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AN ANALYSIS OF A DIRECT RADIATION SOLAR DEHUMIDIFICATION SYSTEM

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ABSTRACT

SERI researchers investigated a desiccant dehumidifier that is regenerated by direct absorption of solar radiation using a simplified numerical model (DESSIM) of the adsorption and desorption processes. This paper presents estimates of the performance of a solar-fired air conditioning system (ventilation cycle) containing the dehumidifier/collector. The researchers also considered the effects of dehumidifier NTUs, heat exchanger performance, and insolation levels. The direct radiation system can operate effectively at low insolation levels and thus may have some advantages in some geographic areas.

NOMENCLATURE

A_c	collector area (m^2)
A_s	surface area in bed section (m^2)
c_a	specific heat of air (J/kg ^oC)
c_b	specific heat of desiccant belt (J/kg ^oC)
c_w	specific heat of water (J/kg ^oC)
C_{air}	thermal capacity rate of air (W/^oC)
C_{bed}	thermal capacity rate of desiccant belt (W/^oC)
CC	C_{min}/C_{max}
C_{max}	larger capacity rate (W/^oC)
C_{min}	minimum capacity rate (W/^oC)
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 ^oC)
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)
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 section (kg)
M_d	mass of desiccant in belt section (kg)
M_{H_2O}	mass of water adsorbed (kg)
N	number of bed sections
NTU	number of transfer units
Q_s	net energy gain from incident solar radiation
T	temperature (^oC)
T_b	temperature of belt section (^oC)
$T_{b,int}$	intermediate temperature of belt section (^oC)
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 ^oC)
α	absorptance of the belt
Δt	time step (s)
ε	effectiveness of exchange process

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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

H_2O water

∞ ambient

1. INTRODUCTION

Dehumidification processes with desiccants such as silica gel have been used in air conditioning for some time. Desiccant systems require periodic regeneration of the desiccant bed. This regeneration is normally achieved by passing hot air through the bed and drying it. In solar energy systems the air is heated in a collector, whereas in gas-driven desiccant systems the air is heated by combustion of the natural gas. 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 Figure 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 regeneration air enters perpendicular to the direction the belt is moving, flows through the space defined between the cover and the belt, and then exits through the front. The desiccant is exposed to solar radiation, which provides the heat for regeneration directly. Desorption occurs in the upper half of the compartment and the desiccant is dried. 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 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.

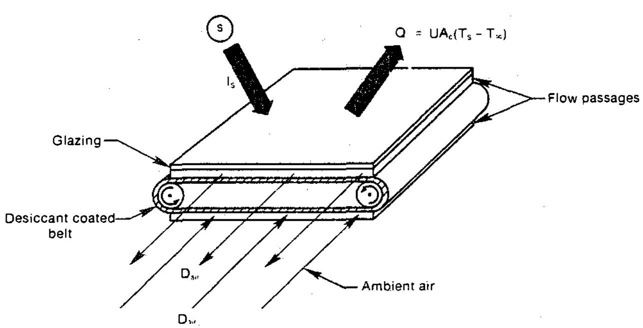

Fig. 1. Desiccant Belt Dehumidifier Collector

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 migrate 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% to 15% greater than the heat of condensation of water. This 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 adsorption process comprises simultaneous heat and mass transfer with thermal energy being generated in the desiccant. With the direct solar concept an additional thermal energy source in the dessicant must be included.

To analyze the adsorption process a computer model called DESSIM was developed. The model considers an element of the desiccant belt of differential width. 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 the 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

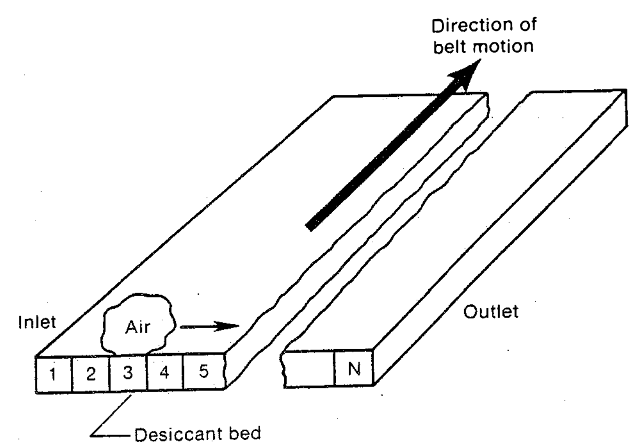

Fig. 2. Discretization of Desiccant Bed Used in DESSIM Calculations

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 come from the manner in which the mass and heat transfer calculations are performed at each node. Although the sorption process is transient, the transfer calculations can be performed in steps with effectiveness equations for steady-state, counter-flow 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 counter-flow 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. The fact that the exchanger equations are analytical solutions to differential equations describing the approach of an exchange process toward a maximum effectiveness allows the use of relatively large time and space increments without using complicated numerical techniques or compromising accuracy.

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 [7]

$$\varepsilon = \frac{1 - \exp\,[-NTU(1 - CC)]}{1 - CC\,\exp\,[-\,NTU(1 - CC)]}\ , \qquad (1)$$

where NTU is the number of transfer units [7] 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.

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

$$NTU_h = h_c A_s / C_{min}\ . \qquad (2)$$

The appropriate capacity rates are

$$C_{air} = M_a c_a / \Delta t\ , \qquad (3)$$

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

and

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

For the mass exchange process we have

$$NTU_g = g A_s / D_{min}\ , \qquad (5)$$

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:

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

and

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

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, as an air parcel is exposed successively to a new section of the belt, is presented next. First, the equilibrium properties m_s, $\partial Y/\partial ms$, and H_{ads}/H_{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

$$m_f = m_i - \varepsilon_g(m_i - m_s)\ , \qquad (8)$$

and the final humidity ratio is

$$w_f = m_f/(1 - m_f)\ . \qquad (9)$$

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

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

and the final loading of the desiccant therefore is

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

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 not to change, and an intermediate temperature for the belt section $T_{b,int}$ is calculated

$$T_{b,int} = \{T_{bi} M_b c_{bi} + M_a[h(T_{ai},W_i) - h(T_{ai},W_f)] + M_{H_2O}(H_{ads} - H_{vap}) + Q_s\}/M_b c_{bf}\ , \qquad (12)$$

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]

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

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

$$Q_h = \varepsilon_{HX} C_{min}(T_{b,int} - T_{ai})\ . \qquad (14)$$

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

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

$$T_{bf} = T_{b,int} - Q_h\Delta t/M_b c_{bf}\ . \qquad (16)$$

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 do cause the mass and heat transfer calculations to be coupled.

A version of DESSIM was previously compared with single-blow experimental results for a packed bed [5] and a parallel passage dehumidifier [6], and good agreement between predictions and experimental data was demonstrated. Another version of DESSIM 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 [10]). Typical errors resulted in using DESSIM instead of MOSHMX. This is shown in Figure 3. For a ventilation cycle with heat exchanger effectiveness of 95% 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%; an error in the humidity ratio of +0.3 g/kg gives an error of -4%. When these errors are combined, the error in COP is approximately 1% because the temperature and humidity errors offset each other. The conclusion, therefore, is that DESSIM is sufficiently accurate for predicting desiccant cooling cycle performance.

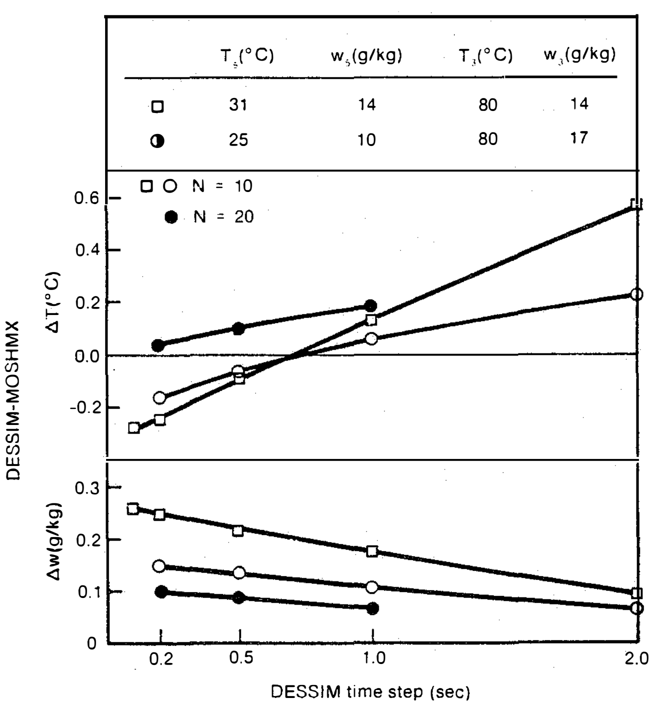

Fig. 3. Errors in Adiabatic Rotary Dehumidifier Outlet States between Predictions of DESSIM and MOSHMX

3. RESULTS AND DISCUSSION

To determine the performance of the direct radiation concept the rotating belt system shown in Figure 1 was incorporated into a desiccant cooling system operating in the ventilation mode as shown in Figure 4a. 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, occurs 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 heat exchanger. After passing through the evaporative cooler, EC-2, it enters the room at state point 8. The operation of the system is illustrated the psychometric chart in Figure 4b.

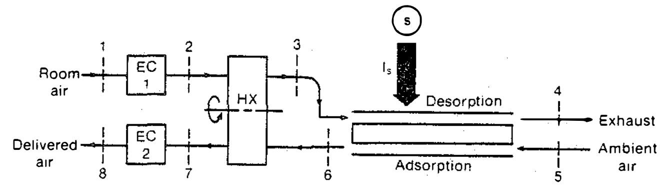

Fig. 4a. Direct Radiation Dehumidifier in Ventilation Mode

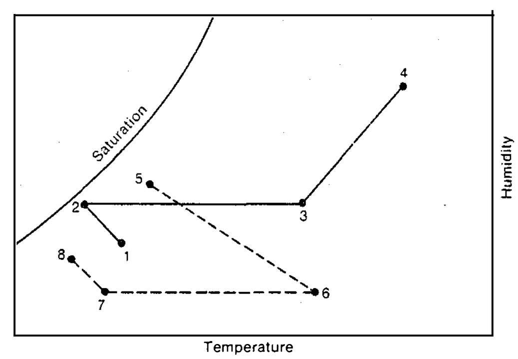

Fig. 4b. Statepoints on Psychometric Chart

The nominal operating parameters for which performance calculations were made are presented in Table 1. The air states represent the 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 the coefficient of performance (COP) was investigated. The COP is defined as the ratio of the cooling effect produced by the system ($h_1 - h_8$ in Figure 4b) to the net solar energy input ($Q_s A_c$, Eq. 13). There is an optimum belt speed that yields a maximum COP as shown in Figure 5. However, this maximum condition is relatively insensitive to the belt

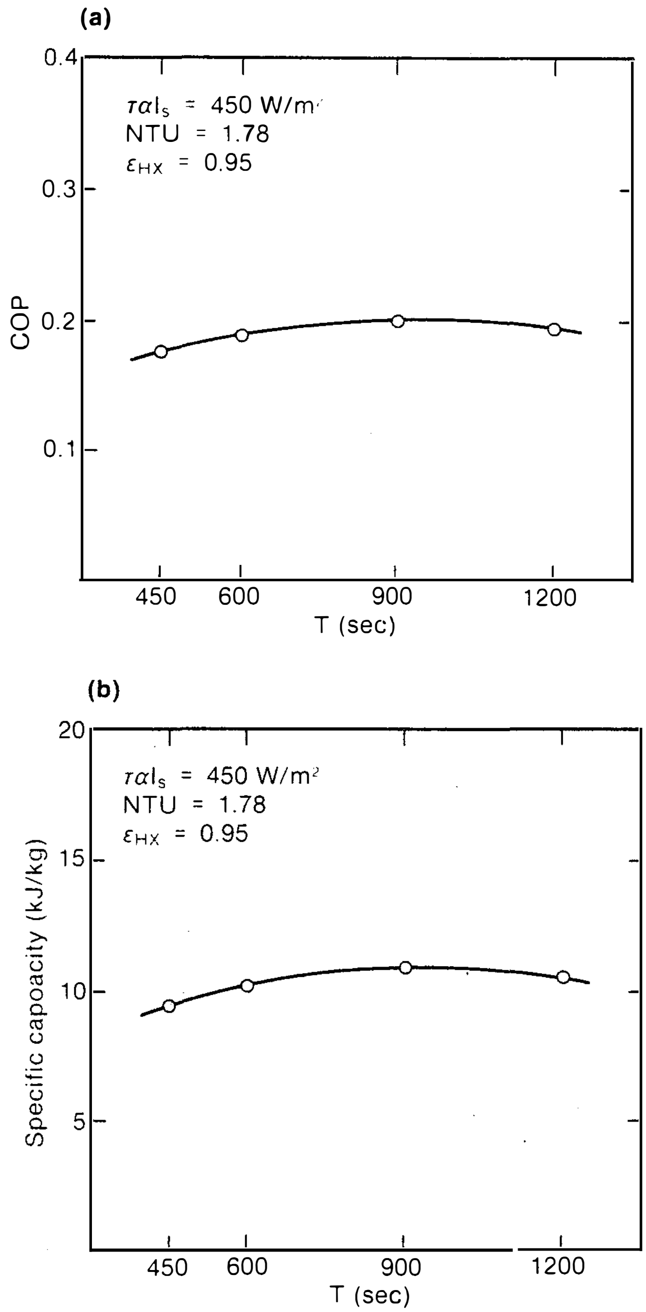

Fig. 5. Effect of Belt Rotation Period on System Performance

speed over a wide range due to 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 6 shows the relationship between COP and heat exchanger effectiveness for a belt speed corresponding approximately to the optimum. The COP and the cooling

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%--silica gel period, 600 s
Collector:	width, 1 m length, 3 m 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

capacity of the system are sensitive to the heat exchanger effectiveness; however, the maximum latent capacity is not. The maximum latent capacity 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.

The COPs of conventional rotary dehumidifier systems are now in the vicinity of 1.0 [6]. The direct radiation system COPs are much less that this. One reason for this difference is because 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 desiccant and air quantities. Another reason for the lower COP is that the NTU for the direct radiation system is only the order of 2 whereas with the parallel passage design NTUs of 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 7 shows the effect of insolation levels. While cooling capacity appears to drop sharply below insolation levels of 350 W/m^2, COP does increase 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 6 and 7 both indicate significant potential improvements in performance of the dehumidifier, if NTUs can be increased. This 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

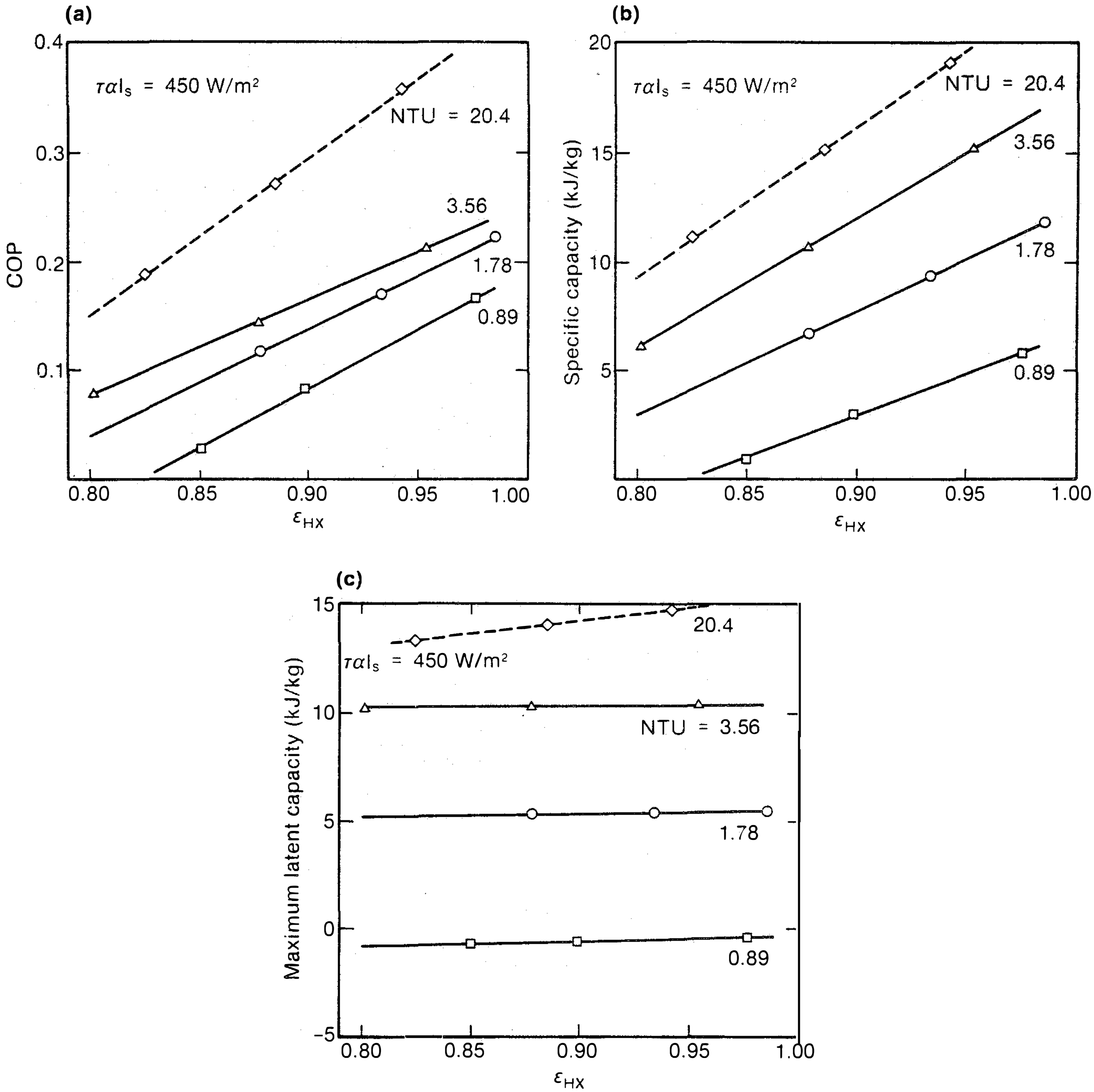

Fig. 6. Effect of Heat Exchanger Effectiveness and Dehumidifier NTUs on System Performance

the direct radiation system is much less than that possible from conventional dehumidifier systems, although the "fuel" is free.

A somewhat similar system to the one treated in this paper has been tested by Ohigoshi et al. [11] using 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. Also 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

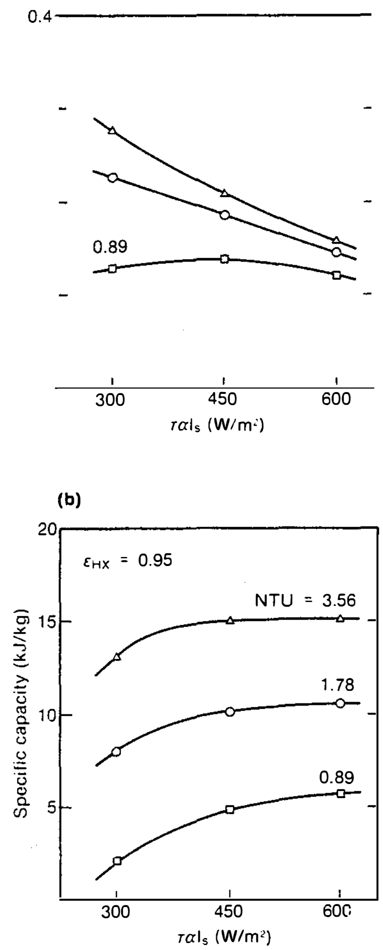

Fig. 7. Effect of Insolation Level on System Performance

process because it can maintain a high COP throughout the day. The results presented in this paper do not substantiate this claim, but if the endless belt dehumidification system described in this paper 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.

CONCLUSIONS

A method for regenerating the desiccant bed used for dehumidification of 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. SERI researchers evaluated the performance of a desiccant cooling system operating in ventilation mode using this direct radiation concept and found that there is an optimum belt speed to give maximum COP. They also 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 lower than that of a cooling system using a parallel-passage rotary dehumidifier regenerated indirectly with solar heat (COP of 1.0). From the effect of the solar radiation level on COP and cooling capacity of the system, they concluded that the direct radiation system can operate quite effectively at low insolation levels and thus may have advantages in some geographic regions.

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