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High Performance Solar Desiccant Cooling System

Performance Evaluations and Research Recommendations

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PREFACE

The DOE Active Solar Heating and Cooling Program's objective is to develop the technology base to allow the private sector to produce efficient, economically competitive solar technology options for the marketplace. The program is composed of a balance of research on systems, components, and materials. The systems research directs the program through analysis of alternative, advanced concepts in heating and/or cooling. Additionally, systems reliability research provides data on critical components and materials that affect long-term performance. The analysis and reliability programs are augmented by a laboratory and field test program that develops data on experimental and state-of-the-art systems. These data identify design and operational problems and, thus, opportunities for research.

The Component and Materials Program performs research on advanced concepts that have been identified by systems analysis to offer promise of being competitive with conventional, nonrenewable energy sources. Key components, such as chillers, dehumidifiers, heat exchangers, and collectors, are evaluated for improved efficiency and cost through innovation in design. Advanced materials for these key components are evaluated/modified to provide improved performance and/or lower cost.

This report presents an assessment of the current status of solar desiccant cooling and makes recommendations for continued research to develop high performance systems competitive with conventional cooling systems.

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SUMMARY

Objective

To assess the current status of solar desiccant cooling and make recommendations for continued research to develop high performance systems competitive with conventional cooling systems.

Discussion

Solar-driven desiccant cooling systems have been investigated extensively as alternatives to conventional cooling systems. Solid desiccant, liquid desiccant, and hybrid systems combining desiccant dehumidifiers with vapor compression units have been considered. Solid systems for residential applications have received the most attention, but further improvements in the dehumidifier are required to reach competitive levels. Accurate models of the dehumidifier exist.

Liquid systems have several potential advantages: direct solar regeneration of the desiccant and chemical energy storage. Modeling of system components is not well developed, and long-term performance has not been analyzed very much.

Hybrid systems for commercial applications have the greatest potential for short-term impact with both solid and liquid systems being considered.

Conclusions and Recommendations

Development of high performance, low pressure solid desiccant dehumidifiers should be pursued along the current research directions. Optimization of the dehumidifier in terms of operating conditions and desiccant properties should be accomplished, and models should be experimentally verified.

Liquid system component development should continue with emphasis on the collector/regenerator and efficient absorbers. Accurate and simplified models for components should be developed, and estimates of long-term system performance should be obtained.

Development of hybrid systems should be pursued and component development continued.

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SECTION 1.0

INTRODUCTION

1.1 DESICCANT COOLING SYSTEMS

Solar-driven desiccant cooling systems have been investigated extensively during the past decade as alternatives to electrically driven vapor compression cooling systems. This interest has been shown for several reasons. The natural gas industry is looking for new markets because of growing supplies [1] and an interest in generating summer loads to balance their annual distribution profiles. The electric utilities are interested in reducing their peak summer loads caused by people using vapor compression air conditioning [2]. Desiccant cooling systems can be fired with solar energy and can aid conservation of nonrenewable energy resources. Desiccant systems can be economically competitive with vapor compression systems and provide the consumer with lower cooling costs.

Desiccants have been used for many years to provide dry air for a variety of industrial and commercial processes, in particular, for situations where very low humidities are required. Desiccant dehumdification for use in air conditioning systems is an extension of this. Both residential and commercial applications have been considered.

Residential applications have received most of the attention to date. The proposed systems typically dry air in a desiccant dehumidifier, cool the air through heat exchange with an available temperature sink, and then cool the air further by humidifying it in an evaporative cooler. These systems use either a solid or liquid desiccant, which is regenerated with solar or other thermal energy sources. Solid desiccant systems are discussed in Section 3.0 and liquid systems in Section 4.0.

Recently, desiccant dehumidification has been considered for use in commercial air conditioning applications. In most of these systems, the dehumidifier removes only enough moisture to meet the latent portion of the cooling load, while the sensible portion is met by a vapor compression cooling system. The desiccant can be regenerated with the heat rejected from the condenser of the vapor compression system in combination with solar and other thermal energy sources. These hybrid desiccant/vapor compression systems are designed to use the strengths of the individual components to maximize system performance. Again, either solid or liquid desiccants can be used. Hybrid systems are discussed further in Section 5.0.

1.2 PURPOSE

A significant amount of work, both analytical and experimental, has been done on desiccant cooling systems. The concept of desiccant cooling has been proven, advances have been made in component and system design and performance, and further improvements are expected. However, the current state-of-the-art desiccant system performance is not yet competitive with conventional systems in general practice. Additional work is needed to obtain further improvements and to implement them in actual components and systems.

The purpose of this report is to assess the current status of solar desiccant cooling and make recommendations for continued research to develop high performance systems. The term "high performance" is taken to include factors that will make desiccant systems cost-effective in terms of owning and operating and economically competitive with vapor compression cooling systems. The process and results of the Active Program Research Requirements (APRR) project [3], discussed in Section 2.0, will be used as the basis for these recommendations.

1.3 APPROACH

Meaningful recommendations can be made only after consideration of the current status of desiccant systems and their potential for improvement. To determine this, we

- Surveyed the current level of desiccant systems development
- Surveyed the tools and techniques available for analyzing the performance of desiccant components and systems
- Surveyed the results of desiccant systems analyses work
- Analyzed system costs and summarized projected fuel savings to identify critical areas for further research.

SECTION 2.0

THE ACTIVE PROGRAM RESEARCH REQUIREMENTS

2.1 BACKGROUND

In 1982, an effort was begun to develop a standard analysis procedure for active solar heating and cooling systems, so research projects aimed at improving system performance or decreasing system cost could be judged on a common, objective basis. The title given this effort, sponsored by the U.S. Department of Energy (DOE) and coordinated by Science Applications, Inc. (SAI), was the Active Program Research Requirements project (APRR) [3].

2.2 PURPOSE AND APPROACH

The purpose of the project is to make recommendations for continued research in improving the cost and performance of active heating and cooling systems. The approach adopted by SAI and carried out by the participants in the first round was to identify "base case" systems that represented the state of the art today; i.e., technology that would be used if a system were purchased today and installed in a residence or commercial building. From system performance data and a standard load profile for residential or commercial buildings, TRNSYS modeling was used to establish energy savings over the conventional, commercially available competition. Using data on energy prices and payback periods, a present-value capital cost for the system was established as an economic goal.

To determine the relative economy of the present technology, a system configuration with detailed equipment lists was prepared and costed according to a standard cost catalog developed by SAI. The ratio of cost-to-cost goal was then available as a means of comparison among technologies and among system concepts within a technology.

Once the system analysis process was complete on the base case systems, participants were asked to propose a path of development to advanced systems that would represent the state of the art in the year 2000. Each step in the path was to be documented by supporting rationale for cost and performance so determinations could be made as to the probability of success of the proposed improvement. Estimates of project duration and cost were also required as a measure of the eventual cost of the proposed technology improvements to the government sponsors. Finally, the advanced systems were to be analyzed by the same procedure applied to the base case systems to provide cost, performance, and cost-to-cost-goal information for technology comparisons.

2.3 CANDIDATE SYSTEMS IN DESICCANT COOLING TECHNOLOGIES

In the first round of the APRR process, nine candidate system configurations were considered in the desiccant cooling technology area:

- Open-cycle solid desiccant
  - residential cooling
  - residential hybrid with vapor compression chiller
  - commercial hybrid with vapor compression chiller
  - commercial hybrid with absorption chiller
  - commercial hybrid with Rankine chiller
- Open-cycle liquid desiccant
  - residential cooling
  - commercial cooling
  - commercial hybrid with vapor compression chiller
- Closed-cycle solid desiccant
  - Residential cooling and heating.

Preliminary investigations of the cost and performance characteristics of the candidate systems allowed the elimination of three concepts. The hybrid systems with absorption and Rankine chillers were dropped because of the extremely high capital costs and relatively small opportunities for improving the cost and performance of the combinations, especially when compared to the hybrid vapor compression system. The closed-cycle solid desiccant system candidate was also eliminated because of similar high capital cost and little chance for improvement, according to the sole manufacturer of such systems, Zeopower, Inc.

The balance of the candidate systems was slated for analysis under the APRR process, to assess their present and future cost-competitiveness and the path for development to economic parity with conventional competitors. The first step in the process was to survey the level of current development and the analysis tools for each of the technologies. The results are discussed in the following sections.

SECTION 3.0

SOLID DESICCANT SYSTEMS

3.1 CURRENT STATUS

Solid desiccant cooling systems have been investigated extensively at the component and system levels both analytically and experimentally. Several system configurations have been proposed. These systems will be briefly described in this section, along with the current status of component and system development.

3.1.1 Systems

Two open-cycle systems that use adiabatic dehumidifiers have received most of the attention to date. These are the ventilation cycle, shown in Figure 3-1, and the recirculation cycle, Figure 3-2. A similar cycle with two heat exchangers, proposed by Dunkle [4], has not received much interest.

In the ventilation cycle, Figure 3-1, as ambient air (1) is dried in the dehumidifier (2) its temperature also increases because of the energy that is released in the absorption process. Typically, this energy is slightly greater than the heat of vaporization. The process air is sensibly cooled (3) in the heat exchanger and then evaporatively cooled (4) to provide the conditioned air for the room (5). Room air is evaporatively cooled (6) to provide a sink for the heat exchanger. This air stream is then passed through the heat exchanger (7) where the energy released during absorption is reclaimed. Additional heating (8) is done with solar or other thermal energy and the air stream is used to regenerate the desiccant (9). The recirculation cycle is similar except that room air is processed and recirculated, and ambient air is used for the regeneration stream.

In an adiabatic dehumidifier the temperatures can become fairly high because of the heat released during the adsorption process. This either limits the amount of dehumidification that can be done or increases the required regeneration temperature. By cooling the dehumidification process, either increased dehumidification (and therefore cooling capacity), Figure 3-3a, or reduced regeneration temperatures (and therefore better solar system performance), Figure 3-3b, can be expected. The ventilation and recirculation cycles are not appropriate for use with a cooled dehumidifier.

An appropriate configuration, shown in Figure 3-4, has been proposed by researchers at the Illinois Institute of Technology (IIT) [5]. Evaporatively cooled ambient air is used to cool the dehumidifier while adsorption is taking place. While this may improve the dehumidification process, it also throws away the heat of adsorption that is reclaimed in the adiabatic cycles. For this reason, overall system performance is not improved [6]. Note also the need for an additional heat exchanger and evaporative cooler in the cooled system. Cooled systems do not appear to have any advantage over adiabatic systems.

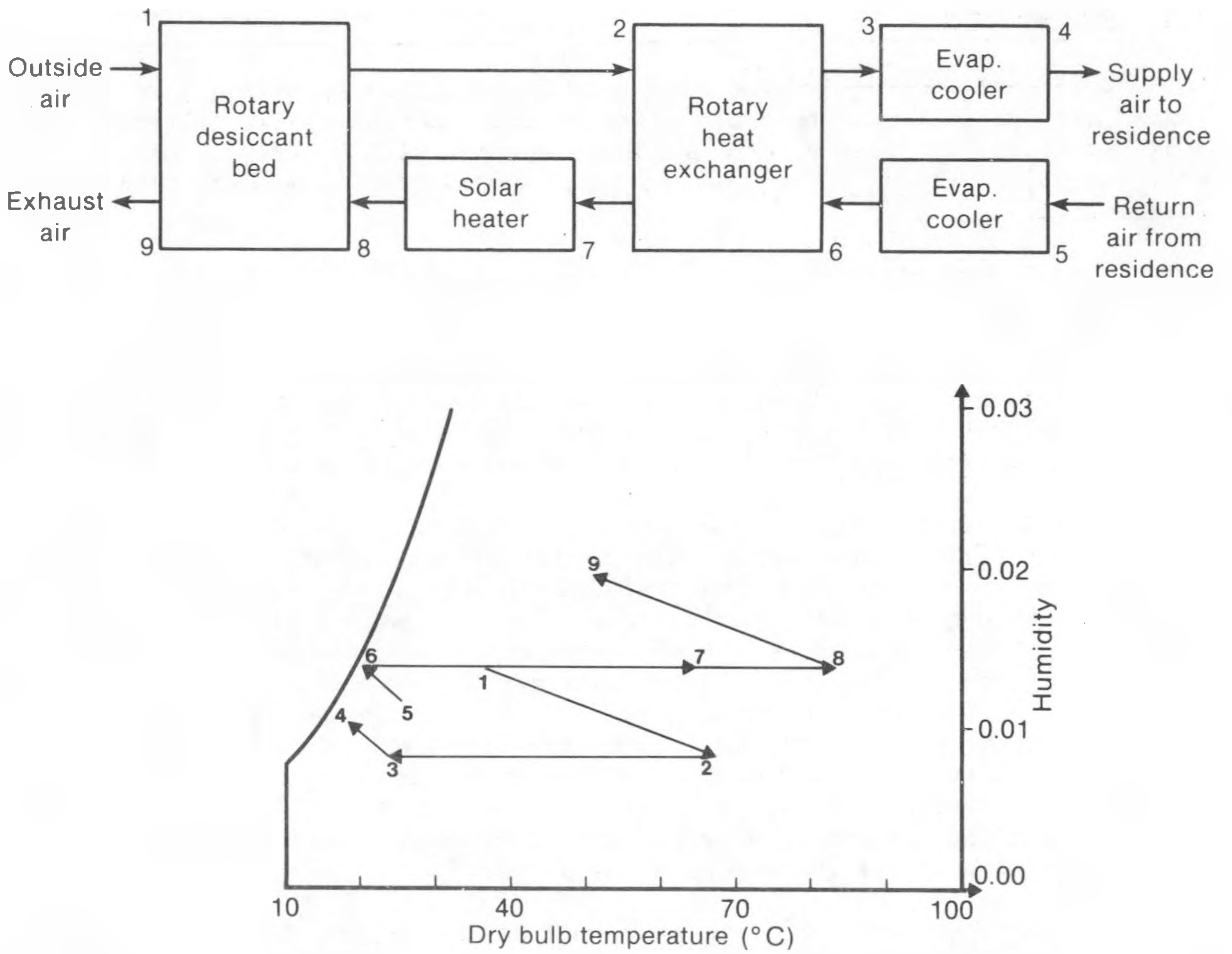

Figure 3-1. Schematic of Ventilation Cycle

3.1.2 Prototype Dehumidifiers

The heart of these systems is the dehumidifier. Several designs have been investigated and various desiccants have been used. An early test unit operated by the Institute of Gas Technology (IGT) used a rotary wheel dehumidifier consisting of lithium chloride (LiCl) impregnated in a corrugated laminar flow asbestos matrix. Later, because the LiCl became deliquescent under high humidities, a molecular sieve was used as the desiccant and fiberglass was used as the matrix for health reasons [7]. A prototype system developed by AiResearch used a packed bed of silica gel beads in a thin rotating drum as the dehumidifier [8]. The cooled dehumidifier developed by

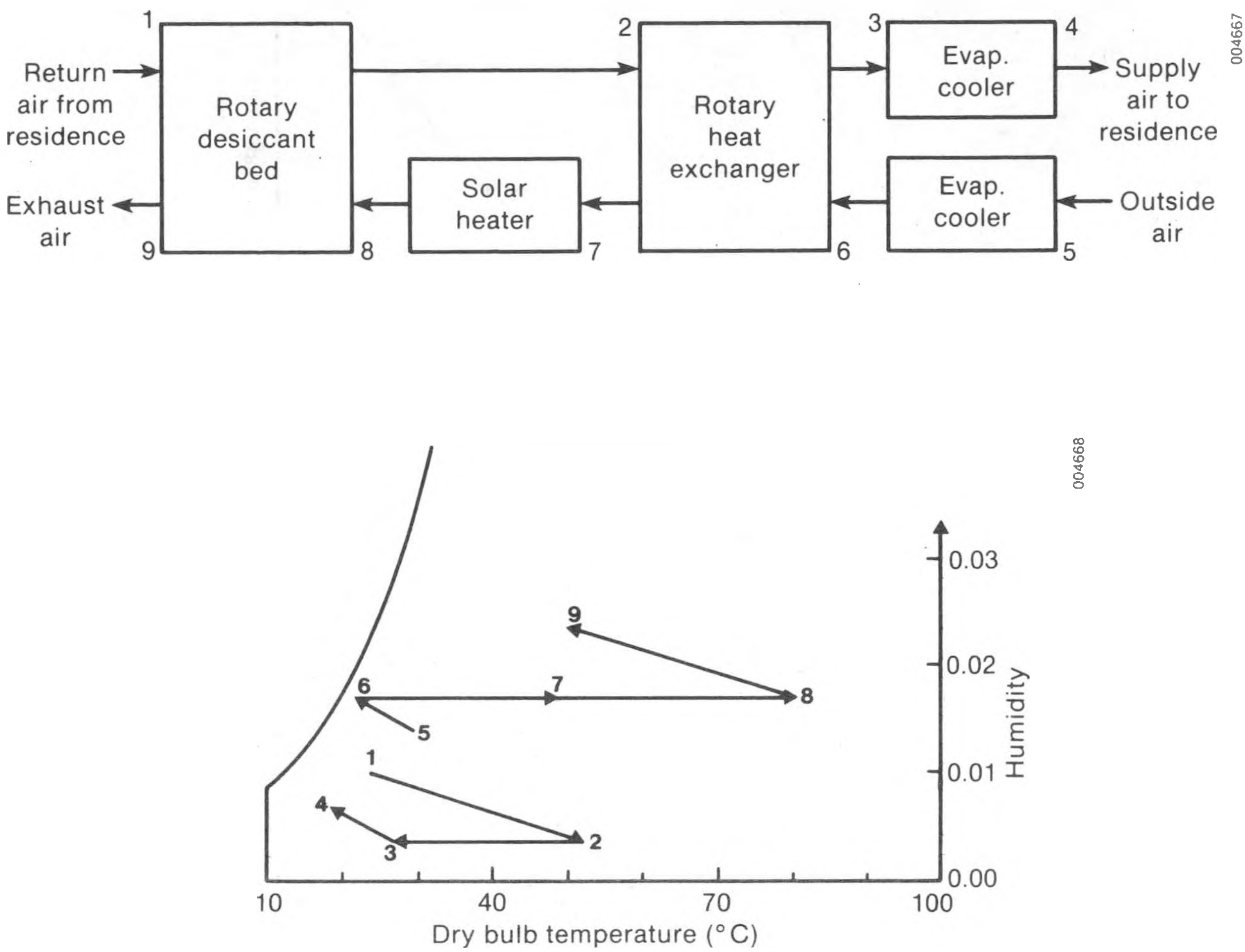

Figure 3-2. Schematic of Recirculation Cycle

IIT consists of two fixed, parallel passage, crossflow exchangers. The process channels are formed by aluminum trays lined with Teflon-bonded silica gel sheets [5].

The dehumidifiers described above have been tested in prototype systems. The performance of these systems is shown in Table 3-1. It has been projected that solid desiccant systems with a thermal coefficient of performance (COP) of 1.2 can be competitive with vapor compression cooling in the 1990-2000 timeframe [10]. None of the current prototype systems can meet this requirement. Prospects for the AiResearch and IIT prototypes have not been sufficient to justify further development at this time. The moisture resistance

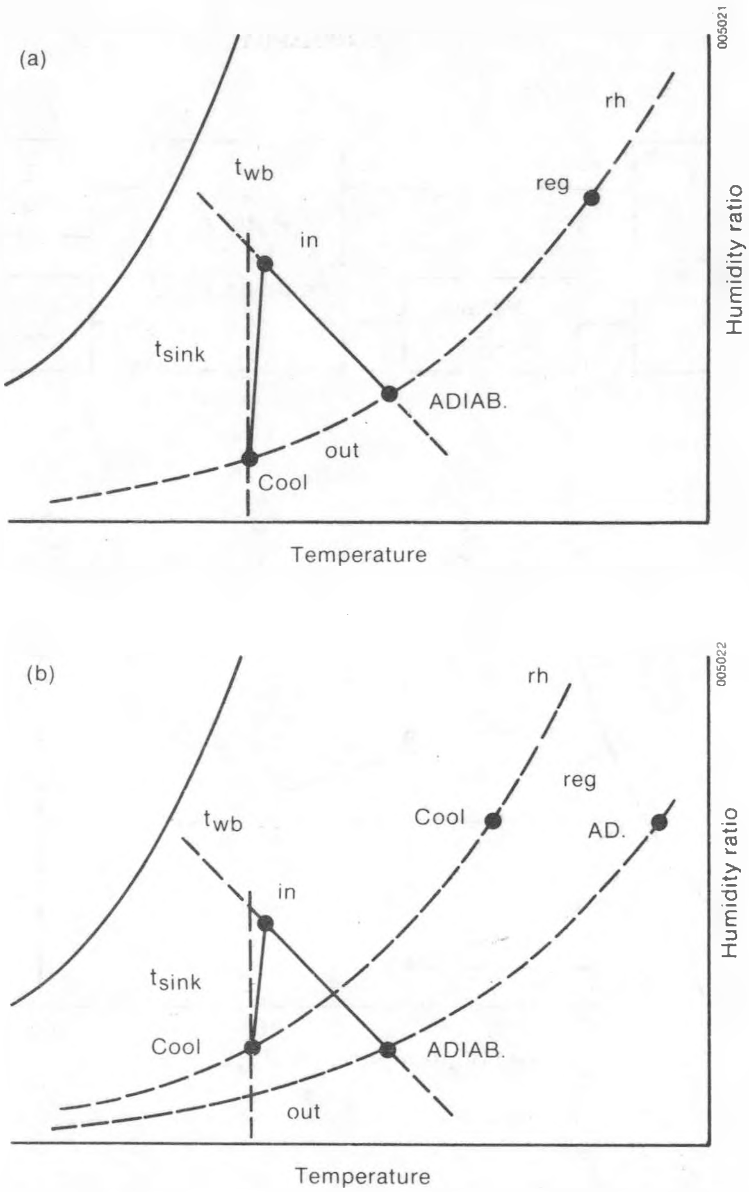

Figure 3-3. Comparison of Idealized Cooled and Adiabatic Dehumidifier Adsorption Process (a) both with same regeneration temperature (b) both with same outlet humidity ratio

in the large silica gel particles and the pressure drop through the packed down of the AiResearch dehumidifier limit the ultimate performance of this unit. The cross-cooling channels in the IIT dehumidifier make fabrication of a compact unit difficult.

Typical heat exchanger effectivenesses in these prototypes have been around 0.90. IGT has projected that the thermal COP of their system at ARI conditions can be raised to 0.95 by increasing the heat exchanger effectiveness to 0.95. Simulations of this improved system show seasonal thermal COPs of 1.2 to 1.5 [9].

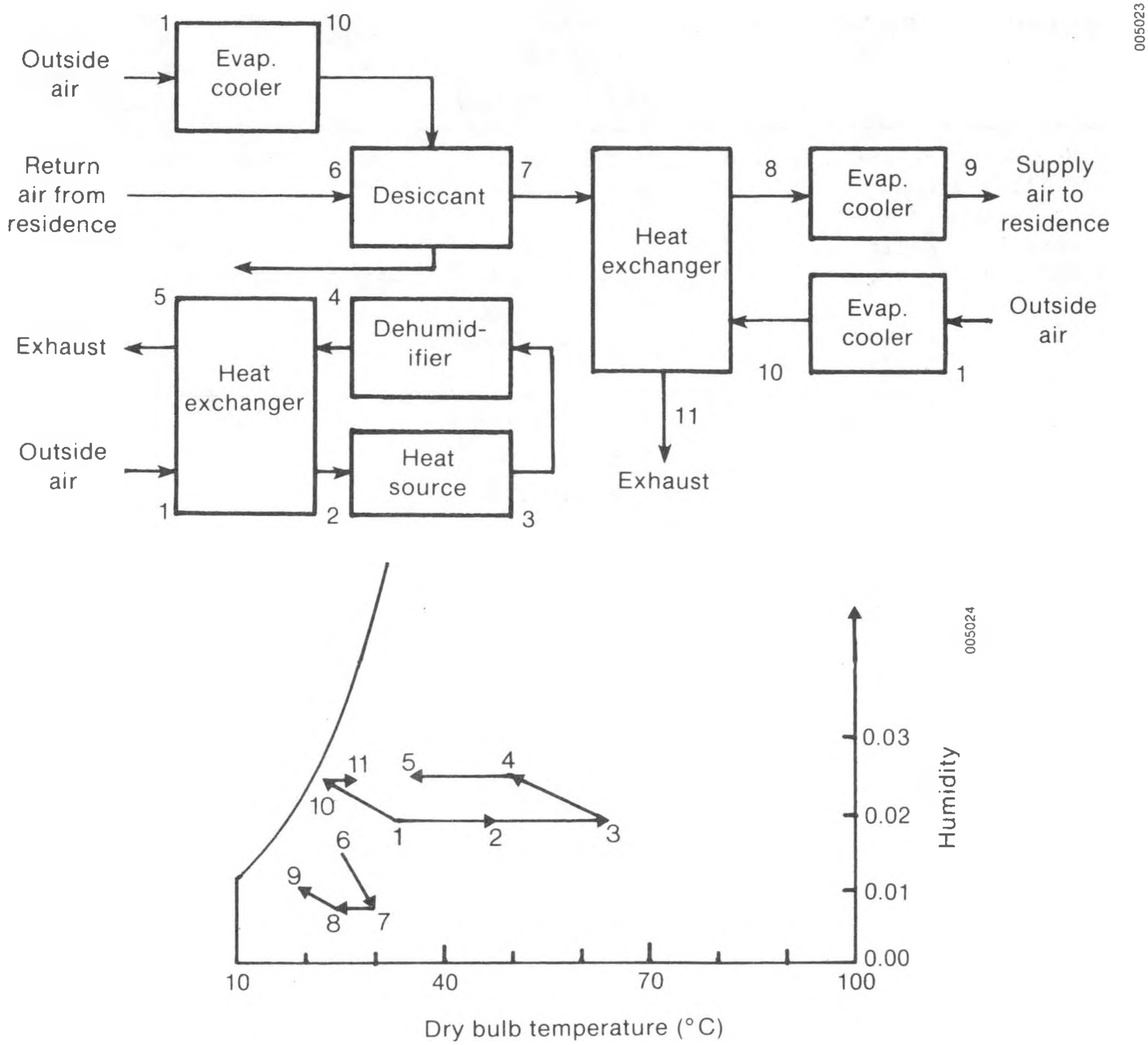

Figure 3-4. Diagram of Illinois Institute of Technology Cooled Desiccant Cycle

Other simulations have also shown the importance of the heat exchanger effectiveness [11,12]. Values of 0.95 will probably be needed to reach thermal COPs of 1.2 and greater. Studies have shown, though, that the greatest potential for reaching these COPs lies in improving the performance of the dehumidifier.

Table 3-1. Performance of Current Prototype Systems

System[b]	Cycle	T_{Reg} (°C)	ARI Design Point[a] Capacity (kW)	ARI Design Point[a] Capacity (kJ/kg)	ARI Design Point[a] COP_{th}	ARI Design Point[a] COP_e	Seasonal Simulation COP_{th}	Seasonal Simulation COP_e
IGT Solar-MEC III [9]	Recirc	103	9.5	21.1	0.46	8.0	0.6-0.7	9.2-11.1
	Vent	95	7.5	16.7	0.51	6.5		
AiResearch SODAC [8]	Recirc	93	4.6	10.9	0.60	5.8	--	--
	Vent	93	3.4	7.9	0.48	4.3	--	--
IIT [5]	Cross-Cooled	65	--	--	0.60	--	--	--

[a]ARI design point: Room 26.7°C 0.0110 kg/kg
Ambient 35.0°C 0.0142 kg/kg

[b]Typical heat exchanger effectiveness is 0.90.

3.1.3 Improved Dehumidifiers

Barlow [13] determined the following desirable characteristics for high performance dehumidifiers and assessed dehumidifier geometries with respect to those characteristics.

1. High heat and mass transfer Ntus
2. High ratio of Stanton number to friction factor
3. Small solid-side resistance to mass transfer
4. Large transfer surface area per unit volume
5. Use of common materials and manufacturing techniques
6. Suitable desiccant properties.

Five geometries were investigated. These included the three prototypes already discussed; IGT's laminar flow corrugated rotary wheel, AiResearch's packed bed thin rotating drum, and IIT's fixed, parallel passage, cross-cooled exchanger. The other two geometries were a fixed exchanger, developed at UCLA [14,15], composed of laminar flow channels formed by sheets of Lexan coated with a layer of silica gel particles, and SERI's parallel passage wheel formed by spirally winding a polyester tape also coated with silica gel particles [16]. It was concluded that the SERI parallel passage wheel offered the best potential for obtaining high thermal COPs (items 1, 3, and 4) with reasonable parasitic power consumption (item 2) in a compact, easy to construct design (item 5).

Although actual tests have yet to be conducted, computer simulation of the SERI wheel has shown an ARI design point thermal COP of 1.07 [16]. This is near the goal of 1.2 and has been obtained under nominal operating conditions that have not been optimized. Exxon, with recent help from the Gas Research Institute (GRI), has also developed a parallel passage dehumidifier wheel. A thermal COP of 1.05 was obtained at ARI conditions from a laboratory test unit, again at conditions that had not been optimized [17]. Computer simulations show the potential of reaching a COP of 1.3 [17]. The projected performance of these near-term advanced dehumidifier systems is summarized in Table 3-2.

The development of laminar flow parallel passage rotary dehumidifiers has significantly aided the improvement of overall system performance and has led to compact system designs. While important, the geometry is only one factor that governs the performance of the dehumidifier.

As noted above, COPs near 1.1 have been obtained under operating conditions that have not been optimized. Certain variables such as process air inlet state and regeneration air humidity ratio cannot be controlled. Regeneration temperature can be controlled, however, and the effects are well studied. Systems with adiabatic silica gel dehumidifiers typically have a maximum thermal COP for regeneration temperatures in the range of 60^{o}–$80^{o}C$ [7,8,11,16,17]. Cooling capacity, and therefore electric COP, increases with regeneration temperature, however. The optimum regeneration temperature in terms of lowest operating costs, therefore, may not coincide with that for maximum thermal COP. A small improvement in performance has been obtained by using a nonuniform temperature profile at the regeneration inlet [9].

Table 3-2. Projected Performance of Near-Term Advanced Dehumidifier Systems

Dehumidifier	Cycle	HX	T_{Reg} ^{o}C	ARI Design Point: Capacity kW	Capacity kJ/kg	COP_{th}	COP_e
IGT HCOP [9]	Vent	0.95	--	8.4	18.7	0.95	6.2(Computer Projection)*
SERI [16]	Vent	0.95	80	--	16.7	1.07	7.2(Computer Projection)*
EXXON/GRI [17]	Vent	0.93	77	2.3	12.9	1.05	5.4(Laboratory Test)*
	Vent	0.93	77	--	15.1	1.3	8.7(Computer Projection)*

*Not completely optimized.

For rotating wheel dehumidifiers, the speed of rotation is important. Optimum values exist and depend on the amount of desiccant present, the process air flow rates, and the dehumidifier inlet states [7,11]. In many cases, the process and regeneration periods of the wheel have different capacities to adsorb and desorb moisture. Unbalancing the flows between the two periods can improve performance. Increases of 10%-40% in thermal COP with losses in capacity of only 5%-10% have been noted [7]. The AiResearch prototype was operated with unbalanced flows, and use of unequal flow areas (i.e., period lengths) was suggested [8]. It is suspected that the IGT systems were operated with unbalanced flows [7,9]. The SERI and Exxon wheels were operated with balanced flow, however, so potential for improvement by completely optimizing these parameters exists. This has yet to be done.

As the desiccant wheel leaves the regeneration period and enters the adsorption period, it is hot, and the outlet state of the adsorption period is hot and wet for a short time. If this part of the adsorption outlet air stream could be removed, the system cooling capacity could be increased. Since this air is hot, it can be reused in the regeneration period to offset some thermal energy and improve the COP. Purging processes such as this have been used in the IGT [9] and AiResearch [8] prototypes and a related preconditioning process has been used in the IIT [5] system. Jurinak [7] has shown that cooling capacity can be increased by up to 25% and thermal COP by up to 10% by purging. The SERI and Exxon wheels did not incorporate purging.

The properties of the desiccant are of obvious importance to the performance of desiccant dehumidifier systems. The prototype dehumidifiers have been limited to commercially available desiccants, although some effort has been expended in attaching these desiccants to the supporting matrix [5,14,16]. Various grades of silica gel have been the most common choice.

Several studies have been done to determine a desirable set of properties for desiccants used in cooling systems. General arguments, based on systems analysis and equilibrium considerations, have been made by Collier et al. [18] for ideal desiccants. Barlow and Collier [19] have considered staging hypothetical desiccants in a dehumidifier and found potential for a 6%-10% increase in performance.

Jurinak [7] has performed a parametric study of the effects of various desiccant properties on the drying performance of the dehumidifier and total system performance. The results are briefly summarized in Table 3-3. Control of items 1, 2, and 5 depends on selection and possible development of appropriate desiccants. The thermal capacitance (item 3) of the dehumidifier can be controlled during construction. The Exxon wheel [17] contains inert material to increase its thermal capacitance and improve thermal COP. A decrease in cooling capacity and greater sensitivity to operating conditions are two disadvantages of this concept, however [7]. The development of parallel passage dehumidifiers using thin layers of desiccant has done much to reduce the problem of moisture diffusion (item 4).

Jurinak [7] concluded that commercially available microporous silica gels are attractive materials for dehumidifiers and that radical changes in properties would be required to significantly improve system performance. Though limited, the search for improved desiccants continues. An activated carbon

Table 3-3. Effects of Desiccant Properties on Dehumidifier and System Performance

Increasing	Dehumidifier Efficiency*	Vent System	
		Capacity	COP_{th}
1. Heat of adsorption	−	−	+
2. Maximum water content	+	+	−
3. Thermal capacitance	−	−	+
4. Moisture diffusion resistance	−	−	−
5. Hysteresis	−	−	−

*Defined as ratio of moisture removed from air stream during adsorption period to the maximum possible.

fiber has been considered as a desiccant for cooling systems [20,21]. Manganese oxides and mixtures of manganese oxides with silica gel (to lower moisture diffusion resistances) are being investigated [22].

3.2 ANALYSIS TOOLS FOR SOLID DESICCANT DEHUMIDIFIERS

The most difficult component to model in solid desiccant cooling systems is the dehumidifier. There are two reasons for this. First, the mathematical formulation of the sorption process is described by partial differential equations for the conservation of mass and energy, for the rate of heat and mass transfer from the air stream to the desiccant, and for the rate of mass transfer within the desiccant. These are all coupled by additional equations describing the equilibrium properties of the desiccant. Second, these equations contain transfer coefficients and other parameters that are difficult to obtain accurately analytically or experimentally.

Attempts to model the adsorption process began with Hougen and Marshall's [23] development of an analytical solution for isothermal adsorption using a linear equilibrium relationship and effective (lumped) gas-side transfer coefficients. Later work extended the analysis to include solid-side mass diffusion [24,25], general isotherms, and adiabatic operation [26,27,28]. These analyses, both analytical and numerical, have modeled single-blow operation of the adsorber starting with uniform initial conditions. To analyze desiccant cooling system operation, models of periodic steady-state dehumidifier operation are needed. The models available for this purpose will be described and assessed in this section.

3.2.1 Detailed Analysis of the Dehumidifier

Different levels of sophistication are needed depending on the model's use. For detailed analysis of the dehumidifier processes and performance, finite difference models have been most popular, although several analytical models have been attempted. The assumptions made by most of these models in deriving the set of equations to be solved are:

- Negligible axial (and circumferential) heat and moisture diffusion
- Negligible gas-side heat and moisture capacity
- Transfer coefficients constant over space and time
- No carryover or leakage from one period to the next
- Gas-side controlled heat transfer (or can be modeled as such)
- Gas-side controlled mass transfer (or can be modeled as such).

Detailed and gas-side resistance models for heat and mass transfer are shown schematically in Figure 3-5. These assumptions hold well for most dehumidifier applications, except that gas-side controlled moisture transfer does not hold well for packed beds using large desiccant particles.

Pla-Barby et al. [29] have developed a finite difference solution to the governing equations of the dehumidifier. Both parallel flow and counterflow rotary steady-state operation, as well as fixed bed, single-blow operation, can be modeled. Effective gas-side transfer coefficients are assumed. Satisfactory agreement is reported between the model and experimental data and

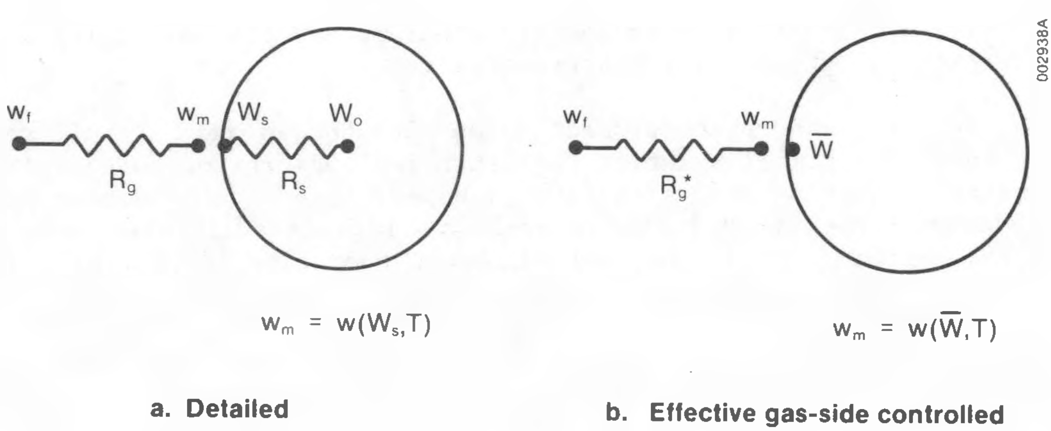

Figure 3-5. Resistance Model for Mass Transfer

the results of other models. The model was used to parametrically study the performance of a balanced flow silica gel dehumidifier in terms of two non-dimensional parameters representing the dehumidifier size, air flow rate, and rotational speed. Average outlet conditions are reported, and optimum operating points are determined.

Barker and Kettleborough [30] have examined fixed bed, single-blow, and parallel flow rotary silica gel dehumidifier operation using a finite difference model very similar to that of Pla-Barby. The model was checked against the results of Bullock and Threlkeld's [27] single-blow model with good agreement. The transient outlet conditions for the startup of a rotary bed from uniform initial conditions are reported. Steady-state operation is obtained by running transient calculations for a long enough period of time.

Holmberg [31] has also modeled a counterflow rotary dehumidifier with a finite difference model. Again, gas-side controlled transfer rates are assumed. A staggered mesh of air and matrix nodes was used. Uneven spacing in the circumferential direction was used to accommodate the rapid changes that occur during the initial portion of each period while being efficient during the rest of the cycle. The model accounts for longitudinal heat conduction in the dehumidifier yet uses a linearized equilibrium relationship and constant heat of adsorption for silica gel. The model has not been compared with experimental data or other models.

Mathiprakasam and Lavan [32] have presented an analytical solution to steady-state dehumidifier operation. Gas-side controlled moisture transfer and a constant heat of adsorption are assumed. The analytical solution is made possible by linearizing the derivative of the equilibrium relationship. The constants in this linearization are chosen based on empirical results of another model. The equations are solved with Laplace transforms. The method is much more rapid than other numerical methods and agrees well with the results of more detailed models.

Mathiprakasam and Lavan [33] have also developed a finite difference model for the steady-state operation of a cross-cooled dehumidifier. Rather than assuming a lumped gas-side mass transfer coefficient, the solid-side moisture diffusion is approximately accounted for by postulating a linear moisture profile in the desiccant. This provides a known relationship between the surface and average moisture contents of the desiccant. The model has been compared with experimental data, and is in good agreement [34].

Maclaine-cross [35] and Banks et al. [36] have developed a detailed finite difference solution of the rotary dehumidifier equations. Two step sizes in the time (or circumferential) direction are used: a small step to adequately follow the rapid changes that occur during the initial portion of the period and a longer step to efficiently cover the rest of the period. The equations are solved for three successively smaller grid sizes, and the known second order accuracy of the difference scheme is used to quadratically extrapolate the results to zero grid size. This model has probably been the most used of any for the detailed analysis of the dehumidifier. It was originally developed for use in checking the analogy method.

Brandemuehl [37] has adapted the program to investigate the effects of non-uniform inlet conditions and transient operation. Jurinak [7] has used the program extensively to model the effects of desiccant properties (including hysteresis), unbalanced flow, and purging on the performance of rotary dehumidifiers. A limitation of the model is that effective gas-side controlled moisture transfer is assumed. The experimental verification that exists has been done for packed bed wheels. The model can be made to agree with the data by varying the effective moisture transfer coefficient used in the model [38]. It is expected that this model will work very well for parallel passage rotary dehumidifiers since solid-side moisture resistance should not be significant.

Maclaine-cross [35] and Banks et al. [36] have developed an approximate analytical model of the heat and mass transfer in the dehumidifier based on an analogy with heat transfer alone. The coupled governing equations are transformed to uncoupled equations in terms of two characteristic potentials. These equations have the same form as those for heat transfer alone to which the solution is known [39]. The transformation assumes that the new potentials are linear combinations of air enthalpy and humidity ratio and that the Lewis number is unity. The analogy method has been compared with Maclaine-cross' finite difference model and found to work well for high performance dehumidifiers with low thermal capacitance, for Lewis numbers of unity, for favorable isotherms (i.e., silica gel), and operated under balanced flow [7]. Although less accurate than the finite difference models for quantitative analysis of the dehumidifier, the major usefulness of the analogy method is in its economy of computation and its qualitative description of the wave nature of the dehumidifier processes.

Barlow [11] has developed a dehumidifier model that does not solve the governing equations directly. Rather, a bed is discretized along its length and each node is modeled as a counterlow heat and mass exchanger. To model a wheel, the bed (a slice of the wheel) is followed as it rotates through the cycle.

In the model, a mass transfer calculation is first performed considering the node to be a counterflow exchanger. The effectiveness is calculated from the mass capacity rate ratio and the mass transfer NTU for that node. The outlet air state from the node is obtained from the effectiveness and the inlet states, the inlet air humidity ratio to the node, and the humidity ratio in equilibrium with the initial desiccant water content of the node. Next, an energy balance is performed to account for the heat of adsorption. Finally, a heat transfer calculation is performed similarly to the mass transfer calculation. The outlet air state from a node is used as the inlet state for the next node. The final desiccant state is used as the initial node state at the next time step.

While the model does not solve the governing equations directly, the calculation procedure is straightforward. Its utility lies in the fact that many geometries, flowstream arrangements, and variations in desiccant properties can be modeled easily. Computation time is comparable to that of MOSHMX. The model has not been validated against experimental data for the case of cyclic operation, but good agreement has been found between model predictions and data for single-blow operation [11,16].

All of the models discussed so far have assumed that moisture transfer can be modeled as gas-side controlled or at best have assumed some empirical moisture profile in the desiccant. The reason for this is that finite difference methods are computationally very expensive. To model solid-side diffusion, finite difference analysis must be applied in an additional direction, and this increases the amount of calculations. While the development of parallel passage dehumidifiers has reduced the problem of solid-side diffusion, knowledge of the mechanisms is important to the complete understanding of dehumidifier performance.

Pesaran [40] and Pesaran and Mills [41] have solved the conservation equations for a dehumidifer bed, including the solid-side diffusion equation, by finite difference techniques. Spherical desiccant particles are assumed, and both surface and Knudsen diffusion are modeled. Good agreement has been obtained between model predictions and experimental data for the adsorption process. The model is limited to single-blow operation because of the computation time required.

The models discussed here are summarized in Table 3-4. Other models have been developed and used by IGT, AiResearch, and Exxon in their programs, but little information is available on them. While further steady-state validation is needed, Maclaine-cross' model is recommended for detailed modeling of adiabatic dehumidifiers because of its direct solution of the governing equations. Modifications exist for modeling purged flows and transient operation. The analogy method remains useful for qualitative descriptions of dehumidifier performance. Barlow's model will be useful for investigating the effects of desiccant property variations along the flow direction and more complicated situations such as the direct solar regenerated dehumidifier [20] or the cross-cooled dehumidifier. Pesaran's model provides a useful tool for analyzing the diffusion processes in the desiccant and their effect on dehumidifier performance [40,41].

3.2.2 Dehumidifier Models for System Simulations

Most of the models described above can not be used in long-term and seasonal simulations of system performance because they require excessive amounts of computing time. The one model from above that is applicable to this use is the analogy method, which has been used extensively in system simulations [7,12,42,43]. Several levels of sophistication are available. Constant effectivenesses can be used to describe generic dehumidifiers and to assess performance in terms of system configuration and climate. A more detailed model can be used to accurately assess the optimum performance of a particular dehumidifier in terms of flow ratios, rotational speeds, etc.

Another method that has been used to model the performance of a particular dehumidifier in a systems context is to use correlations derived from the results of finite difference codes [44,45]. These models are limited in scope so a reasonable number of parameters can be used. Each different dehumidifier requires a correlation to be generated.

Van Den Bulck [46] and Van Den Bulck et al. [47] have reduced this effort somewhat. They have generated correlations for a temperature and humidity ratio effectiveness based on the results of Maclaine-cross' finite difference solution and an equilibrium model of the dehumidifier. A large set of operating conditions is covered. However, these correlations apply only to a

Table 3-4. Comparison of Detailed Dehumidifier Models

	Type of Model (a)	Fixed Bed		Rotary Wheel		Mass Transfer Resist		Computing Time (b)	Validation (c)
		Single Blow	Periodic Steady State	Steady State	Transient	Effective Gas-Side	Solid-Side		
Adiabatic									
Pla-Barby [29]	F			X	X	X		H	L
Barker and Kettleborough [30]	F	X		X	X	X		H	L
Holmberg [31]	F			X		X		H	N
Mathiprakasam & Lavan [32]	A		X			X		M	N
Maclaine-cross [35,7]	F			X	X	X		H	L
Banks et al. [36]	A			X		X		L	L
Barlow [11]	O	X	X	X		X		M	G
Pesaran [40,41]	F	X					X	H	G
Cross-cooled									
Mathiprakasam & Lavan [33]	F		X				(d)	H	G

(a) F = Finite difference A = Analytical O = Other

(b) H = High M = Medium L = Low

(c) G = Good L = Limited N = None

(d) Empirically modeled.

dehumidifier composed of all silica gel. New correlations must be generated for other sets of desiccant properties.

An equilibrium model has been used to analyze the effect of additional inert heat capacity in the dehumidifier on system performance [48]. With the assumption of constant velocity wave fronts, overall mass and energy balances are used to calculate the ideal outlet dehumidifier conditions. At this time, the model is limited to idealized dehumidifiers.

3.2.3 Desiccant Properties and Other Parameters

Accurate results can be obtained from any of these models only if accurate data are provided to them. The equilibrium properties of the desiccant, properties of the construction materials, heat and mass transfer coefficients, and other design parameters must be known.

The equilibrium properties of common silica gels have been well studied, yet significant variances exist and questions remain [11]. Of particular interest is the question of hysteresis in the equilibrium properties from adsorption to desorption. Tests are currently planned to quantify the hysteresis in silica gels and other desiccants [49]. Another area of uncertainty is the parameters that describe the moisture transfer process in the desiccant (when modeling solid-side diffusion) and the transfer processes on the gas-side. The heat transfer coefficient most commonly used for parallel passage wheels is for fully developed laminar flow with a constant temperature boundary. No methods exist for determining the effective gas-side mass transfer coefficient except to match experimental data empirically.

3.3 OTHER COMPONENT MODELS

Heat exchangers and evaporative coolers are the other major components that need to be considered in solid desiccant cooling systems. Both of these components have been well studied in the past and can be easily and accurately modeled. For the purpose of systems modeling, the assumption of a constant temperature effectiveness for the heat exchanger and a constant saturation effectiveness for the evaporative cooler can be used.

3.4 RESEARCH ISSUES AND GOALS

Although solid desiccant systems have been investigated extensively, the current level of performance is not quite competitive with vapor compression cooling. Yet the gains made in the past couple of years indicate a high probability for success. High performance, low pressure drop designs have been developed for heat exchangers and dehumidifiers. Continued development of desiccant materials provides further promise. Optimization of dehumidifier operation in terms of regeneration temperatures, flow rates, rotational speed, purging, etc., may improve the performance of current dehumidifier designs sufficiently to make them competitive in the near term.

SECTION 4.0

LIQUID DESICCANT SYSTEMS

Unlike solid desiccant cooling systems, which are viewed as solar-assisted, cooling only systems, liquid desiccant systems are being considered for both heating and cooling and as strictly solar-fired. The characteristics of liquid systems offer several advantages over solid systems. Liquid desiccants can be regenerated on thin-film, open flow collectors that would be inexpensive to build. Energy can be stored as chemical energy in the form of concentrated desiccant solution rather than thermal energy. This allows for greater energy storage and reduced reliance on auxiliary thermal energy sources.

4.1 CURRENT STATUS

As of this time, the heating aspects of liquid desiccant systems have been investigated very little, so only cooling will be addressed here. Overall, liquid desiccant systems have not received as much attention as solid systems and so development is not as advanced. Analytical and experimental work has concentrated mainly on the collector/regenerator. Little overall systems analysis has been done.

4.1.1 Systems

The liquid desiccant systems proposed use an externally cooled absorber/conditioner similar in concept to the cross-cooled solid desiccant dehumidifier of the IIT system [5]. This cooling can be provided by water from a cooling tower, directly from a well or other source, or by evaporatively cooled air.

Griffiths [50] and Robison [51] have proposed the system configuration shown in Figure 4-1. Return air from the conditioned space (1) contacts concentrated liquid desiccant solution in the absorber and is dehumidified. Simultaneously, the heat of absorption and possibly some sensible heat is rejected to the heat sink. This dried and cooled air (2) is then evaporatively cooled (3) and delivered to the conditioned space. During periods of favorable insolation, weak solution is pumped from storage and heated in the collector. Ambient air (6) is then contacted with the solution, removing moisture (7) and concentrating the solution.

Robison [51] has proposed using a thin-film, open flow collector for the collector and regenerator as shown in Figure 4-2. A complete system using this concept was designed and installed in a test house. A counterflow packed tower was used as the absorber and well water was used as the heat sink. The collector/regenerator was constructed of plywood coated with neoprene liquid roofing material and covered with low-iron glass. Calcium chloride solution was used as the desiccant. The system was operated for two cooling seasons. The average ratio of cooling supplied to incident solar energy was 0.6 and the electric COP was 2.9 [52]. Several problems occurred with the roofing

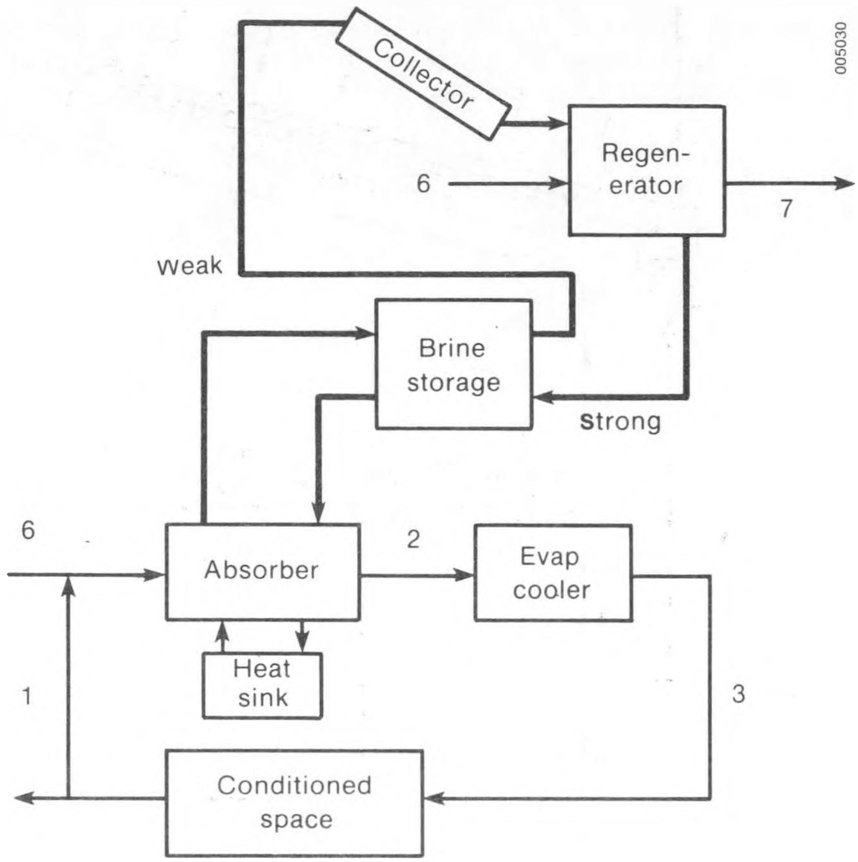

Figure 4-1. Liquid Desiccant Cooling System Proposed by Griffiths [50]

materials, insulation, and storage tanks, but overall the performance was satisfactory [53].

Turner [54] and Peng and Howell [55] have proposed a system configuration similar to Griffith's in which part of the air stream leaving the absorber is used in a packed cooling tower to produce the cooling water for the absorber. Ambient air is taken in with the recirculated room air to maintain the mass balance. Peng and Howell [55] have proposed using recirculated room air to produce the absorber cooling water. Numerical analysis of these systems shows thermal COPs on the order of 0.5 [55].

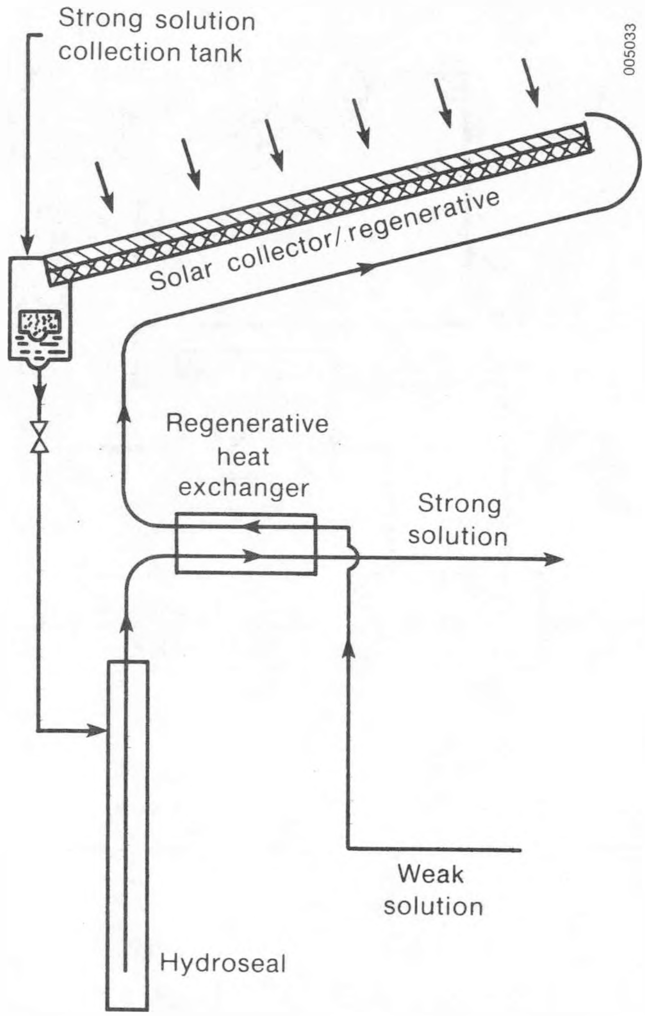

Figure 4-2. Thin-Film, Open Flow Collector Proposed by Robison [51]

4.1.2 Absorber/Conditioners and Regenerator/Concentrators

The two most important components in liquid desiccant cooling systems are the absorber and the regenerator. The absorber has received little attention so far. Turner [54] proposed using a finned-tube surface in which desiccant solution is sprayed over fins and air is blown through the fin passages. Cooling water is then pumped through the tubes. The absorber installed in Robison's system [51] consisted of a packed column rather than fins. This was an adaptation of the current technology in commercial liquid desiccant drying systems.

In Griffith's system, a packed column is also proposed for the regenerator with the solar collector preheating the desiccant solution. These functions can be combined by using an open flow collector in which the desiccant solution flows over the collector surface in a thin film in direct contact with the ambient air (Figure 4-2). Unglazed collector/regenerators have been built and tested in both dry [56] and humid [57,58] climates. Both collectors were constructed as conventional asphalt shingled roofs with some precautions taken to protect against corrosion by the desiccant solution. This results in a very inexpensive collector. Satisfactory operation of the collector/regenerators was obtained with only minor problems due to contamination by dust, bugs, and birds. The major problem encountered was rain. The desiccant was either diluted if rain was drained to storage to save the desiccant washed from the roof or lost if rain was drained away.

Adding glazing has been recommended to reduce the problems caused by rain and contamination, which Robison [51] did in constructing his collector. Adding the glazing may cause materials problems because of the higher temperatures involved, however.

4.1.3 Liquid Desiccants

Several desiccant solutions have received interest as being applicable to liquid desiccant cooling systems. Triethylene glycol and other glycols have been used in commercial and industrial drying processes and have been considered for liquid desiccant cooling systems [54,55]. Carryover of glycol into the conditioned air stream and evaporation and loss of glycol in the regenerator are disadvantages, however.

Lithium chloride [58] and calcium chloride [51] solutions have received the most attention. Their thermodynamic properties are excellent for applications in liquid desiccant cooling systems. They tend to be corrosive to metals, though, and lithium salts are expensive.

4.2 ANALYSIS TOOLS FOR LIQUID DESICCANT SYSTEMS

As with solid desiccant systems, analysis tools are needed to effectively investigate liquid desiccant cooling systems.

4.2.1 Absorber

Little attention has been given to absorbers specifically designed for use in liquid desiccant cooling systems. Adaptations of the packed column technologies used in commercial liquid desiccant drying and other processes have been considered. Modeling of absorption in packed columns is well developed, but application of the general principles to particular applications may not be as straightforward as desired; therefore, more effort may be needed in this area. Peng [59] has developed a finite difference model for a finned tube absorber and has used this model to optimize absorber design. The model has not been checked against experimental data. Siebe [60] has proposed a simplified two parameter model of a generic absorber. It is useful for long-term simulations of system performance.

4.2.2 Collector/Regenerator

An analytical closed-form solution to the unglazed, thin-film, open flow collector has been developed by Collier [61]. Several assumptions, including small evaporation rates, constant transfer coefficients and heat of absorption, and a particular mathematical form for the equilibrium relationship, were required to obtain the solution. Good agreement was obtained with available experimental data. Siebe et al. [56] and McCormick and Theker [58] have compared the results of Collier's model with their experimental data and also found good agreement. Siebe et al. [56] were able to obtain improved agreement with the experiments by empirically varying the solar absorptivity and convection transfer coefficients.

Finite difference models have been developed for the unglazed open flow collector by Peng [59] and McCormick and Theker [58]. Siebe et al. [56] has compared the predictions of Peng's model with experimental data and Collier's model and found good agreement. McCormick and Theker [58] found that the thermal capacitance of the roof can cause a transient affect not accounted for in the steady state model, although daily average performance is predicted very well by the model.

The Collier and finite difference models provide similar insight into collector performance. Although the finite difference models are much easier to calculate than those of the solid desiccant dehumidifier, the Collier model is faster and almost as accurate. For these reasons, Siebe [60] has used Collier's model for the unglazed collector in long-term simulations of system performance.

Robison [51] presents a simplified model for a glazed, open flow collector. The overall evaporation rate is calculated based on the arithmetic average of the inlet and outlet conditions. This model does not account for any temperature or concentration profiles in the solution or air stream over the length of the collector. No verification of the model is indicated.

Gandhidasan [62] has derived the differential equation describing the water evaporation rate in a glazed open flow collector with parallel flow between the solution film and the air. Constant transfer coefficients and heat of absorption were assumed as well as a particular mathematical form of the equilibrium relationship. The equation is solved numerically. The model has been used to parametrically study the effect of ambient conditions, initial solution concentration, and insolation level on evaporation rates from the regenerator. The model has not been experimentally verified.

Howell and Shephard [63] have developed a finite difference model for a glazed collector with air and solution film in counterflow. An iterative solution is required to obtain steady-state performance. The air is assumed to be in turbulent flow and the thermal entry length is accounted for. A parametric analysis of the effects of inlet conditions, insolation, and flow rates on the concentration and temperature profiles in the regenerator has been done. Again, the model has not been verified against experimental data.

4.2.3 Desiccant Properties and Other Parameters

The equilibrium properties of the commonly used liquid desiccants are well known. The problems of corrosiveness and solidification at high concentrations need to be solved and desiccants that can improve system performance need to be found. The search for improved desiccants has not been extensive, however.

The accuracy of a model depends on the accuracy of the parameters used in the model. As noted with Collier's regenerator model, problems appear to exist in theoretically determining the proper transfer coefficients for unglazed regenerators. The effects of turbulence in the wind and film flow over rough surfaces (such as a shingled roof) may not be adequately described by available correlations. These problems exist for the finite difference models as well. Other such problems may turn up in attempts to accurately model the absorber.

4.3 ISSUES AND GOALS

Liquid desiccant systems are not as well developed as solid systems. Current liquid system performance is poor, especially in terms of electrical power consumption. The packed column absorber may be a major problem in this regard. More work is needed in this area.

Other major issues appear to be materials related. The corrosiveness of some liquid desiccant solutions needs to be dealt with. It appears that glazed open flow regenerators will be required to prevent problems with rain and contamination. Low-cost roofing materials that can withstand the increased temperatures involved need to be found.

SECTION 5.0

HYBRID SYSTEMS

Vapor compression systems are very good at providing sensible cooling. To generate a latent cooling capacity, however, they must cool the air past the dew point to condense and remove the excess moisture as shown in Figure 5-1a. Desiccant dehumidifier cooling systems are very good at producing latent cooling but must overdry the air so sufficient sensible and latent cooling capacity remains after the evaporative cooling process. A portion of the solid desiccant recirculation cycle processes is shown in Figure 5-1b.

By combining a desiccant dehumidifier to meet the latent load and a vapor compression system to meet the sensible load, a very efficient system can be built. As shown in Figure 5-1c, the dehumidifier in the hybrid system is required to remove less moisture than in the desiccant system. This allows the desiccant to be regenerated at a lower temperature, either using solar energy more efficiently or using waste heat rejected from the condenser of the vapor compression unit. The vapor compression unit need handle only the sensible load, so a smaller unit may be used. Performance of the unit is also improved by operating at a higher evaporator temperature since it does not have to condense moisture.

5.1 CURRENT STATUS

The desiccant cooling systems discussed in Sections 3.0 and 4.0 have been proposed for residential applications. Most of the hybrid systems proposed to date have been for commercial applications.

A cooperative effort among ThermoElectron, GRI, and Jewel Supermarkets has resulted in the installation of a hybrid system in a Jewell store [64]. In a food store, much of the sensible cooling is provided by the freezer and refrigeration cases. Additional dehumidification is often required, however. By using a desiccant dehumidifier to remove this moisture, the size of the vapor compression unit and the electrical power consumed can be substantially reduced. Additionally, the freezer and refrigeration cases can be operated more efficiently since less frost buildup should occur.

The simple configuration of components shown in Figure 5-2 is used in the Jewell store. An adiabatic silica gel dehumidifier is used to dry ambient ventilation air. After passing through a heat exchanger to remove a portion of the heat of adsorption, it is mixed with recirculated store air and sensibly cooled by the air conditioner to the required inlet temperature. Many problems were encountered during the first year of operation, yet definite indications of reduced energy use and economic viability were obtained. Optimization of system performance and control remain to be made as the study continues. A similar system has been installed by Cargocaire in another supermarket [64]; however, the mixed recirculation air stream is dehumidified.

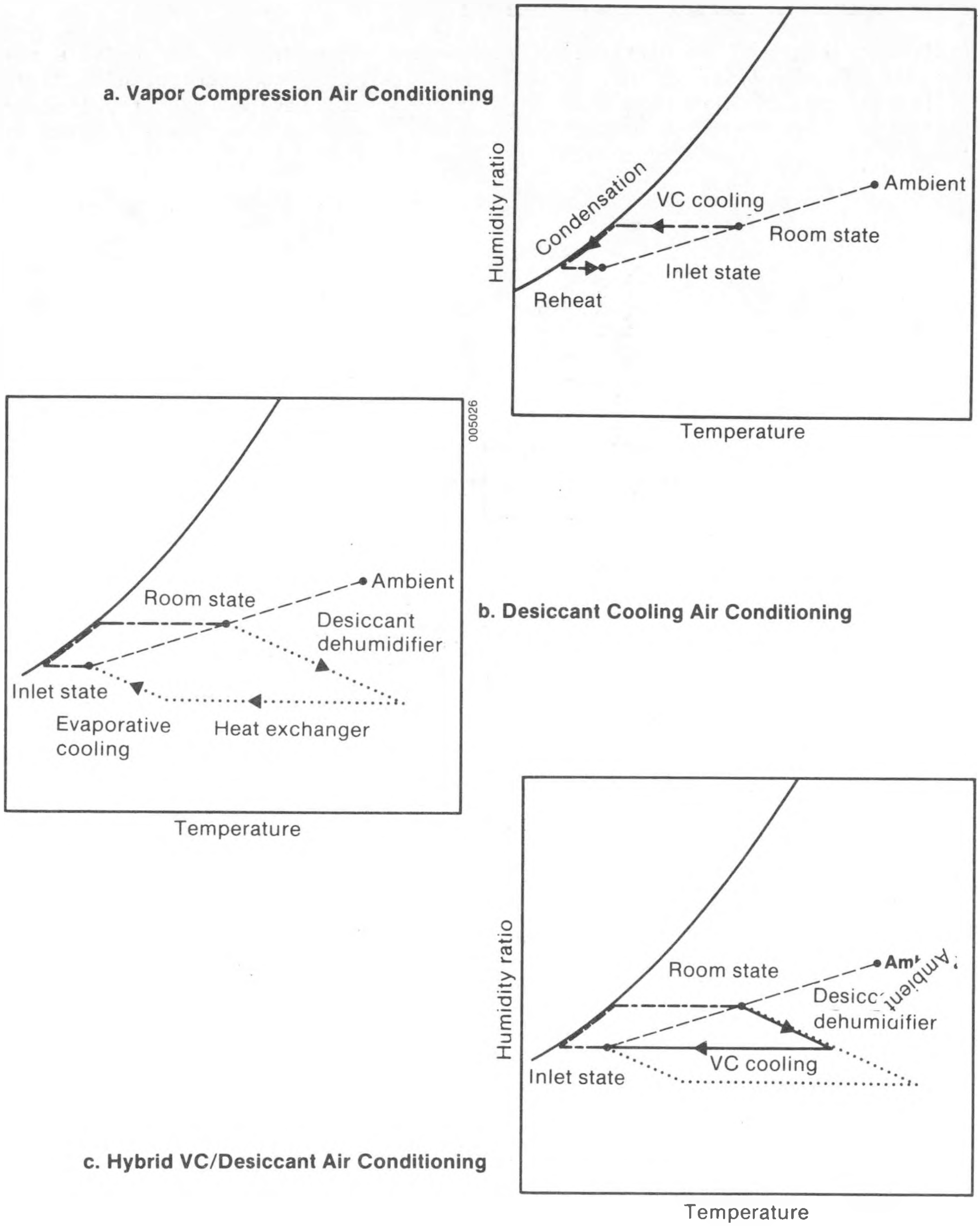

Figure 5-1. Psychrometric Representation of Vapor Compression, Adiabatic Desiccant Dehumidifier, and Hybrid System Cooling Processes

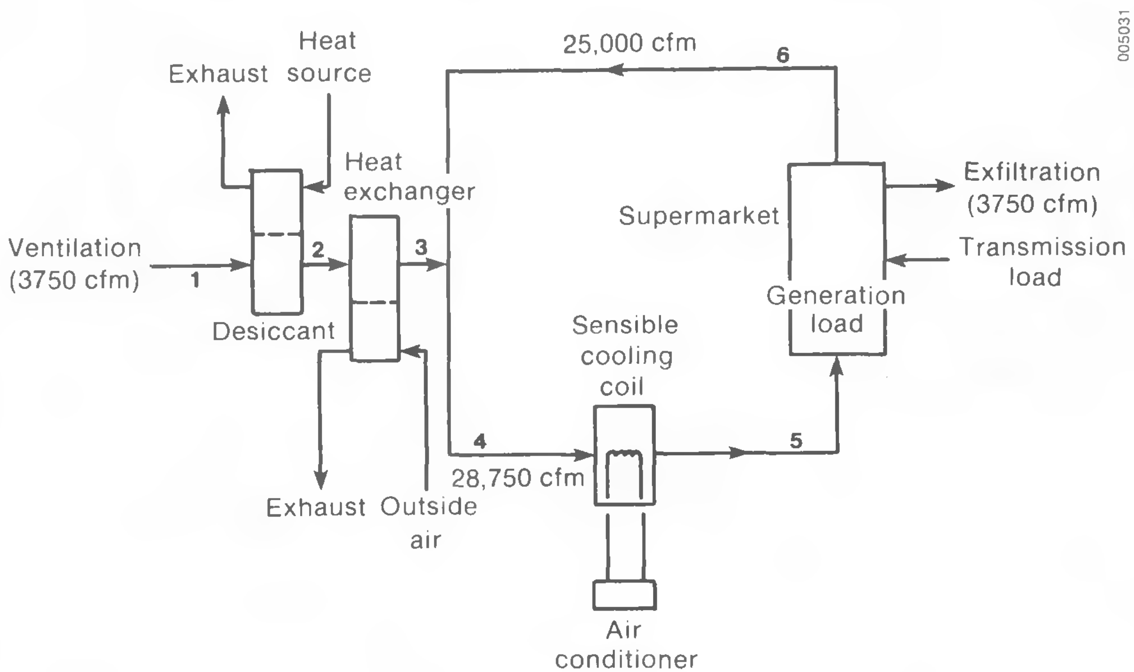

Figure 5-2. Desiccant Cooling System Used in a Supermarket

Hybrid systems can also use liquid desiccant dehumidifiers. Gershon Meckler (of Haines, Lundberg, and Waehler in New York) has designed and installed the system shown in Figure 5-3 in a Veterans Administration hospital [65]. Outside ventilation air is dried by a lithium chloride solution (with external cooling) and passed on to the vapor compression unit. The solution is regenerated by heating it with solar energy or recovered waste heat and subsequent contact with a heated air stream. This hybrid system is expected to reduce building cooling costs by 25%-50%.

Current installations have tended to be somewhat specialized applications because of special cooling requirements and favorable economics. Analytical

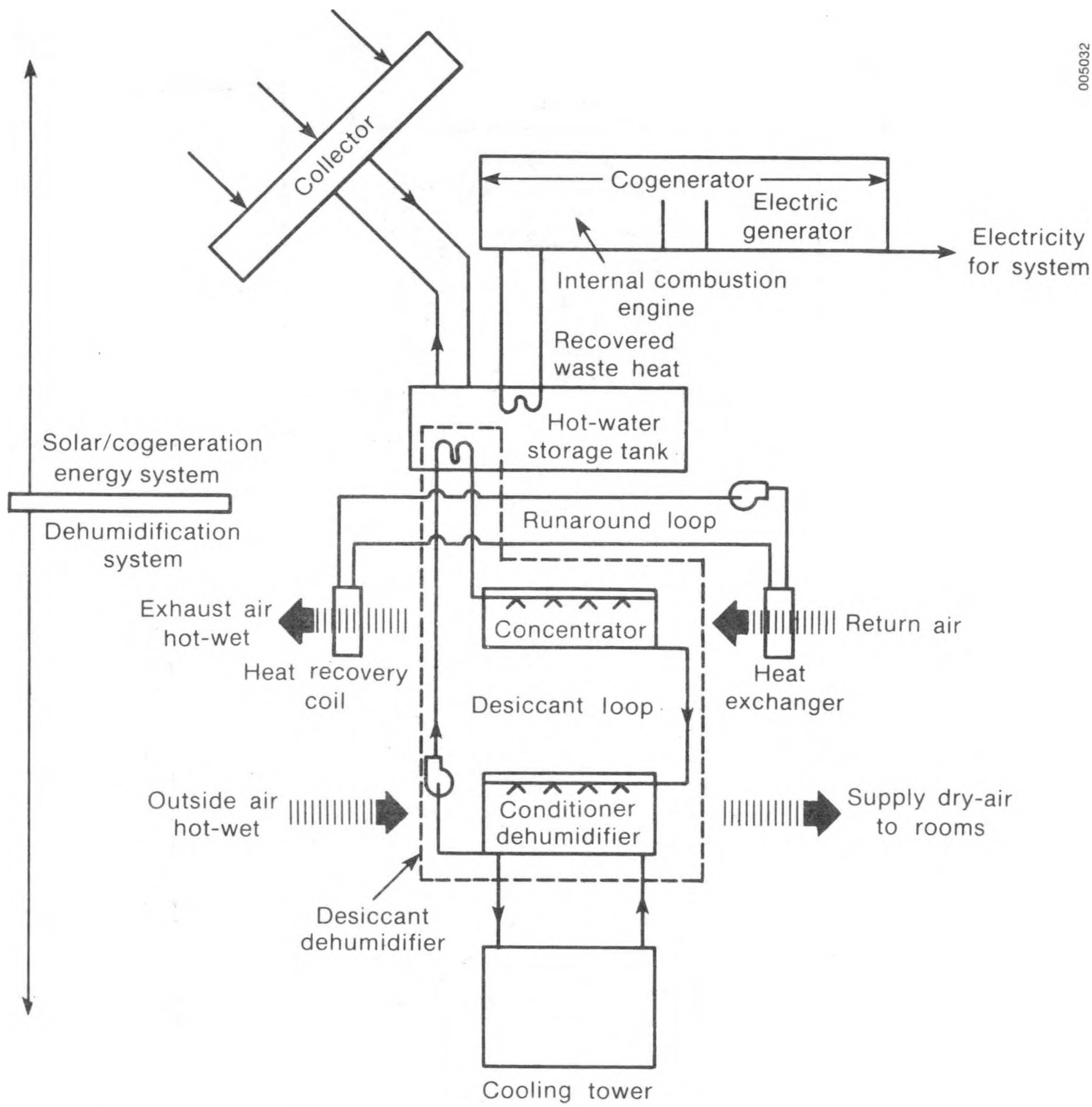

Figure 5-3. Hybrid Liquid Desiccant System Used in a Hospital

studies of a hybrid system for general small office building applications have been done [66,67,68]. The system has an adiabatic silica gel dehumidifier and is shown in Figure 5-4. Room air (1) is mixed with the required amount of ventilation air (6) and dehumidified (3). The air is then cooled (4) to near the ambient wet-bulb temperature (9) in an indirect evaporative cooler. This removes much of the cooling load from the vapor compression unit and is thermally "free" cooling. The dehumidifier is regenerated using combinations of solar energy, condenser waste heat, and auxiliary energy.

Simulations of this hybrid system show that resource energy consumption can be reduced to 35%-55% from the standard vapor compression system [67,68]. The

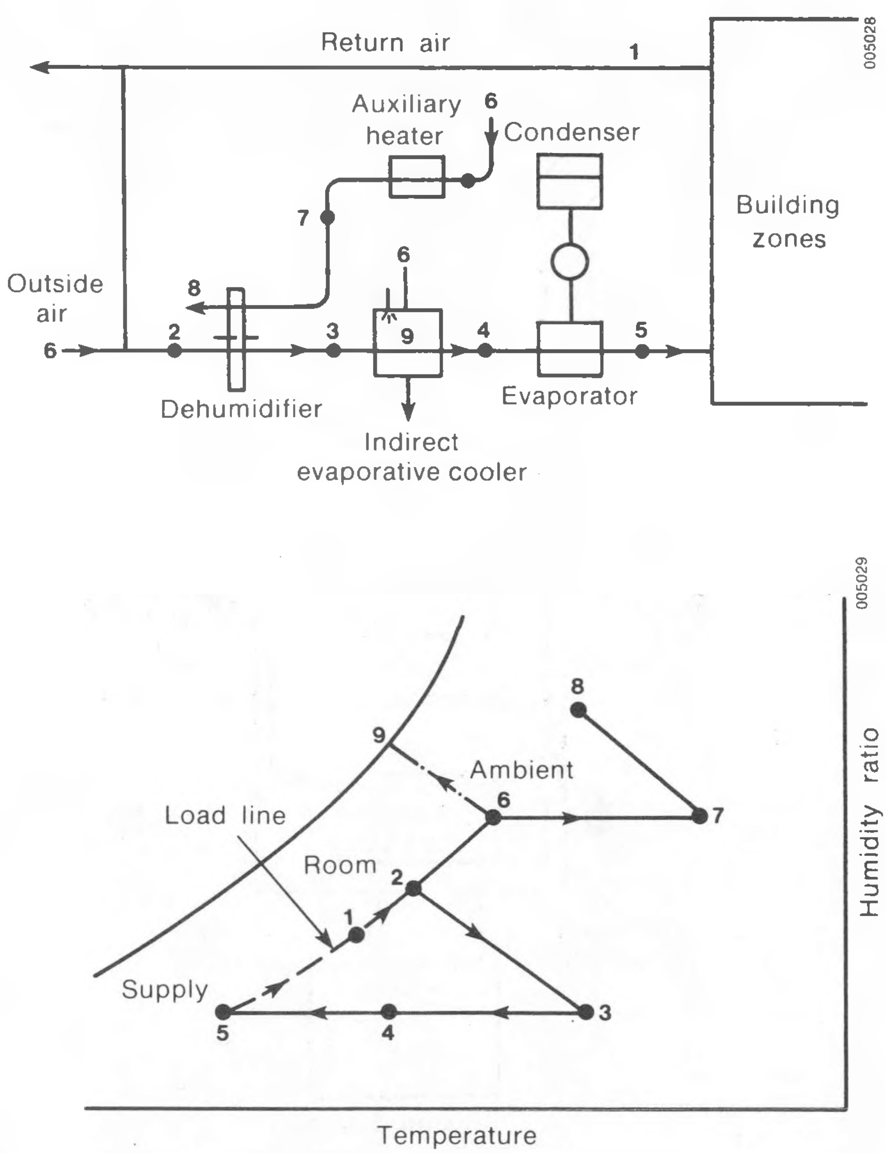

Figure 5-4. Hybrid System for Small General Office Buildings

reduction depends significantly on how much cooling can be obtained from the indirect evaporative cooler. The indirect evaporative cooler is very effective in dry climates, and, therefore, energy savings can be large. However, the indirect evaporative cooler may provide too much sensible cooling. In this case, insufficient heat would be rejected from the condenser to fully regenerate the dehumidifier, requiring auxiliary energy input.

The current supermarket installations do not include solar systems, and the simulations of general application solid desiccant hybrid systems have not been favorable to solar energy since most of the regeneration energy can be supplied by the condenser rejection heat. Only the liquid based systems seem suitable for solar regeneration. Many problems concerning system performance and control remain to be worked out, yet it is very encouraging that actual hybrid systems have been designed and installed in settings where economics are a major factor.

5.2 ANALYSIS TOOLS FOR HYBRID SYSTEMS

The desiccant dehumidifiers in hybrid systems are the same as those in desiccant cooling systems. Therefore, the analysis tools discussed in Section 3.0 for solid desiccant dehumidifiers and in Section 4.0 for liquid desiccant absorbers and regenerators can be used. Although the same tools can be used, the results of analyses of desiccant systems using these tools may not be directly applicable to hybrid systems. For example, a solid desiccant dehumidifier designed to optimize performance of a residential cycle may not optimize hybrid system performance.

The other major component in a hybrid system is the vapor compression unit. Current models consist of curves fitted to manufacturers' catalog data [67]. This provides an adequate and easily calculated model for systems analysis. Vapor compression cooling technology is well established. Improvements in unit performance at the operating conditions in hybrid systems are best handled by manufacturers.

The performance of the hybrid system in Figure 5-4 depends strongly on the performance of the indirect evaporative cooler. Pescod [69] has analyzed and tested a plate type crossflow heat exchanger in which one of the air streams is evaporatively cooled by spraying water in the channel. This provides a low temperature sink for the other stream. For systems analysis, a simple effectiveness based on the degree of approach of the process air stream to the wet-bulb temperature of the evaporatively cooled air stream can be used [67,68]. Effectivenesses of 0.90 to 0.95 can be achieved by this type of unit.

5.3 RESEARCH ISSUES AND GOALS

The development of hybrid systems is tied directly to that of desiccant cooling systems through the dehumidifiers. The research issues identified previously for the dehumidifiers also apply for hybrid systems. Additional issues involve the performance of the vapor compression unit and the development of high effectiveness indirect evaporative coolers.

Many applications and many potential configurations exist for hybrid systems, making it difficult to quantitatively determine performance goals. The potential for the success of hybrid systems appears to be high, however, as economic considerations have led to the installations of both liquid and solid desiccant/vapor compression hybrid systems.

SECTION 6.0

RESEARCH RECOMMENDATIONS

6.1 APRR RESULTS

The previous sections have detailed the state of the art in technology and analysis tools. Because of the status of systems analysis at the time the APRR process was under way, the detailed analysis necessary for complete consideration was limited to the open-cycle solid desiccant cooling and hybrid vapor compression systems. The liquid desiccant-based systems were reserved for later evaluation when TRNSYS modeling capabilities are available and validated using the results of ongoing experimental and monitoring projects.

6.1.1 Open-Cycle Desiccant Cooling

Comparisons of the base case and advanced systems for the open-cycle solid desiccant cooling concept are made in Table 6-1. The base case system used the established performance of the IGT Solar-MECI system with a design-point thermal COP of 0.52 [9]. Evacuated-tube collectors were specified to supply the regeneration temperatures necessary for the molecular sieve desiccant. The system used natural gas as the energy source for back-up regeneration. Based on IGT's performance and cost data, the base case system had a cost-to-cost-goal ratio of 11.3, far from economically competitive with the conventional alternatives.

Table 6-1. Open-Cycle Solid Desiccant Cooling Systems, Base Case vs. Advanced Residential System

	Base Case	Advanced
System Specifications		
Unit thermal COP (kW/kW)	0.52	1.1
Unit electrical COP (kW/kW)	5.0	8.2
Collectors used	Evacuated tube	Thin film
Collector area (m^2)	40	26
System Energy Use		
Electricity (MJ/yr)	13.8	9.2
Gas (MJ/yr)	48.0	21.8
Conventional Energy Use		
Electricity (MBtu/yr)	40.8	46.3
Gas (MBtu/yr)	35.3	0.0
Installed Cost		
Solar	$30,103	$12,539
Conventional	2,636	3,655
Cost/Cost Goal	11.3	2.8

The advanced system used the recent research data on advanced dehumidifier prototypes at SERI to project the performance of the system to a thermal COP of 1.1 and an electrical COP of 8.2. Because the regeneration temperature for silica gel based dehumidifier is 60^{o}-80^{o}C, the advanced collector subsystem uses low-cost, thin-film plastic/metal collectors that are matched to the lower temperatures. Based on conservative estimates of system costs, the cost-to-cost-goal ratio is 2.8, clearly much closer to cost-comptetitiveness, and with further technical improvements it can approach the long-term goals.

The key component development activities identified to improve the base case system to the advanced system were research and development of high-performance dehumidifiers; low-cost, low temperature thermal storage; and low-cost, flat plate collectors. Development of high-performance dehumidifiers would increase the thermal and electrical coefficients of performance of the system and help lower initial system costs. First, the collector area needed to drive the system is reduced by almost one-half by doubling the thermal COP. Second, the efficiency of the system in using auxiliary energy is improved. Less natural gas (or other back-up fuel) is used when the system is operating in auxiliary mode, and less electricity is used to drive the fans for the system because of the low-pressure-drop dehumidifier designs now identified. Development of the low-cost thermal storage, typified by a plastic-lined sheet metal storage tank, is a capital cost reduction feature with no penalty in performance. The low-cost, flat-plate collector subsystem also reduces capital cost with little change in performance.

Further improvements in cost and performance are anticipated by technical improvements in the dehumidifier component to raise the thermal COP levels to 1.5-1.8. Improvements in heat and mass transfer and in desiccant materials will be investigated in experimental programs at the SERI Desiccant Cooling Test Facility to identify the paths to these major goals.

6.1.2 Open-Cycle Solid Desiccant Hybrid with Vapor Compression

Comparisons of the base case and advanced systems for the open-cycle solid desiccant hybrid system with a vapor compression chiller are shown in Table 6-2. The base case system used the IGT dehumidifier design in conjunction with performance data for present vapor-compression chillers. As shown in the Table 6-2, the combination of desiccant dehumidification for the latent load and vapor compression chilling for sensible load leads to a significant improvement in system performance when compared to either of the systems alone. The cost-effectiveness was still not satisfactory, at a cost-to-cost-goal ratio of 8.3.

As for the desiccant cooling concept, the advanced system used the higher efficiency desiccant dehumidifier, low-cost thermal storage, and low-cost collector subsystem to achieve improvements in cost and performance. In addition, the system is aided by the improvements identified for the vapor compression subsystem by SAI for the conventional HVAC systems. These technical improvements combine to bring the cost-to-cost-goal ratio down to 1.7, the closest of any of the solar technologies considered in the APRR project to reaching cost-competitiveness in the year 2000.

Table 6-2. **Open-Cycle Solid Desiccant Hybrid with Vapor Compression, Base Case vs. Advanced Commercial System**

	Base Case	Advanced
System Specifications		
Desiccant unit COP (kW/kW)	0.52	1.1
VC chiller COP (kW/kW)	1.4	1.54
Collectors used	Evacuated tube	Thin film
Collector area (m^2)	123	95
System Energy Use		
Electricity (MJ/yr)	147.2	80.5
Gas (MJ/yr)	0	0
Conventional Energy Use		
Electricity (MJ/yr)	50.1	166.1
Gas (MJ/yr)	217.9	0
Installed Cost		
Solar	$98,837	$53,107
Conventional	35,106	35,100
Cost/Cost Goal	8.3	1.7

6.2 RESEARCH RECOMMENDATIONS

As outlined in Section 6.1, the key components critical to the development of economic desiccant cooling concepts are (1) high-efficiency dehumidifiers, (2) low-cost, low temperature thermal storage, and (3) low-cost, mid-temperature collectors. In identifying the necessary research linked to these components it is clear that efforts aimed at further increasing the efficiency of the dehumidifier have the greatest potential for impacting the cost and efficiency of the systems as a whole. As part of the central core of both the desiccant cooling and hybrid systems, improvements in this component can reduce both the use of auxiliary energy for back-up and parasitics and the cost by cutting the collector area and storage capacity needed. Therefore, it is recommended that research on high-efficiency dehumidifiers and the associated heat and mass transfer and desiccant materials research areas be given the highest priority.

The development of the other components needed to reduce system costs should also be given consideration for the benefit that will accrue not only for desiccant systems but also for all solar heating and cooling systems. The APRR project identified these as key areas for further research on this basis [3].

SECTION 7.0

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