Reprinted from February 1987, Vol. 109, Journal of Solar Energy Engineering

K. Schultz[2]

R. Barlow[3]

A. Pesaran

F. Kreith

Solar Energy Research Institute,
Golden, CO 80401

An Analysis of a Direct Radiation Solar Dehumidification System[1]

A desiccant dehumidifier regenerated by direct absorption of solar radiation was investigated using a simplified numerical model (DESSIM) of the adsorpotion and desorption processes. This paper presents estimates of the performance of an air conditioning system in the ventilation mode using direct solar radiation regeneration. The effects of dehumidifier NTUs, heat exchanger performance, and insolation levels were also analyzed. The direct radiation regeneration system was found to have a COP less than that of other types of regeneration schemes.

1 Introduction

Desiccants such as silica gel have been used in air conditioning dehumidification processes for some time. Desiccant systems require periodic regeneration of the desiccant bed, which is normally achieved by passing hot air through the bed and drying it. This paper describes and analyzes a method for regenerating the desiccant bed using direct solar radiation.

The system analyzed in this paper is shown schematically in Fig. 1. It consists essentially of a rotating belt coated on the outer side with a desiccant material. The belt is installed in an enclosure into which solar radiation can pass through a transparent cover on the top side. The enclosure is divided into two compartments by the belt. In the upper compartment regenerated air enters perpendicular to the direction of the moving belt, flows through the space defined between the cover and the belt, and then exits through the front. This orientation of air flow and belt motion was chosen because it is similar to the configuration of a counterflow rotary dehumidifier for which numerical models exist. The desiccant is exposed to solar radiation, which provides the heat for regeneration directly. Desorption, which occurs in the upper half of the compartment, dries the desiccant. In the lower half of the compartment, wet process air enters through the front and interacts with the desiccant material on the belt for adsorption of water; the dry air then exits in the back.

The desorption-adsorption problem has been treated mathematically by several researchers [1,2,3,4]. The conventional approach has been to derive a set of differential equations describing conservation of mass and energy in the desiccant bed and to solve these coupled equations using finite difference techniques. The model used in this paper has been developed by Barlow [5]. It also uses a finite difference method in that the solution moves through discrete time and space steps. However, rather than directly solving differential equations, the model determines the amount of water vapor and thermal energy transferred between the airstream and the desiccant by using simple effectiveness equations from the theory of steady-state, mass and heat exchangers. This approach simplifies the mathematics by eliminating the need for a transformation of variables or sophisticated numerical techniques and makes the model easy to adapt to investigate a variety of adsorption problems, such as the rotating belt treated here.

2 Analysis and Computer Model

When humid air contacts desiccant particles, water molecules in the air at the surface of the particles are adsorbed by the desiccant. This creates a humidity gradient in the airstream and causes other water molecules to diffuse toward the surface where they, in turn, are adsorbed. The adsorption process releases an amount of energy, which for silica gel, a common adsorbent, is about 5 percent to 15 percent greater than the heat of condensation of water. The heat of adsorption elevates the temperature of the desiccant particles. Some of the heat is transferred to the airstream, and some is retained in the bed. Thus, the absorption process comprises simultaneous heat and mass transfer with thermal energy being generated in the desiccant.

The numerical model used in this analysis was originally developed to provide a simple means of studying the qualitative behavior of silica gel in packed beds. Primary variables were used to facilitate a physical understanding of the adsorption process. Initially, a simple, conventional finite difference scheme was used; however, the very small times steps dictated by stability requirements resulted in very long computation times. The technique to be described here was developed as an easily implemented alternative to available finite difference models for coupled heat and mass transfer in porous media. The technique is not based on deriving a mathematical solution to the equations for conservation of mass and energy in the transient adsorption process, but on solving equations for a discretized conceptual analog to the adsorption process. This approach has been validated by comparison with experimental data for a packed bed [5].

---

[1]This work was supported by the US Department of Energy, Office of Solar Heat Technologies.

[2]Currently at the Mechanical Engineering Department, University of Wisconsin, Madison, Wis.

[3]Currently at the Mechanical Engineering Department, Stanford University, Stanford, CA.

Contributed by the Solar Energy Division for publication in the JOURNAL OF SOLAR ENERGY ENGINEERING. Manuscript received by the Solar Energy Division, July, 1985.

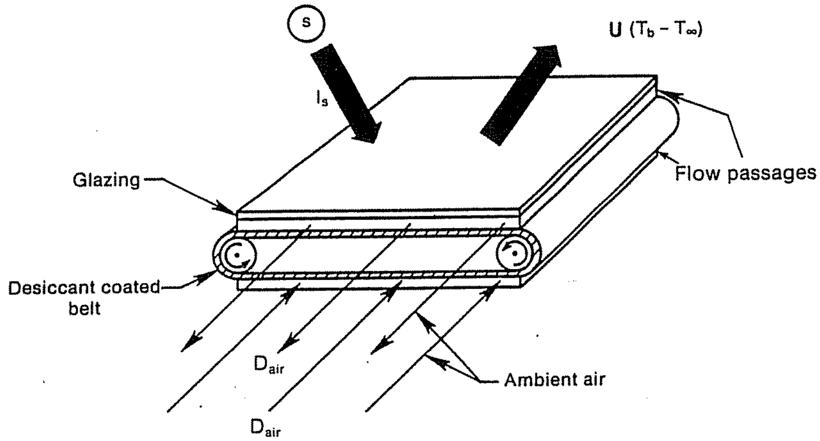

Fig. 1 Desiccant belt dehumidifier collector

The computer model for this simplified approach, called DESSIM, considers differential widths of the desiccant belt. The length in the flow direction is divided into N sections (typically 10 to 20). As illustrated in Fig. 2, the computer program can be thought of as carrying one parcel of air at a time through the system. Each increment in time corresponds to a certain mass of air passing any point on the belt. As each parcel of air is successively exposed to each desiccant section, mass transfer and heat transfer calculations are performed. One begins with initial conditions of temperature and water content along the belt inlet or boundary conditions of fixed air temperature and humidity ratio. The calculation procedure moves in space along the belt in the air flow direction, then advances in time, following the element as it rotates, and repeats the process.

The simplicity and flexibility of the model are a result of the mass and heat transfer calculations being performed at each node. Although the sorption process is transient, the transfer

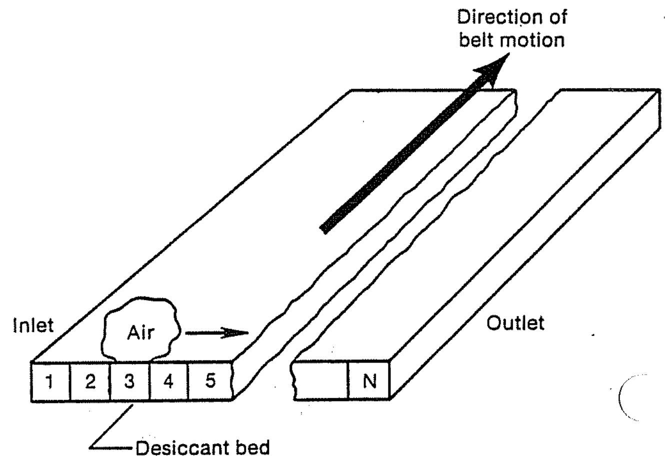

Fig. 2 Discretization of desiccant bed used in DESSIM calculations

Nomenclature

A_c = collector area (m^2)
A_s = transfer surface area in bed node (m^2)
c_a = specific heat of air (J/kg°C)
c_b = specific heat of desiccant belt (J/kg°C)
c_w = specific heat of water (J/kg°C)
C_{air} = thermal capacity rate of air (W/°C)
C_{bed} = thermal capacity rate of desiccant belt (W/°C)
CC = C_{min}/C_{max}
C_{max} = larger capacity rate (W/°C)
C_{min} = minimum capacity rate (W/°C)
COP = thermal coefficient of performance, cooling effect/net solar energy input
D_{air} = capacity rate of air (kg/s)
D_{bed} = capacity rate of desiccant belt (kg/s)
D_{min} = minimum capacity rate (kg/s)
g = mass transfer coefficient (kg/m^{2}s)
h = enthalpy of moist air (J/kg)
h_b = enthalpy of moist desiccant (J/kg)
h_c = heat transfer of coefficient (W/m^2°C)
H_{ads} = heat of adsorption of water (J/kg)
H_{vap} = heat of vaporization of water (J/kg)
I_s = incident solar radiation (W/m^2)
Le = effective Lewis number
m = mass fraction of water vapor in air
m_s = mass fraction of water vapor in air at equilibrium with the desiccant surface (function of T,X)
M_a = mass of dry air parcel (kg)
M_b = mass of dry bed node (kg)
M_d = mass of desiccant in belt node (kg)
M_{H_2O} = mass of water adsorbed (kg)
N = number of bed nodes
NTU = number of transfer units
Q_s = net energy gain from incident solar radiation (W/m^2)
T = temperature (°C)
T_b = temperature of belt node (°C)
$T_{b,int}$ = intermediate temperature of belt node (°C)
w = humidity ratio (kg/kg dry air)
X = desiccant water content (kg water/kg dry desiccant)
Y = kg water/(kg water + kg desiccant) $= X/(1+X)$
U = heat loss coefficient (W/m^2°C)
α = absorptance of the belt
Δt = time step (s)
ϵ = effectiveness of exchange process
ϵ_{HX} = effectiveness of system heat exchanger
τ = transmittance of the cover of the collector

Subscripts

a = air state
b = bed state
f = final state
g = mass transfer
h = heat transfer
i = initial state
∞ = ambient

calculations can be performed in steps with effectiveness equations for steady-state mass and heat exchangers. Final moisture contents and temperatures for the air parcel and the bed section at the end of each time step are taken to be the same as the outlet moisture contents and temperatures from exchangers that have steady flows of air and desiccant material with inlet conditions equal to the initial conditions of the air parcel and the bed section. Applying these exchanger equations over time steps and bed sections that are small compared to the scale of the overall process preserves the basic physics of the transient problem.

DESSIM was originally formulated to model adsorption in fixed beds of silica gel particles. Consider a discrete heat or moisture exchange process in which the mass of air corresponding to the flow during a single time step passes through the mass of silica gel contained in one node of the desiccant bed. An approximate physical analog to this process is the flow of air and silica gel through a counterflow heat or mass exchanger. The effectiveness of a heat or mass exchange process is defined as the actual heat or mass transfer divided by the maximum possible. The analytical expression for the effectiveness of a simple, steady-state counter-flow exchanger is [6]

$$\epsilon = \frac{1-\exp[-\mathrm{NTU}(1-CC)]}{1-CC\exp[-\mathrm{NTU}(1-CC)]}, \tag{1}$$

where NTU is the number of transfer units per node [6] and CC is the ratio of the minimum capacity rate $C_{\min}$ to the maximum capacity rate $C_{\max}$ on the two sides of the exchanger. Even though the flow configuration suggests a crossflow pattern, the relation for counterflow effectiveness was used. This simplification is justified because Kays and London [7] have shown that for NTUs smaller than one for a particular node, the difference between crossflow and counterflow effectivenesses is negligible. In this study, the typical NTU per node was between 0.1 to 0.25.

For the heat exchange process we can obtain NTU from the equation

$$\mathrm{NTU}_h = h_c A_s / C_{\min}. \tag{2}$$

The appropriate capacity rates are

$$C_{\mathrm{air}} = M_a c_a / \Delta t, \tag{3}$$

where M_a is the mass of dry air that passes through the bed during time step Δt and

$$C_{\mathrm{bed}} = (M_b c_b + M_d X c_w)/\Delta t. \tag{4}$$

For the mass exchange process we have

$$\mathrm{NTU}_g = g A_s / D_{\min}, \tag{5}$$

where g is an effective gas-side mass transfer coefficient modified to account for a resistance-to-moisture diffusion within the solid particles as shown in Barlow [5]. $D_{\min}$ is the smaller of the following two capacity rates:

$$D_{\mathrm{air}} = M_a(1+w_i)/\Delta t \tag{6}$$

and

$$D_{\mathrm{bed}} = M_b(1+X_i)\left[\frac{\partial Y}{\partial m_s}\right]/\Delta t. \tag{7}$$

The partial derivative of Y with respect to m_s, $\partial Y/\partial m_s$, replaces the inverse of the Henry Number used in gas/liquid mass exchanger analysis [8] and is analogous to the specific heat used in the expressions for capacity rates in heat exchangers.

The computation procedure of air parcels being exposed sucessively to new sections of the belt is presented next. First, the equilibrium properties m_s, $\partial Y/\partial m_s$, and $H_{\mathrm{ads}}/H_{\mathrm{vap}}$ for the desiccant are calculated, based on the initial temperature and water content of the belt section. The equations for these properties for silica gel are presented in Barlow [5].

The maximum possible mass transfer is defined by the difference between the initial vapor mass fraction in the air parcel and the vapor mass fraction of air at equilibrium with the bed at its initial loading and temperature. Thus, the final vapor mass fraction of the air parcel is

$$m_f = m_i - \epsilon_g(m_i - m_s), \tag{8}$$

and the final humidity ratio is

$$w_f = m_f/(1-m_f). \tag{9}$$

The mass of water transferred between the air parcel and the belt section is

$$M_{H_2O} = M_a(w_i - w_f), \tag{10}$$

and the final loading of the desiccant is, therefore,

$$X_f = (X_i M_d + M_{H_2O})/M_d. \tag{11}$$

Second, an energy balance is performed to account for the effect of the heat of adsorption and the net energy gain by direct solar radiation. Here, the air temperature is assumed to remain the same, and an intermediate temperature for the belt section $T_{b,\mathrm{int}}$ is calculated

$$\begin{aligned} T_{b,\mathrm{int}} = {} & \{T_{bi}M_b c_{bi} + M_a[h(T_{ai},w_i) - h(T_{ai},w_f)] \\ & + M_{H_2O}(H_{\mathrm{ads}} - H_{\mathrm{vap}}) + Q_s \Delta t A_s\}/M_b c_{bf}, \end{aligned} \tag{12}$$

where c_{bi} and c_{bf} are specific heats for the belt based on the initial and final water contents, and Q_s is the net energy gain from the incident solar radiation to the belt [9]

$$Q_s = [I_s \alpha\tau - U(T_{b,\mathrm{int}} - T_\infty)]. \tag{13}$$

The ease with which the solar energy gain has been included in the calculations is an example of DESSIM's versatility, which has greatly simplified the modeling of the direct radiation dehumidifier.

Third, the heat transfer between the air parcel and the belt section is calculated

$$Q_h = \epsilon_{HX} C_{\min}(T_{b,\mathrm{int}} - T_{ai}). \tag{14}$$

The final temperatures are given by an energy balance between the dessicant node and the air parcel as

$$T_{af} = T_{ai} + Q_h \Delta t / M_a c_{af} \tag{15}$$

$$T_{bf} = T_{b,\mathrm{int}} - Q_h \Delta t / M_b c_{bf}. \tag{16}$$

The belt section temperature and loading are stored, and the air temperature and humidity ratio are passed to the next belt section and used as initial values for the next mass and heat transfer calculations. Final air conditions leaving the last belt section are averaged over all time steps to give the average outlet state of the airstream for one side of the belt.

The temperature and loading profiles along the belt are preserved and used as the initial conditions of the dehumidifier as it rotates into the other airstream. This completes a single cycle of the belt. To predict the steady-state performance of the dehumidifier, the above calculations are carried through several cycles until the solution converges. The amount of water adsorbed by the desiccant belt as it rotates through the dehumidification period is used as the convergence criterion. Although the mass and heat transfer calculations are performed uncoupled, the iterative nature of the solution, together with the small time and spacial steps used, cause the mass and heat transfer calculations to be coupled.

A version of DESSIM for analyzing adsorption in packed beds was validated by comparing experimental data from the literature with data from SERI's desiccant test laboratory [5]. Figures 3 and 4 show typical results from these comparisons and demonstrate that the model captures the physics of the adsorption process, producing good predictions in spite of its somewhat ad hoc nature. Measured and predicted results for adsorption in a fixed, laminar-flow-channel dehumidifier have also shown good agreement [10]. Another version of DESSIM

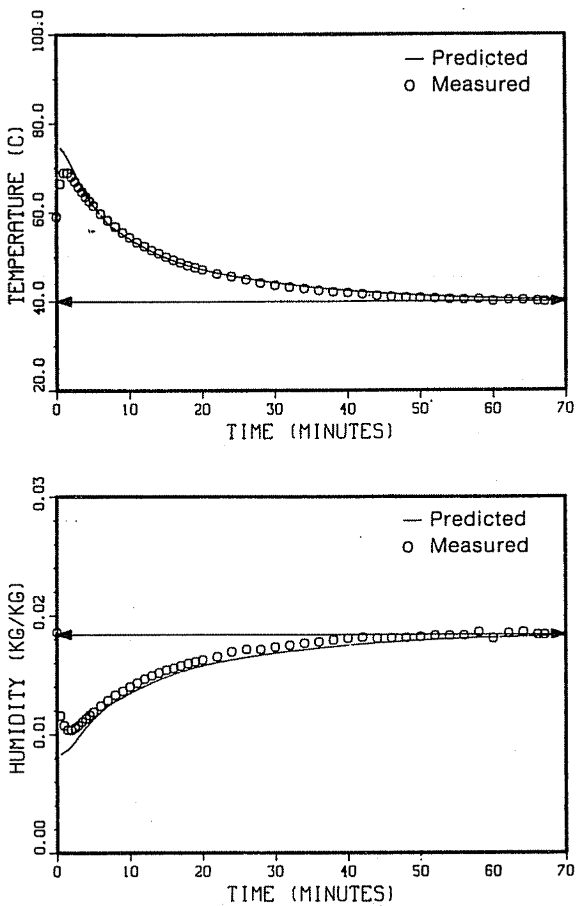

Fig. 3 Measured and DESSIM predicted outlet air temperature and humidity (data from [5])

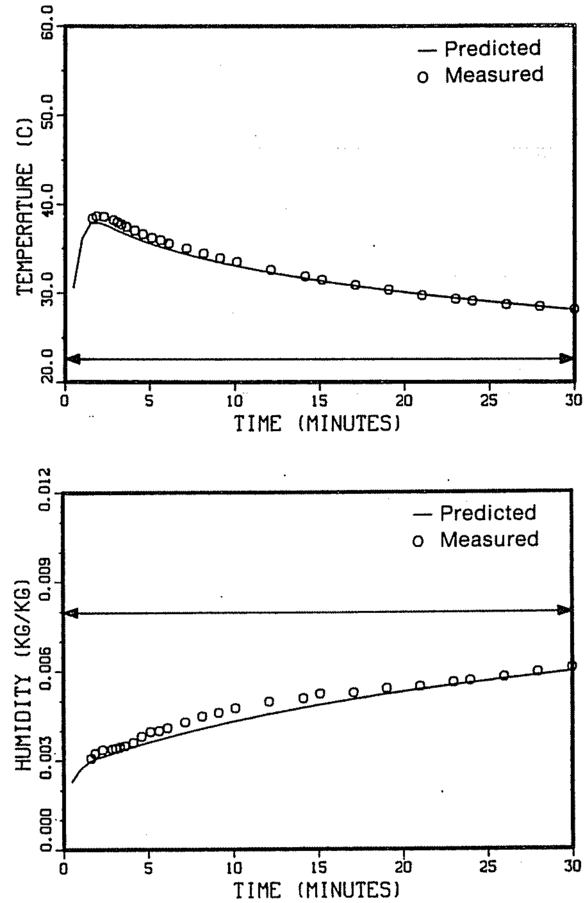

Fig. 4 Measured and DESSIM predicted outlet air temperature a... humidity (data from [5])

that models the steady-state operation of an adiabatic rotary dehumidifier has been compared with a direct finite difference solution of the governing equations (MOSHMX [11]). Typical errors resulting from using DESSIM instead of MOSHMX are shown in Fig. 5. For a ventilation cycle with heat exchanger effectiveness of 95 percent and a thermal COP near 1.0, an error in dehumidifier outlet temperature of $+0.5°C$ results in an error in COP of approximately $+3$ percent; an error in the humidity ratio of $+0.3$ g/kg gives an error of -4 percent. When these errors are combined, the error in COP is approximately 1 percent because the temperature and humidity errors offset each other. The DESSIM concept is being thoroughly analyzed and compared with MOSHMX [12] to determine DESSIM's accuracy and limitations over a wide range of operating conditions. Current indications suggest that DESSIM is sufficiently accurate for predicting gross desiccant cooling cycle performance.

3 Results and Discussion

To determine the performance of the direct radiation concept, the rotating belt system shown in Fig. 1 was incorporated into a desiccant cooling system operating in the ventilation mode as shown in Fig. 6(*a*). In this system, room air enters at point 1, passes through the evaporative cooler EC-1, and enters the rotary heat exchanger HX at point 2 where it reclaims energy from the hot adsorption airstream. The desorption process, driven by solar radiation, takes place in the belt between state points 3 and 4. The heated airstream and the heated air leaving the belt passage at state point 4 is discharged into the atmosphere.

Ambient air enters the lower part of the belt system at state point 5, exits at 6, and then is cooled as it passes through the

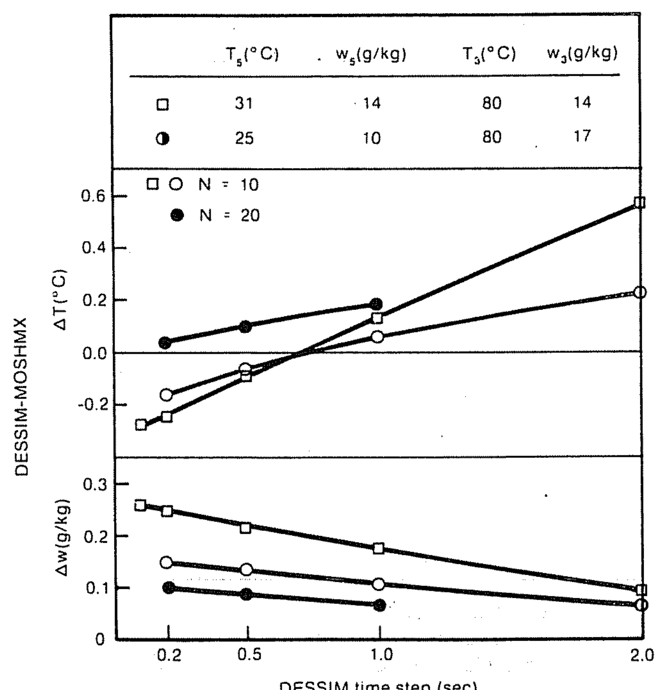

Fig. 5 Errors in adiabatic rotary dehumidifier outlet states betw... predictions of DESSIM and MOSHMX for Le = 1

heat exchanger. After passing through the evaporative cooler EC-2, it enters the room at state point 8. The psychometric chart in Fig. 6(*b*) illustrates the operation of the system.

Table 1 presents the nominal operating parameters for which performance calculations were made. The air states represent the American Refrigeration Institute (ARI) design points. The desiccant belt properties are similar to those considered by Barlow [5] in the analysis of a parallel-plate dehumidifier. Note the very low NTUs (approximately 2) of the belt collector design. This is due to the need to expose all of the surface area of the belt to the incident solar radiation.

During the initial calculation, the effect of belt speed on COP was investigated. The COP is defined as the ratio of the cooling effect produced by the system [D_{air} $(h_1 - h_8)$ in Fig. 6(*b*)] to the net solar energy input ($Q_s A_c$, equation (13)). An optimum belt speed yields a maximum COP, as shown in Fig. 7; however, this maximum condition is relatively insensitive to the belt speed over a wide range because of the low NTUs of the dehumidifier in this system. Note that evaporative coolers have an effectiveness of 0.9 in this study.

In a second series of runs the significance of the heat exchanger effectiveness on the COP, specific cooling capacity, and maximum latent capacity was investigated. Figure 8 shows the relationship between COP and heat exchanger effectiveness for a belt speed corresponding approximately to the optimum. The COP and the cooling capacity of the system are sensitive to the heat exchanger effectiveness; however, the maximum latent capacity is not. The maximum latent capacity

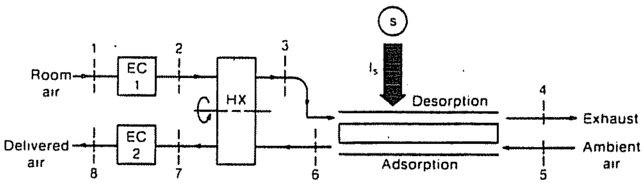

Fig. 6(a) Direct radiation dehumidifier in ventilation mode

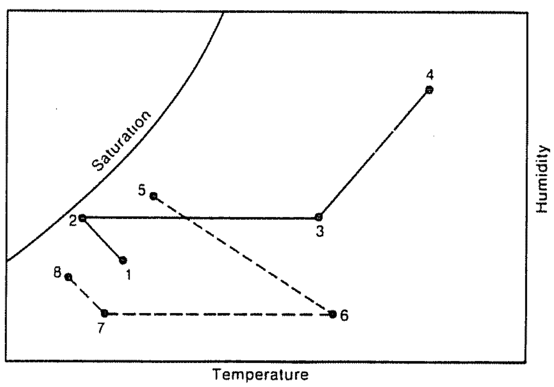

Fig. 6(*b*) State points on psychometric chart

is determined by the dryness of the dehumidifier process outlet, which is mainly a function of the effective regeneration temperature. This indicates that the solar gain is creating a very high effective regeneration temperature, so the temperature at the collector inlet (state 3) is not important. Therefore, the main function of the heat exchanger is to cool the adsorption airstream from state 6 toward state 2. This contrasts with the conventional rotary dehumidifier system where the energy reclamation of the heat exchanger between states 2 and 3 is very important.

COPs of conventional rotary dehumidifier systems are now in the vicinity of 1.0 [10]. Direct radiation system COPs are much less than this. One reason for this difference is that in the direct radiation system the air used for regeneration leaves the system at a much higher temperature than in a comparable parallel-passage system. Since the heat used to increase the air temperature is not used for any useful purpose, it is lost and reduces the COP. This indicates that the solar gain present may be excessive compared with the requirements for desiccant regeneration and air flow rates. Another reason for the lower COP is that the NTU for the direct radiation system is only on the order of 2, whereas with the parallel passage design NTUs on the order of 20 are possible. The reason for this is that in the parallel-passage design, many flow passages per unit frontal area can be incorporated within the system, whereas in the direct radiation design only one passage can be used. Since NTU is proportional to transfer area the parallel passage design is more favorable from this perspective.

Figure 9 shows the effect of insolation levels. While cooling capacity appears to drop sharply below insolation levels of 350 W/m^2, COP increases for NTUs greater than 1 as insolation levels decrease. The small gain in cooling capacity and drop in COP above 350 W/m^2 indicates again that excessive solar gain may be present. The direct radiation system operates quite effectively at low insolation levels and thus may have advantages in some geographic regions.

Figures 8 and 9 both indicate that significant potential improvements in performance of the dehumidifier can be

Table 1 Nominal operating parameters

Ambient state:	35°C, 0.014 kg/kg
Room state:	27°C, 0.011 kg/kg
Belt:	total mass, 0.35 kg/m^2 desiccant mass fraction, 62 percent – silica gel period, 600 s
Collector:	width, 1 m length, 3m air flow rate, 3.33×10^{-3} m^3/m^2 s NTU, 1.78 flow passage spacing, 1.5 cm heat loss coefficient U, 5.0 W/m^2 °C

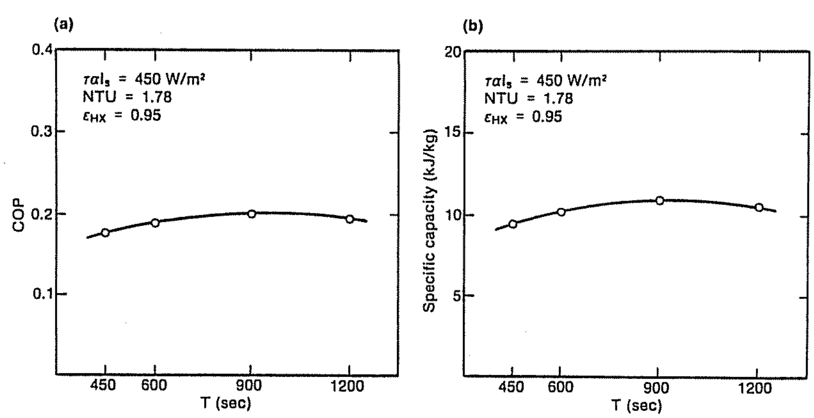

Fig. 7 Effect of belt rotation period on system performance

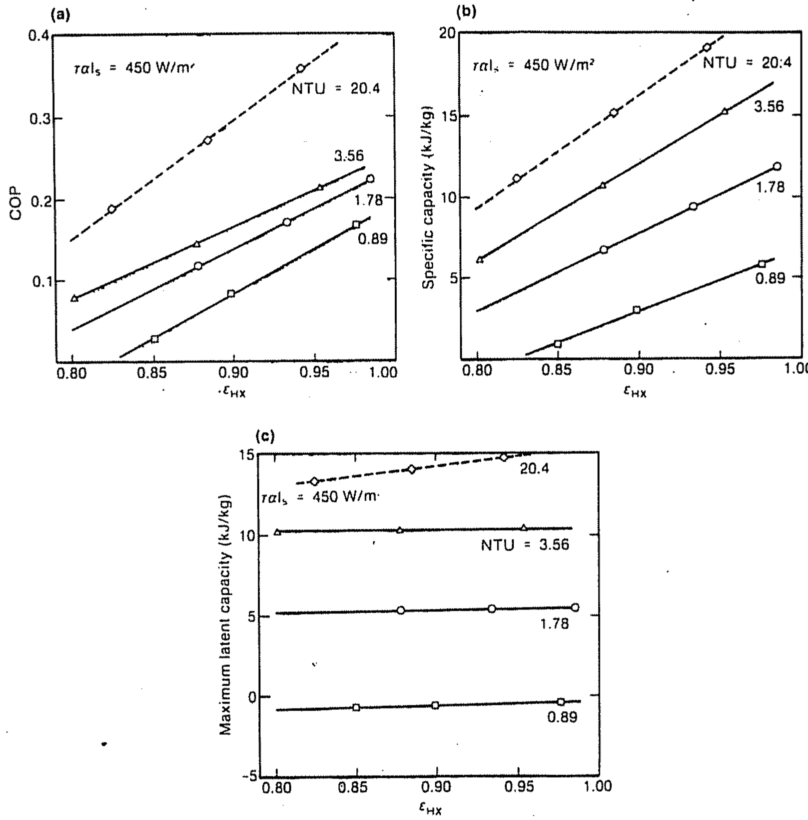

Fig. 8 Effect of heat exchanger effectiveness and dehumidifier NTUs on system performance

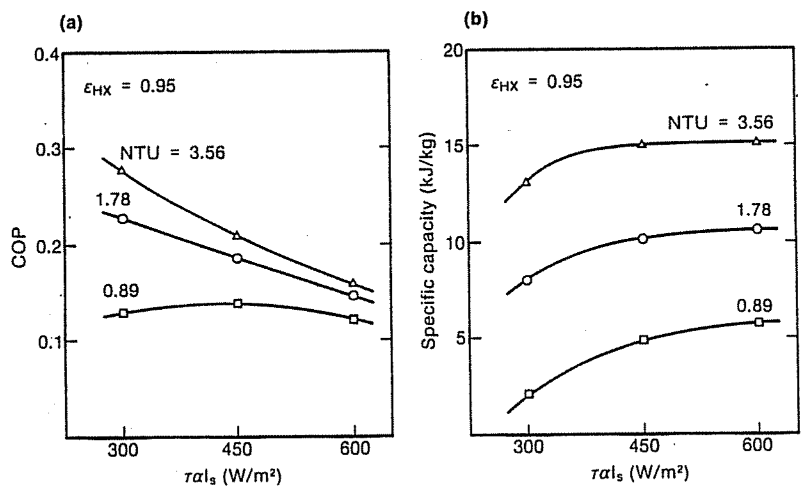

Fig. 9 Effect of insolation level on system performance

achieved if NTUs can be increased. These improvements could be accomplished by reducing the size of the flow passage or by using heat transfer promoters. The improvements possible with this approach are limited by the increase in pumping power accompanying such enhancement measures. Even if NTUs on the order of 20 could be reached, the COP of the direct radiation system is much less than that possible from conventional dehumidifier systems, although the "fuel" is free.

The above analysis is limited to an effective Lewis number of unity. It is expected that effective Lewis numbers greater than unity (i.e., significant diffusional resistance in the desiccant) would reduce performance, both specific capacity and COP. This was not quantified because of the already low performance of the above system.

A somewhat similar system to the one treated in this paper has been tested by Ohigoshi et al. [13], who used a porous belt made of fibrous activated carbon as the desiccant. In this Japanese design the air passed through the porous belt, which was exposed to direct radiation from above for regeneration. The COPs achieved by this approach were of the same order of magnitude as those obtained for the system presented in this paper. Furthermore, Ohigoshi et al. observed that little or no reduction in performance occurred with lower solar radiation input, and the authors claimed great advantages for the direct radiation dehumidification process because it can maintain a high COP throughout the day. The results presented here do not substantiate their claim, but if the endless belt dehumidification system described here were used during the winter as a solar collector for heating by simply turning off the moving belt, a relatively compact, combined heating and cooling system could be constructed. In the final analysis, of course, economic considerations will decide if such a combined system is viable.

4 Conclusions

A method for regenerating a desiccant bed used for dehumidifying air using direct solar radiation was described and analyzed. A simplified numerical model (DESSIM) incorporating steady-state equations of heat and mass exchangers was used in the analysis. The performance of a desiccant cooling system operating in a ventilation mode using this direct radiation concept was evaluated and the optimum belt speed that gives maximum COP was determined. It was found that the COP and the cooling capacity of the system increase when the heat exchanger effectiveness increases. The COP of the system is from 0.1 to 0.3, which is considerably less than that of a cooling system using a parallel-passage rotary dehumidifier regenerated indirectly with solar heat. From the effect of the solar radiation level on COP and cooling capacity of the system, it was concluded that the direct radiation system operates quite effectively at low insolation levels and may have potential in some geographic regions.

References

1 Chi, C. W., and Wasan, D. T., *AIChE Journal, Vol. 16*, 1972, pp. 23–31.

2 Maclaine-cross, I. L., and Banks, P. J., *Int. J. Heat Mass Transf., Vol. 15*, 1972, pp. 1225–1242.

3 Pesaran, A. A., "Moisture Transport in Silica Gel Particle Beds," Ph.D. Dissertation, University of California, Los Angeles, CA, 1983.

4 Mathiprakasam, B., and Lavan, Z., ASME JOURNAL OF SOLAR ENERGY ENGINEERING, Vol. 102, 1980, pp. 73–79.

5 Barlow, R. S., "An Analysis of the Adsorption Process and of Desiccant Cooling Systems: A Pseudo-Steady-State Model for Coupled Heat and Mass Transfers," SERI/TR-631-1330, Solar Energy Research Institute, Golden, CO, 1981.

6 Kreith, F., *Principles of Heat Transfer*, 3rd Ed., Harper and Row, New York, 1978.

7 Kays, W. M., London, A. L., *Compact Heat Exchangers*, McGraw-Hill, New York, 1964.

8 Edwards, D. K., Denny, V. E., and Mills, A. F., *Transfer Processes*, Holt, Rinehart and Winston, Inc., New York, 1973.

9 Kreith, F., and Kreider, J. F., *Principles of Solar Engineering*, McGraw-Hill, New York, 1978.

10 Schlepp, D., "A High Performance Dehumidifier for Solar Desiccant Cooling Systems," SERI/TP-252-1979, Solar Energy Research Institute, Golden, CO, 1984.

11 Maclaine-cross, I. L., "A Theory of Combined Heat and Mass Transfer in Regenerators," Ph.D. Thesis, Monash University, Clayton, Victoria, Australia, 1974.

12 Schultz, K., personal communication, University of Wisconsin, Madison, WI, Apr. 1986.

13 Ohigoshi, T., Kishimoto, T., Imamura, Y., Shimada, M., and Komagata, H., *Progress in Solar Energy*, Vol. 6, Proc. of the Annual Meeting of ASES, Minneapolis, MN, 1983, pp. 187–191.