Comparison of the DESSIM Model With a Finite Difference Solution for Rotary Desiccant Dehumidifiers

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A heuristic "pseudo-steady-state" model of the heat and mass transfer occurring in desiccant dehumidifiers, embodied in the program DESSIM, has been proposed as a conceptually and numerically simple analysis tool (Barlow, 1982). A comparison is made with a finite difference solution to determine the accuracy and limitations of the pseudo-steady-state model. The comparison indicates that the pseudo-steady-state model can produce accurate results of dehumidifier performance relative to the finite difference solution when used carefully, although at greater computational expense. Substitution of the finite difference solution into the overall DESSIM program results in a potentially accurate and useful analysis tool.

1 Introduction

Steady progress in the development of high-performance solid desiccant dehumidifiers has resulted in desiccant-based air conditioning systems that are becoming competitive with conventional vapor compression machines. Desiccant dehumidifer design and estimates of performance are often based on analytical models. It is therefore important to establish valid models for desiccant dehumidifiers.

A variety of models have been employed in the analysis, development, and design of rotary solid desiccant dehumidifiers. Equilibrium solutions to the governing wave equations have been developed by Van den Bulck et al. (1985) and Epstein et al. (1985). Maclaine-cross and Banks (1972) and Mathiprakasam and Lavan (1980) have proposed approximate analytical solutions to the governing conservation and rate equations. Finite difference approximations to the conservation and rate equations have proven useful for detailed analyses and several solutions have been developed, for example, Pla-Barby (1978), Holmberg (1979), Maclaine-cross (1974), and Pesaran and Mills (1984). The solutions have been derived from essentially the same set of partial differential equations. These programs should therefore generate nearly identical results if carefully implemented.

Barlow (1982) has proposed a pseudo-steady-state model, incorporated in the program DESSIM, which considers discrete sections of the dehumidifier to act as simple steady-state heat and mass exchangers. The program provides a flexible tool for analyzing dehumidifier performance and has been used intensively by some groups to study the effects of desiccant properties (Barlow and Collier, 1981; Collier et al., 1986; Collier, 1989), to estimate desiccant cooling system performance (Schlepp and Barlow, 1984), and has been modified to model the performance of a direct solar regenerated collector/dehumidifier (Schultz et al., 1987). Fair agreement of the model with single-blow experimental data has been obtained (Schlepp and Barlow, 1984).

The DESSIM model has met its goal as a tool to evaluate different concepts in a quick and efficient manner. However, the model is heuristic in nature and its validity has never been fully established. The purpose of this paper is to examine the validity of the pseudo-steady-state model by comparing it with a model of a rotary counterflow desiccant dehumidifier that is based on fundamental principles.

2 The Pseudo-Steady-State Model

The pseudo-steady-state (PSS) model considers a differential angular slice of a dehumidifier wheel that is discretized along the flow length. The model follows the slice in discrete time-steps as it rotates from one air stream to the next. Rather than obtaining a set of finite difference equations from the governing partial differential equations, each discrete control volume or node is considered as a simple steady-state counterflow heat and mass exchanger, shown schematically in Fig. 1. A summary of the DESSIM concept is presented here in non-

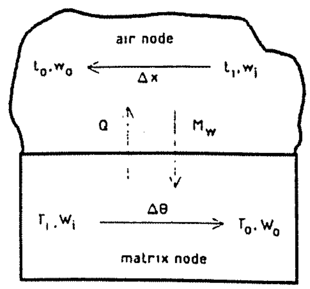

Fig. 1 Schematic representation of the heat and mass transfer processes occurring in a discrete section of a dehumidifier

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dimensional form; further detail is contained in Schultz (1987). Barlow (1982) presents the original derivation in dimensional terms; this can also be found in Schultz et al. (1987).

Barlow first considers each node to be an isothermal counterflow mass exchanger, in effect locally uncoupling the mass transfer from the heat transfer. The driving force for mass transfer is taken to be water vapor mass fraction,

$$y_o = y_i - E_m (y_i - y_{E_i}) \tag{1}$$

where E, the counterflow effectiveness, is (Kays and London, 1984)

$$E = \frac{1 - e^{-\Lambda(1-C)}}{1 - Ce^{-\Lambda(1-C)}} \tag{2}$$

and the nodal parameters for mass transfer are given by

$$\Lambda_m = Ntu \cdot \Delta x \qquad C_m = \frac{1}{\sigma_m \beta \Gamma} \cdot \frac{\Delta\theta}{\Delta x} \tag{3}$$

$$\sigma_m = \left(\frac{1+W}{1+w}\right) \cdot \left(\frac{\partial Y}{y_E}\right)_T = \left(\frac{1+w_E}{1+w}\right) \left(\frac{1+W_E}{1+W}\right) \cdot \left(\frac{\partial W}{\partial w_E}\right)_T$$

where Ntu is a number of mass transfer units for the side of the wheel under consideration, Γ is the ratio of the "mass flow rate" of dry desiccant to mass flow rate of dry air for that side of the wheel, and σ_m is the moisture capacity ratio and is analogous to the ratio of specific heats of the matrix and the air for heat transfer. The water adsorbed (desorbed) is then determined by a simple mass balance,

$$W_o = W_i - \frac{1}{\beta\Gamma} \cdot \frac{\Delta\theta}{\Delta x} (w_o - w_i). \tag{4}$$

Second, an energy balance is performed on the matrix alone assuming no heat transfer to the air to account for the heat released by the adsorbed water. The resulting intermediate matrix temperature is given by

$$T_* = T_i + (h_s/c_M)(W_o - W_i) \tag{5}$$

$$c_M = c_{DM} + W_o c_{WL} \tag{6}$$

where the temperature dependence of the integral heat of wetting term in the moist matrix enthalpy function has been indirectly neglected due to an incorrect assumption made in the derivation of equation (5). This is shown through a comparison of equation (5) with equation (32) (developed later). This introduces only very small errors in general.

Third, the heat transfer from the matrix to the air stream is calculated considering the node to be a counterflow heat exchanger,

$$t_o = t_i - E_h (t_i - T_*) \tag{7}$$

where equation (2) is used for E with the following parameters,

$$\Lambda_h = \mathrm{Le}_o Ntu \cdot \Delta x \qquad C_h = \frac{1}{\sigma_h \beta\Gamma} \cdot \frac{\Delta\theta}{\Delta x} \qquad \sigma_h = c_M/c_A \tag{8}$$

where Le_o is the nondimensional ratio of the local lumped heat-transfer coefficient to the lumped mass-transfer coefficient. Collier et al. (1986), and Collier (1989) have chosen to use the relations for parallel flow effectiveness. A final energy balance then determines the "outlet" matrix temperature at the end of the time-step,

$$T_o = T_* + \frac{1}{\sigma_h \beta\Gamma} \cdot \frac{\Delta\theta}{\Delta x} (t_i - t_o). \tag{9}$$

The outlet states of a node are calculated by one pass through the above set of equations resulting in a potentially efficient numerical procedure. Property evaluations are made at the "initial" conditions that exist when the calculations are performed. For comparison with the finite difference equations, developed later, the set of pseudo-steady-state equations are rewritten below in which equation (5) has been substituted into equations (7) and (9),

$$y_o = y_i - E_m (y_i - y_{E_i}) \tag{10}$$

Nomenclature

c = specific heat (J/kg-°C)
C = local capacity rate ratio for a node ( )
E = counterflow effectiveness factor (equation (2)) ( )
$\mathfrak{g}$ = lumped mass-transfer coefficient (kg_W/m²-sec)
$\mathfrak{h}$ = lumped heat-transfer coefficient (W/m²-sec)
h = specific enthalpy of moist air (J/kg_{DA})
h_s = heat of sorption (J/kg_W)
H = specific enthalpy of the moist desiccant (J/kg_{DD})
ΔH_w = integral heat of wetting (J/kg_W)
K = finite difference effectiveness factor (equation (29)) ( )
Le_o = overall Lewis number, ratio of the lumped heat and mass transfer coefficients ( ), $= \mathfrak{h}/\mathfrak{g}_A c$
Ntu = number of mass transfer units for the side of the wheel under consideration ( ), $= \mathfrak{g}A/m_A$
t = temperature of moist air (°C)
T = temperature of desiccant matrix (°C)
w = humidity ratio ($\mathrm{kg}_W/\mathrm{kg}_{DA}$)
w_E = humidity ratio of air in equilibrium with moist desiccant ($\mathrm{kg}_W/\mathrm{kg}_{DA}$)
W = water content of dry desiccant ($\mathrm{kg}_W/\mathrm{kg}_{DD}$)
x = nondimensional axial length coordinate relative to depth of the wheel ( )
Δx = nondimensional spacial step-size ( )
y = mass fraction of water in moist air, $w/(1 + w)$
y_E = mass fraction of water in moist air in equilibrium with moist desiccant
Y = mass fraction of water in moist desiccant, $W/(1 + W)$
β = fraction of total rotation period occupied by side of the wheel under consideration ( )
Γ = ratio of flow rate of dry desiccant to flow rate of dry air for side of the wheel under consideration ( )
θ = nondimensional time coordinate, $t/\mathcal{T}$ ( )
$\Delta\theta$ = nondimensional time step-size ( )
Λ = nondimensional nodal-transfer coefficient ( )
σ = specific capacity ratio ( )
$\mathcal{T}$ = total rotational period (sec)

Subscripts

A = moist air
DA = dry air
DD = dry desiccant
DM = dry matrix
ex = extrapolated state
h = heat transfer
i = inlet state
L = large time step-size
m = mass transfer
M = moist matrix
o = outlet state
P = process period
R = regeneration period
S = small time step-size
W = water
WL = liquid water
$*$ = intermediate state

$$W_o = W_i - \frac{1}{\beta\Gamma} \cdot \frac{\Delta\theta}{\Delta x}(w_o - w_i) \tag{11}$$

$$t_o = t_i - E_h(t_i - T_i) + E_h(h_s/c_M)(W_o - W_i) \tag{12}$$

$$T_o = T_i + (h_s/c_M)(W_o - W_i) - \frac{1}{\sigma_h \beta\Gamma} \cdot \frac{\Delta\theta}{\Delta x}(t_o - t_i). \tag{13}$$

To determine the periodic steady-state solution of a dehumidifier, an initial profile of matrix states is assumed. Together with the known inlet air condition, the outlet states from the first node can be calculated using the above equations. The air outlet state from that node becomes the inlet state to the next node, etc. The matrix outlet state becomes the "inlet" state for that node at the next time-step, etc. The average air outlet state for a whole period is the time-step-weighted average of the air outlet states from the last node as it rotates through the period. The periodic steady-state condition is reached when the average air outlet state from one revolution is within a small tolerance of the state from the previous revolution.

3 Modifications to the Pseudo-Steady-State (PSS) Model

The PSS energy conservation equations are nonlinear in the dependent variables and, therefore, equations (5) and (9) do not exactly conserve energy. This situation is easily corrected by replacing equations (5) and (9) with the thermodynamic relations

$$H_* = H(T_i, W_i) - \frac{1}{\beta\Gamma} \cdot \Big(h(t_i, w_o) - h(t_i, w_i)\Big)$$

$$T_* = T(H_*, W_o) \tag{14}$$

$$H_o = H_* - \frac{1}{\beta\Gamma} \cdot \Big(h(t_o, w_o) - h(t_i, w_o)\Big)$$

$$T_o = T(H_o, W_o), \tag{15}$$

respectively, where enthalpy is used as an intermediate dependent variable and the corresponding temperatures are found from the property relations. Implementation of this modification will be referred to as the PSSe (energy conserving) model.

Local coupling of the heat and mass transfer can be introduced by replacing equation (1) with

$$y_o = y_i - E_m(y_i - y_E(T, W_i))$$

which can also be written as

$$y_o = y_i - E_m(y_i - y_{E_i}) + \frac{1}{2} E_m \cdot \left.\frac{\partial y_E}{\partial T}\right)_Y (T_o - T_i) \tag{16}$$

where y_E has been expanded using a Taylor's series and T is taken to be the average of the inlet and outlet states. An iterative procedure is now required to solve for the outlet states at each node. A similar approach appears to have been taken by Collier et al. (1986) in later verions of DESSIM.

4 Fundamental Heat and Mass Transfer Model

For heat and mass transfer processes between the air stream and the matrix that are described by lumped transfer coefficients and where the heat and mass storage capacity of the air is negligible relative to that of the matrix, the following set of conservation and transfer rate equations can be derived (Maclaine-cross, 1974; Jurinak and Mitchell, 1984; Schultz, 1987),

$$\frac{\partial w}{\partial x} = Ntu \cdot (1 + w)(w_E - w) \tag{17}$$

$$\frac{\partial w}{\partial x} + \beta\Gamma \frac{\partial W}{\partial \theta} = 0 \tag{18}$$

$$\frac{\partial t}{\partial x} = \mathrm{Le}_o Ntu \cdot (T - t) \tag{19}$$

$$\frac{\partial h}{\partial t} \cdot \frac{\partial t}{\partial x} + \frac{\partial h}{\partial w} \cdot \frac{\partial w}{\partial x} + \beta\Gamma \frac{\partial H}{\partial T} \cdot \frac{\partial T}{\partial \theta} + \beta\Gamma \frac{\partial \mathrm{H}}{\partial \mathrm{W}} \cdot \frac{\partial \mathrm{W}}{\partial \theta} = 0. \tag{20}$$

To be compatible with the pseudo-steady-state model development, the mass transfer coefficient is taken to be defined for mass fraction as the driving potential. These equations are nonlinear and coupled through the equilibrium relationship for the air-water vapor-desiccant system,

$$W = W(T, w_E) \quad H = H(T, W) \quad h = h(t, w). \tag{21}$$

The description of the steady-state problem is completed by specifying the inlet air states and periodic boundary conditions on the matrix as it rotates.

Equation (17) can also be written on a wet-mass basis,

$$\frac{\partial y}{\partial x} = Ntu \cdot (1 - y)(y_E - y) \tag{22}$$

and equation (22) can be combined with equation (18) on a wet-mass basis to give

$$\frac{\partial y_E}{\partial \theta} - \frac{1}{(1 + w_E)^2} \cdot \left.\frac{\partial w_E}{\partial T}\right)_W \frac{\partial T}{\partial \theta} = \frac{1}{\beta\Gamma \dfrac{(1 + w_E)^2}{1 + w} \cdot \left.\dfrac{\partial W}{\partial w_E}\right)_T} \cdot Ntu \cdot (y - y_E) \tag{23}$$

where $Y(T, y_E)$ has been expanded in terms of T and y_E.

A set of difference equations can be obtained from the partial differential equations by expressing the inlet and outlet states of a node as a Taylor's series about the "center" of the node, for example,

$$t_i = \bar{t} - \frac{\partial t}{\partial x} \cdot \frac{\Delta x}{2} + \frac{\partial^2 t}{\partial x^2} \cdot \frac{\Delta x^2}{8} - \frac{\partial^3 t}{\partial x^3} \cdot \frac{\Delta x^3}{48} + \ldots \tag{24}$$

where t is the temperature at the center of the node. Subtraction of equation (24) from a similar equation for t_o gives a second-order accurate estimate of the derivative at the center of the node. A second-order accurate estimate of the state of the center of the node is obtained by adding these same equations. Substitution of these and similar results for the other dependent variables into the partial differential equations results in the following set of difference equations,

$$y_o = y_i - K_m(y_i - y_{E_i}) + \frac{1}{2} K_m \cdot \frac{1}{(1 + w_E)^2} \cdot \left.\frac{\partial w_E}{\partial T}\right)_W (T_o - T_i) \tag{25}$$

$$W_o = W_i - \frac{1}{\beta\Gamma} \cdot \frac{\Delta\theta}{\Delta x}(w_o - w_i) \tag{26}$$

$$t_o = t_i - K_h(t_i - T_i) + \frac{1}{2} K_h(h_s/c_M)(W_o - W_i) \tag{27}$$

$$T_o = T_i + (h_s/c_M)(W_o - W_i) - \frac{1}{\sigma_h \beta\Gamma} \cdot \frac{\Delta\theta}{\Delta x}(t_o - t_i) \tag{28}$$

where the "effectiveness", K, is given by

$$K = \frac{\Lambda}{1 + (\Lambda/2)(1 + C)}. \tag{29}$$

This equation is identical in form to that obtained by Lambertson (1954) in his finite difference analysis of rotary heat exchangers. The appropriate parameters for mass and heat transfer are

$$\Lambda_m = \frac{Ntu}{1+w} \cdot \Delta x \quad C_m = \frac{1}{\sigma_m \beta \Gamma} \cdot \frac{\Delta\theta}{\Delta x} \quad \sigma_m = \left(\frac{1+w_E}{1+w}\right)^2 \left.\frac{\partial W}{\partial w_E}\right)_T \tag{30}$$

$$\Lambda_h = \mathrm{Le}_o Ntu \cdot \Delta x \qquad C_h = \frac{1}{\sigma_h \beta \Gamma} \cdot \frac{\Delta\theta}{\Delta x} \qquad \sigma_h = c_M / c_A \tag{31}$$

$$c_M = \left.\frac{\partial H}{\partial T}\right)_W = c_{DM} + W c_{WL} + \left.\frac{\partial \Delta H_w}{\partial T}\right)_W \tag{32}$$

where the integral heat of wetting is correctly accounted for in the moist matrix enthalpy function. Stability of the numerical solution is maintained by choosing Δx and $\Delta\theta$ such that K is less than unity.

Equations (25)-(28) describe the heat and mass transfer processes for a node and are of a form with which the PSS equations can be easily compared. Because equation (25) depends on the temperature calculated in equation (28), an iterative procedure is needed to find the solution. Convergence of various iteration schemes was found to be slow, resulting in long computation times in determining the steady-state performance of a rotary dehumidifier.

By choosing humidity ratio, w, rather than mass fraction, y, as a dependent variable, Maclaine-cross (1974) arrived at the following set of second-order difference equations in matrix form,

differences are in the "effectiveness" relations, equation (2) and equation (29), that are used, and the absence of a factor of 1/2 in the third term of equation (12) as compared with equation (27). Minor differences also occur because of slight variations in the definitions of the effectiveness parameters, Λ and C (equations (3), (8), (30), and (31)), and in the states used to evaluate properties. Modification of the PSS model using equation (16) results in the correct coupling of the heat and mass transfer, however, the other differences remain.

Figure 2 presents a comparison of the PSS and finite difference effectiveness relations as a function of Λ and C. The finite difference and counterflow effectivenesses are identical for $C = 1$. There is very little difference between the three relationships at the same C for $\Lambda < 1$, although the deviation of the parallel flow effectiveness is somewhat greater for $C = 0$. However, with the proper choice of sufficiently small space and time-steps, the use of the counterflow or parallel flow effectiveness in DESSIM should not lead to large errors.

As the space step Δx decreases, the node effectiveness also decreases and so the difference between the node outlet and inlet states becomes small. In equations (25) and (27), this means that the last term goes to zero like K^2 as Δx goes to zero, while the second term goes to zero like K. The last term becomes negligible compared with the second term at sufficiently small space-steps. Equations (10) and (12) then become equivalent to equations (25) and (27), respectively. Therefore,

$$\begin{pmatrix} \frac{\mathrm{Le}_o}{2} + \frac{1}{Ntu \cdot \Delta x} & 0 & -\frac{\mathrm{Le}_o}{2} & 0 \\ 0 & \frac{1}{\Delta x} & 0 & \frac{\beta\Gamma}{\Delta\theta} \\ 0 & 1 + \frac{2(1+w_E)}{Ntu \cdot \Delta x} & -\left.\frac{\partial w_E}{\partial T}\right]_W & -\left.\frac{\partial w_E}{\partial W}\right]_T \\ \frac{1}{\Delta x} \cdot \left.\frac{\partial h}{\partial t}\right]_w & \frac{1}{\Delta x} \cdot \left.\frac{\partial h}{\partial w}\right]_t & \frac{\beta\Gamma}{\Delta\theta} \cdot \left.\frac{\partial H}{\partial T}\right]_W & \frac{\beta\Gamma}{\Delta\theta} \cdot \left.\frac{\partial H}{\partial W}\right]_T \end{pmatrix} \cdot \begin{pmatrix} t_o - t_i \\ w_o - w_i \\ T_o - T_i \\ W_o - W_i \end{pmatrix} + \begin{pmatrix} \mathrm{Le}_o (t_i - T_i) \\ 0 \\ 2(w_i - w_{E_i}) \\ 0 \end{pmatrix} = \begin{pmatrix} 0 \\ 0 \\ 0 \\ 0 \end{pmatrix} \cdot \tag{33}$$

Through various matrix row operations, this system of equations is placed in upper triangular form and solved by back substitution without need for iteration. Two passes are used, however. In the first pass, properties are evaluated at the inlet states and estimates of the outlet states are calculated. In the second pass, properties are evaluated at the average of the inlet and estimated outlet states and the outlet states are recalculated. This procedure is analogous to a second-order Runge-Kutta method and maintains the second-order accuracy of the difference scheme (Maclaine-cross, 1974). Because the water conservation equation, equation (18), is linear, the finite difference scheme explicitly conserves water. Although the energy conservation equation, equation (20), is nonlinear, the scheme has been found to adequately conserve energy also. Equations (33) have been shown to adequately represent actual dehumidifier performance given sufficient design information (Schultz, 1987; Schultz and Mitchell, 1987).

The two sets of difference equations, equations (25)-(28) and equations (33), predict nearly identical dehumidifier outlet states when the same discretization is used ($\Delta t < \pm .0005$ °C, $\Delta w < \pm .0003$ g/kg) as should be expected. Equations (33) represent a more efficient computational procedure (by up to a factor of 5 over the iterative scheme) and are easily inserted into the overall DESSIM program.

5 Comparison of Equations

A comparison of the PSS equations, equations (10)-(13), and the finite difference equations, equations (25)-(28), shows that the conservation equations are identical in form. A major difference occurs in the PSS mass transfer equation, equation (10), which lacks the temperature term that couples the heat transfer to the mass transfer in equation (25). The other major

the PSS concept and the finite difference solution converge as the space discretization becomes small.

6 Comparison of Solutions

The dehumidifier performance predicted by the PSS and finite difference formulations can be easily compared by implementing the appropriate sets of equations in the DESSIM framework. The formulations are referred to here as DES-PSS (the pseudo-steady-state model), DES-PSSe (incorporating the energy conserving modification), and DES-FD (the finite difference solution represented by equations (33)). The dehumidifier considered consists of regular density silica gel with a zero heat capacity supporting matrix.

The results of the finite difference solution for different

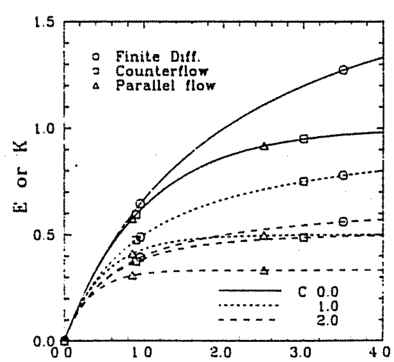

Fig. 2 Comparison of the counterflow (E) and finite difference (K) effectiveness relations

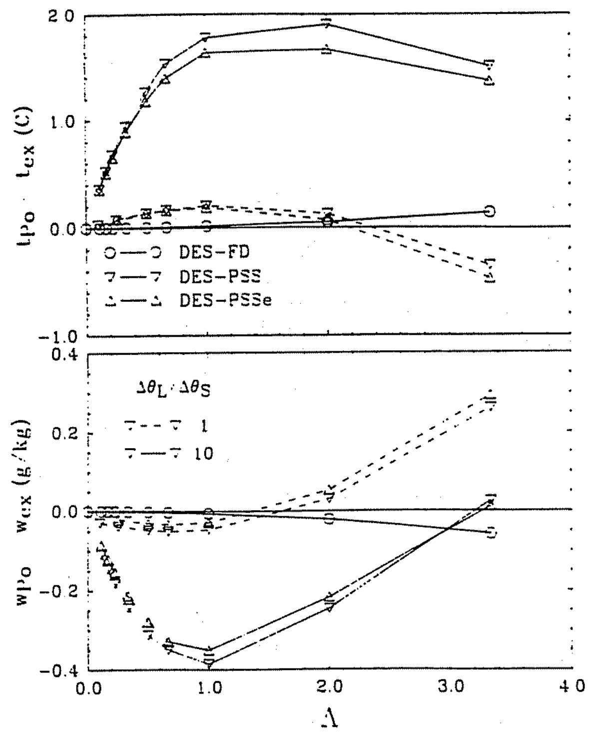

Fig. 3 Differences in process stream outlet states predicted by the PSS model (energy conserving and nonconserving versions) and the finite difference solution as a function of grid size. The finite difference solution has been extrapolated to zero grid size. *Ntu* = 10.

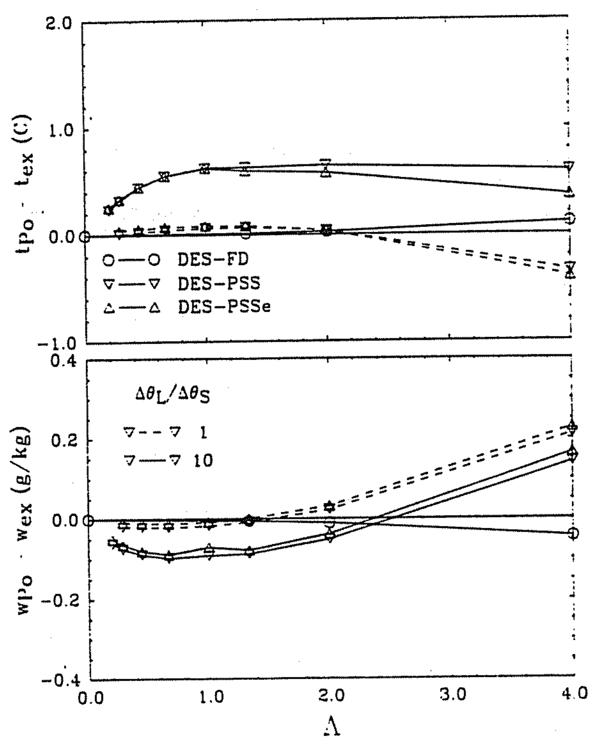

Fig. 4 Same as Fig. 3, with *Ntu* = 20

discretizations in which the ratio of $\Delta x/\Delta\theta$ is maintained a constant can be extrapolated to zero grid size because of the second-order nature of the scheme (Maclaine-cross, 1974; Carnahan, et al., 1969). Using the extrapolated results as a basis, Figs. 3 and 4 show the differences in the steady-state outlet temperatures and humidity ratios predicted by the PSS and finite difference solutions as a function of grid size. A silica gel dehumidifier has been modeled at two levels of perform-

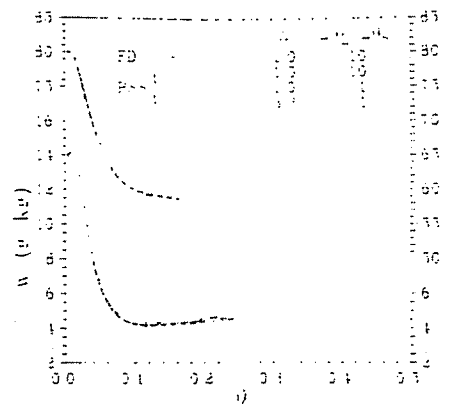

Fig. 5 Process stream outlet state profiles as predicted by the PSS model (energy conserving version) and the finite difference solution

Table 1 Description of silica gel dehumidifier modeled

Dehumidifier description and operating parameters:	
$t_P = 30.0°C$	$t_R = 80.0°C$
$w_P = 14.0$ g/kg	$w_R = 14.0$ g/kg
$Ntu_P = 10, 20$	$Ntu_R/Ntu_P = 1.0$
$\Gamma_P = 0.2$	$\Gamma_R/\Gamma_P = 1.0$
$Le_o = 1.0$	$c_{DM} = 921$ J/kg$_{DD}$-°C
Model parameters:	
$\dfrac{\Delta\theta_S}{\Delta x\beta_P\Gamma_P} = 0.454$	$\dfrac{\Delta\theta_L}{\Delta\theta_S} = 1, 10$

Extrapolated finite difference solution outlet states for the process period:

$Ntu = 10$:	$t_{Po} = 58.744°C$	$w_{Po} = 5.471$ g/kg
$Ntu = 20$:	$t_{Po} = 60.707°C$	$w_{Po} = 4.823$ g/kg

ance: moderate (Ntu = 10, equivalent to Ntu_o = 5 in Kays and London (1984) notation) and high (Ntu = 20 or Ntu_o = 10). Further description of the dehumidifier is shown in Table 1. The small time-step size was chosen to approximately follow the fast moving thermal wave in the process period; the larger time-step size, when used, was based on the fastest portion of the slower moving mass transfer wave (Maclaine-cross and Banks, 1972). The determination of appropriate time-step sizes is not currently an integral part of the DESSIM program and was done separately here. Without the aid of the wave theory, the choice of appropriate time-step sizes must be done by numerical experimentation.

The behavior of the PSS solutions as the grid size becomes small confirms that the PSS formulation does indeed converge to the finite difference solution as discussed in Section 5. Errors caused by the nonconservation of energy tend to be small. However, the discretization errors tend to be larger for the PSS model than for the finite difference solution, especially when a large time-step is used. For small Δx where the effectivenesses E and K are nearly equal, the absence of the temperature term in the PSS mass transfer equation, equation (25), is responsible for much of the error. Without this term, the PSS model tends to overpredict the amount of water adsorbed in the process period, resulting in a lower outlet humidity ratio. Because more water is transferred, more adsorption energy is released, resulting in a higher process outlet temperature from the PSS equations. For large Δx, the effectiveness E_m can be sufficiently less than K_m that the PSS model underpredicts the amount of water adsorbed in the process period, resulting in a larger outlet humidity ratio. In addition, the heat transfer is also underpredicted, resulting in a lower outlet temperature. Results for other inlet air states show similar errors. However, the effect of other matrix thermal and sorption properties has not been tested.

Figure 5 shows the process stream outlet temperature and humidity profiles as a function of rotation angle. The PSSe formulation predicts the proper outlet profiles in general. This is because errors passed from one space or time-step to the next tend to produce compensating errors. The PSS solutions predict an initial outlet state that is greater than the regeneration inlet state. For large step sizes, this initial overprediction is significant and adversely affects the profile for the rest of the period. For large changes in time-step size, the DESSIM procedure begins to show some apparent instabilities. The finite difference scheme is not subject to these errors for the range of grid sizes considered. In addition, a comparison of the profiles inside the wheel reveals that the PSSe model errors are even greater there than those shown in Fig. 5. These errors are caused by the PSS model's uncoupling of the mass transfer from the heat transfer, i.e., by the absence of the temperature term in equation (25).

Experimentally, the process stream outlet temperature and humidity ratio can be determined to within $\pm 0.5°C$ and $\pm .2$ g/kg, respectively (Schultz, 1987). Therefore, the differences between the PSS solution when done carefully (i.e., using small space-steps and only one small time-step) and the finite difference solution appear not to be significant. However, the use of small space and time-step results in a computationally inefficient procedure, even though only one set of calculations need be done at each node. Experience shows that the DES-PSS model requires about 80 percent more computation time than does the DES-FD model to achieve the same level of precision.

7 Application of the Model to a Direct Solar-Regenerated Collector/Dehumidifier

Several researchers have proposed regenerating the desiccant by direct exposure to solar radiation rather than relying on convection heat transfer from a solar heated air stream. Schultz et al., (1987) studied a desiccant-coated "belt" that rotates through a solar collector where the regeneration energy is supplied by direct absorption of solar radiation. To model this device, the PSS formulation was modified to account for the solar radiation gain by including an additional term in the intermediate energy balance, equation (5),

$$T_* = T_i + (h_s(W_o - W_i) + Q_s)/c_M \tag{34}$$

where

$$Q_s = (F_R(\tau\alpha)I_s - F_R U_L(T_i - t_\infty))\Delta\theta T/(M_{DD}/A_s) \tag{35}$$

is the net solar energy gained by the node during the time-step. This provided a simple straightforward means for a first study of the collector/dehumidifier. This approach (with additional effort) could also be inserted into the finite difference solution.

During the regeneration period, the energy gain from the solar radiation can be 2–5 times the magnitude of the energy removed as water is desorbed from the matrix. This results in changes in the matrix temperature for the time step that are 1–4 times that of the adiabatic case and are in the opposite direction. The errors introduced in the PSS model by locally uncoupling the mass transfer from the heat transfer will thus be larger than in the adiabatic case. However, these errors again produce compensating effects at the next step, so the overall qualitative description provided by the model should be satisfactory.

8 Conclusions

A comparison of the system of equations used in the pseudo-steady-state (PSS) model with a set of finite difference equations has shown that the PSS model neglects the local coupling of the heat and mass transfer processes that occur in a desiccant dehumidifier. It appears that the greatest consequence of this is on the numerical stability of the computational scheme as the resulting errors are sensitive to the sizes of the space and time-steps used. By careful choice of space and time-step sizes, accurate predictions of dehumidifier performance can be obtained from the PSS model relative to a finite difference solution; at greater computational expense, however.

The reasonable agreement shown between the PSS model and the finite difference solution indicates that conclusions drawn from past work using the PSS model are valid. However, it is recommended that future research and development efforts use a model with a more sound theoretical basis. This should be especially true for detailed investigations of dehumidifier design and performance and for situations, such as the collector/dehumidifier, which depart from that of an adiabatic wheel. The finite difference scheme presented here is a solution to a particular set of partial differential equations which have been widely used to describe desiccant dehumidifiers. Substitution of equations (33) into the overall DESSIM program is straightforward and results in a potentially accurate and useful tool for the investigation of these devices.

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