

# A Single-Blow Test Procedure for Compact Heat and Mass Exchangers

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*This paper discusses a single-blow test procedure for estimating the overall heat and mass transfer coefficients of compact dehumidifier matrices. The procedure consists of three sequential experimental procedures for obtaining, respectively, the core geometry of the test matrix, the active mass of sorbent within the matrix, and the distributions of the temperature and humidity ratio responses with time and distance in the flow direction. The analysis technique paired to the experimental procedure is based upon the transformation of the model partial differential equations into a set of ordinary differential equations. The temperature and mass-fraction distributions are then modeled by a system of nonstiff ordinary differential equations, which can be easily integrated numerically. The Lewis number, defined as the ratio of the overall heat to mass transfer coefficients, determines the shape of the distributions with a dimensionless flow coordinate  $x^+$ . With a curve fit method, the experimental Lewis number can be determined with acceptable accuracy. The procedure is illustrated with selected experimental results for humid air and a silica gel matrix.*

## 1 Introduction

In single-blow experiments, an exchanger test matrix is conditioned at specified initial conditions and a step change is applied to the inlet conditions of the fluid stream passing through the matrix. The distributions in time of the fluid stream outlet conditions form the experimental data that characterize the heat and/or mass transfer performance of the test matrix. Retrieving quantitative information, i.e., the heat and/or mass transfer coefficients, is the objective of the analysis. Whereas the methods for heat transfer alone have evolved into de facto standards, the methods for isothermal or adiabatic mass transfer are less developed.

**Single-Blow Methods for Heat Transfer.** The single-blow method is widely used for evaluating the performance of compact heat exchanger geometries (Heggs, 1983). Almost all analysis techniques are based upon the *porous-matrix* or *insulated-duct* equations established by Anzelius (1926) and later by Schumann (1929). These equations express an overall energy conservation equation for the adiabatic system {fluid-stream + matrix} and a linear transfer rate equation based upon an overall transfer coefficient.

There exist two classes of solution method for single-blow experiments. In the *point matching solutions* the principle is to match one parameter derived directly from the experimental data with a theoretical curve showing that parameter as a function of the exchanger *Number of Transfer Units* (e.g., Baclic et al., 1986). These parameters can be (Heggs, 1983): the effectiveness at the point of breakthrough, the maximum slope of the breakthrough curve, the initial temperature rise, the second moment of the distribution curve, and the 20–80 percent distribution width. The *curve matching technique* generally employs finite difference solutions of the insulated-duct equations (e.g., Elliot et al., 1986). Curve matching is typically an iterative procedure in which the *NTU* parameter is determined by minimizing the rms error of the experimental dis-

tribution data with a theoretical curve. For heat transfer alone, there is usually only one curve-fit parameter.

**Single-Blow Methods Involving Mass Transfer.** The general classification of the various techniques for heat transfer alone can be used for isothermal mass transfer in adsorbers. However, the mass conservation equation involves a property equilibrium relationship that is generally nonlinear. Eagleton and Bliss (1953) successfully used a maximum-slope point matching technique to obtain the mass transfer coefficient from experimental breakthrough curves with a nonlinear isotherm. Jefferson (1972) discusses the method of moments for dilute (i.e., linear), binary systems. Schneider and Smith (1968) experimentally studied the isothermal mass transfer of hydrocarbons on silica gel in a packed-bed geometry. Their analysis incorporates three dimensionless parameters: the inter- and intraparticle diffusivity and the sorption rate constant. They state that the curve matching procedure is unsatisfactory because good agreement between the experimental data and a theoretical response curve can be obtained with more than one set of these three constants.

Standard techniques for analyzing single-blow, adiabatic sorption experiments are even less developed. Single-blow methods for mass exchangers usually rely on global curve-matching procedures, however; the description of the transient process invariably includes a fair number of adjustable parameters for the transfer rate and equilibrium constants, and the effects of these parameters on the position and shape of the response curves are intermixed. Chi and Wasan (1970) use empirical correlations for the transfer coefficients, developed by Hougen and Marshall (1947), in their numerical model for the adiabatic adsorption of water vapor on silica gel in a packed-bed geometry. They conclude a fair agreement between their theoretical curves and experimental breakthrough curves published by Bullock and Threlkeld (1966).

Clark et al. (1981) report experiments on the adiabatic sorption of water vapor on silica gel-packed beds using a model similar to Chi and Wasan and the Hougen–Marshall correlations. However, the agreement between their theoretical and experimental breakthrough curves is poor and the authors conclude that the discrepancy cannot be explained by experimental

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errors. Biswas et al. (1984) and Kim et al. (1985) present a theoretical and experimental investigation of the adiabatic sorption of water vapor on regular density silica gel in a parallel plate geometry. Their model also follows the Anzelius approach. The agreement between experimental and predicted breakthrough curves is only fair, and a sensitivity study shows that several factors can explain the discrepancy.

Pesaran et al. (1986) performed a theoretical and experimental analysis of adiabatic single blow experiments of laminar flow of humid air through a parallel passage silica-gel matrix. The model is based upon the method of moments applied to the two Riemann invariants of the mass and energy conservation equations for infinite transfer coefficients. However, the analysis of the experimental data yields erroneous heat and mass transfer coefficients due to experimental inaccuracies and the inappropriate assumption of linear equations.

Clearly, the conventional single-blow methods developed for exchange processes involving heat transfer alone are limited if mass transfer is occurring. Adsorbers often have nonlinear thermodynamic equilibrium properties. The curve-matching techniques applied to mass transfer involve many parameters, and agreement with experimental data can often be obtained with more than one set of these parameters. Thus it seems that the single-blow technique may not be a useful experimental tool for obtaining the performance characteristics of heat and mass exchangers. This paper introduces a single-blow analysis technique that makes use of the nonlinear character of the conservation equations. Although the technique can be applied to any binary sorption system, it is explained next for the case of humid air flowing through a silica gel matrix because experimental data for this system are available.

## 2 Model Formulation and Assumptions

The model follows the classical Anzelius model. The situation is that of laminar parallel flow of humid air through a matrix containing a uniform distribution of desiccant material. The equations of change are based upon overall conservation equations for the air stream and test matrix. The transfer-rate equations are linear and expressed in terms of overall heat and mass transfer coefficients. Diffusion equations are not included in the model.

### Assumptions

- 1 The matrix-flow area, the transfer area per unit length, and the distribution of desiccant mass within the test matrix are constant or uniform with respect to position in the flow direction.
- 2 The axial diffusion of heat and mass in the air stream

is negligible compared to the transport by forced convection. Axial diffusion in the desiccant layer is neglected because the layer consists of isolated particles with small dimensions.

3 The pressure drop through the bed is small with respect to absolute pressure and is neglected for evaluating the sorption equilibrium conditions.

4 The rate of heat and mass transfer can be modeled with constant overall heat and mass transfer coefficients.

5 The temperature distributions in the desiccant mass and structural parts of the test matrix are assumed to be uniform, and the average temperatures of both systems are equal at all times.

Assumption 4 converts the model equations from Rosen-type diffusion equations (Rosen, 1954) into Anzelius-type overall rate equations and it simplifies the resulting equations substantially. Van den Bulck (1987) has experimentally investigated the effect of nonuniform, transient distributions of mass and energy fluxes on the air-side and desiccant-side Nusselt and Sherwood numbers. His analysis shows that the effect can be neglected for the distributions that are generally encountered with mass transfer.

With assumptions 1-5, the overall equations of change for the air stream and test matrix can be written as follows.

#### 1 Water vapor conservation

$$\dot{m}_f L \frac{\partial w_f}{\partial x} + \dot{m}_f \theta_d \frac{\partial w_f}{\partial \theta} + M_d \frac{\partial W_d}{\partial \theta} = 0$$

#### 2 Energy conservation

$$\dot{m}_f L \frac{\partial i_f}{\partial x} + \dot{m}_f \theta_d \frac{\partial i_f}{\partial \theta} + M_d \frac{\partial I_m}{\partial \theta} = 0 \quad (1)$$

#### 3 Rate of water vapor mass transfer

$$M_d \frac{\partial W_d}{\partial \theta} = h_m A (w_f - w_d)$$

#### 4 Rate of thermal energy transfer

$$M_d \frac{\partial I_m}{\partial \theta} = hA (t_f - t_m) + h_m A (w_f - w_d) i_w$$

$x$  measures the distance in the flow direction and  $\theta$  measures the real time.  $w_d$  is the humidity ratio of the air stream in equilibrium with the silica gel at the temperature  $t_m$  and water content  $W_d$ .

Equations (1) form a system of nonlinear, coupled evolution equations, and the behavior of such systems is well known.

## Nomenclature

$A$ = transfer area of exchange matrix	$L$ = length of flow passage	$w_d$ = humidity ratio of air in equilibrium with desiccant
$A_c$ = free flow area of exchange matrix	$Le$ = Lewis number as defined by equation (6)	$w_f$ = humidity ratio of air stream
$c_p$ = fluid thermal capacitance	$\dot{m}_f$ = dry-air mass flow rate	$W_d$ = water content of dry desiccant
$D_h$ = hydraulic diameter of flow passage	$M_d$ = mass of dry desiccant contained within exchanger	$x$ = coordinate measuring position in the flow direction
$h$ = heat transfer coefficient	$M_f$ = fluid mass entrained in flow passages	$x^+$ = dimensionless position coordinate
$h_m$ = mass transfer coefficient	$Nu$ = Nusselt number	$z$ = flow coordinate
$i_f$ = specific enthalpy of fluid stream (per unit mass of dry air)	$Re$ = passage flow Reynolds number	$\alpha_f$ = thermal diffusivity of fluid stream
$i_w$ = specific enthalpy of water vapor	$S$ = dimensionless wave speed of the breakthrough curves	$\theta$ = real-time variable
$I_m$ = specific enthalpy of matrix (per unit mass of dry desiccant)	$t_f$ = fluid stream temperature	$\theta_d$ = dwell time of fluid within matrix
	$t_m$ = matrix temperature	$\tau$ = dimensionless time

Whereas the Anzelius equations model the propagation of the distribution of the air stream temperature, equations (1) model the propagation of two confined property distributions: temperature and humidity. Ruthven (1984) discusses in detail the propagation and dispersion of the distributions, also called waves. There exists a thermal wave, during which most of the sensible energy is exchanged between the air stream and the matrix, and a mass transfer wave, which incorporates the majority of the mass exchange. The thermal wave is narrow and propagates at high speed through the matrix, whereas the mass transfer wave is wider and slower. The respective wave speeds are proportional to the ratio of thermal and mass capacitances of the fluid stream and matrix. The two distributions are usually completely separated, with a well-defined intermediate state between the two transfer zones.

For heat exchangers, the energy conservation equation is linear and as a result the thermal wave is an expansion wave, continuously spreading as it progresses through the exchanger passages. The shape of this wave never becomes *fully developed*, and therefore depends at all times on its initial shape, that is, the change of the inlet conditions with time. For mass transfer, however, the nonlinearity of the sorption isotherm introduces favorable wave patterns, which are not encountered in heat transfer. For selected matrix-initial and fluid-inlet states, the property distributions are *constant-pattern waves*, i.e., the shape of the distributions is preserved as the waves progress through the matrix.

For the system regular-density silica gel and humid air, theoretical analyses by Van den Bulck et al. (1985) experimentally verified by Van den Bulck (1987), show that constant pattern waves occur during regeneration of a wet desiccant with a hot air stream. Constant pattern waves become rapidly fully established after they originate at the inlet face of the matrix. The shape and width of the mass transfer wave in desorption processes are constant, and, for reasonable-quality inlet steps and sufficiently long bed lengths, are independent of the sharpness of the step change in inlet conditions. Furthermore, the wave dispersion is well defined and independent of the air stream mass flow rate.

Michaels (1952) and Garg and Ruthven (1975) successfully used a specific transformation technique introduced by Glueckauf (1947) for analysis of experimental breakthrough curves with constant patterns. This transformation technique is based upon the argument that an observer translating at the speed of the wave will not measure a change of shape of the wave. For the system of equations (1), the physical speed of the second wave is given by

$$\frac{dx}{d\theta} = \frac{\frac{\dot{m}_f L}{M_d} S}{1 + \frac{\dot{m}_f \theta_d}{M_d}} \quad (2)$$

where  $S$  is a constant dimensionless wave speed dependent only upon the thermodynamic properties of the fluid and matrix system, and the inlet and initial conditions. Based upon equation (2) a new position coordinate  $z$  can be defined as

$$z = x - \frac{\frac{\dot{m}_f L}{M_d} S}{1 + \frac{\dot{m}_f \theta_d}{M_d}} \theta \quad (3)$$

The partial differential equations expressed in the stationary coordinate system  $(x, \theta)$  can be written in terms of the moving coordinates  $(z, \theta)$ . The condition for a constant pattern wave translates into  $\frac{\partial}{\partial \theta} z = 0$ . Hence, the system of partial differential equations (1) reduces to a system of ordinary differential

equations in the  $z$  coordinate, moving with the wave. This system can be further simplified by using the time variable  $\tau$  introduced by Anzelius (1926) for modeling fluid flow through porous matrices

$$\tau = - \left( 1 + \frac{\dot{m}_f \theta_d}{M_d} S \right) \frac{hA}{c_p \dot{m}_f} z \quad (4)$$

In terms of the dimensionless time  $\tau$ , the model equations (1) become:

#### 1 Water vapor conservation

$$\frac{dw_f}{d\tau} - S \frac{dW_d}{d\tau} = 0$$

#### 2 Energy conservation

$$\frac{dt_f}{d\tau} - S \frac{dI_m}{d\tau} = 0 \quad (5)$$

#### 3 Rate of water vapor mass transfer

$$\frac{dw_f}{d\tau} = \frac{1}{Le} (w_f - w_d)$$

#### 4 Rate of energy transfer

$$\frac{dt_f}{d\tau} = (t_f - t_m)$$

where  $Le$  is the effective, overall Lewis number, which is defined for this situation as:

$$Le = \frac{h}{h_m c_p} \quad (6)$$

and indicates the magnitude of the resistance for mass transfer relative to that for heat transfer. A compact heat and mass exchanger is characterized by the Nusselt number and Lewis number, i.e., the heat and mass transfer coefficients. The Nusselt number is mainly determined by the geometry of the flow passages, whereas  $Le$  is determined by the geometry of the desiccant layer relative to the dimensions of the flow passage. The model equations (5) indicate that the temperature and humidity distributions can be expressed in terms of the new time variable  $\tau$ , with  $Le$  as parameter. Let  $\Delta\tau$  measure the width of the wave in dimensionless form. Numerical integration of equations (5) indicates that the breakthrough curves become more asymmetric with increasing  $Le$ , and  $\Delta\tau$  increases almost linearly with  $Le$ . Thus, the Lewis number is a measure of the skewness of the curves.

### 3 Experimental Results

A full description of a single-blow experimental apparatus and test procedures is given by Van den Bulck (1987). The test loop configuration is that of a conventional single-blow test facility. A humid air stream with controlled properties and mass flow rate is passed through a test matrix arranged as a vertical stack of 74 parallel rectangular passages with hydraulic diameter  $D_h = 2.0 \pm 6$  percent mm, a width of  $126 \pm 0.9$  percent mm, and a flow length  $L = 0.203 \pm 0.2$  percent m. The walls of the flow passages are coated with regular-density silica gel particles with diameter  $D_p = 0.177$ –297 mm, yielding a total mass of dry desiccant  $M_d = 0.49 \pm 3$  percent kg. The total transfer area  $A = 3.65 \pm 4.5$  percent  $m^2$  and the free flow area  $A_c = 0.00927 \pm 3$  percent  $m^2$ . The  $\pm$  values in these and the following statements denote uncertainties.

A series of 14 dynamic desorption experiments is reported by Van den Bulck (1987). The temperature and water content of the desiccant matrix are initialized at selected conditions, and a step change in the inlet temperature of the air stream is introduced. The air stream mass flow rate is varied in a sys-

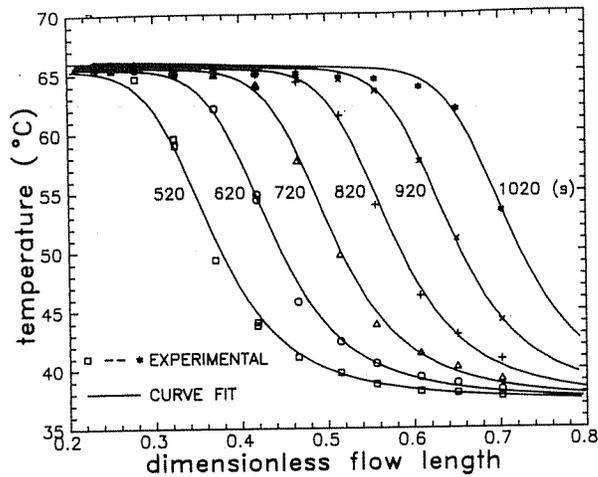


Fig. 1(a) Experimental temperature distributions  $t$ , with dimensionless flow length  $x^+$  and measuring time as parameter; each sequence with time corresponds to one single-blow experiment

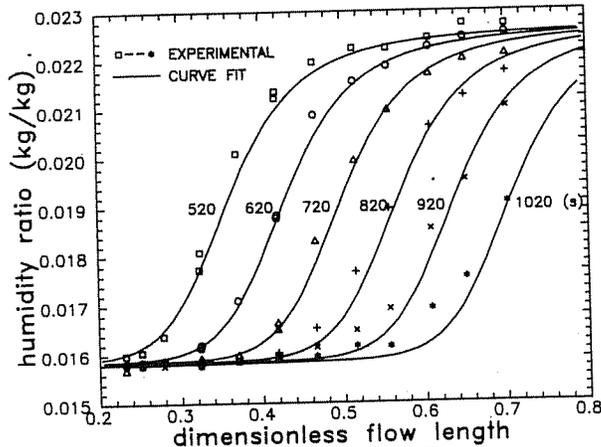


Fig. 1(b) Experimental humidity ratio distributions  $w$ , with dimensionless flow length  $x^+$  and measuring time as parameter; the interpolating curves are identical in width and shape

tematic sequence from  $\dot{m}_f = 0.013$  to  $0.028 \pm 2.5$  percent kg/s. Each experiment has the same pair of matrix-initial and air-inlet conditions, ( $t_{f,0} = 29.6 \pm 0.8^\circ\text{C}$ ,  $w_{f,0} = 0.0141 \pm 0.0006$  kg/kg,  $W_{d,0} = 0.228 \pm 0.005$  kg/kg) and ( $t_{f,2} = 67.1 \pm 0.2^\circ\text{C}$ ,  $w_{f,2} = 0.0157 \pm 0.0004$  kg/kg,  $W_{d,2} = 0.047 \pm 0.001$  kg/kg), respectively. For these conditions, the thermal and mass transfer waves are fully separated, and the experimental intermediate state properties are ( $t_{f,1} = 37.1 \pm 0.3^\circ\text{C}$ ,  $w_{f,1} = 0.0227 \pm 0.0004$  kg/kg,  $W_{d,1} = 0.233 \pm 0.005$  kg/kg). The Reynolds number  $Re$  based upon the fluid inlet state properties ranges from 140 to 425 for these experiments; providing a passage flow well in the laminar regime.

The experimental results are presented as graphs showing the air stream outlet conditions, i.e., temperature and humidity ratio, versus the dimensionless flow length  $x^+$  defined for each of the experiments as

$$x^+ = \frac{L}{D_h Re} \quad (7)$$

This dimensionless flow length or Graetz variable  $x^+$  ranges from 0.23 to 0.70 with steps of 0.05 for the alternate experiments. Equations (4) and (7) indicate that  $\tau$  and  $x^+$  are inversely proportional to  $\dot{m}_f$ , and thus  $x^+$  is proportional to  $\tau$ . The constant pattern condition allows the measured fluid outlet property responses with time to be interpreted as the temporal

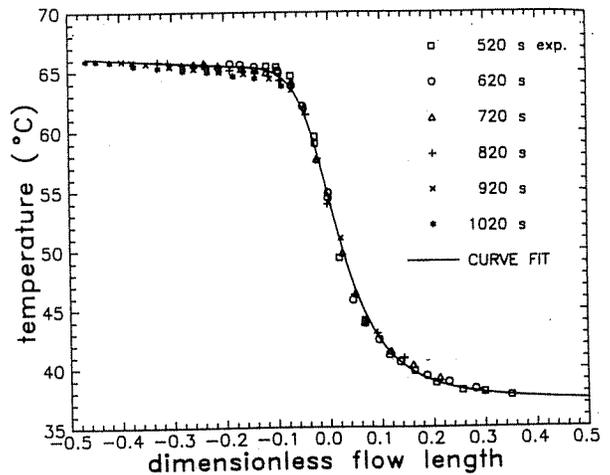


Fig. 2(a) Shifted experimental temperature distributions with dimensionless flow length  $x^+$  showing the constant-pattern behavior of the breakthrough curve

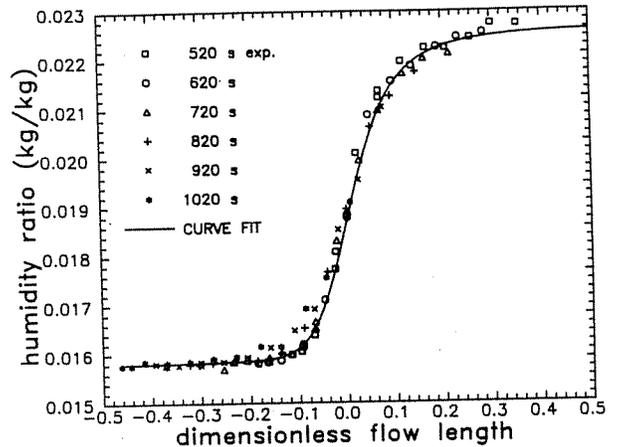


Fig. 2(b) Shifted experimental humidity ratio distributions

distributions with position in the flow direction. The analysis presented here focuses on the mass transfer wave of the desorption experiments because the property distributions associated with this wave are constant-pattern waves and experimentally well defined.

Figures 1(a, b) show the experimental air-stream outlet conditions for each of the 14  $x^+$  values with the real measuring time  $\theta$  as parameter. The conditions on the left of Figs. 1(a, b) are those of the inlet state ( $t_{f,2}$ ,  $w_{f,2}$ ), and the conditions on the right are the intermediate state properties ( $t_{f,1}$ ,  $w_{f,1}$ ). Using least-square curve fitting techniques with cubic splines and variable knots, the experimental distributions for the six values of the sampling time  $\theta$  in Figs. 1(a, b) can be shifted to the left over a distance  $x_c$  proportional to  $\theta$  to overlap with a single distribution curve. This shift is linear with the wave speed  $S$ , and can be determined from the experimental distributions in Fig. 1 with high accuracy. The shifted experimental distributions are shown in Figs. 2(a, b) and are centered about the point of maximum slope of the curves. The rms scatter of the experimental data about this "best fit" is  $0.34^\circ\text{C}$  for temperature and  $0.00012$  kg/kg for humidity. These values are of the order of the accuracy of the measurements, and therefore show that the distributions satisfy the constant-pattern condition.

The time variable  $\tau$  in equations (5) is proportional to  $x^+$  and these equations can thus be integrated with respect to  $x^+$ , with  $L$  as a parameter. The results of this integration with a

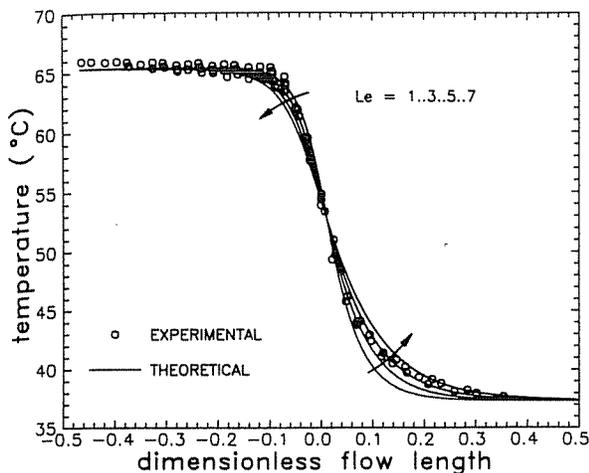


Fig. 3(a) Shifted experimental temperature distributions of Fig. 2(a) in comparison with theoretical breakthrough curves and  $Le$  as parameter

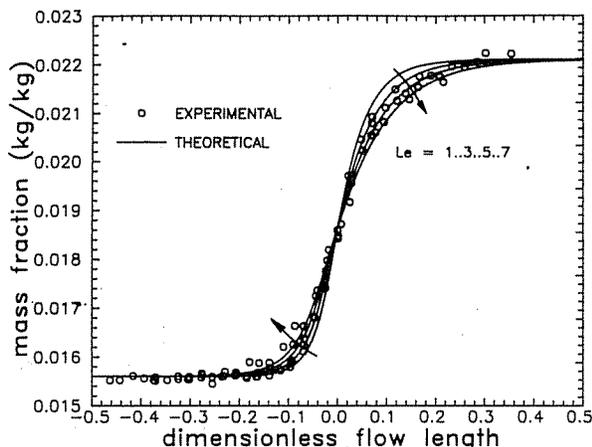


Fig. 3(b) Shifted experimental water vapor mass fraction distributions derived from Fig. 2(b) in comparison with theoretical breakthrough curves and  $Le$  as parameter

fourth-order Runge-Kutta scheme are shown in Figs. 3(a, b) in comparison with the experimental data of Figs. 2(a, b).

The alternate curves in Figs. 3(a, b) illustrate that the Lewis number acts as a skewness parameter for the distributions, whereas the Nusselt number is rather a scale parameter as indicated by equation (9). As a result, it is difficult to extract  $Le$  from the experimental results with high accuracy. For the branch connecting the distribution curves with the inlet state, the scatter of the data is larger than the variation caused by varying Lewis numbers, and no conclusion with respect to the magnitude of this parameter can be made. However, for the opposite branch, connecting the curves with the intermediate state, the scatter of the experimental data is confined within a range corresponding to  $Le = 3..7$  for both temperature and humidity.

The close agreement of the theoretical curves with the experimental breakthrough curves in Figs. 3(a, b) validate the model based upon overall heat and mass transfer coefficients. A Lewis number greater than unity implies that the mass transfer resistance is predominantly determined by the interparticle transient diffusion process. However, the time scale of the overall transient exchange process is about 600 s, which is much larger than the time constant of the interparticle diffusion process, which is  $\pm 100$  s. For these conditions the latter process can be described by a linear transfer rate equation,

and the overall transfer rates can thus be modeled with overall transfer coefficients.

The large uncertainty level of the Lewis number is mainly due to the scatter of the experimental data points about average values. This scatter is not as much due to inaccuracies of the model or measurement inaccuracies as it is to the limit with which the initial conditions of the desorption experiments were repeatable with the experimental facility.

## 4 Discussion

**Experimental Procedure and Analysis Technique.** The principle of the analysis technique is based upon the transformation of the model partial differential equations into a set of ordinary differential equations for the constant-pattern condition of the desorption wave. The transformation is defined by equation (4). The nature of  $\tau$  can be shown by relating  $\tau$  and  $\theta$  in terms of dimensionless groups. Inserting equation (4) into equation (3) gives

$$\Delta\tau = \frac{hA}{c_p M_d} (S\Delta\theta) \quad (8)$$

where  $\Delta\theta$  measures the elapsed real measuring time at a specified location within the flow channel, e.g., the exit face of the matrix. Using the conventional definition expressions for the Nusselt number  $Nu$  and the transfer area  $A$  in terms of  $D_h$  and the thermal diffusivity of the fluid stream,  $\alpha_f$  (Kays and London, 1984), equation (8) can be written as

$$\Delta\tau = Nu \frac{M_f}{M_d} \frac{4\alpha_f}{D_h^2} S\Delta\theta \quad (9)$$

$Nu$  is a parameter depending upon the geometry of the flow passage. Due to the high thermal conductivity of the solid phase, the heat transfer resistance of the desiccant particles is negligible compared to the convective resistance in the fluid stream. The overall Nusselt number is thus approximately the fluid-side Nusselt number.

All other parameters appearing in the conversion formula (9) can be experimentally determined. The transformation technique requires then the following sequence of experiments.

**1 Pressure Drop Experiments.** The presence of granular desiccant particles with irregular shape and nonuniform dimensions affects the flow geometry of compact dehumidifiers in such a manner that often this geometry cannot be directly measured or computed. A series of pressure drop experiments should be performed to determine the hydraulic diameter  $D_h$ , flow area, transfer area, and porosity of the passages.

**2 Sorption Experiments With Varying Matrix-Initial and Fluid-Inlet States.** The adhesion of the desiccant particles to the surface of the wall passages may render part of the desiccant mass inactive. A series of dynamic, single-blow adsorption and desorption experiments should be performed to determine the active mass of desiccant  $M_d$  within the test matrix. The overall mass balance requires an accurate expression for the isotherm correlation, which should be established preferably by static sorption experiments.

**3 Dynamic Sorption Experiments With Varying Mass Flow Rate.** A series of repeated desorption experiments should be performed with identical matrix-initial and fluid-inlet conditions, and varying process air-mass flow rates. The choice of the fluid-inlet and matrix-initial conditions is determined by a compromise between the operating range of the experimental facility, the pattern type of the breakthrough wave, and the strength of the outlet response. The accuracy of the analysis of the experimental data increases with the magnitude of the swing in outlet conditions. The range of inlet and initial conditions that produce constant-pattern waves can easily be determined by the equilibrium theory for equations (1) (e.g., Van

den Bulck et al., 1985). The inverses of the sequential mass flow rates should be chosen to form an arithmetic series of numbers because the dimensionless flow length  $x^+$  or  $\tau$  is inversely proportional to the mass flow rate.

The wave speed  $S$  appearing in equation (9) is a semithermodynamic property of the fluid and matrix system, dependent upon the matrix-initial and fluid-inlet conditions. The conservation equations in equations (5) can be readily integrated because  $S$  is constant. Along the continuous-property distributions of the wave, the condition for constant pattern translates as

$$\frac{(w_f - w_{f,1})}{(W_d - W_{d,1})} = \frac{(w_{f,2} - w_f)}{(W_{d,2} - W_d)} = \frac{(w_{f,2} - w_{f,1})}{(W_{d,2} - W_{d,1})} = S$$

$$\frac{(i_f - i_{f,1})}{(I_m - I_{m,1})} = \frac{(i_{f,2} - i_f)}{(I_{m,2} - I_m)} = \frac{(i_{f,2} - i_{f,1})}{(I_{m,2} - I_{m,1})} = S \quad (10)$$

where the subscripts 1 and 2 refer to the conditions at the start and finish of the wave, respectively. For the first (thermal) wave, 1 is the matrix initial state and 2 is the intermediate state. For the second (mass transfer) wave, 1 is the intermediate state and 2 is the fluid stream inlet state. Although equation (10) could be used to compute  $S$  from experimental data,  $S$  is usually obtained from equation (2).  $dx/d\theta$  is the physical wave speed in real-time coordinates. This wave speed can be obtained directly with high accuracy from the series of dynamic sorption experiments with constant matrix-initial and fluid-inlet states and varying fluid mass flow rate. Equation (10) should rather be used as a check on the accuracy of the thermodynamic property relationships and measurements.

**Design Criteria for Experimental Facilities.** The design criteria for the experimental facility for isothermal or adiabatic mass transfer are different from those for heat transfer alone in the following ways:

1 Of utmost importance is the repeatability of the experimental facility for mass transfer. Repeatability creates the possibility of repeated experiments with identical inlet and initial conditions, and varying process air-mass flow rates. Such a series of experiments is necessary for the analysis technique paired with the experimental procedure, because the wave speed  $S$  and the shifted distributions are determined directly from the experimental data. The sample analysis shows that a high-quality experimental apparatus is needed to predict third-order parameters, such as the Lewis number.

2 The squareness of the step change in inlet conditions is less important, because the effect of the inlet step disappears when the wave becomes *fully developed*. The change of inlet conditions with time is determined by the temperature and humidity controllers of the experimental facility. The constant pattern behavior allows the specification requirements for the controllers to be less stringent, and therefore the controllers to be less expensive.

## 5 Conclusion

The conventional single-blow analysis techniques for compact heat exchangers are reviewed and their limitations for compact heat and mass exchangers are indicated. An alternative single-blow technique for compact dehumidifiers is presented. The governing partial differential equations are reduced to a set of ordinary differential equations by a coordinate transformation. This transformation is based upon the constant-pattern condition, which occurs for selected test conditions with desiccant matrices with nonlinear equilibrium isotherms. The core geometry and effective desiccant capacity of the test matrix are needed to perform the transformation. These specifications can be experimentally determined from

conventional pressure drop experiments and dynamic or static sorption experiments.

There are two parameters that determine the width and shape of the temperature and humidity ratio distributions with a dimensionless time coordinate  $\tau$ . These parameters are the passage Nusselt number and the overall Lewis number. The Nusselt number can be estimated for the core geometry of the matrix with conventional heat transfer tools, and the overall Lewis number can be determined from a curve matching technique applied to experimental distributions. These distributions need to be obtained from a sequence of dynamic sorption experiments with identical initial and inlet conditions, and varying mass flow rates. Repeatability is crucial to achieve accuracy in estimating the Lewis number.

This single-blow technique is easy to perform. The step in inlet conditions does not need to be "near" square. The computations involved are minimal.

## References

- Anzelius, A., 1926, "Über Erwärmung vermittels durchstromender Medien," *Ztschr. f. angew. Math. und Mech.*, Vol. 6, No. 4, pp. 291-294.
- Bacic, B. S., Heggis, P. J., and Abou Ziyah, H. Z. Z., 1986, "Differential Fluid Enthalpy Method for Predicting Heat Transfer Coefficients in Packed Beds," *Proceedings, 8th International Heat Transfer Conference*, San Francisco, CA, pp. 2617-2622.
- Biswas, P., Kim, S., and Mills, A. F., 1984, "A Compact Low-Pressure Drop Desiccant Bed for Solar Air Conditioning Applications: Analysis and Design," *ASME Journal of Solar Energy Engineering*, Vol. 106, pp. 153-158.
- Bullock, X. X., and Threlkeld, Y. Y., 1966, "Dehumidification of Moist Air by Adiabatic Adsorption," *Trans. Am. Soc. Heat Refrig. Air-Cond. Engrs.*, Vol. 72, pp. 301-313.
- Chi, C. W., and Wasan, D. T., 1970, "Fixed Bed Adsorption Drying," *AIChE Journal*, Vol. 16, No. 1, pp. 23-31.
- Clark, J. E., Mills, A. F., and Buchberg, H., 1981, "Design and Testing of Thin Adiabatic Desiccant Beds for Solar Air Conditioning Applications," *ASME Journal of Solar Energy Engineering*, Vol. 103, pp. 89-91.
- Eagleton, L. C., and Bliss, H., 1953, "Drying of Air in Fixed Beds," *Chem. Eng. Progress*, Vol. 49, No. 10, pp. 543-548.
- Elliot, M. P., Rapley, C. W., and Webb, A. I. C., 1986, "Application of the Single Blow Experimental Method to Square Duct Heat Transfer," *Proceedings, 8th International Heat Transfer Conference*, San Francisco, CA, pp. 981-986.
- Garg, D. R., and Ruthven, D. M., 1975, "Performance of Molecular Sieve Adsorption Columns: Combined Effects of Mass Transfer and Longitudinal Diffusion," *Chem. Engng. Sci.*, Vol. 30, pp. 1192-1194.
- Glueckauf, E., 1947, *J. Chem. Soc.*, pp. 1302-1329.
- Heggis, P. J., 1983, "Experimental Techniques and Correlations for Heat Exchanger Surfaces: Packed Beds," in: *Low Reynolds Number Flow Heat Exchangers*, S. Kakac, R. K. Shah, and A. E. Bergles, eds., Hemisphere Publishing Corp., New York, pp. 341-368.
- Hougen, O. A., and Marshall, W. R., Jr., 1947, "Adsorption From a Fluid Stream Flowing Through a Stationary Granular Bed," *Chemical Engineering Progress*, Vol. 43, No. 4, pp. 197-208.
- Jefferson, C. P., 1972, "Prediction of Breakthrough Curves in Packed Beds: I. Applicability of Single Parameter Models," *AIChE Journal*, Vol. 18, No. 2, pp. 409-420.
- Kays, W. M., and London, A. L., 1984, *Compact Heat Exchangers*, 3rd ed., McGraw-Hill, New York, pp. 79-101.
- Kim, S., Biswas, P., and Mills, A. F., 1985, "A Compact Low-Pressure Drop Desiccant Bed for Solar Air Conditioning Applications—II. Bench Scale Tests," *ASME Journal of Solar Energy Engineering*, Vol. 107, pp. 120-127.
- Michaels, A. S., 1952, "Simplified Method for Interpreting Kinetic Data in Fixed-Bed Ion Exchange," *Ind. Eng. Chem.*, Vol. 44, No. 8, pp. 1922-1930.
- Pesaran, A. A., Maclaine-cross, I. L., and Van den Bulck, E., 1986, "Measurements on Promising Dehumidifier Materials and Geometries," Rep. SERI/TR-252-2898, Solar Energy Research Institute, Golden, CO.
- Rosen, J. B., 1954, "General Numerical Solution for Solid Diffusion in Fixed Beds," *Ind. Eng. Chem.*, Vol. 46, No. 8, pp. 1590-1594.
- Ruthven, D. M., 1984, *Principles of Adsorption & Adsorption Processes*, Wiley, New York, pp. 124-166.
- Schneider, P., and Smith, J. M., 1968, "Chromatographic Study of Surface Diffusion," *AIChE Journal*, Vol. 14, No. 6, pp. 886-895.
- Schumann, T. E. W., 1929, "Heat Transfer: A Liquid Flowing Through a Porous Prism," *J. Franklin Inst.*, Vol. 208, pp. 405-416.
- Van den Bulck, E., Mitchell, J. W., and Klein, S. A., 1985, "Design Theory for Rotary Heat and Mass Exchangers," *Int. J. Heat Mass Transfer*, Vol. 28, No. 8, pp. 1575-1595.
- Van den Bulck, E., 1987, "Convective Heat and Mass Transfer in Compact Regenerative Dehumidifiers," PhD Thesis, University of Wisconsin—Madison, Madison, WI.