooling systems have the ability to provide efficient humidity and temperature control while reducing the electrical energy requirement for air conditioning as compared to a conventional system. Naturally, the desiccant air dehumidification process greatly influences the overall performance of the desiccant system. Therefore, the effects of variables such as air and desiccant flow rates, air temperature and humidity, desiccant temperature and concentration, and the area available for heat and mass transfer are of great interest. Due to the complexity of the dehumidification process, theoretical modeling relies heavily upon experimental studies. However, a limited number of experimental studies are reported in the literature. This paper presents results from a detailed experimental investigation of the heat and mass transfer between a liquid desiccant (triethylene glycol) and air in a packed bed absorption tower using high liquid flow rates. A high performance packing that combines good heat and mass transfer characteristics with low pressure drop is used. The rate of dehumidification, as well as the effectiveness of the dehumidification process are assessed based on the variables listed above. Good agreement is shown to exist between the experimental findings and predictions from finite difference modeling. In addition, a comparison between the findings in the present study and findings previously reported in the literature is made. The results obtained from this study make it possible to characterize the important variables which impact the system design.

Introduction and Background

Air conditioning requires efficient control of both temperature and humidity. In hot and humid climates, conventional vapor compression air conditioning systems cool the air below its dew point to reduce the moisture content, followed by reheat of the air to a comfortable temperature before it is introduced into the conditioned space. Hence, the evaporator in the vapor compression system operates at a lower temperature than what is required to meet the sensible cooling load, resulting in a lower coefficient of performance (COP). Furthermore, energy efficient vapor compression systems designed to operate at higher evaporator temperatures, have been found unable to maintain the indoor relative humidity within a comfortable range in hot and humid climates (Marsala et al., 1989). Therefore, separating the control of humidity and temperature by means of desiccants could result in energy savings, as well as improved humidity control. The largest energy requirement associated with the use of a desiccant dehumidifier is low temperature heat that could be provided by solar energy or waste heat.

The use of liquid desiccants may be advantageous compared to solid desiccants. According to Howell (1987), the pressure drop through a liquid desiccant system is smaller than the pressure drop through a solid desiccant wheel. Also, the ability to pump the liquid makes it possible to connect several small dehumidifiers to one large regeneration unit (Harriman, 1990), which may be advantageous in large buildings. Finally, concentrated desiccant may be stored for use during the times when no suitable source of regeneration heat is available.

The driving force for mass transfer between the air and the desiccant is the difference in vapor pressure between the air and the desiccant. Hence, the desiccant must have as low a vapor pressure as possible. Liquid desiccants commonly used are aqueous solutions of lithium bromide, lithium chloride, calcium chloride, salt mixtures, and triethylene glycol (TEG). As cool and concentrated desiccant is brought in contact with air, water vapor in the air is absorbed by the desiccant, i.e., water condenses into the desiccant. During this process, heat is evolved due to the latent heat of condensation of the water, and the heat of mixing. Equipment commonly employed in desiccant systems includes packed bed absorption towers (e.g., Gandhidasan, 1994, Kinsara et al., 1996, Sick et al., 1988, and Thornbloom and Nimmo, 1995) and spray chambers containing finned cooling coils (e.g., Johannsen, 1984, Mahmoud and Ball, 1988, Robison, 1977, and Scalabrin and Scaltriti, 1990). In a sprayed cooling coil dehumidifier, air is dehumidified as it is brought in contact with the desiccant film flowing over the coil. Cooling water or refrigerant flowing through the coil removes the heat evolved during the absorption, allowing for an isothermal process. Packed towers offer a larger area for heat and mass transfer per unit volume than coil dehumidifiers. However, the heat evolved is usually not removed with the result that the desiccant temperature may increase throughout the tower, reducing the potential for mass transfer. Pressure drop through a packed bed may also be higher than in a coil dehumidifier. However, modern packings are being designed for low pressure drop. The possibility of designing compact air dehumidifiers makes the packed bed absorption towers very attractive as a contact device. Thus, a packed bed absorption tower was chosen as the dehumidifier for this investigation.

Naturally, the effectiveness of the desiccant air dehumidification process greatly influences the overall performance of the

[1] Author to whom all correspondence should be addressed.

Contributed by the Solar Energy Division of The American Society of Mechanical Engineers for publication in the ASME Journal of Solar Energy Engineering. Manuscript received by the ASME Solar Energy Division, Nov. 1997; final revision, July 1998. Associate Technical Editor: W. M. Worek.

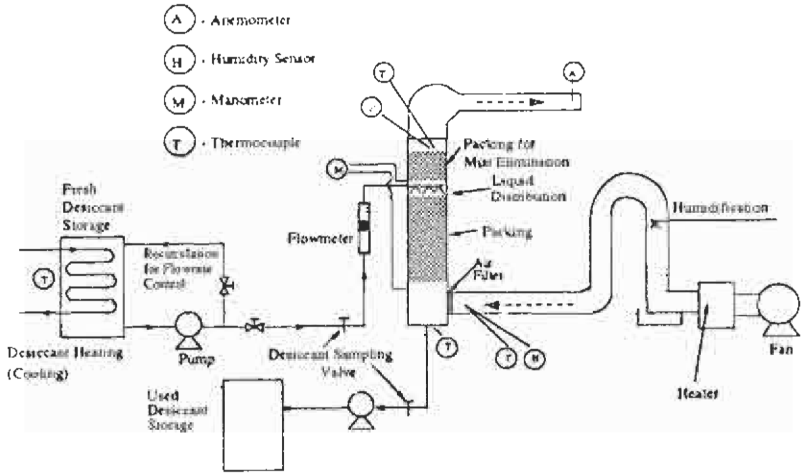


Fig. 1 Experimental facility

desiccant system. Therefore, the impact of variables such as air and desiccant flow rates, air temperature and humidity, desiccant temperature and concentration, and the area available for heat and mass transfer on the performance of the dehumidifier is of great interest. A number of studies of the heat and mass transfer in the dehumidifier have been presented. The performance of a packed bed dehumidifier as a function of design variables has been modeled by Gandhidasan et al. (1987), Khan (1996 and 1994), and Ullah et al. (1988). Due to the complexity of the dehumidification process, theoretical modeling relies heavily upon experimental data. However, a limited number of studies which include experimental findings are reported in the literature. Chen et al. (1989), Chung et al. (1993), McDonald et al. (1992), Patnaik et al. (1990) and Potnis and Lenz (1996) carried out experiments on packed bed dehumidifiers, using salt solutions as the desiccants. Chung et al. (1995) reported some experimental findings using triethylene glycol as the desiccant.

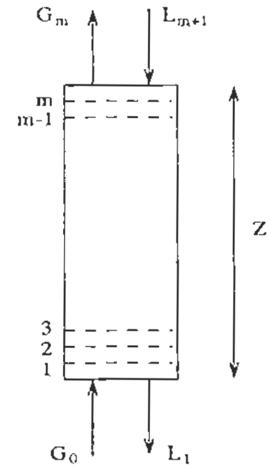
The objective of the present investigation is to provide additional experimental data to aid in the design of desiccant systems. Therefore, a thorough experimental analysis was carried out, exploring the influence of all the variables previously listed. Compared to most of the experimental studies listed above, higher liquid flow rates were used in the present investigation. A preliminary set of experiments showed these higher flow rates to be necessary to ensure adequate wetting of the packing. Due to its lower corrosivity and lower surface tension as compared to salt solutions, 95% by weight triethylene glycol was chosen as the desiccant. The performance of the dehumidification process was evaluated in terms of the water condensation rate (i.e., the rate of moisture removal from the air), and the dehumidification effectiveness (concept introduced in a later section). The experimental findings were compared to those obtained from theoretical modeling, as well as other experimental findings reported in the literature.

Nomenclature

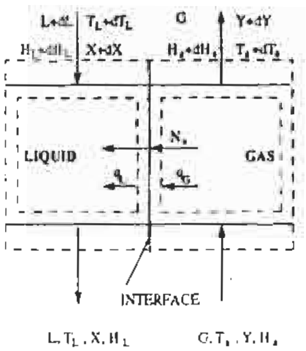
a_t = specific surface area of packing (m^2/m^3)
a_w = wetted surface area of packing (m^2/m^3)
COP = coefficient of performance
c_p = specific heat (kJ/kg-°C)
D = diffusivity (m^2/s)
d_p = diameter of sphere with the same surface area as a single packing particle (m)
F_G = gas phase mass transfer coefficient ($kmol/m^2$-s)
F_L = liquid phase mass transfer coefficient ($kmol/m^2$-s)
G = superficial air (gas) flow rate (kg/m^2-s)
g = acceleration of gravity (m/s^2)
H = enthalpy (kJ/kg)
h_G = gas side heat transfer coefficient (kJ/m^2-s)
j_h = dimensionless heat transfer group (Eq. 13)
j_m = dimensionless mass transfer group (Eq. 13)
K_G = overall gas side mass transfer coefficient ($kmol/m^2$-s)
k_G = gas phase mass transfer coefficient ($kmol/m^2$-s-Pa)
k_L = liquid phase mass transfer coefficient (m/s)
L = superficial desiccant flow rate (kg/m^2-s)
M = molar mass (kg/kmol)
m = flow rate (kg/s) or (g/s)
N_v = molar vapor mass transfer flux ($kmol/m^2$-s)
P = total pressure (Pa)
Pr = Prandtl number
p = vapor pressure (Pa)
q = heat transfer flux (kW/m^2)
Sc = Schmidt number
T = temperature (°C)
TEG = triethylene glycol
X = desiccant concentration (kg TEG/kg solution)
x = desiccant mole fraction (kmol TEG/kmol solution)
x_{SM} = logarithmic mean solvent mole fraction difference between the bulk liquid and interface values (kmol TEG/kmol solution)
Y = air humidity ratio (kg water/kg dry air or g water/kg dry air)
y = water mole fraction (kmol water/kmol air)
Z = tower height (m)
γ = surface tension (N/m)
ϵ = effectiveness
λ = latent heat of condensation/vaporization (kJ/kg)
μ = viscosity (Ns/m^2)
π = dimensionless vapor pressure difference (Eq. 15)
ρ = density (kg/m^3)

Subscripts

a = air
c = critical
cond = water condensation
equ = equilibrium
G = gas phase
IN = inlet
i = interface
L = desiccant or liquid phase
OUT = outlet
v = vapor
w = water
Y = humidity
0 = reference state



(a)



(b)

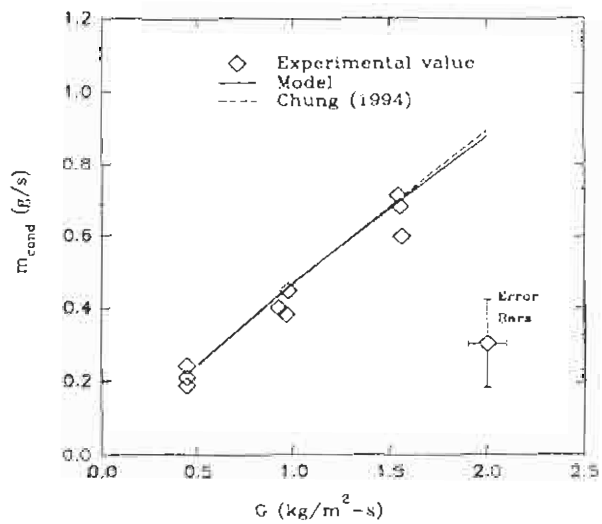
Fig. 2 Packed bed dehumidifier: (a) overview; (b) differential segment

Experimental Procedure

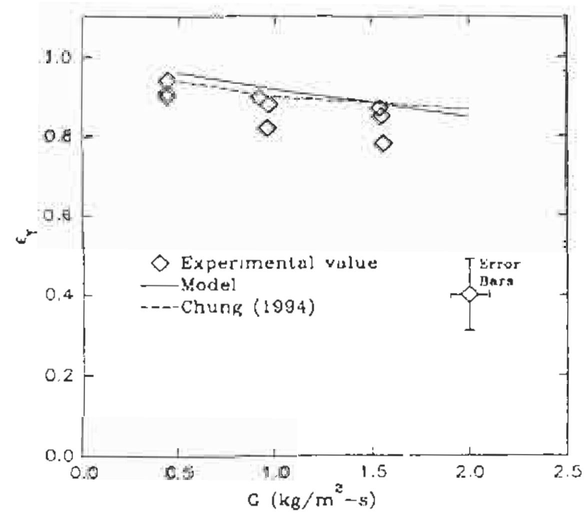
The rate of moisture removal from the air (water condensation rate) as well as the effectiveness of the dehumidification process were studied experimentally as a function of the following variables: air and desiccant flow rates; air temperature and humidity ratio; desiccant temperature and concentration; and the height of the packed bed.

A schematic of the experimental facility is shown in Fig. 1. The packed bed absorption tower was constructed from a 25.4 cm (10 in) diameter acrylic tube to allow for flow visualization. The tower was made in sections so that the bed height could be varied without changing the distance from the liquid distribution to the top of the bed. The inner diameter of the tower was 0.24 m. The packing used was 2.54 cm (1 in) polypropylene Rauschert Hiflow® rings with a specific surface area of 210 m^2/m^3. Fresh, unused triethylene glycol was stored in a tank, and its temperature was adjusted by circulating cold or warm water through a submerged copper coil. Before each experiment, the desiccant was allowed to recirculate to remove any temperature and concentration gradients. Air was blown past an air heater and through a humidifying chamber to adjust its temperature and relative humidity before it entered the packed tower. At the air inlet, an air filter was installed to prevent air borne particles and water droplets from entering the dehumidifier. When the desired air and desiccant conditions were obtained, the desiccant was allowed to flow through the tower. The desiccant was distributed over the packing by three spray heads evenly spaced in an equilateral triangular configuration. A section of packing (20 cm) was placed above the spray heads, before the air outlet to minimize the loss of desiccant through the air outlet. Once steady state was obtained, measurements were taken for 15 to 20 minutes using a PC-based data acquisition system. These measurements included inlet and outlet temperatures of the desiccant and the air using copper-constantan thermocouples (±0.5°C accuracy), as well as inlet and outlet air relative humidities (±1% accuracy). In addition, samples of the desiccant entering and leaving the dehumidifier were taken during the experiment and analyzed for water content using Karl Fischer titration. The used desiccant was pumped over to a separate storage tank so that the inlet desiccant concentration did not change during the experiment. The liquid flow rate was set approximately using a flow indicator. However, it was measured accurately by a catch-bucket method. The air velocity was measured using an vane anemometer at the air outlet (±5% accuracy). Finally, the air pressure drop over the packed bed (not including the mist eliminating section) was determined by an air-over-oil manometer.

Experiments were conducted for each variable at three levels (low, intermediate, and high value) while keeping the other



(a)



(b)

Fig. 3 Influence of air flow rate on: (a) condensation rate; (b) dehumidification effectiveness

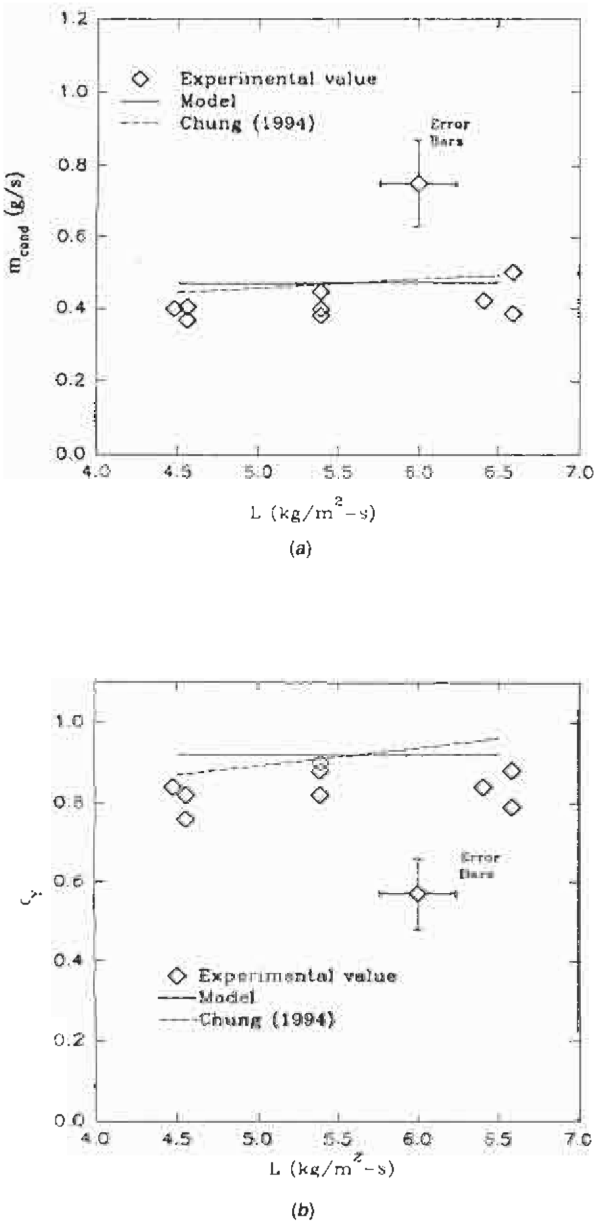


Fig. 4 Influence of desiccant flow rate on: (*a*) condensation rate; (*b*) dehumidification effectiveness

variables constant at their intermediate value. Three experiments were conducted at each level.

Theoretical Model of the Packed Bed Absorption Tower

For this study, a finite difference model similar to those used by Factor and Grossman (1980) and Gandhidasan et al. (1987) was utilized. This model is essentially based on the model for adiabatic gas absorption presented by Treybal (1969) with the exception that the resistance to heat transfer in the liquid phase is neglected. In summary, the assumptions made in this study are: adiabatic absorption; concentration and temperature gradients in the flow direction (Z-direction, referring to Fig. 2) only; only water is transferred between the air and the desiccant; the interfacial surface area is the same for heat transfer and mass transfer, and it is equal to the specific surface area of the packing; the heat of mixing is negligible as compared to the latent heat of condensation of the water; and the resistance to heat transfer in the liquid phase is negligible.

Figure 2*a* gives an overview of the packed bed absorption tower. For the finite difference model, the packed bed height Z is divided into small segments, dZ (Fig. 2*b*), and the mass and energy balances are solved for each segment, from the bottom to the top of the tower. The governing equations that describe the changes in air humidity and air temperature, desiccant temperature and desiccant concentration, and desiccant flow rate across a segment are given below. A detailed derivation of these equations is given by Treybal (1969).

Change in air humidity across the segment:

$$\frac{dY}{dZ} = -\frac{M_w F_G a_i}{G} \ln\left(\frac{1 - y_i}{1 - y}\right) \tag{1}$$

where the interfacial gas phase concentration is given by

$$y_i = 1 - (1 - y)\left(\frac{x}{x_i}\right)^{F_L/F_G} \tag{2}$$

The vapor-liquid equilibrium data for the triethylene glycol-

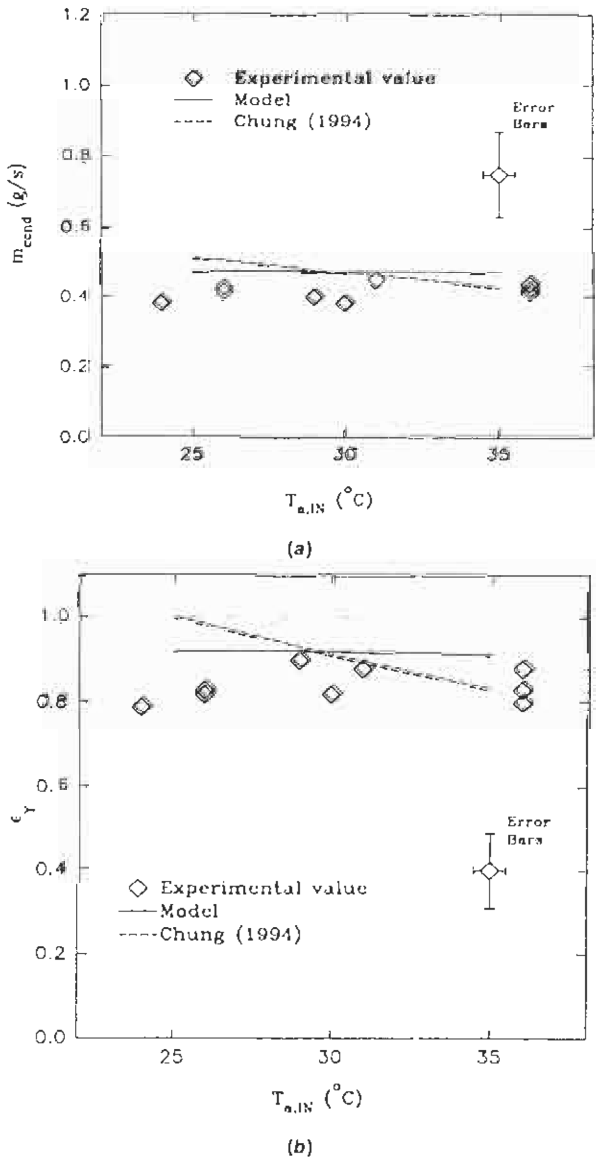


Fig. 5 Influence of inlet air temperature on: (*a*) condensation rate; (*b*) dehumidification effectiveness

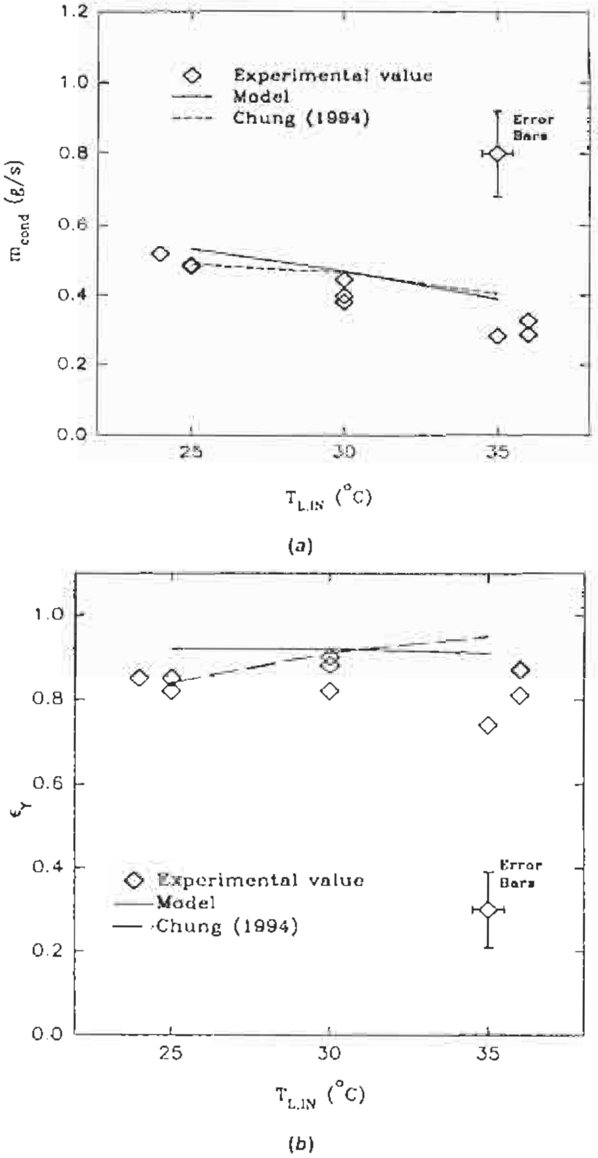


Fig. 6 Influence of inlet desiccant temperature on: (*a*) condensation rate; (*b*) dehumidification effectiveness

water system (Dow Chemical Company, 1992) were used along with Eq. (2) to solve for the interface concentrations in the gas and liquid phases.

Change in air temperature across the segment:

$$\frac{dT_a}{dZ} = \frac{-h_G a'(T_a - T_L)}{G(c_{p,a} + Yc_{p,v})} \tag{3}$$

where $h_G a'$ is the heat transfer coefficient corrected for simultaneous heat and mass transfer (Eq. (4)).

$$h_G a' = \frac{-Gc_{p,v}\dfrac{dY}{dZ}}{1 - \exp\left(\dfrac{Gc_{p,v}\dfrac{dY}{dZ}}{h_G a_i}\right)} \tag{4}$$

- Change in desiccant temperature across the segment:

$$dT_L = \frac{G}{c_{p,L}L}\{(c_{p,a} + Yc_{p,v})dT_a + [c_{p,v}(T_a - T_0) - c_{p,L}(T_L - T_0) + \lambda_0]dY\} \tag{5}$$

- Change in desiccant concentration across the segment:

$$dX = -\frac{G}{L}XdY \tag{6}$$

- Change in desiccant flow rate across the segment:

$$dL = GdY \tag{7}$$

Correlations for the mass transfer coefficients obtained for packed bed desiccant air dehumidifiers and regenerators are available in the literature. For instance, Potnis and Lenz (1996) presented dimensionless liquid phase mass transfer correlations based on experimental results on packed bed liquid desiccant contactors using a lithium bromide solution as the desiccant. However, since the present investigation use TEG as the desiccant, it was decided to use empirical correlations by Onda et al. (1967 and 1968) for the gas and liquid phase heat and mass transfer coefficients (Eqs. (8), (9), and (10)). These correlations were chosen as they had predicted data within ±20% for a range of operating and system conditions, using

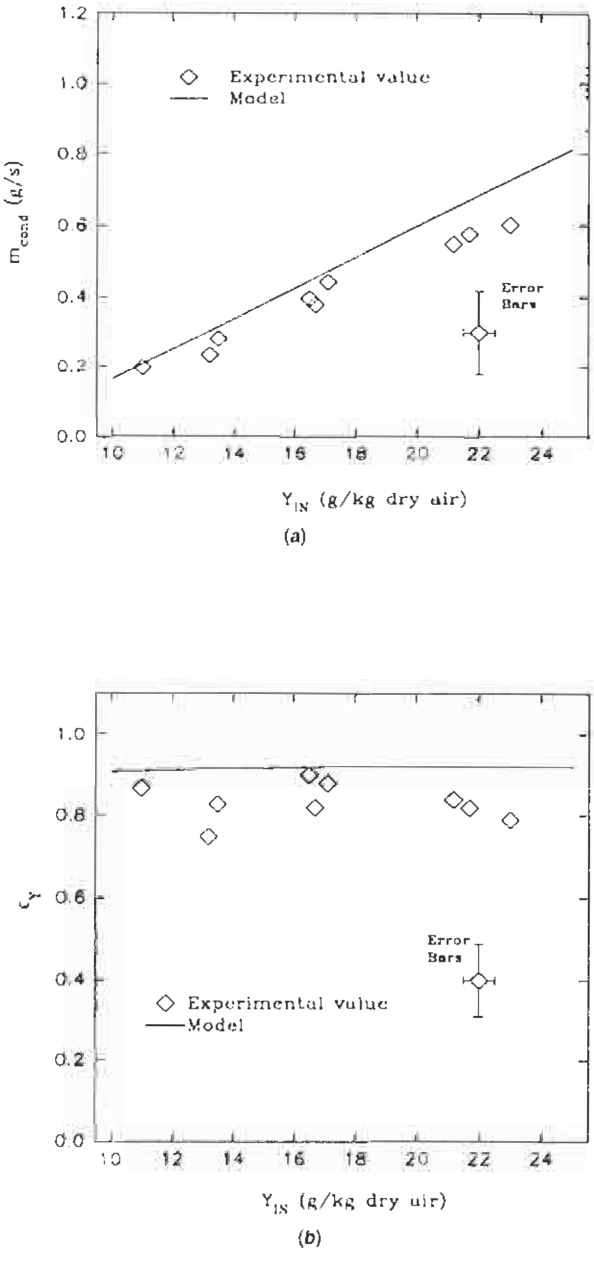
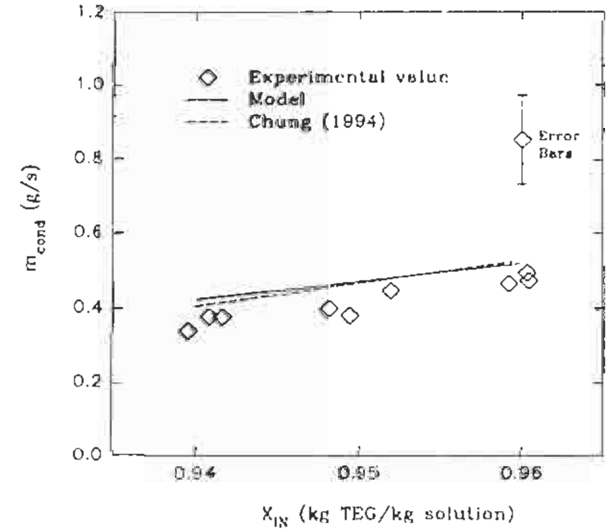
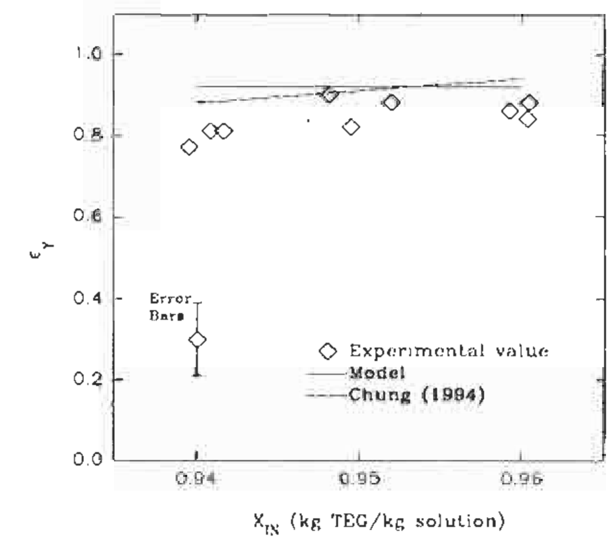


Fig. 7 Influence of inlet air humidity ratio on: (*a*) condensation rate; (*b*) dehumidification effectiveness



(a)



(b)

Fig. 8 Influence of inlet desiccant concentration on: (a) condensation rate; (b) dehumidification effectiveness

organic solvents, as well as water (Onda et al., 1967, and Onda et al., 1968).

$$k_L = 0.0051\left(\frac{\mu_L g}{\rho_L}\right)^{1/3}\left(\frac{L}{a_w \mu_L}\right)^{2/3}\left(\frac{\rho_L D_L}{\mu_L}\right)^{1/2}(a_t d_p)^{0.4} \quad (8)$$

$$\frac{a_w}{a_t} = 1 - \exp\left[-1.45\left(\frac{\gamma_c}{\gamma_L}\right)^{0.75}\left(\frac{L}{a_t \mu_L}\right)^{0.1} \times \left(\frac{L^2 a_t}{\rho_L^2 g}\right)^{-0.05}\left(\frac{L^2}{\rho_L \gamma_L a_t}\right)^{0.2}\right] \quad (9)$$

$$k_G = 5.23\,\frac{a_t D_G}{RT}\left(\frac{G}{a_t \mu_G}\right)^{0.7}\left(\frac{\mu_G}{\rho_G D_G}\right)^{1/3}(a_t d_p)^{0.2} \quad (10)$$

These k-type mass transfer coefficients can be converted to F-type coefficients by Eqs. (11) and (12) (Treybal, 1980 and Treybal, 1969).

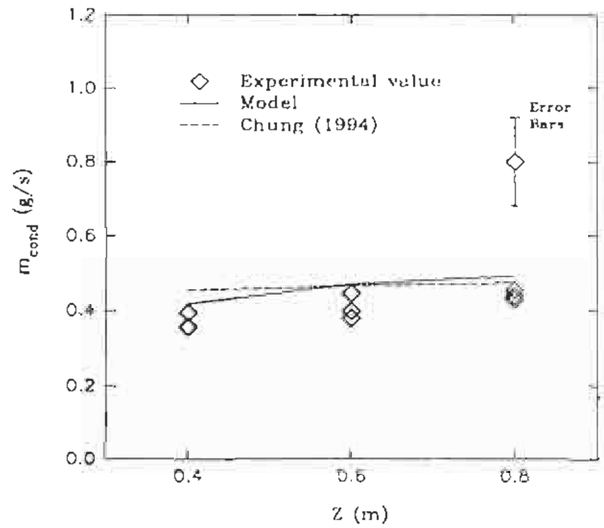
$$F_L = k_L x_{SM}\,\frac{\rho_L}{M_L} \quad (11)$$

$$F_G = k_G P \quad (12)$$

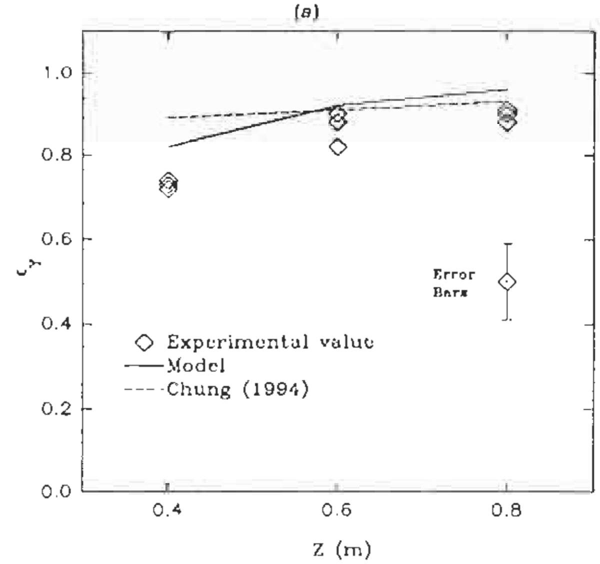
In the present investigation, the solutions was assumed to be sufficiently dilute with respect to water so that $x_{SM} = 1$. The gas phase heat transfer coefficient is found by applying the heat and mass transfer analogy (Eq. (13)).

$$j_h = \frac{h_G}{c_{p,a} G}\,\mathrm{Pr}^{2/3} = j_m = \frac{F_G M_a}{G}\,\mathrm{Sc}^{2/3} \quad (13)$$

A FORTRAN computer program was written to carry out the finite difference analysis with the bed height Z divided into 1000 segments. An under-relaxation iterative procedure was utilized to promote convergence. The criteria for convergence was ±0.05°C for the inlet desiccant temperature, and ±0.0001 kg TEG/kg solution for the inlet desiccant concentration.



(a)



(b)

Fig. 9 Influence of packed bed height on: (a) condensation rate; (b) dehumidification effectiveness

Results and Discussion

The results from the experimental study and theoretical modeling are depicted graphically in Figs. 3 to 9. These figures show the water condensation rate (moisture removal rate), m_{cond}, and the dehumidification effectiveness, ϵ_Y, as a function of air and desiccant flow rates and inlet temperatures, desiccant concentration, inlet air humidity ratio, and packed bed height. Uncertainties of the experimental measurements were calculated using the method by Kline and McClintock (1953). Error bars obtained from these calculations are shown in the figures. To cross-check the consistency of the data, a water mass balance across the dehumidifier was calculated, yielding $\pm 3\%$ deviation between the amount of water entering and leaving the dehumidifier. Similarly, an energy balance across the dehumidifier gave deviations of $\pm 6\%$. Thus, the assumption of adiabatic absorption is satisfactory. The pressure drop across the packed bed varied between 30 and 210 Pa/m packing, depending on the air flow rate.

The dehumidification effectiveness, ϵ_Y, is defined as the ratio of the actual change in moisture content of the air flowing through the dehumidifier to the maximum possible under the same operating conditions (Ullah et al., 1988).

$$\epsilon_Y = \frac{Y_{IN} - Y_{OUT}}{Y_{IN} - Y_{equ}} \tag{14}$$

Here, Y_{IN} and Y_{OUT} are the humidity ratios at the air inlet and outlet, respectively, and Y_{equ} is the humidity ratio in equilibrium with the desiccant at the local solution temperature and concentration. For counter flow arrangement, Y_{equ} would be the humidity ratio of the air in equilibrium with the desiccant at the desiccant inlet. Ullah et al. (1988) presented a curve fit for ϵ_Y as a function of the inlet desiccant and air temperatures, and the desiccant concentration for a given tower height, liquid and air flow rates, geometry, and desiccant. A more general correlation of ϵ_Y as a function of air and liquid flow rates, column and packing dimensions, and equilibrium properties of the desiccant was suggested by Chung (1994). This correlation was obtained using experimental data available in the literature. A parameter, π, representing the equilibrium properties of the desiccant was defined as the ratio of the vapor pressure depression to the vapor pressure of pure water (Eq. (15)). The correlation by Chung (1994) is given in Eq. (16).

$$\pi = \frac{p_w - p_L}{p_w} \tag{15}$$

$$\epsilon_Y = \frac{1 - \left\{ \dfrac{0.205\left(\dfrac{G_{IN}}{L_{IN}}\right)^{0.174} \exp\left[0.985\left(\dfrac{T_{a,IN}}{T_{L,IN}}\right)\right]}{(aZ)^{0.484}\pi^{1.686}} \right\}}{1 - \left\{ \dfrac{0.152 \exp\left[-0.686\left(\dfrac{T_{a,IN}}{T_{L,IN}}\right)\right]}{\pi^{3.388}} \right\}} \tag{16}$$

Results predicted with this correlation are shown together with the experimental findings from the present study in Figs. 3 to 9.

The experimental findings agree well with the predictions from the finite difference model described in this study. Only a slight discrepancy can be seen (approximately $\pm 15\%$), and the difference is consistently within the error bars of the experiments. Also, the repeatability of the experiments for each set of variables is better than what the error bars indicate. In cases where a discrepancy is apparent, the finite difference model generally over-predicts the performance of the dehumidifier. This is presumably due to the assumption that the area available for heat and mass transfer is equal to the total specific surface area of the packing. Even though the liquid flow rates are high as compared to the air flow rate, complete wetting of the packing is difficult to obtain. Therefore, the mass transfer area is less than the packing surface area. Also, it should be kept in mind that the correlations used for the transfer coefficients are empirical, and they were obtained for liquid-gas systems and packings other than those used in the present study. Although the correlation by Chung (1994) predicted the performance of the dehumidifier within 10% of the experimental findings, the finite difference model appears to predict the influence of design variables more accurately; i.e., trends shown by the experimental values were also predicted by the finite difference model. More specifically, the decrease in dehumidifier performance with increasing inlet air temperature (Fig. 5) predicted by the correlation of Chung (1994) cannot be seen from the experiments or the finite difference model. Also, Chung's correlation (1994) showed a dependency on the inlet desiccant temperature which was not found in the present investigation (Fig. 6).

The following variables were found to have the most significant effect on the dehumidifier performance: air flow rate, inlet desiccant temperature, inlet air humidity ratio, inlet desiccant concentration, and the area available for heat and mass transfer, i.e., the height of the packed bed. The condensation rate increased with the air flow rate (Fig. 3*a*). The change in humidity ratio through the tower decreased with an increase in the air flow rate due to the reduced residence time for the air in the dehumidifier. Hence, the dehumidification effectiveness decreased with an increase in the air flow rate (Fig. 3*b*). Increasing the desiccant temperature decreased the condensation rate (Fig. 6*a*). A higher desiccant temperature gives a lower potential for mass transfer in the dehumidifier resulting in a lower condensation rate. However, the dehumidification effectiveness was not affected by the desiccant temperature (Fig. 6*b*). This is because the lowest possible humidity ratio that can be obtained at the air outlet, Y_{equ}, is directly dependent on $T_{L,IN}$, making the effectiveness somewhat normalized with respect to the desiccant temperature. Similarly, the condensation rate increased slightly with the desiccant concentration, but the desiccant concentration did not change the effectiveness significantly (Fig. 8). An increase in the area available for heat and mass transfer, obtained by increasing the height of the packed bed, increased the condensation rate and the effectiveness (Fig. 9). A taller bed makes it possible for the air to reach a humidity ratio closer to the equilibrium value, Y_{equ}, at the air outlet.

The influence of design variables is summarized in Table 1 along with the experimental findings previously reported in the literature. The table shows the desiccant used, the parameters describing the performance, the independent variables and the ranges examined. Under each independent variable, the influence of the variable on the performance parameter is indicated by up and down arrows. As shown, the present study used liquid flow rates significantly higher than the previous studies with the exception of the work by Chung et al. (1995). Initial experiments at lower flow rates showed poor performance compared to the predictions from the theoretical model. This was presumably due to inadequate wetting of the packing. Therefore, it was decided to carry out the experiments at higher flow rates. Chung et al. (1995) had similar reasons for using high liquid flow rates. Indeed, Patnaik et al. (1990) and Chen et al. (1989) found the condensation rate to increase with liquid flow rate. They explained this partly by the increased wetting of the packing with increased flow rates. However, in the present study no dependency on the liquid flow rate was found. Hence, it may be concluded that the flow rates used in this study were sufficient to achieve maximum wetting for the present system. Potnis and Lenz (1996) also found the condensation rate to increase with increasing desiccant flow rate. In their study, this increase was due to the resistance to mass transfer being dominating in the liquid phase.

Patnaik et al. (1990) found that the condensation rate decreased as the inlet air temperature increased. They explained this dependency on the increase in liquid temperature due to sensible heat transfer from the air to the desiccant. The present

Table 1 Packed bed dehumidifier performance

Author	Desiccant	Performance Parameter	Independent Variables						
Oberg and Goswami (present study)	TEG		L (kg/m²-s)	X_{IN} (kg/kg)	$T_{L,IN}$ (°C)	G (kg/m²-s)	$T_{a,IN}$ (°C)	Y_{IN} (g/kg)	Z (m)
			4.5-6.5	0.94-0.96	25-35	0.5-2.0	25-35	11-22	0.4-0.8
		m_{cond}	↑↓	↑	↓	↑	↑↓	↑	↑
		ϵ_Y	↑↓	↑↓	↑↓	↓	↑↓	↑↓	↑
Chen et al. (1989)	LiCl		L* (kg/m²-s)	X_{IN} (kg/kg)	$T_{L,IN}$ (°C)	G* (kg/m²-s)	$T_{a,IN}$ (°C)	Y_{IN} (g/kg)	Z (m)
			0.1-1.0	0.3-0.4	26-39	0.2-1.2	25-35	17-22	0.1-0.6
		m_{cond}	↑	↑	↓	↑	↑	↑	↑
Chung et al. (1995)	TEG		L* (kg/m²-s)		X_{IN} (kg/kg)			G* (kg/m²-s)	
			6-11		0.9-0.95			0.8-1.2	
		ϵ_Y	↑		↑			↓	
		$K_G a$	↑		↑			↑	
McDonald et al. (1992)	mixture of LiCl and $CaCl_2$		L* (kg/m²-s)		X_{IN} (kg/kg)	$T_{L,IN}$* (°C)	$T_{a,IN}$* (°C)	Y_{IN} (g/kg)	Z* (m)
			1.5-3.9		0.4-0.45	28-43	31-36	21-41	0.5-1.4
		$T_{a,OUT}$	↓		↑↓	↑	↑↓	↑	↑↓
		Y_{OUT}	↓		↑↓	↑	↑↓	↑	↑↓
Patnaik et al. (1990)	LiBr		L* (kg/m²-s)		X_{IN} (kg/kg)	$T_{L,IN}$ (°C)	G* (kg/m²-s)	$T_{a,IN}$ (°C)	Y_{IN} (g/kg)
			0.6-1.7		0.45-0.58	24-32	1.3-1.9	24-36	9-22
		m_{cond}	↑		↑	↑↓	↑↓	↓	↑
Potnis and Lenz (1996)	LiBr		L (kg/m²-s)						
			0.8-2.5						
		m_{cond}	↑						

* Values converted to this unit by present authors.
↑ performance parameter increases with increasing variable
↓ performance parameter decreases with increasing variable
↑↓ variable has no significant effect on the performance parameter

study showed no dependency of the condensation rate on the inlet air temperature. The reason for this observation is that the desiccant flow rate was significantly higher than the air flow rate. Therefore, the sensible heat transfer from the air to the desiccant was too small to increase the desiccant temperature significantly. Chen et al. (1989) found that the condensation rate increased with the inlet air temperature. This was probably due to the fact that they used relative humidity as a variable instead of humidity ratio. For a constant relative humidity, the warmer the air, the higher the humidity ratio, which gives a higher condensation rate.

Based on the comparison between the experimental and the theoretical results in this study, it is believed that the finite difference model described herein gives good predictions for design and performance simulation.

Conclusions

Design variables found to have the greatest impact on the performance of the dehumidifier are the air flow rate and the humidity ratio, the desiccant temperature and concentration, and the packed bed height. The liquid flow rate and the inlet air temperature did not have a significant effect on the dehumidifier performance; however, the liquid flow rate must be high enough to ensure wetting of the packing. In this study, the liquid flow rate was higher as compared to the flow rates used in the majority of studies previously reported.

The results obtained in this study compare reasonably well with other experimental investigations. Contrary to the findings of this study, some studies suggest that the liquid flow rate and air temperature influence the performance. This is presumably due to the lower liquid flow rate used in those investigations.

The dehumidifier performance predicted with the finite difference model described in this paper shows a good agreement with the experimental findings. Thus, for a detailed study of the absorption process, this model gives accurate performance predictions based on fundamental equations, minimizing the assumptions and use of empirical correlations.

Acknowledgment

This work was funded by the Florida Energy Office, Department of Community Affairs, Tallahassee, Florida. In addition, the authors would like to express their gratitude to Rauschert Industries, Madisonville, Tennessee, for the generous donation of the packing material used in this study.

References

Chen, L. C., Kuo, C. L., and Shyu, R. J., 1989, "The Performance of a Packed Bed Dehumidifier for Solar Liquid Desiccant Systems," *Proceedings of the 11th Annual ASME Solar Energy Conference*, San Diego, California, pp. 371–377.

Chung, T.-W., Ghosh, T. K., Hines, A. L., and Novosel, D., 1995, "Dehumidification of Moist Air With Simultaneous Removal of Selected Pollutants by Triethylene Glycol Solutions in a Packed-Bed Absorber," *Separation Science and Technology*, Vol. 30, pp. 1807–1832.

Chung, T.-W., 1994, "Predictions of the Moisture Removal Efficiencies for Packed-Bed Dehumidification Systems," *Gas Separation & Purification*, Vol. 8, pp. 265–268.

Chung, T.-W., Ghosh, T. K., and Hines, A. L., 1993, "Dehumidification of Air by Aqueous Lithium Chloride in a Packed Column," *Separation Science and Technology*, Vol. 28, pp. 533–550.

Dow Chemical Company, 1992, "*A Guide to Glycols*," Dow Chemical Company, Midland, Michigan.

Factor, H. M., and Grossman, G., 1980, "A Packed Bed Dehumidifier/Regenerator for Solar Air Conditioning With Liquid Desiccants," ASME JOURNAL OF SOLAR ENERGY ENGINEERING, Vol. 24, pp. 541–550.

Gandhidasan, P., 1994, "Performance Analysis of an Open-Cycle Liquid Desiccant Cooling System using Solar Energy for Regeneration," *International Journal of Refrigeration*, Vol. 17, pp. 475–480.

Gandhidasan, P., Ullah, M. R., and Kettleborough, C. F., 1987, "Analysis of Heat and Mass Transfer Between a Desiccant-Air System in a Packed Tower," ASME JOURNAL OF SOLAR ENERGY ENGINEERING, Vol. 109, pp. 89–93.

Harriman, L. G., 1990, *The Dehumidification Handbook*, 2nd edition, Munters Cargocaire, Amesbury, MA.

Howell, J. R., 1987, "A Survey of Active Solar Cooling Methods," *Progress in Solar Engineering*, Goswami, D. Y., editor, pp. 171–182.

Johannsen, A., 1984, "Performance Simulation of a Solar Air-Conditioning System With Liquid Desiccant," *International Journal of Ambient Energy*, Vol. 5, pp. 59–89.

Khan, A. Y., 1996, "Parametric Analysis of Heat and Mass Transfer Performance of a Packed Type Liquid Desiccant Absorber at Part-Load Operating Conditions," *ASHRAE Transactions*, Vol. 102, pt. 1, pp. 349–357.

Khan, A. Y., 1994, "Sensitivity Analysis and Component Modeling of a Packed-Type Liquid Desiccant System at Partial Load Operating Conditions," *International Journal of Energy Research*, Vol. 18, pp. 643–655.

Kinsara, A. A., Elsayed, M. M., and Al-Rabghi, O. M., 1996, "Proposed Energy-Efficient Air-Conditioning System Using Liquid Desiccant," *Applied Thermal Engineering*, Vol. 16 No. 10, pp. 791–806.

Kline, S. J., and McClintock, F. A., 1953, "Describing Uncertainties in Single-Sample Experiments," *Mechanical Engineering*, Vol. 75, pp. 3–8.

Mahmoud, K. G., Ball, H. D., 1988, "Liquid Desiccant Systems for Cooling Applications," *Proceedings of the Intersociety Energy Conversion Engineering Conference*, Denver, Colorado, pp. 149–152.

Marsala, J., Lowenstein, A., and Ryan, W. A., 1989, "Liquid Desiccant for Residential Applications," *ASHRAE Transactions*, Vol. 95, pt. 1, pp. 828–834.

McDonald, B., Waugaman, D. G., and Kettleborough, C. F., 1992, "A Statistical Analysis of a Packed Tower Dehumidifier," *Drying Technology*, Vol. 10, pp. 223–237.

Onda, K., Takeuchi, H., and Okumoto, Y., 1968, "Mass Transfer Coefficients Between Gas and Liquid Phases in Packed Columns," *Journal of Chemical Engineering of Japan*, Vol. 1, pp. 56–62.

Onda, K., Takeuchi, H., and Koyama, Y., 1967, *Kagaku Kōgaku*, Vol. 31, pp. 126–134.

Patnaik, S., Lenz, T. G., and Löf, G. O. G., 1990, "Performance Studies for an Experimental Solar Open-Cycle Liquid Desiccant Air Dehumidification System," *Solar Energy*, Vol. 44, pp. 123–135.

Potnis, S. V., and Lenz, T. G., 1996, "Dimensionless Mass Transfer Correlations for Packed-Bed Liquid-Desiccant Contactors," *Industrial and Engineering Chemistry Research*, Vol. 35, pp. 4185–4193.

Robison, H., 1977, "Liquid Sorbent Solar Air Conditioner," *Alternative Energy Sources*, 2, Veziroğlu, T. N., editor, Hemisphere Publishing Company, Washington, pp. 761–779.

Scalabrin, G., and Scaltriti, G., 1990, "A Liquid Sorption—Desorption System for Air Conditioning With Heat at Lower Temperature," ASME JOURNAL OF SOLAR ENERGY ENGINEERING, Vol. 112, pp. 70–75.

Sick, F., Buschulte, T. K., Klein, P., Northey, P., and Duffie, J. A., 1988, "Analysis of the Seasonal Performance of Hybrid Liquid Desiccant Cooling Systems," *Solar Energy*, Vol. 40, pp. 211–217.

Thornbloom, M., and Nimmo, B., 1995, "An Economic Analysis of a Solar Open Cycle Desiccant Dehumidification System," *Proceedings of the 13th Annual ASME Solar Engineering Conference*, Hawaii, Vol. 1, pp. 705–709.

Treybal, R. E., 1980, *Mass Transfer Operations*, 3rd edition, McGraw-Hill, New York.

Treybal, R. E., 1969, "Adiabatic Gas Absorption and Stripping in Packed Towers," *Industrial Engineering and Chemistry*, Vol. 61 No. 7, pp. 36–41.

Ullah, M. R., Kettleborough, C. F., and Gandhidasan, P., 1988, "Effectiveness of Moisture Removal for an Adiabatic Counterflow Packed Tower Absorber Operating with $CaCl_2$-Air Contact System," ASME JOURNAL OF SOLAR ENERGY ENGINEERING, Vol. 110, pp. 98–101.