

Design theory for rotary heat and mass exchangers—II. Effectiveness–number-of-transfer-units method for rotary heat and mass exchangers

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Abstract—Analytical performance correlations for rotary heat and mass exchangers with infinite transfer coefficients have been presented in part I. A finite-difference model for performance prediction of rotary dehumidifiers with finite transfer coefficients is used in combination with the ideal dehumidifier model to establish effectiveness correlations. Correlations for the humidity and enthalpy effectiveness for silica gel regenerators are given as functions of the dehumidifier number of transfer units (NTU). An Effectiveness–NTU model, incorporating the correlations for the effectiveness and ideal dehumidifier performance, allows rapid calculation of the dehumidifier performance. The correlations are valid for a wide range of operating conditions and account for the effect of unbalanced flow and high Lewis numbers.

1. INTRODUCTION

OPEN cycle desiccant cooling systems utilize dehumidifiers to provide a cool and dry air stream for use in industrial processes [1] and residential air conditioning [2]. Regeneration energy required to drive these systems may be supplied by combustion of natural gas [3] or may be provided by flat plate solar collectors [4]. To assess the performance of solar-fired desiccant cooling systems at a particular location, the year-round operation of these systems needs to be simulated using weather data representative for the climate at that location. Such system simulations require models that allow rapid and accurate prediction of individual component performance [5].

Models for dehumidifiers based on finite-difference schemes [6, 7] are not suited for use in yearly simulations because of the large computation time required for performance prediction. Analytical models exist [8, 9], but the approximations involved limit the range of applicability to dehumidifiers with a Lewis number equal to one. Maclaine-cross and Banks [9] and Banks [10] have introduced the idea of using an effectiveness technique to compute the outlet states of the heat and mass exchangers. Their model is based on the analogy theory, in which the heat and mass exchanger is related to a regenerator for heat transfer alone. In the analogy theory, the dehumidifier number of transfer units (NTU) are transformed into analogous dimensionless transfer coefficients. The temperature effectiveness of a thermal regenerator with these analogous transfer coefficients will then approximate the effectiveness of the combined heat and mass potentials of the corresponding dehumidifier [9–11]. These potential effectivenesses do not approach unity with increasing transfer coefficients.

This paper presents an alternative effectiveness

technique to compute dehumidifier performance. An analytical theory for modeling rotary dehumidifiers with infinite transfer coefficients is presented in part I of this paper [11]. A finite-difference model for performance prediction of rotary dehumidifiers with finite transfer coefficients is used in combination with the ideal dehumidifier model to establish effectiveness correlations. Correlations for the humidity and enthalpy effectiveness for silica gel regenerators are given as functions of the NTU. These correlations allow rapid prediction of the dehumidifier performance and are valid for a wide range of inlet conditions, operating parameters and transfer coefficients. The accuracy of the presented model is discussed and compared to existing analytical models.

2. DEPENDENT VARIABLES AND DIMENSIONLESS GROUPS

The nomenclature and governing conservation and transfer rate equations for the rotary heat and mass exchanger are given in part I [11, equations (1)–(6)]. Air stream mass flow rates, wheel rotational speed and mass of desiccant are combined into two dimensionless capacitance rate parameters (Γ_j). The NTU that appear in the expressions for the rate of transfer of heat and mass are defined as conventional:

$$NTU_{i,j} = \frac{h_{i,j} A_j}{\dot{m}_j c_p} \quad (1)$$

$$NTU_{w,j} = \frac{h_{w,j} A_j}{\dot{m}_j}$$

The moist air specific heat, c_p , is defined as

$$c_p = \left(\frac{\partial i_f}{\partial t_f} \right)_{\bar{w}_f} \quad (2)$$

NOMENCLATURE

A_j	total heat transfer area of the dehumidifier matrix in period j [area]	w	moist air humidity ratio [dimensionless]
c_p	moist air specific heat [energy/(dry mass-temperature)]	W	matrix water content [dimensionless].
C^*	ratio of minimum to maximum fluid capacity rates for a thermal regenerator; or, ratio of capacitance rate parameters [dimensionless]	Greek symbols	
F_{Le}	correction factor for NTU_w based on Lewis number [dimensionless]	Γ_j	j th capacitance rate parameter of the rotary dehumidifier [dimensionless]
h^*	ratio of the isothermal differential heat of adsorption to the heat of vaporization [dimensionless]	δ	perturbation of variable
h_i	heat transfer coefficient [energy/(area-time-temperature)]	ε	effectiveness [dimensionless]
h_w	mass transfer coefficient [mass/(area-time)]	ε_{cf}	counterflow heat exchanger effectiveness [dimensionless]
i	moist air enthalpy [energy/dry mass]	ε_i	enthalpy effectiveness of the rotary dehumidifier [dimensionless]
i_s	isothermal differential heat of adsorption [energy/dry mass]	ε_R	rotary heat exchanger effectiveness [dimensionless]
i_v	water vapor enthalpy [energy/mass]	ε_t	temperature effectiveness of the rotary dehumidifier [dimensionless]
i_{vap}	heat of vaporization [energy/mass]	ε_w	humidity ratio effectiveness of the rotary dehumidifier [dimensionless]
I	matrix enthalpy [energy/dry mass]	θ	time
K	effectiveness correction factor [dimensionless]	θ_j	duration of period j [time]
Le	overall Lewis number [dimensionless]	λ_i	i th characteristic wave speed [dimensionless].
\dot{m}	air stream mass flow rate [mass/time]	Subscripts	
NTU	number of transfer units [dimensionless]	f	evaluated at fluid state
NTU_0	overall number of transfer units [dimensionless]	i	enthalpy
NTU_i	number of transfer units for heat transfer, $h_i A / \dot{m} c_p$ [dimensionless]	id	ideal outlet state
NTU_w	number of transfer units for mass transfer, $h_w A / \dot{m}$ [dimensionless]	j	period index
P_{ws}	saturation pressure of water vapor [force/area]	m	evaluated at, or in equilibrium with, the matrix state
r	relative humidity of moist air [dimensionless]	R	regeneration
t	temperature	t	heat transfer or temperature
		v	vapor
		vap	vaporization
		w	mass transfer or moisture.
		Superscripts	
		.	rate
		—	average value for a period.

The Lewis number for period j of the regenerator is defined as the ratio of the overall resistance for mass transfer to the overall resistance for heat transfer:

$$Le_j = \frac{NTU_{t,j}}{NTU_{w,j}} \quad (3)$$

In the analysis, it is assumed that the Lewis numbers in both periods are equal.

Dehumidifier performance is determined by the outlet state of the processed air stream. The solutions to the conservation and transfer rate equations and the initial and boundary conditions may be used to compute the mean outlet state properties of the two air

streams. For given desiccant and fluid property relationships, correlations for the fluid stream outlet states can be expressed in the following functional form:

$$\bar{w}_{j,2} = \bar{w}_{j,2}(t_{11}, w_{11}, t_{21}, w_{21}, \Gamma_1, \Gamma_2, NTU_{t,1}, NTU_{t,2}, Le) \quad j = 1, 2. \quad (4)$$

$$\bar{t}_{j,2} = \bar{t}_{j,2}(t_{11}, w_{11}, t_{21}, w_{21}, \Gamma_1, \Gamma_2, NTU_{t,1}, NTU_{t,2}, Le)$$

The periodic initial and boundary conditions show that

the outlet states are functions of the inlet conditions and capacitance rate parameters Γ_j . The NTU for heat transfer and the Lewis number enter into the functional form (4) through the expressions for the rate of transfer of heat and mass within the desiccant matrix.

3. THERMODYNAMIC PROPERTIES OF MOIST AIR AND SILICA GEL

The effectiveness correlations that are presented in this text are valid for nominal silica gel as the desiccant and a mixture of water vapor and air as the fluid streams. In the analysis, moist air is treated as an ideal gas mixture [12]. The various equations that are used for evaluation of the properties of moist air are obtained from [13]. The following conventional assumptions are made with respect to property evaluation of desiccants [14]:

- (1) the solid adsorbent is completely inert. All its thermodynamic properties are the same in the presence as in the absence of the adsorbate (adsorbed phase).
- (2) surface area, structure and specific heat of the adsorbent are independent of temperature and pressure.

Table 1 defines the state properties of nominal silica gel, as established by Brandemuehl [15].

4. THE ϵ -NTU METHOD FOR ROTARY HEAT AND MASS EXCHANGERS

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

$$\epsilon_w = \frac{w_{11} - \bar{w}_{12}}{w_{11} - (\bar{w}_{12})_{id}}, \quad (5a)$$

Table 1. Thermodynamic properties of silica gel (from ref. [15])

Heat of sorption	
$h^* = 1.0 + 0.2843 \exp(-10.28 W_m)$	
$i_s = h^* i_{vap}$	
Equilibrium isotherm	
$r = (2.112 W_m)^{h^*} (29.91 P_{wa})^{h^* - 1}$	
Enthalpy	
$I_m = 921.096 t_m + \int_0^{w_m} (i_v - i_s) dW_m$	
Units	
$W_m = [\text{kg/kg dry desiccant}]$	
$I_m = [\text{J/kg dry desiccant}]$	

and for enthalpy

$$\epsilon_i = \frac{i_{11} - \bar{i}_{12}}{i_{11} - (\bar{i}_{12})_{id}}. \quad (5b)$$

The subscript 'id' refers to the performance of a dehumidifier operating at the same inlet conditions, the same Γ_j parameters, and with infinite overall transfer coefficients for heat and mass:

$$\begin{aligned} (\bar{w}_{12})_{id} &= \lim_{\substack{NTU_{w,j} \rightarrow \infty \\ NTU_{i,j} \rightarrow \infty}} (\bar{w}_{12}) \\ (\bar{i}_{12})_{id} &= \lim_{\substack{NTU_{w,j} \rightarrow \infty \\ NTU_{i,j} \rightarrow \infty}} (\bar{i}_{12}). \end{aligned} \quad (6)$$

In part I [11], it is shown that the outlet state of an ideal dehumidifier can be obtained analytically without the need to solve the conservation and transfer rate equations and to take the limit (6). To predict the performance of a dehumidifier with finite transfer coefficients, a general computer code, MOSHMX, has been developed by Maclaine-cross [6] that solves these governing equations numerically. The program is based on a second-order finite-difference method with extrapolation to zero grid size. In this paper, the ideal dehumidifier model is combined with performance evaluation by MOSHMX [6] to establish empirical effectiveness correlations for humidity and enthalpy. Using the definition of effectiveness (5) and the correlations provided by the ideal dehumidifier model, the outlet states may then be calculated provided that ϵ_w and ϵ_i are readily computable. Equations (4) show that the effectivenesses are functions of the inlet conditions, operating parameters, transfer numbers and the Lewis number.

4.1. Effectiveness correlation for humidity ratio

The effectiveness ϵ_R of a rotary regenerator with heat transfer alone may be correlated by [16]:

$$\epsilon_R = K_\epsilon \epsilon_{cf} \quad (7)$$

where ϵ_{cf} is the expression for the effectiveness of a counterflow recuperator:

$$\epsilon_{cf} = \frac{1 - \exp[-NTU_0(1 - C^*)]}{1 - C^* \exp[-NTU_0(1 - C^*)]}. \quad (8)$$

C^* is the heat capacity rate ratio, which is always less than or equal to 1, and NTU_0 is the overall NTU. K_ϵ is an empirical correction factor less than 1, which takes into account the rotary nature of the regenerator [16].

In a dehumidifier, the humidity ratio of the air streams can only change by mass transfer. Similarly, in a rotary heat exchanger the temperature of the air streams can only change by heat transfer. Hence, it may be expected by analogy that ϵ_w may be correlated with an expression analogous to equation (7). Humidity ratio effectiveness values obtained from the computer code MOSHMX [6] and the ideal dehumidifier model [11] are compared to the expression for the counterflow recuperator effectiveness (8) in Fig. 1. In

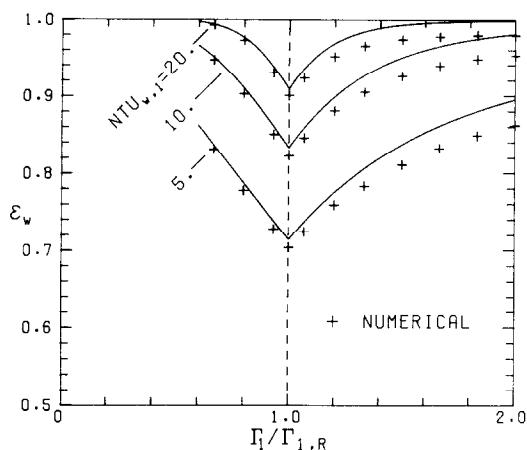


Fig. 1. Comparison of dehumidifier effectiveness values with the counterflow heat exchanger effectiveness correlation.

this figure, C^* and NTU_0 are set as follows:

for $\Gamma_1 \leq \Gamma_{1,R}$,

$$C^* = \frac{\Gamma_1}{\Gamma_{1,R}} \quad (9a)$$

$$\frac{1}{NTU_0} = \frac{C^*}{NTU_{w,1}} + \frac{1}{NTU_{w,2}}$$

$\Gamma_1 > \Gamma_{1,R}$,

$$C^* = \frac{\Gamma_{1,R}}{\Gamma_1} \quad (9b)$$

$$\frac{1}{NTU_0} = \frac{1}{NTU_{w,1}} + \frac{C^*}{NTU_{w,2}}$$

where $\Gamma_{1,R}$ is given by the regeneration line correlation [11, equation (18) and Fig. 7]. For Γ_1 equal to $\Gamma_{1,R}$, each desiccant particle will just reach complete equilibrium with the regeneration air inlet state during the regeneration period, for infinite transfer coefficients for heat and mass. $\Gamma_{1,R}$ is dependent on the inlet conditions and capacitance rate factor Γ_2 . A functional form for $\Gamma_{1,R}$ is given in part I of this paper [11].

Figure 1 shows that there is a similarity between the ratio $\Gamma_1/\Gamma_{1,R}$ for a heat and mass exchanger and the heat capacity rate ratio, C^* , for a counterflow recuperator. The humidity ratio effectiveness is a minimum at $\Gamma_1 = \Gamma_{1,R}$. In part I it is shown that the derivative of $(\bar{w}_{12})_{id}$ is discontinuous with respect to Γ_1 for $\Gamma_1 = \Gamma_{1,R}$. Finite transfer coefficients tend to smooth abrupt changes of the outlet state properties. Hence, the deviation of \bar{w}_{12} from $(\bar{w}_{12})_{id}$ will be largest for $\Gamma_1 = \Gamma_{1,R}$. Figure 1 shows further that, for $\Gamma_1 < \Gamma_{1,R}$ the agreement between equations (8, 9) and the computed points is within a few percent, but for $\Gamma_1 > \Gamma_{1,R}$, equation (8) overpredicts the actual humidity effectiveness.

Similar to the counterflow heat regenerator, the deviation of ε_w from correlation (8) and (9) is absorbed in the empirical correction factor K_e . For the case in

which $\Gamma_1 < \Gamma_{1,R}$, a constant value for K_e is an adequate choice as follows from Fig. 1.

If the dehumidifier is operated far enough into the region of complete regeneration (i.e. $\Gamma_1 > \Gamma_{1,R}$), eventually ε_w will become independent of Γ_2 and $NTU_{w,2}$. The adsorption process during the processing period will then resemble the transient response of a porous matrix, initially at uniform state. This behavior is analogous to the insulated duct problem for heat transfer alone [17]. In order to obtain an analogy with the insulated duct problem, a dimensionless time has to be introduced depending on the speed at which the inlet state propagates through the matrix. In the region of complete regeneration, only the first wave is important for a heat and mass exchanger, hence the corresponding dimensionless time is given by $\bar{\lambda}_1/\Gamma_1$, where $\bar{\lambda}_1$ is the average wave speed of the first rarefaction wave during processing [11]. This ratio is used in the expression for the correction factor K_e for $\Gamma_1 > \Gamma_{1,R}$. However, the analogy with the insulated duct problem only holds if Γ_1 is larger than $\Gamma_{1,R}$ to an extent depending on the transfer parameter $NTU_{w,1}$. In the effectiveness correlation, it is therefore assumed that the correction factor K_e changes smoothly with Γ_1 , from the constant value valid outside the region of complete regeneration to the limiting value provided by the insulated duct analogy. The final moisture effectiveness is given in Table 2.

To incorporate the effect of the Lewis number into equations (7)–(9), an effective NTU, $NTU_{w,j}^*$, is defined for each period, based on an empirical correction factor F_{Le} that is a function of the Lewis number. Numerical constants that appear in the expressions in Table 2 reflect the empirical character of the correlation.

The accuracy of this correlation has been estimated by comparing effectiveness values obtained from that correlation with effectiveness values provided by MOSHMX [6] and the ideal dehumidifier theory [11]. Based on 62 points covering the range of parameters listed in Table 3, the mean value of the error in humidity effectiveness (i.e. the bias) is 0.002 and the standard deviation is 0.013.

4.2. Effectiveness correlation for enthalpy

In a dehumidifier, enthalpy is exchanged by both heat and mass transfer while moisture is exchanged only by the latter. As a result, correlating the enthalpy effectiveness, ε_h , is far more challenging than correlating the humidity ratio effectiveness, ε_w . However, the ratio of the change of enthalpy of the process air stream from inlet to exit to the enthalpy difference of the inlet air streams is typically about 0.1 for well-defined dehumidifiers. An accurate expression for ε_h is therefore not needed. The error in exit temperature of the process air stream, $\delta \bar{t}_{12}$, due to an error in enthalpy effectiveness, $\delta \varepsilon_h$, may be expressed as [18]

$$\delta \bar{t}_{12} \simeq 10 \delta \varepsilon_h \quad (10)$$

where the units of δt are °C. Enthalpy effectiveness values obtained from MOSHMX [6] and the ideal

Table 2. Correlation for process-air humidity ratio effectiveness

$\epsilon_w = K_\epsilon \epsilon_{cf}$	
for $C^* < 1$, $\epsilon_{cf} = \frac{1 - \exp[-NTU_{w,0}(1 - C^*)]}{1 - C^* \exp[-NTU_{w,0}(1 - C^*)]}$	for $\Gamma_1 \leq \Gamma_{1,R}$, $K_\epsilon = 0.99$
$C^* = 1$, $\epsilon_{cf} = \frac{NTU_{w,0}}{1 + NTU_{w,0}}$	$\Gamma_1 > \Gamma_{1,R}$, $K_\epsilon = K + (0.99 - K) \exp\left[-\frac{\left(\frac{1}{C^*} - 1\right)}{b}\right]$
for $\Gamma_1 \leq \Gamma_{1,R}$, $C^* = \Gamma_1/\Gamma_{1,R}$	$K = a \exp\left[-\frac{\left(\frac{\bar{\lambda}_1}{\Gamma_1} - 1\right)^2}{80.0}\right]$
$\frac{1}{NTU_{w,0}} = \frac{C^*}{NTU_{w,1}^*} + \frac{1}{NTU_{w,2}^*}$	$a = 1.0 - \exp\left(-\frac{NTU_{w,1}^*}{1.52}\right)$
$\Gamma_1 > \Gamma_{1,R}$, $C^* = \Gamma_{1,R}/\Gamma_1$	$b = 4.6 \ln\left(\frac{5.88}{NTU_{w,1}^* 0.48}\right)$
$\frac{1}{NTU_{w,0}} = \frac{1}{NTU_{w,1}^*} + \frac{C^*}{NTU_{w,2}^*}$	$\bar{\lambda}_1 = \frac{1}{2}(\lambda_{1,21} + \lambda_{1,in})$
$NTU_{w,j} = NTU_{t,j}/Le$	
$F_{Le} = 1.0 + 0.498 [\ln(Le)]^{1/1.546}$	
$NTU_{w,j}^* = NTU_{w,j} F_{Le}$	

dehumidifier model [11] show that, for $Le = 1$ and $NTU_{t,j} > 10$, the effectiveness ϵ_i differs from 1.0 by not more than 4%. Hence if the enthalpy effectiveness is assumed to be constant and equal to unity, then it follows from equation (10) that the temperature error is within 0.4°C.

If the Lewis number is higher than unity, the enthalpy effectiveness may be substantially higher than 1.0. For increasing resistance to mass transfer, the dehumidifier performance approaches that of a thermal regenerator. It may be shown that, for the case of heat transfer alone, the enthalpy change of the process air stream is typically about 50% higher than for the case of both heat and mass transfer [6, 16]. The parameter of importance to include the Lewis number effect therefore is

$$\epsilon_i = \epsilon_i \left[\frac{(\Delta \bar{i})_{id}}{(\Delta \bar{i}_{t,w})_{id}} \right], \tag{11}$$

$(\Delta \bar{i})_{id}$ is the enthalpy change of the process air stream

Table 3. Investigated range for effectiveness correlations		
t_{11}	20.0...35.0	[°C]
w_{11}	0.010...0.020	[kg kg ⁻¹]
t_{21}	70.0...100.0	[°C]
w_{21}	0.010...0.020	[kg kg ⁻¹]
Γ_1	0.10...0.40	
Γ_2	0.10...0.40	
$NTU_{t,1}$	10.0...40.0	
$NTU_{t,2}$	0.5...2.0	
$NTU_{t,1}$		
Le	1.0...20.0	

from inlet exit for heat transfer alone while $(\Delta \bar{i}_{t,w})_{id}$ is the enthalpy change for both heat and mass transfer. Infinite transfer coefficients may be assumed in both cases, thus $(\Delta \bar{i})_{id}$ may be calculated using the ideal regenerator theory [16], and $(\Delta \bar{i}_{t,w})_{id}$ is calculated using the ideal dehumidifier theory.

Table 4. Correlation for process-air enthalpy effectiveness

$\epsilon_i = 1.0 + ab \left[\frac{(\Delta i)_{id}}{(\Delta i_{t,w})_{id}} - 1.0 \right]$
$(\Delta i_{t,w})_{id} = (\bar{i}_{12})_{id} - i_{11}$
$(\Delta i)_{id} = \Gamma_1 c_m (t_{21} - t_{11})$
$c_m = 921.0 + 4187.0 \bar{W}_m$
$\bar{W}_m = W_m(\bar{t}_t, \bar{w}_t) \quad \text{where} \quad \bar{t}_t = \frac{t_{11} + t_{21}}{2}$
$\bar{w}_t = \frac{w_{11} + w_{21}}{2}$
$a = \exp\left[-0.5\left(\frac{NTU_{t,0}}{10.0} - 1.0\right)\right]$
$b = 1.0 - \exp\left(-\frac{Le - 1}{25.0}\right)$
for $\Gamma_1 \leq \Gamma_{1,R}$, $C^* = \Gamma_1/\Gamma_{1,R}$
$\frac{1}{NTU_{t,0}} = \frac{C^*}{NTU_{t,1}} + \frac{1}{NTU_{t,2}}$
$\Gamma_1 > \Gamma_{1,R}$, $C^* = \Gamma_{1,R}/\Gamma_1$
$\frac{1}{NTU_{t,0}} = \frac{1}{NTU_{t,1}} + \frac{C^*}{NTU_{t,2}}$

The final correlation for the enthalpy effectiveness is given in Table 4. Numerical constants that appear in the expressions reflect the empirical character of the proposed correlation. The same accuracy analysis as conducted for humidity ratio shows that the mean error for this effectiveness is 0.006 and the standard deviation is 0.035. The temperature effectiveness, ϵ_t , may now be calculated based on the correlations for ϵ_w and ϵ_t . The mean error for this outlet temperature effectiveness, based on 62 points, is 0.003 and the standard deviation is 0.015. These values are again valid for the range of parameters as given in Table 3.

5. EFFECTIVENESS CHARTS

Based on the correlations presented in Tables 2 and 4, the effectiveness may be plotted against the overall NTU with the inlet conditions and the capacitance rate factors, Γ_j , as parameters. Figures 2 and 3 show such effectiveness charts for a rotary silica gel dehumidifier. The overall NTU for heat transfer, $NTU_{t,0}$, and capacitance rate ratio, C^* , are defined as

for $\Gamma_1 \leq \Gamma_{1,R}$

$$C^* = \Gamma_1 / \Gamma_{1,R}$$

$$\frac{1}{NTU_{t,0}} = \frac{C^*}{NTU_{t,1}} + \frac{1}{NTU_{t,2}}$$

for $\Gamma_1 > \Gamma_{1,R}$

$$C^* = \Gamma_{1,R} / \Gamma_1$$

$$\frac{1}{NTU_{t,0}} = \frac{1}{NTU_{t,1}} + \frac{C^*}{NTU_{t,2}},$$

(12)

Figure 2 shows the temperature and humidity effectiveness as functions of the overall NTU for heat transfer, with C^* as parameter. C^* incorporates the effect of inlet conditions, wheel speed and air stream mass flow rates. $\Gamma_{1,R}$ is approximately proportional to Γ_2 , increases with process air temperature and humidity, and decreases with increasing regeneration

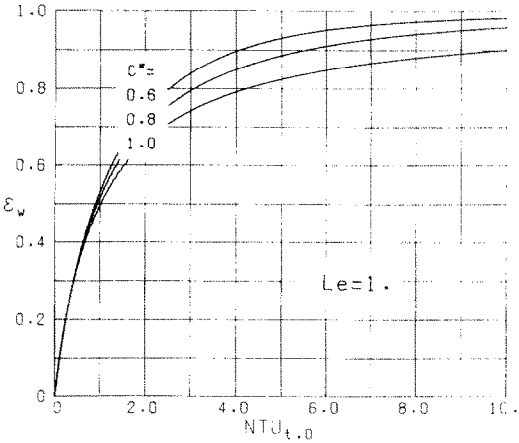


FIG. 2b. Effect of C^* on process-air humidity ratio effectiveness.

air temperature and humidity [18]. The humidity of the process air outlet stream is minimal for $\Gamma_1 = \Gamma_{1,R}$ for ideal dehumidifiers. For dehumidifiers with finite transfer coefficients, the outlet humidity is minimal for $\Gamma_1 > \Gamma_{1,R}$ to an extent depending on the dehumidifier transfer coefficients. Well-defined desiccant cooling systems operate the dehumidifier so that the outlet humidity of the processed air is minimum [3]. In Fig. 2, C^* is defined for $\Gamma_1 \geq \Gamma_{1,R}$. Both the temperature and humidity ratio effectiveness increase with decreasing C^* .

Figure 3 illustrates the effect of the Lewis number on the dehumidifier effectiveness. The effect on the effectiveness for humidity is more substantial than for the temperature effectiveness. While for $Le < 5$, moisture effectiveness values of 0.90 may still be achieved, for $Le > 10$, this effectiveness drops substantially. The ratio of the $NTU_{t,j}$ parameters has only a slight effect on the effectiveness. This has also been reported for the case of regenerators with heat transfer alone [16]. Figures 2 and 3 are valid for the

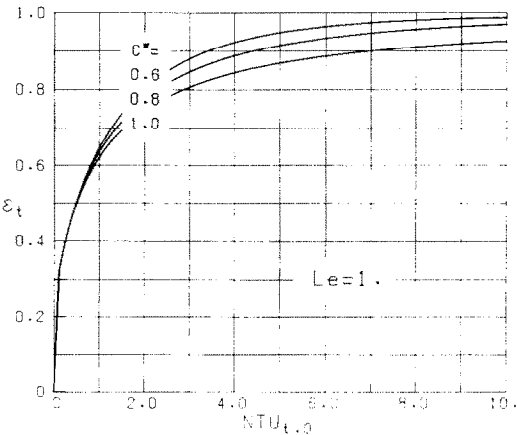


FIG. 2a. Effect of C^* on process air temperature effectiveness.

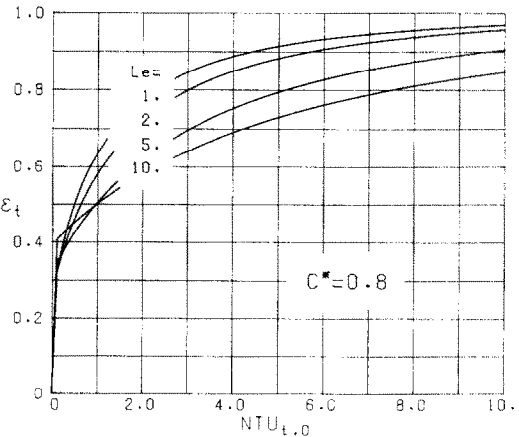


FIG. 3a. Effect of Lewis number on process air temperature effectiveness.

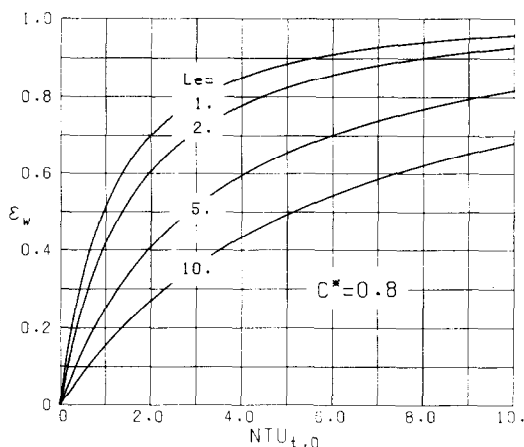


FIG. 3b. Effect of Lewis number on process-air humidity ratio effectiveness.

following range of NTU -ratio:

$$0.5 \leq NTU_{t,1}/NTU_{t,2} \leq 2. \quad (13)$$

To get good prediction of the effectivenesses of the dehumidifier, the number of transfer coefficients should be larger than 5.0 for both periods.

6. DISCUSSION

For given capacitance rate parameters, the equations for regeneration and processing line indicate whether the heat and mass regenerator is operated in the mode of complete regeneration or processing, or in the intermediate mode. In the first two cases, the appropriate asymptotic solutions to the conservation equations that model the exchanger with infinite transfer coefficients in part I give the ideal outlet states. In the latter case, a quadratic interpolation between both asymptotic solutions is adequate. The effectiveness for heat and mass may then be computed using the expressions given in part II to yield the outlet states of an actual dehumidifier with finite transfer coefficients. The computation time per outlet state using this Effectiveness- NTU approach is about 0.5–1.0% of the computation time needed by a finite difference method in which the governing partial differential equations are solved numerically.

The Effectiveness- NTU method offers several improvements over the analogy theory [9,10] to model rotary heat and mass exchangers. In the analogy theory, the effectivenesses are complicated combined functions of the capacitance rate parameters and transformed transfer coefficients. The effectiveness method presented in this paper clearly identifies the various effects of the different parameters that determine the performance of the dehumidifier. The effect of mass flow rates and flow unbalance is incorporated in the ideal dehumidifier analysis. The effectiveness correlations describe the effect of finite transfer coefficients and use an overall NTU that is based upon the actual transfer coefficients.

The effectiveness charts may be used for the design of rotary dehumidifiers for use in desiccant cooling and dehumidification systems. Such studies have shown that, for good overall system performance, the dehumidifier NTU_o should be high [19] and the flow rate of the regeneration air stream should be 60–80% of the flow rate of the air stream that is being dried [3]. Numerical analyses of the analogy theory [19,20] have indicated that the accuracy of the analogy theory decreases with increasing NTU [20] and flow imbalance [19]. For optimum system operating conditions, the average error in prediction by the analogy theory of the temperature of the process air outlet state is estimated to be 3.8%, and for humidity ratio 4.8% [19]. The accuracy of the effectiveness model that is presented in this paper increases with NTU and is independent of flow unbalance. For optimum operating conditions, the errors in prediction of outlet state properties may be estimated within 1.0%.

In the analysis, the heat and mass transfer coefficients are treated as parameters. The Lewis number is defined as the ratio of the overall resistance for mass transfer to the overall resistance for heat transfer. The Lewis number is nearly one for combined heat and mass transfer processes in which convective transfer through the air dominates the transport [21]. If there is additional resistance for mass transfer by internal diffusion within the desiccant particles, the Lewis number may differ from one. In this case, the Lewis number depends on the flow geometry of the dehumidifier. Proposed geometries for dehumidifiers include flow through granular packed beds and parallel plate channels.

Based on experimental data for the adiabatic adsorption of water vapor from air by silica gel for flow through stationary granular beds, Hougen and Marshall [22] developed empirical correlations for the overall Nusselt numbers for heat and mass transfer. Their correlations indicate that the overall Nusselt numbers are 3.6 times less than the values for gas-side resistance only and the overall Lewis number is approximately one. Chi and Wasan [23] implemented the empirical correlations of Hougen and Marshall in a mathematical model for fixed bed adsorption drying with silica gel. The numerical results of the model were compared with the experimental results of Bullock and Threlkeld [24] and good agreement was reported. Allander [25] investigated analytically and experimentally the external flow of moist air past single silica gel particles and through packed beds. His analysis shows that the overall heat transfer coefficient is approximately equal to the external heat transfer coefficient at the air side. The overall mass transfer coefficient is given by the air-side mass transfer coefficient multiplied by an effectiveness factor which angles from 0.5 to 1.0 depending on the particle-water content. Allander's analysis was verified experimentally but does not agree with Hougen and Marshall's analysis with respect to the actual values for the overall transfer coefficients. However, they both indicate that

the Lewis number for flow through packed beds is of the order of 1.0 to 2.0.

Whereas reasonably well-established correlations are developed for the overall heat and mass transfer coefficients for flow through packed beds, much less is known for flow through parallel channels. Combined heat and mass transfer for isothermal flow of humid air through parallel desiccant plates has been studied analytically and experimentally by Ghezelayagh and Gidaspow [26]. The mass-transfer Nusselt number for the air flow in the channel was computed and agreed with the theoretical Graetz value of 1.346. The local Nusselt number for mass transfer into the desiccant sheet varies substantially with time and position in the channel. Their experiments indicate that there can be a considerable resistance for mass transfer due to internal diffusion within the desiccant, and that the Lewis number might be greater than one. Their study shows further that the assumption of constant overall transfer coefficients throughout the dehumidifier may not be valid, and that the computation of these transfer coefficients for this type of flow may not be straightforward. Further research on combined heat and mass transfer for flow through parallel passages remains to be done.

Mathiprakasam and Lavan [8] have presented a linear solution method for regenerators that is valid for Le ranging from 1 to 2. Manipulations of the analogy theory [9, 10] to include the effect of the Lewis number were found to be unsatisfactory [19]. The effectiveness expressions that are presented in this paper are valid for Le ranging from 1 to 20 within the indicated range of accuracy.

7. CONCLUSIONS

A finite-difference program [6] and the ideal dehumidifier performance correlations [11] were used to establish empirical correlations for the effectiveness of a dehumidifier with finite transfer coefficients. Correlations of the moisture and enthalpy effectiveness of a silica gel rotary dehumidifier are presented. The principles that are used to generate these correlations are general and may be applied to dehumidifiers using other desiccants. An ϵ -NTU model, incorporating the correlations for the effectiveness and the ideal dehumidifier performance, allows rapid and accurate calculation of the dehumidifier performance. The correlations are valid for a wide range of inlet conditions, operating parameters and transfer coefficients and are suitable to be used in detailed long term performance simulations of desiccant cooling systems.

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THEORIE DE DIMENSIONNEMENT DES ECHANGEURS ROTATIFS DE CHALEUR ET DE MASSE—II. EFFICACITE; METHODE DU NOMBRE D'UNITES DE TRANSFERT POUR LES ECHANGEURS ROTATIFS DE CHALEUR ET DE MASSE

Résumé—Des formules analytiques de performance des échangeurs rotatifs de chaleur et de masse avec coefficients infinis de transfert ont été présentées dans la partie I. Un modèle de différence finie pour la prévision de performance des deshumidificateurs rotatifs avec des coefficients de transfert finis est utilisé en combinaison avec le modèle du deshumidificateur idéal pour établir des formules d'efficacité. Des relations pour l'efficacité d'humidité et d'enthalpie avec des régénérateurs à silicagel sont données en fonction du nombre d'unités de transfert. Un modèle efficacité—NUT, incorporant les formules pour l'efficacité la performance idéale du deshumidificateur, fournit un calcul rapide des performances des deshumidificateurs. Les corrélations sont valables pour un large domaine de conditions opératoires et elles tiennent compte de l'effet d'un écoulement non conservé et des nombres de Lewis élevés.

AUSLEGUNGSTHEORIE FÜR ROTIERENDE WÄRME- UND STOFFAUSTAUSCHER— TEIL II. WIRKUNGSGRAD-NTU-VERFAHREN FÜR ROTIERENDE WÄRME- UND STOFFAUSTAUSCHER

Zusammenfassung—Analytische Beziehungen für das Leistungsvermögen von rotierenden Wärme- und Stoffaustauschern mit unendlich großen Transportkoeffizienten wurden in Teil I vorgestellt. Ein Modell auf der Grundlage von Finiten Differenzen für die Vorausberechnung des Leistungsvermögens von rotierenden Trocknern mit endlichen Transportkoeffizienten wird in Verbindung mit dem idealen Trocknermodell verwendet, um Wirkungsgrad-Korrelationen aufzustellen. Beziehungen für den Feuchte- und Enthalpiewirkungsgrad für Silicagel-Regeneratoren werden als Funktionen der Trockner-NTU angegeben. Ein Wirkungsgrad-NTU-Modell erlaubt durch die Verwendung der Beziehungen für den Wirkungsgrad und das ideale Trocknerverhalten eine schnelle Berechnung des realen Trocknerverhaltens. Die Beziehungen gelten für einen großen Bereich der Betriebsbedingungen und berücksichtigen den Einfluß von schwankender Strömung und hohen Lewis-Zahlen.

ТЕОРИЯ ВРАЩАЮЩИХСЯ ТЕПЛОМАСООБМЕННИКОВ. II. ЭФФЕКТИВНОСТЬ. NTU-МЕТОД РАСЧЕТА ДЛЯ ВРАЩАЮЩИХСЯ ТЕПЛОМАСООБМЕННИКОВ

Аннотация—В первой части представлены аналитические выражения для характеристик вращающихся тепломассообменников с бесконечными коэффициентами переноса. Для определения эффективности используется конечно-разностный метод расчета характеристик вращающихся осушителей с конечными коэффициентами переноса в сочетании с моделью идеального осушителя. Выражения для эффективности по влажности и энтальпии силикагельных регенераторов даны как функции NTU осушителей. NTU-модель, включающая выражения для эффективности характеристик идеального осушителя, позволяет быстро рассчитать рабочие характеристики осушителя. Выражения справедливы в широком диапазоне рабочих условий и учитывают влияние несбалансированного течения и большие числа Льюиса.