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AN OVERVIEW OF OPEN-CYCLE DESICCANT COOLING SYSTEMS AND MATERIALS

R. COLLIER
F. ARNOLD
R. BARLOW

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Solar Energy Research Institute
A Division of Midwest Research Institute

1617 Cole Boulevard
Golden, Colorado 80401

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INTRODUCTION

Desiccant cooling systems process water vapor in the earth's atmosphere to produce cooling. Since mass transfer occurs between the system and its environment, they are commonly referred to as "open-cycle" systems. These systems all use a liquid or solid material called a desiccant to remove water vapor from the air. The process by which water is removed is most often adsorption on the solid desiccants and absorption in the liquid desiccants.

Desiccant systems are presently used in industrial air-drying applications. There are solid systems marketed by Bry-Air® and Cargocaire® and liquid systems marketed by Niagara® and Kathabar®. The first two use a desiccant-laden wheel in which air may flow in the axial direction only. The solid desiccant (lithium chloride salt or silica gel) is impregnated into the wheel material or encapsulated as a packed bed. Air to be dried flows through one side of the wheel, while the desiccant on the other side of the wheel is being dried by an externally heated air stream. These two air streams must be kept physically separate in order to maintain the distinctly separate functions of air drying and desiccant regeneration.

The liquid systems use two separate spray chambers for the processes of drying and regeneration. The "strong," concentrated solution (triethylene glycol or a lithium chloride solution in water) is sprayed over cooling coils whose temperature is maintained by water from a cooling tower or some other low-temperature source. Air is dried as it passes through the spray. When the desiccant solution absorbs sufficient moisture from the air, it is pumped to another chamber where it is sprayed into an externally heated air stream or over heated coils. This regenerates the solution, which is used again in the drying process.

These commercial systems are intended as air driers only and do not produce a significant net cooling. There are important differences in design philosophy between solar-regenerated desiccant cooling systems and commercial desiccant air driers. The most important difference involves thermal and electrical coefficients of performance (COPs). Commercial desiccant air drier manufacturers have chosen markets in which vapor-compression equipment cannot compete: applications where extremely dry air (≈2-10% relative humidity) is required. The energy requirements to achieve these conditions have not been a major concern, and, as a consequence, the desiccant dehumidifiers often have very low thermal and electrical COPs. Since solar-regenerated desiccant cooling machines must compete with vapor compression as well as all other space-cooling technologies, both electrical and thermal COPs are of primary concern.

Desiccant cooling systems are attractive areas of research because of their inherent ability to use air as the working fluid. In addition, they do not have to be hermetically sealed and are adaptable to applications with high ventilation loads.

One of the earliest solar-regenerated desiccant systems was built and tested by Löf [1]. This was a liquid, triethylene-glycol system using air heated directly in a solar collector for generating the desiccant. A problem with this system was the migration of glycol into the building space. More recently, Johannsen [2] has reported on a glycol system in which the weak solution is regenerated in the solar collector itself. The solution flows as a fluid film in contact with a solar absorbing surface and the ambient air. The heated solution releases its absorbed moisture to the atmosphere. A similar system using calcium chloride and water as the desiccant solution has been reported by Mullick & Gupta [3].

Various solid-desiccant systems employing rotary desiccant wheels or drums have been proposed and studied [4,5,6,7,8]. In addition, researchers [9,10] have also considered stationary beds that are cycled between drying and regeneration. In all these systems, it is possible to use solar collectors to provide heat for the regeneration of the desiccant. Solar air heaters are particularly suited to these desiccant systems because air is the regeneration medium. An additional heat exchanger would be needed if a liquid were used as the working fluid in the solar collectors.

SERI report

THERMODYNAMIC ANALYSIS

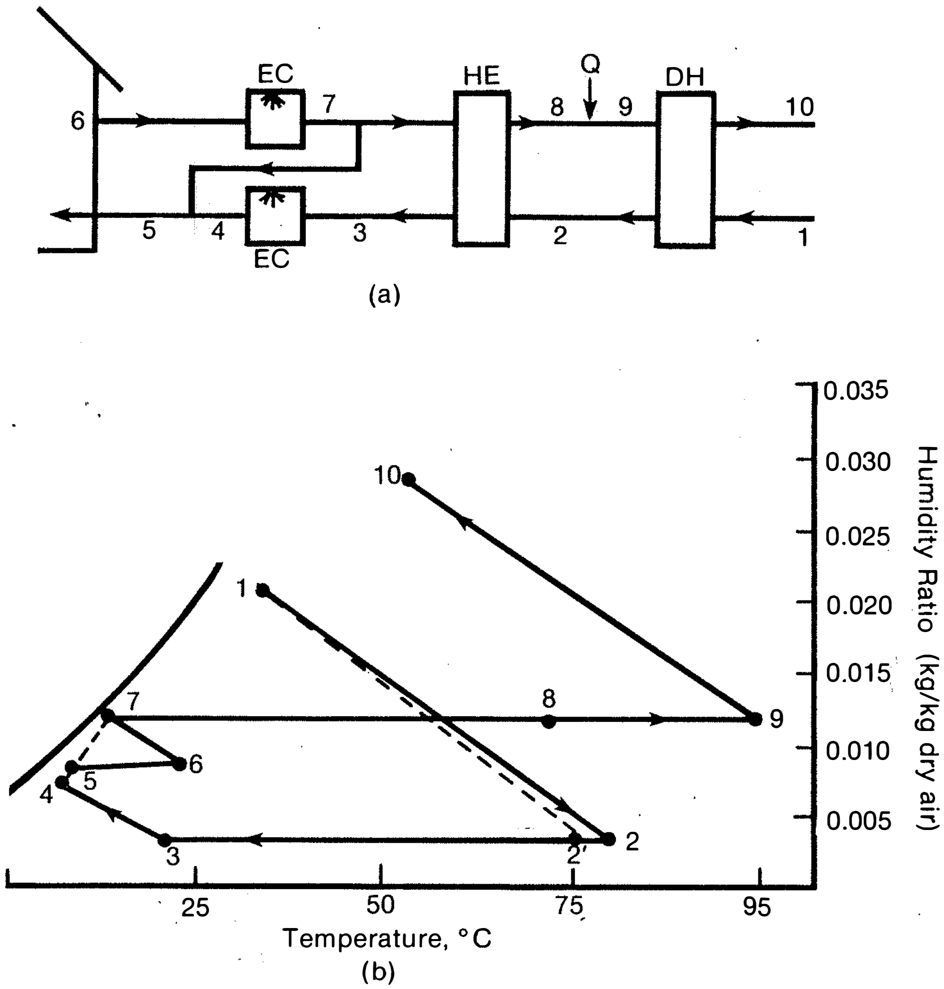
Block diagrams of the most common rotating solid-desiccant systems are shown in Figs. 1(a), 2(a) and 3(a) [11]. The symbols EC, DH and HE indicate evaporative coolers, dehumidifiers, and heat exchangers, and Q represents heat input. Psychrometric charts with state points corresponding to locations on the block diagrams are shown in Figs. 1(b), 2(b) and 3(b). These were calculated for ambient conditions of 35°C and 0.021 kg/kg absolute humidity (95°F, 60% RH), room conditions of 24°C and 0.009 kg/kg absolute humidity (75°F, 50% RH) and for a regeneration temperature of 95°C (203°F). The effectiveness of all components is 90% for the three cycles.

The cycle shown in Fig. 1 is called the ventilation mode. In this case, room air is used to regenerate the dehumidifier bed and outdoor air is cooled. The air leaving the room (6) is evaporatively cooled (7) and used as the cold sink for the dried room-return air. The room-exit air is heated during the heat-exchange process (8). It is then further heated by an external source of energy (Q) for desiccant drying (9). Drying the desiccant cools and humidifies the air (10). To supply room make-up air, ambient air (1) is dried by the desiccant (2) and cooled by heat exchange (3) with room-exit air. This dried and cooled ambient air is then further chilled by evaporative cooling (4). Just before point 5, remix air is introduced. This scheme, proposed by Dunkle [6], mixes evaporatively cooled room air with the cooled and dried room make-up air in order to control the sensible-heat factor.

The recirculation mode shown in Fig. 2 uses the same components as the ventilation mode except that room air is constantly reconditioned in a closed loop, and outdoor air is used for regeneration. Thermodynamically, this cycle has the advantage of processing air with greater availability for cooling. It has the disadvantage of having a higher cold-sink temperature than the ventilation mode. The desirability of either cycle is a trade-off which depends upon room and ambient conditions. This trade-off will be discussed in greater detail later in the paper. Another important difference between this mode and the ventilation mode is that there is no direct fresh-air supply, whereas the ventilation mode uses all fresh air. For the recirculation mode, as well as for most vapor-compression cycles, fresh air to the building space is supplied by normal infiltration. In an era of tighter buildings, systems which do not allow and control ventilation will be at a disadvantage.

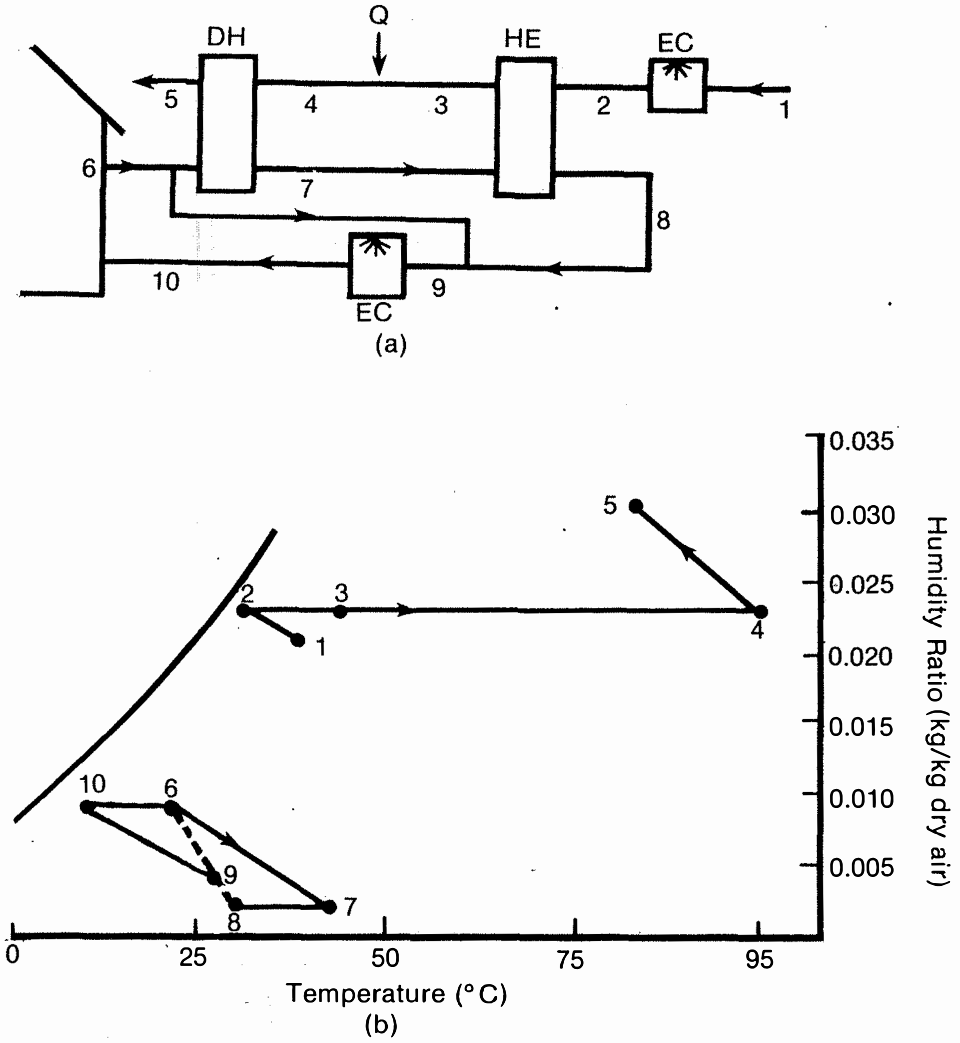
The Dunkle cycle shown in Fig. 3 is an attempt to combine the thermodynamic advantages of both the ventilation and recirculation modes. The cycle uses the advantage of processing the higher cooling availability room air as in the recirculation mode, while retaining the lower cold-sink temperature of the ventilation mode. This advantage in performance comes at the cost of increased complexity and an additional sensible heat exchanger. As with the recirculation mode, the lack of controlled fresh air to the building space may be a disadvantage.

The cooling capacity and thermal COP of these cycles can be calculated with the aid of the psychrometric diagrams. The amount of cooling delivered to the building for the ventilation mode would be defined as (see Fig. 1(b))



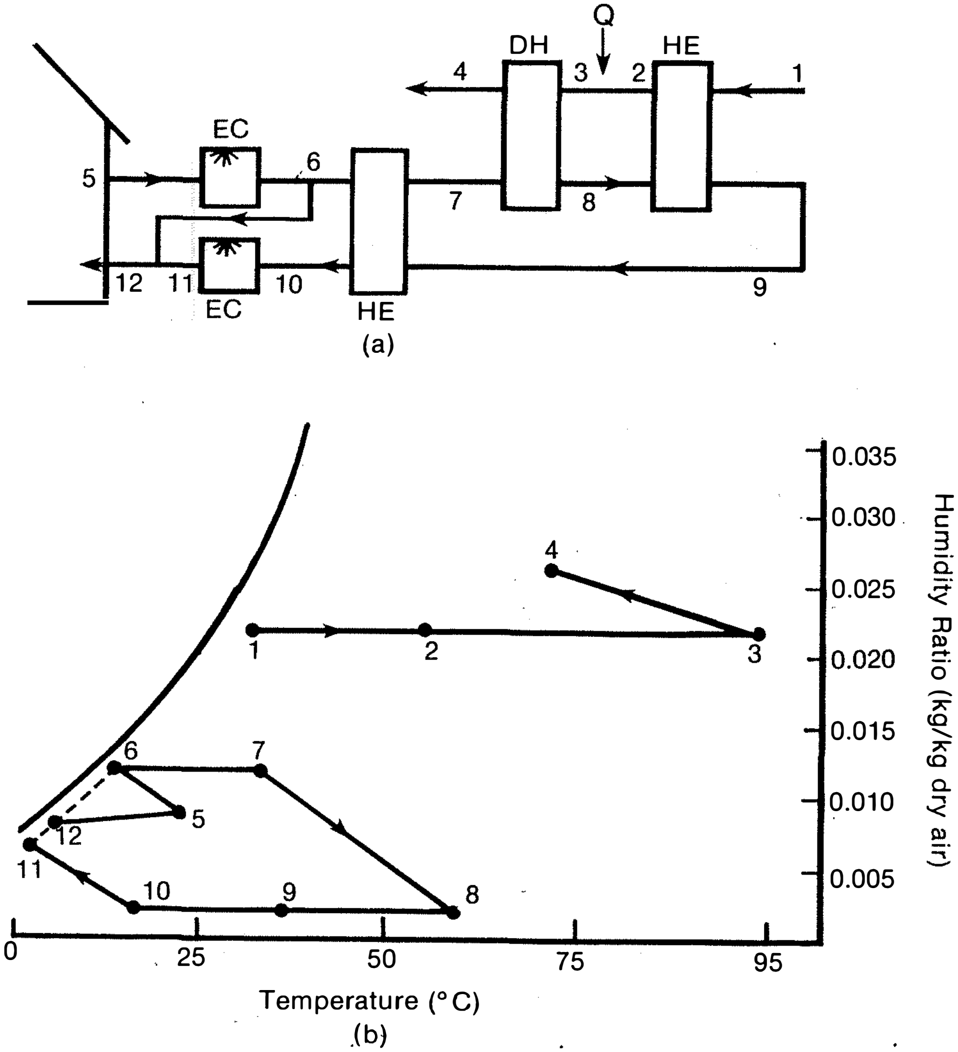
EC = evaporative cooler
HE = heat exchanger
DH = dehumidifier

Figure 1. Ventilation Mode



EC = evaporative cooler
HE = heat exchanger
DH = dehumidifier

Figure 2. Recirculation Mode



EC = evaporative cooler
HE = heat exchanger
DH = dehumidifier

Figure 3. Dunkle Cycle

$$Q_c = \dot{m}_6 h_6 - \dot{m}_5 h_5 \tag{1}$$

where $\dot{m}$ is the mass flow rate of dry air.

If the mass flow rate out of the building equals the mass flow rate into the building, then Eq. 1 may be rewritten as

$$Q_c = \dot{m}_5 (h_6 - h_5) \tag{2}$$

Due to the remix feature shown, the mass flow rate at 5 is equal to the mass flow rate at 4 plus a portion of the flow at 7.

$$\dot{m}_5 = \dot{m}_4 + \dot{m}_R \tag{3}$$

Since the enthalpy at 7 is nearly equal to that at 6, the energy flow at 5 can be expressed as

$$\dot{m}_5 h_5 = \dot{m}_R h_6 + \dot{m}_4 h_4 \tag{4}$$

Substitution of equations (3) and (4) into equation (2) yields

$$Q_c = \dot{m}_4 (h_6 - h_4) \tag{5}$$

The COP is defined as

$$\text{COP} = \frac{\text{cooling capacity}}{\text{heat input}} = \frac{Q_c}{Q_h} \tag{6}$$

and the heat input would be

$$Q_h = \dot{m}_8 (h_9 - h_8) \tag{7}$$

Therefore, the system COP will be

$$\text{COP} = \frac{\dot{m}_4 (h_6 - h_4)}{\dot{m}_8 (h_9 - h_8)} \tag{8}$$

From our previous assumptions that $\dot{m}_6 = \dot{m}_5$ and no leakage occurs, $\dot{m}_4 = \dot{m}_8$, and

$$COP = \frac{(h_6 - h_4)}{(h_9 - h_8)} \qquad (9)$$

For example, the COP for the ventilation mode shown in Fig. 1 is 0.6.

Equations (5) and (9) along with the psychrometric chart of Fig. 1(b) can give us insight into the desirable characteristics of the system, as well as its thermodynamic limits. Notice that lowering the temperature of point 3 increases cooling capacity as well as COP. For the ventilation system, point 7 is the cold-sink temperature. Thus, the component parameter that most noticeably affects system performance is the effectiveness of the sensible heat exchanger. As the heat exchanger effectiveness varies, states 7 and 9 will not change; however, states 3 and 8 will move along lines of constant moisture content. An increase in heat exchanger effectiveness will lower the temperature of state 3, and state 8 will move closer to the dry bulb temperature of state 2. This reduces $h_9 - h_8$, the heat input to the system. Thus, an increase in the sensible heat exchanger effectiveness increases cooling capacity and decreases heat input, producing a doubly positive effect on the system COP. When the heat exchanger effectiveness is unity, states 3 and 8 will achieve the same dry bulb temperatures as states 7 and 2, respectively. Thus, both cooling capacity and system COP will be maximized for the degree of dryness (state 2) achieved. Consequently, prototype solar desiccant cooling systems have been designed with heat exchangers having 0.90 to 0.95 effectiveness. Further increase in heat exchanger effectiveness is unlikely.

Let us now examine how the effectiveness of the evaporative coolers will influence system performance. Lowering the effectiveness of the evaporative cooler between states 3 and 4 increases the dry-bulb temperature at state 4 and decreases moisture content. This has no effect on either the cooling capacity or the COP. However, it does affect the sensible heat factor and the amount of remix air required to match the latent load of the building. On the other hand, since state 7 is the cool sink temperature, evaporative cooler effectiveness between states 6 and 7 determines the minimum dry-bulb temperature achievable at state 3. The moisture ratio for the regeneration stream is lowered, but the enthalpy difference between states 8 and 9 remains almost the same. Therefore, increasing the evaporative cooler effectiveness between states 6 and 7 increases cooling capacity but does not change the heat input. This increases the COP but not as much as an equivalent increase in the sensible heat exchanger effectiveness.

The ultimate dryness achievable (state 2) is the overriding parameter determining system performance. It will determine the minimum enthalpy achievable at state 3 for any given sink temperature (state 7). The characteristics of the air stream required at point 9 are determined by the adsorption properties of the desiccant material and the cooling capacity desired. The dry-bulb temperature of state 9, for the absolute humidity determined by state 7, must be sufficiently large to dry the desiccant material to the point that state 2 is achievable. As the dryness requirement of state 2 increases (moisture content decreases), the temperature necessary at state 9 increases. Since the amount of temperature rise required depends on the desiccant material, it is not possible to make a general statement as to the effect of state 2 on heat input. Notice that from thermodynamic considerations only, the dry-bulb temperature of state 9 need only be incrementally higher than for state 2, and state 10 incrementally higher than state 1. Conservation of mass requires that the moisture cycled from 1-2 must equal the moisture cycled from 9-10. Therefore, for equal mass-flow rates of air, state 10 will be along a line of approximately constant enthalpy with state 9. The moisture content at state 10 will be equal to the amount of moisture cycled between states 1 and 2 plus the moisture content of state 9. Notice that constant-enthalpy lines on a psychrometric chart have steeper slopes for larger absolute humidities. This is due to the fact that moist air has a higher heat capacity than dry air. It insures that state 10 will always be at a dry-bulb temperature greater than state 1, and therefore the cycle is thermodynamically possible. This defines the minimum heat input necessary and hence the maximum COP attainable for the system.

For a thermodynamically idealized cycle, the maximum theoretical COP can be several times larger than one, the theoretical limit for single-stage closed systems. This is due to the fact that we are using the earth's atmosphere as a sink. The extremely moist air at state 10 does not have to be processed by the cycle but is replaced from the earth's atmosphere so that the original cycle inlet conditions of state 1 are maintained. This processing by the earth's atmosphere is "free" as it affects the COP of the cycle, and machine efficiency is accordingly increased. In reality, we do not have desiccant materials that would allow us to approximate the thermodynamically idealized cycle described. For existing materials, state 9 is at a much higher dry bulb temperature than state 2, as shown in Fig. 1(b). This is necessary in order to match the adsorption characteristics of the particular desiccant to the requirements of the cycle. However, it should be emphasized that theoretical COPs greater than one can be obtained with existing desiccant materials. For regeneration by conventional means, (electricity or fossil fuel) the temperature of state 9 will only affect cycle COP. However, for solar regeneration, the state-9 temperature will also affect collector efficiency. Besides reducing cycle COP, an increase in the state 9 temperature will also reduce the overall system efficiency due to a reduction in the solar-collector efficiency.

It is apparent that the adsorption characteristics of the desiccant can have a large influence on the cooling capacity and COP of the cycle. The nature of these adsorption properties will be discussed in detail in the following section.

The effect of ambient conditions on the cycle can now be seen quite readily. As the outdoor humidity increases, the dry-bulb temperature required by state 2 for the same moisture content increases. This increases the dry-bulb requirement of state 9, but also increases the temperature achieved by state 8. The most important effect is the relative change in state 3. With a high effectiveness sensible heat exchanger, state 3 will not change appreciably, and hence the cooling capacity will not change. However, as the sensible-heat exchanger effectiveness decreases, so does the cooling capacity. A similar situation exists if the outdoor dry-bulb temperature increases. The lesson here is that fluctuations in outdoor conditions can be tolerated effectively if high-effectiveness heat exchangers are used. In actual practice, heat-exchanger effectiveness on the order of 95% [12] can be achieved. The importance of high-effectiveness components has also been verified by Jurinak & Beckman [13].

The previous analysis dealt specifically with the ventilation mode depicted in Fig. 1. The same analysis can be applied to the recirculation mode and the Dunkle cycle depicted in Figs. 2 and 3. The results are qualitatively the same as those for the ventilation mode; however, the relative importance of the various parameters is different.

DESICCANT MATERIALS

The desirable properties for any desiccant material used in an open-cycle cooling applications are:

(1) Mechanical and Chemical Stability. The material does not deliquesce (for solid systems) or undergo hysteresis when cycled.

(2) Large Moisture Capacity per Unit Weight. It is desirable to cycle as much water as possible for a given amount of desiccant. This reduces the amount of desiccant required and the size of the cooling system.

(3) Large Adsorption/Absorption Capacity at Low Water Vapor Pressures. The moisture capacity should not deteriorate at very low water vapor pressures. This increases the relative dryness achievable, which will have a strong effect on reducing fan power requirements.

(4) Low Heat of Adsorption/Absorption. A low heat of adsorption increases cooling capacity and COP.

(5) Ideal Isotherm Shape. This lowers regeneration temperature and thus increases COP.

Properties 1-3 are well-known and have been reported in the literature [14]. However, an understanding of properties 4 and 5 is equally important in determining the performance of an open-cycle system. The heat of adsorption/absorption is the energy released when water is converted from a gaseous state to an adsorbed/absorbed state. It is identical with the energy input required to convert water from an adsorbed/absorbed state to a gaseous one. For most materials, this energy is greater than the heat of condensation/vaporization of water. This means that the actual dehumidification and regeneration processes do not follow constant-enthalpy lines on the psychrometric chart, as other researchers have stated. Referring again to Fig. 1(b), the line 1-2' is a line of constant enthalpy, while the line 1-2 depicts the proper process line for silica gel. As shown, the additional energy released by adsorbing the water vapor results in a higher dry-bulb temperature for a given moisture content. This will have the same net effect on the cycle as an increase in the outdoor dry-bulb temperature: minimal effect on performance of systems with very high-effectiveness heat exchangers but reduced cooling capacity for systems with low-effectiveness heat exchangers. For most desiccants in use today, the heat of adsorption/absorption is only 5-10% greater than the heat of condensation/vaporization of water for commonly encountered ranges of desiccant water content. Therefore, the cycle diagrams which follow lines of constant enthalpy are close to being correct for many desiccants. Notable exceptions are molecular sieves which can have adsorption energies 50%-100% higher than the heat of condensation/evaporation for water.

To ascertain an "ideal" or preferred isotherm shape it is helpful to recall the discussion of state 9 for the ventilation mode. In order to minimize heat input to the system, we would like state 9 to be as close to the dry-bulb temperature of state 2 as possible. However, the water vapor pressure is much higher at state 9 than at state 2. Therefore, the first requirement of our "ideal" desiccant material would be that adsorption/absorption capacity be independent of water vapor pressure. We then want the desiccant to have a very high adsorbed/absorbed water capacity for dry-bulb temperatures between states 1 and 2, but a very low water capacity for dry-bulb temperatures above state two. The isotherms for such an "ideal" desiccant are shown in Fig. 4, where the critical temperature T_{cr} is equal to T_2 in this case. On the other hand, the desiccant must maintain low capacity throughout the range of regeneration process temperatures between states 9 and 10. Of course, no single material can achieve both goals, since the temperatures during air drying and desiccant regenerating overlap. However, consider a desiccant bed, consisting of a large number of "ideal" materials, each with a different critical temperature, staged in series. During sorption the air leaving each stage will be at a temperature below the critical temperature of the next stage. Conversely, during counterflow regeneration the air leaving each stage will be above the critical temperature of the next stage. All the stages will sorb or desorb water simultaneously during the appropriate process. In this manner, the temperature difference between states 2 and 9 could be minimized and maximum theoretical capacity and COP would be approached.

Unfortunately, the heat of adsorption is not independent of the isotherm shape. The isosteric, or differential, heat of adsorption is defined by the Clausius-Clapeyron relation as:

$$\Delta H = R\left(\frac{\partial(\ln P)}{\partial(1/T)}\right)_X$$

Thus, the heat of adsorption at a given loading will be proportional to the slope of isosteres plotted on ln P versus 1/T coordinates. As a consequence of the Clausius-Clapeyron relation, a material with a weak pressure dependence must also have a weak temperature dependence in order to keep the heat of adsorption low. Conversely, a strong temperature dependence must be accompanied by a strong pressure dependence. It is important to keep the heat of adsorption of the desiccant material reasonably low. Even though most of this energy can be recaptured by the sensible-heat exchanger, a high heat of adsorption leads to the requirement of a high regeneration temperature. The collector then becomes too hot and reduces the efficiency of the system.

As a consequence of Clausius-Clapeyron, the "ideal" desiccant would be impractical because the heat of adsorption would be infinite at T_{cr}. There are limitations, however, on the applicability of the Clausius-Clapeyron relation. For instance, phase equilibrium must exist between the gas and adsorbed phases for Clausius-Clapeyron to apply. Hypothesizing the existance of systems which are not limited by Clausius-Clapeyron is beyond the scope of this paper. However, it is clear that if such a system is possible, the thermal COPs of open cycle desiccant systems could be increased dramatically.

Even though the "ideal" desiccant material proposed is too idealized to be practical, certain qualitative judgements about the desirability of those desiccant materials currently available can be made. Figures 5-8 show the adsorption isotherms for natural zeolite, molecular sieve, silica gel and lithium chloride/water. In Fig. 8, the deviation of the $LiCl/H_2O$ isotherms from the ideal behavior is immediately apparent: the equilibrium capacity at a given temperature varies tremendously with water vapor pressure. The minimum temperature allowable for state 9 is that which results in a desiccant

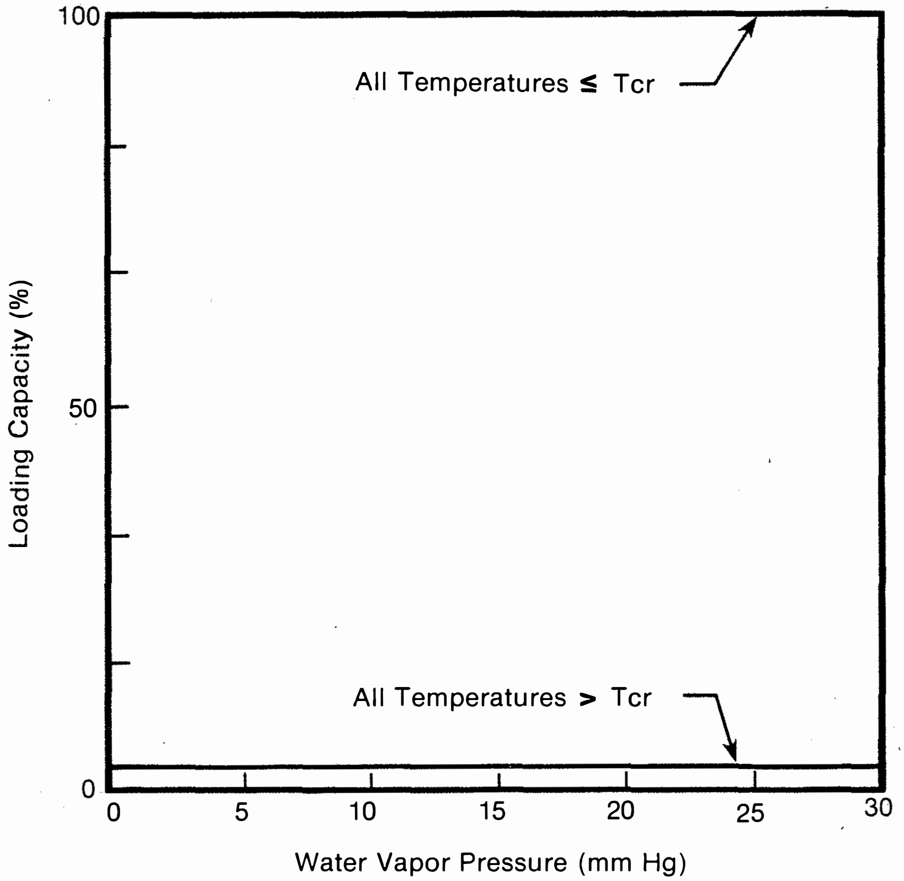


Figure 4. Adsorption/Absorption Isotherms for the "Ideal" Desiccant Material

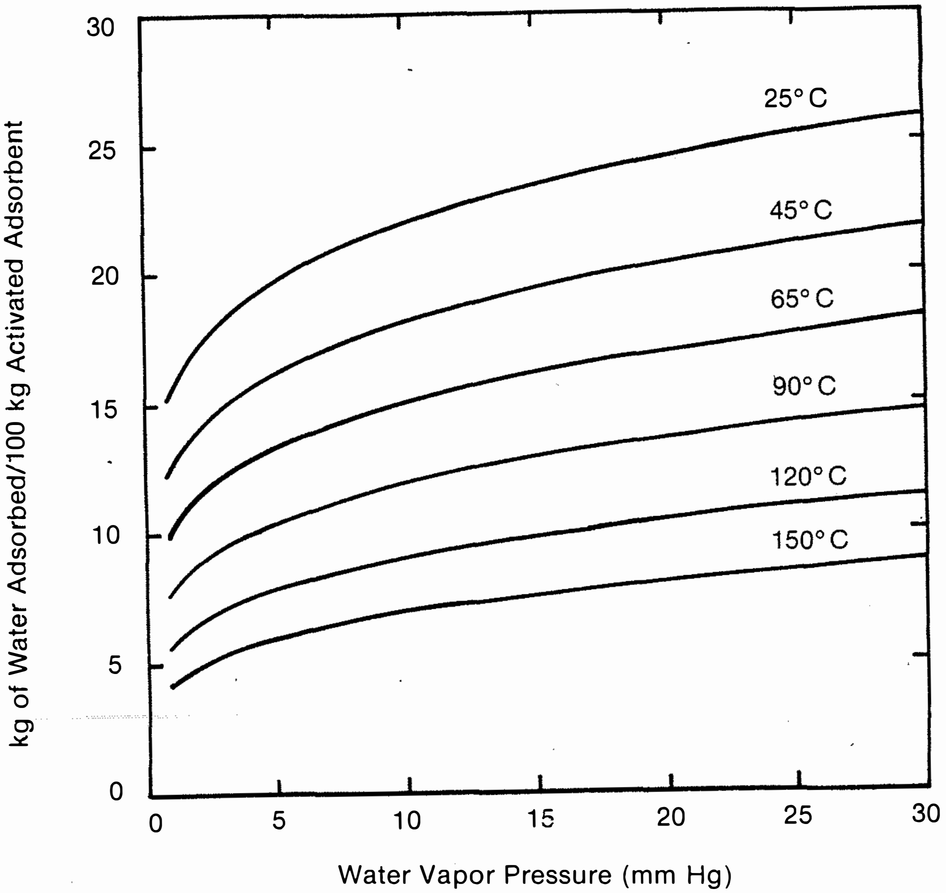


Figure 5. Adsorption Isotherms for a Natural Zeolite

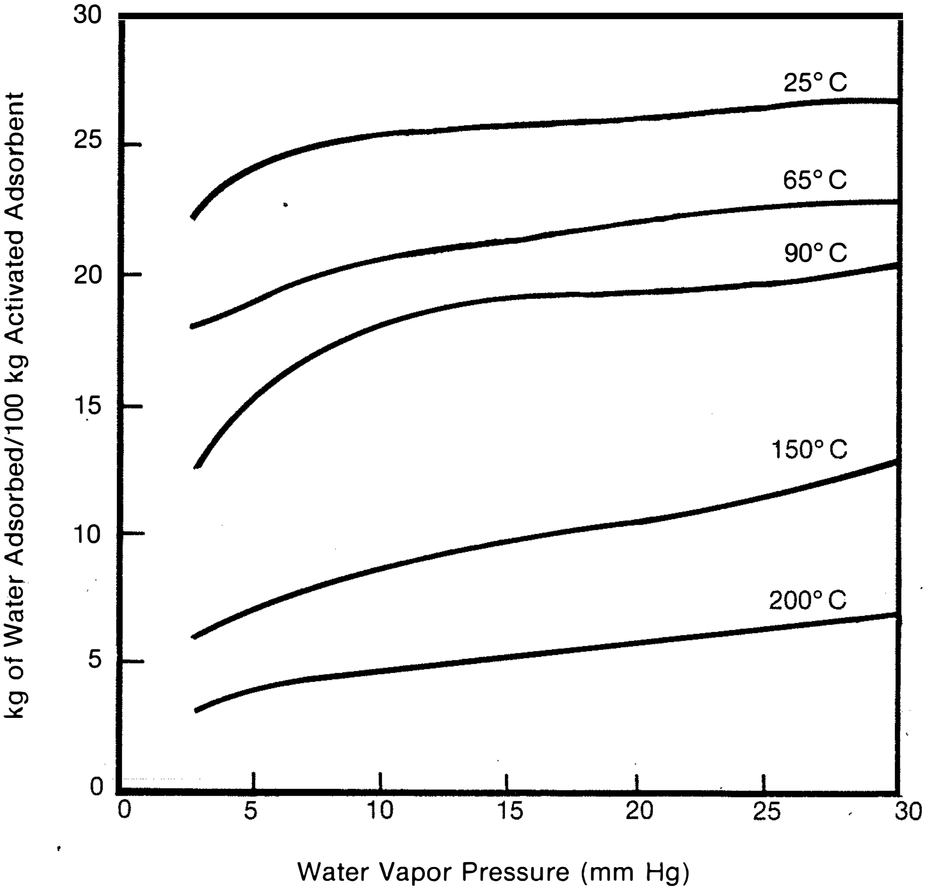


Figure 6. Adsorption Isotherms for Molecular Sieve

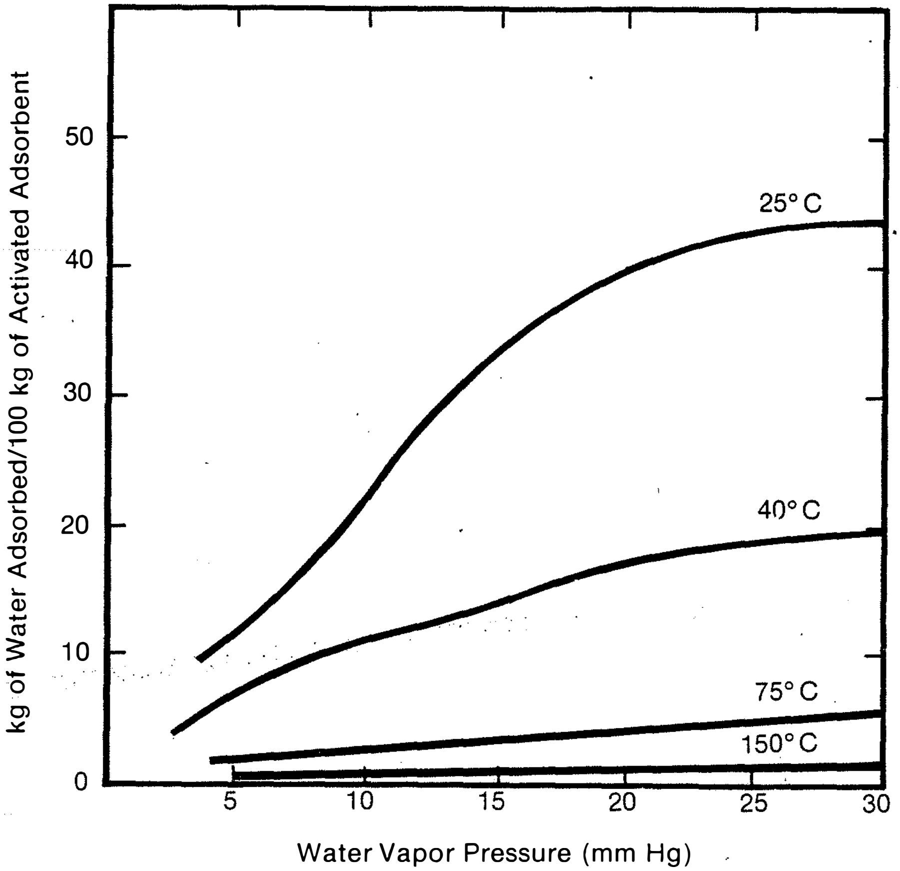


Figure 7. Adsorption Isotherms for Silica Gel.

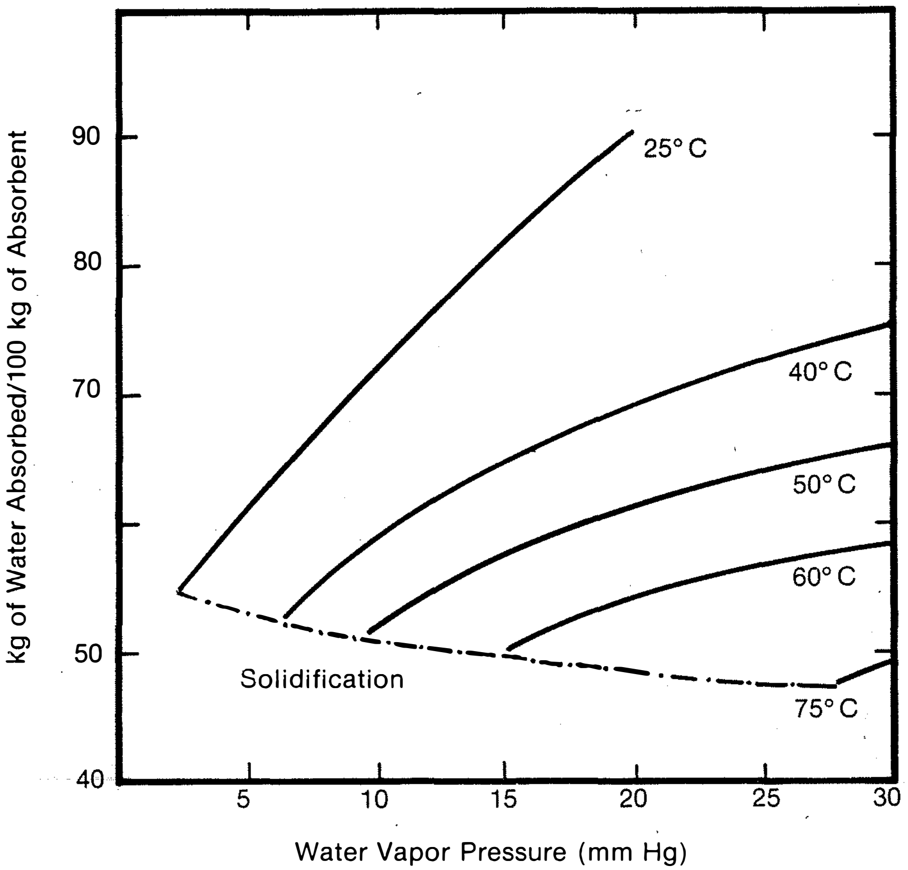


Figure 8. Adsorption Isotherms for Lithium Chloride Solutions

loading equivalent to state 2. For a desiccant material like LiCl/H_2O, the temperature of state 9 must be raised much higher than state 2 in order for the desiccant loadings to be equal at the large difference in water vapor pressures (3.7 versus 16.7 mmHg).

Silica gel also shows considerable vapor pressure dependence. This dependence is reduced as the temperature increases, but the loading capacity is so low that this beneficial effect is largely negated. The isotherms for natural zeolite and molecular sieve show much less water vapor pressure dependence. At first glance, these materials would appear to exhibit very close to ideal behavior. However, these materials must undergo very large temperature variations in order to experience minimal changes in water content. Silica gel, on the other hand, can experience large variations in water uptake for much smaller temperature swings. This behavior is a result of the relatively low heat of adsorption of water on silica gel. It can be shown that a hypothetical desiccant material with the pressure dependence of molecular sieve and the temperature dependence of silica gel would have an adsorption energy of over 7000 Btu/lbm H_2O. In actual practice, molecular sieve, natural zeolite, and silica gel produce about the same net cooling effect per unit weight of material with nearly the same adsorption energies. What silica gel loses in its pressure dependence, it seems to gain in its temperature dependence. The temperature insensitivity of molecular sieve and natural zeolites is advantageous during the dehumidification process; it is a disadvantage during the regeneration process. For silica gel, the temperature sensitivity is a disadvantage during dehumidification but an advantage during regeneration. This observation has led to several attempts [8,10] to make the dehumidification process isothermal instead of adiabatic. The CEM machine [8] used staged drying and cooling wheels which alternately dried and cooled the process air stream. The increase in complexity of this machine far outweighed the additional capacity and further work was suspended. IIT [10] is developing a stationary silica gel bed that uses cross-cooling in order to reduce the dehumidified outlet air temperature. This research is not far enough along to make judgements as to its viability; however, the same question must be asked concerning this machine or any other "novel" approach: does the increased performance justify the increased complexity?

Figure 9 is a diagram of the equilibrium adsorption isobars (constant pressure lines) for silica gel. It is useful to plot the desiccant cycle on this diagram in order to understand the problems of designing a desiccant cooling machine. Assume the following conditions: outdoor temperature 35°C (26°C DP), building space temperature 24°C (13°C DP). For the purposes of this example, we will assume equilibrium between the air and the desiccant. Assuming that the machine is operating in the simplified recirculation mode shown on the psychrometric chart in Fig. 10, the room air and the desiccant will be in equilibrium at point A in Fig. 9. For a desiccant exit dew point

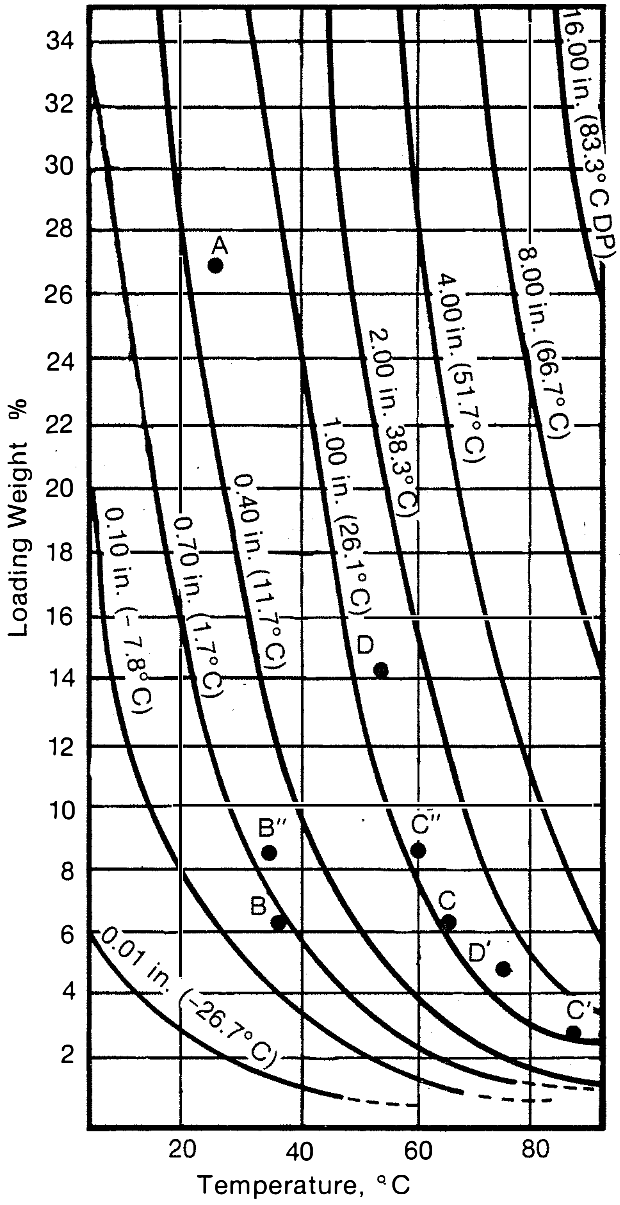


Figure 9. Adsorption Isobars for Silica Gel.

of 0°C, the exit air and the desiccant will be in equilibrium at point B.* Notice that the minimum regeneration temperature necessary to maintain this cycle is about 65°C (point C in Fig. 9). This point is found by taking the same bed loading as at point B (≈6 wt%) and the dew point of the saturated outdoor air (≈28°C) . The air drying cycle will process about 0.005 kg H_2O/kg air (changing the dew point from 13°C to 0°C). The outlet regeneration equilibrium condition which corresponds to cycling 0.005 kg H_2O/kg air is point D (54°C, 31°C DP). This means that the silica gel particles at the bed entrance for air drying (the bed exit for regeneration) will cycle about 13 wt% moisture content while the bed exit for air drying (the bed entrance for regeneration) will cycle no moisture at all. The average amount of moisture cycled, assuming a linear distribution within the bed, will be 6-7 wt%. If the collector regeneration temperature is raised to 90°C, the regeneration inlet condition moves to C', and the regeneration outlet condition moves to D'. The moisture cycled by the silica gel now ranges from 22 to 3 wt% for an average of 12-13 wt%. The assumption of a linear concentration profile within the desiccant bed is not conventional or proper for the thick desiccant beds commonly used in industry. Due to constraints on pressure drop and physical size, desiccant cooling beds are much thinner than those used in industrial air-drying applications. Packed beds as thin as 32 mm are being used for cooling applications [15]. For these thin beds, the length of the theoretical mass transfer zone is several times the bed thickness, and a linear concentration profile is not a poor assumption. A more detailed description of this equilibrium analysis and a discussion of its assumptions can be found in Ref. 16.

One can now appreciate the effect of regeneration temperature on the amount of silica gel necessary to attain a given cooling capacity at a given air mass flow rate. The number of situations which can be modeled with the aid of the equilibrium adsorption properties in Fig. 9 are too numerous to demonstrate in this paper. It is instructive, however, to consider the consequence of a regeneration temperature less than 65°C. Consider, for example, a regeneration temperature of 60°C. Point C now becomes point C", and point B becomes B". The net effect is an increase in the dew point of the dried air, with a reduction in cooling capacity, and a decrease in the average moisture cycled by the silica gel. Notice that the machine did not "turn off," as would happen with absorption machines. The original cooling capacity could be retained by increasing the air flow rates and the amount of silica gel. This would increase the machine size, the parasitic power requirements, or both. This is exactly the dilemma facing developers of commercial hardware today. If this

*Assuming a heat of adsorption equal to the heat of condensation. For this case, the dehumidification process line follows a line of constant enthalpy in Fig. 10. The slope of a process line for heats of adsorption not equal to the heat of condensation is given by

$$\text{slope} = \frac{\text{heat of condensation}}{\text{heat of adsorption}} \times \text{slope of constant enthalpy line}$$

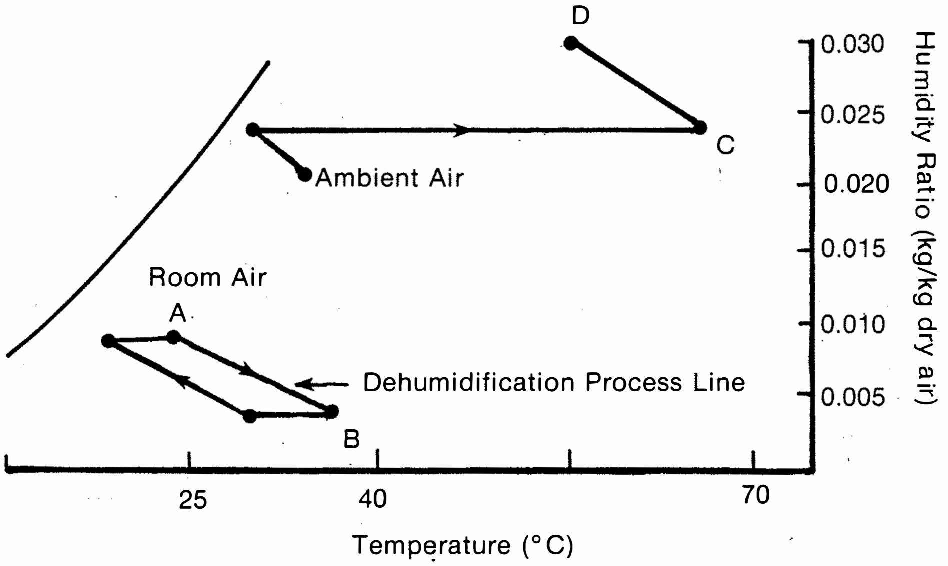


Figure 10. Simplified Recirculation Mode

situation is to be improved, it may be necessary to develop a desiccant material with different adsorption properties.

The preceding example was based on the assumption that both heat and mass transfer equilibrium conditions would be reached. In reality, the process kinetics are very important in determining the actual heat and mass transfer conditions within the desiccant bed. Depending on the desiccant, it can take an hour or more to reach the equilibrium conditions depicted in Fig. 9. For this reason, the modeling of actual desiccant bed performance is very sophisticated and requires solution on a digital computer. Qualitatively, however, the previous example is illustrative of the effect of the adsorption behavior of the desiccant material on machine performance.

The superiority of either solid or liquid desiccant systems has yet to be determined. Each has strengths and weaknesses which affect its desirability and system performance. Solid materials allow much more flexibility in tailoring adsorption properties than do present liquid systems. By changing pore size, particle size, and dopants, solid materials can be engineered to yield a variety of adsorption isotherm shapes. Liquids are presently limited to two-component solutions with fixed isotherm shapes. Liquids can be transported more easily than solids, which could be a significant advantage if research on open regeneration proves fruitful. As mentioned earlier, liquid systems can use the solar collector for both energy input and regeneration. If the solar collector can be glazed or unglazed asphalt roofing, the system cost can be quite low. In fact, lower COPs could then be tolerated because of the reduced cost of the collector/regenerator. Another advantage to liquid systems is their ability to store chemical potential energy rather than thermal energy. By storing "strong" solution in an uninsulated tank the storage costs are potentially low, and there is no thermal interaction between storage and the load to degrade system performance. There has been a good deal of research to develop solid desiccant systems compatible with solar regeneration; research efforts in the area of solar-regenerated liquid systems are just beginning.

CONCLUSIONS

This paper has presented a review of the thermodynamics of three desiccant cooling cycles: the ventilation cycle, the recirculation cycle, and the Dunkle cycle. For the ventilation cycle the qualitative effects of changes in the effectiveness of individual components were analyzed. On the basis of this analysis we conclude that:

- COPs greater than 1.0 for desiccant cooling systems are possible if component performance can be improved.
- Heat exchanger effectiveness has an important influence on COP. However, heat exchangers in current prototype coolers are approaching 0.95 effectiveness and no significant improvements seem feasible.
- Increasing the effectiveness of evaporative coolers in the system would have relatively little effect on performance.
- Improving the performance of the dehumidifier has a significant potential for an increase in COP.

There are two possible paths to improved dehumidifier performance: changing the design of dehumidifiers using currently available desiccants so as to increase effectiveness without increasing parasitic losses, or developing new desiccants specifically tailored for solar cooling applications. The later part of this paper has considered the second option. A list of desirable desiccant properties was defined, properties of currently used solid and liquid desiccants were compared to this list, and a hypothetical desiccant type that would give improved system performance was discussed.

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