

ANALYSIS OF THE SEASONAL PERFORMANCE OF HYBRID LIQUID DESICCANT COOLING SYSTEMS

F. SICK, T. K. BUSHULTE, S. A. KLEIN, P. NORTHEY, and J. A. DUFFIE
Solar Energy Laboratory, University of Wisconsin-Madison, 1500 Johnson Drive,
Madison, WI 53706, U.S.A.

Abstract—A simulation model for the liquid desiccant component of a hybrid liquid desiccant cooling system was developed. Seasonal simulations were performed on different operational modes of a hybrid liquid desiccant cooling system including regeneration by solar energy. The seasonal thermal and electrical energy use and operational costs were compared to those of conventional cooling systems. For the system configuration investigated, the study shows that the cooling energy required by the conditioner exceeds the cooling requirements of the load. Although the conditioner cooling water is provided at a higher temperature (and therefore at a higher COP) than required in a conventional system, the additional requirement of regeneration energy cause the operating costs to be comparable to greater than those of a conventional chilled water cooling system. A preferred liquid desiccant configuration from an operating cost standpoint is proposed.

1. INTRODUCTION

Conventional air conditioning systems adjust both temperature and humidity to the desired values by passing outdoor air through cooling coils. The air is cooled below its dew point so that water vapor condenses to meet the humidity specifications. For a typical humidity ratio of 0.007 kg/kg, this occurs at 9° C, which is usually below the desired air temperature. Therefore, the air has to be reheated. Although this is generally done by using free waste heat or by mixing with return air, the cooling process itself requires more energy than a thermodynamically optimal process with a direct path from the outdoor air state to the set point. Hybrid desiccant cooling systems can follow this direct path more closely by splitting up the conditioning task into cooling (sensible load) and dehumidification (latent load).

Figure 1 shows the configuration of the liquid desiccant cooling subsystem investigated in this study. Precooled desiccant solution flows countercurrently to the air stream through the conditioner where it absorbs water vapor and cools the air to the desired set temperature. The water taken from the air goes into the liquid desiccant solution and is removed from the solution in the regenerator. The process in the regenerator is the reverse of that in the conditioner. Return air from the building absorbs water from the preheated solution, which becomes more concentrated and is pumped back to the conditioner. The humidified air is exhausted. The conditioner and the regenerator are connected by a heat exchanger (interchanger) that pre-cools the solution entering the conditioner while preheating the solution flow to the regenerator.

The combination of conventional and liquid desiccant air conditioning equipment forms a hybrid liquid desiccant cooling system. One such system that uses a LiCl-water solution as the desiccant is installed at the Science Museum in Richmond, VA (SMVA). The

energy inputs into a liquid desiccant system consist of cooling energy for the conditioner, heat for the regenerator, and possibly increased parasitic electrical demands. The cost savings potential for this type of liquid desiccant results in two ways. First, the cooling energy needed by the conditioner is supplied at a higher temperature (12.8° C in this study) and therefore higher efficiency than for a conventional chilled water system that provides dehumidification. Second, the regeneration energy is supplied as low temperature heat, approximately 60° C, and can be inexpensively provided in several ways. The system installed at the SMVA uses a conventional chiller to produce cold water for the conditioner. Both regeneration heat and electricity can be supplied by a gas-cogenerator. There is also a heat pump that delivers hot water and meets part of the cooling load simultaneously. A supplementary boiler produces additional heat if needed. Two 18,500-liter tanks are available for hot water storage. The installation of flat-plate solar collectors as hot water source has also been considered. These elements are the basis for the simulations conducted in this study. A design description of the SMVA system was presented by Meckler[1] and a steady state analysis has been developed by Buschulte[2].

2. THE LIQUID DESICCANT COMPONENT MODEL

Figure 1 shows the liquid desiccant subsystem. Known variables are inputs like weather data and hot and cold water source temperatures, equipment parameters (e.g., pump flow rates), and the conditioned air temperature set point, which is assumed to be 19.3° C. The variable load resulting from variable outside air conditions is met by modulating the flow of cold water through the cooler. The hot water flow rate on the regenerator side is constant when the regenerator is operating.

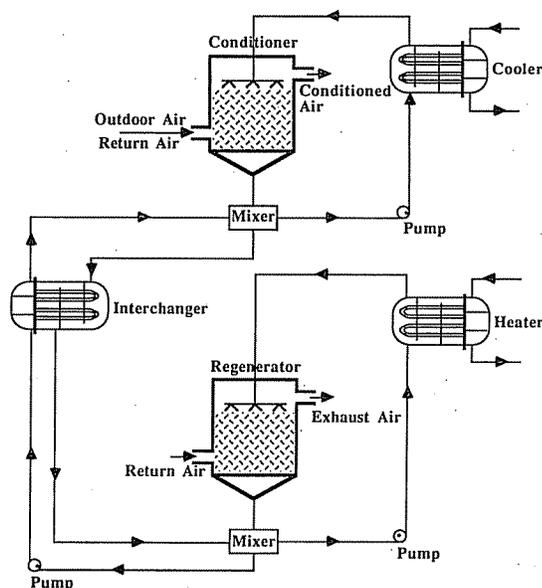


Fig. 1. Schematic of a liquid desiccant subsystem.

The conditioner and regenerator components are both two-phase packed contact devices with simultaneous heat and mass transfer and can be described by the same model. Several models of these components were formulated. The most detailed involved a one-dimensional finite difference approximation wherein the contact area was divided into as many as 200 elements[2]. The finite difference model is similar to that described by Factor and Grossman[3] and Peng and Howell[4]. In this analysis, it was assumed that the air and solution flows are countercurrent, operation is adiabatic, the solution is fully mixed within each element, and the vapor in immediate contact with the solution is in thermodynamic equilibrium with the solution. The heat and mass transfer within each element was assumed to be gas-phase controlled. Several heat transfer coefficient relationships were investigated; the mass transfer coefficient was calculated assuming a Lewis number of 0.87.

Because of the countercurrent flow, the finite difference model required an iterative solution. The computational effort needed to employ this model in a seasonal simulation of the liquid desiccant equipment was prohibitive. As a result, the K-factor model suggested by the equipment manufacturer[5] was employed. The basis of this model is:

1. The air humidity leaving the contact device is equal to the equilibrium humidity of the desiccant solution at its entering concentration and leaving air temperature.
2. The ratio of the difference in temperatures of the inlet streams to the enthalpy difference of the inlet and outlet air is a constant, called the K factor. The K factor is a function of flow rates. For the conditions at SMVA, the values are 0.040 and 0.019 kg-K/kJ for the conditioner and regenerator, respectively[5].

The finite difference and K-factor models were

compared for a range of solution inlet temperatures and flow rates. The K-factor model was found to compare well with the finite difference model for the expected range of operating conditions with the K-factor model requiring significantly less computational effort.

In the model, the mass of the desiccant in the sumps is assumed to be entirely in the regenerator sump. The conditioner sump is treated as a T-piece with no volume. The regenerator sump is assumed to be fully mixed, since the flow rates are high. The liquid desiccant component is controlled by the level (i.e., the amount of water) in the sump. The conditioner adds water to the system that must be removed by the regenerator. If the regenerator cannot keep up with the conditioner, the water level will rise and eventually the controls will turn the conditioner off until the regenerator lowers the sump level to a preset value. In this case, auxiliary conventional cooling has to be supplied to meet the load. On the other hand, if there is no load (i.e., the conditioner is off), the water level will drop and the regenerator will be turned off at a specified lower limit. For the simulation results which follow, the difference in the upper and lower sump level limits represents 500 kg of water.

The two exchange chambers, the conditioner sump are fully described by a set of 19 algebraic equations that include 6 mass balances, 6 energy balances, 3 log-mean temperature difference equations for the heat exchangers, and 4 equations to describe the heat and mass transfer in the conditioner and regenerator with the K-factor model. The regenerator sump is described by 2 ordinary coupled differential equations for the mass and energy balances. The differential equations are solved analytically at the beginning of each simulation time-step followed by solution of the remaining 19 algebraic equations. A detailed description of these equations can be found in[6].

Westerberg, Hutchison, Motard, and Winter[7] present an approach to find solving procedures for sets of linear and nonlinear algebraic equations. The algorithm of Sargent and Westerberg[8] rearranges the set of equations, if physically possible, into smaller blocks, each of which can be solved independently if done in the correct sequence. This procedure is called partitioning and precedence ordering. In this case, partitioning and precedence ordering reduced the problem of solving a single system of 19 equations and 19 unknowns to the simpler task of solving 9 systems ("partitions") of order 5 and smaller. The computer model for the liquid desiccant component is structured according to these partitions and was written to be compatible with TRNSYS[9]. It was found that a simple iteration method with successive substitution and Wegstein (modified secant method) acceleration is sufficiently fast to allow simulation of the cooling season performance with reasonable computing effort. The number of iteration steps is on the order of 10. A listing of the computer model and a description can be found in[6].

A package of physical property subroutines for air-water mixtures and LiCl-water solutions was written by Buschulte[2] for calculation of enthalpy, concentration, or temperature as function of the two other variables. These relationships were compiled from graphical data given in references [10–15] and were found to compare well with the correlations given in[16].

3. SYSTEM SIMULATIONS

System simulations were conducted for the period of April 1 through October 31 in Cape Hatteras, NC, and Sterling, VA. For these simulations, the load on the system was assumed to result from a constant flow of 3.4 kg/s of outdoor air 24 h per day through the conditioner. The set point at the conditioner outlet was fixed at 19.3°C; humidity level at the conditioner outlet was not directly controlled. Other parameter values assumed in the simulations are listed in Table 1. The equipment sizes used in the simulations were such that the liquid desiccant system had sufficient capacity to provide the entire load in both locations. The total load and the heating and cooling energy requirements of the liquid desiccant system to meet this load (with a 60°C regenerator inlet water temperature) for the April–October period are given in Fig. 2.

Three operational modes of the hybrid liquid desiccant system have been examined. In addition, a conventional system consisting of a chiller-cooling tower combination has been simulated in order to serve

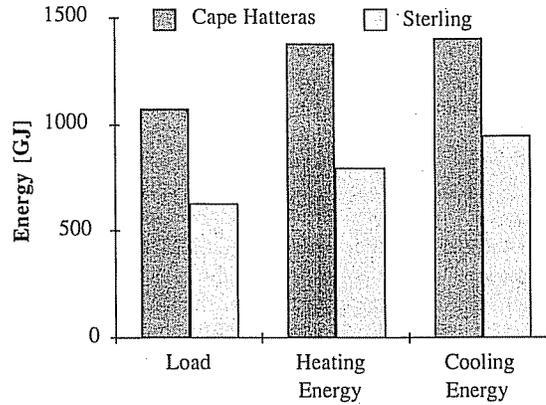


Fig. 2. Total load, regenerator heating energy (at 60°C) and conditioner cooling energy for April–October in Cape Hatteras, NC, and Sterling, VA.

as basis for the comparisons. This configuration is called the “conventional mode.” The three hybrid modes are defined by the following characteristics:

Chiller mode. A gas cogenerator provides heat for the regenerator and electricity to drive a chiller that supplies the cold water for the conditioner. A supplementary boiler provides additional heat if needed. Also, additional electricity may be purchased. Thus, it is guaranteed that the energy inputs required by the liquid desiccant system are provided at any instant of time. The energy demands of the liquid desiccant component in every simulation timestep are split up into the available energy sources and their corre-

Table 1. Simulation parameter values

Conditioner/Regenerator	
Conditioner heat exchanger overall energy transfer coefficient	22.7 kW/K
Regenerator heat exchanger overall energy transfer coefficient	11.2 kW/K
Interchanger heat exchanger overall energy transfer coefficient	11.2 kW/K
Conditioner K-factor value	0.0401 kg-K/kJ
Regenerator K-factor value	0.0191 kg-K/kJ
Solution flowrate into conditioner	8.2 kg/s
Solution flowrate into regenerator	8.3 kg/s
Solution flowrate between conditioner and regenerator	0.55 kg/s
Maximum water accumulation in sump	500 kg
Regenerator air inlet temperature	36.1°C
Regenerator air inlet humidity	0.0093 kg/kg
Regenerator air flowrate	2.8 kg/s
Regenerator hot water flow rate	4.3 kg/s
Conditioner air flow rate	3.4 kg/s
Conditioner water inlet temperature	12.8°C
Solar mode	
Solar collector intercept value $F_R(\tau\alpha)$	0.8
Solar collector overall energy loss coefficient $F_R U_L$	4.7 W/m ² ·°C
Solar storage tank volume	19 m ³
Solar collector slope	35°
Parasitics	
Building supply and return fans (both conventional and desiccant)	59.8 kW
Conditioner and regenerator fans	4.5 kW
Solution pumps in liquid desiccant system	4.5 kW
Pumps for heater and cooler	4.4 kW
Chilled water pumps (both conventional and desiccant)	13.4 kW
Cogenerator pump	1.5 kW

sponding costs for various cogenerator sizes. A zero-capacity cogenerator is equivalent to an energy supply solely by the boiler and the electrical power plant.

Heat pump mode. The chiller is replaced by a heat pump that produces hot water for the regenerator and cold water for the conditioner. In all other respects, the heat pump mode is treated like the chiller mode.

Solar mode. The chiller mode is modified such that the heat for the regenerator is partially supplied by flat-plate solar collectors via thermal storage. For a zero-capacity gas-cogenerator, the energy is supplied entirely by the collector, a boiler, and the power plant.

For each operation mode, a TRNSYS component was written to handle the control of the energy supplies. This component receives the system energy demands for cooling, heating, and electricity and distributes these loads to the gas and electricity supply according to the available equipment and to the desired control. Except for the solar mode, the model assumes that the loads can always be met through the supply of extra heat by a boiler and purchased electricity. Thus, no storage tank model needs to be included in chiller and heat pump modes, although storage may physically exist. Energy supplied to the system from the tank is assumed to be immediately replaced.

For all the simulations, the cogenerator was modeled to convert one-third of its gas input into useful heat and one-third into electricity. The last third is lost to the surroundings. The boiler loses one-third of its input capacity to the environment as well, while the remaining two-thirds provide useful heat.

The three operation modes were simulated varying the cogenerator capacity (over a range from 0 to 625 kW), the regeneration temperature from 60 to 70°C, and, for the solar mode, the collector area from 250 to 750 m². The influence of the electricity to gas price ratio was studied as well. Two estimates were made of the seasonal energy use costs of the conventional modes. In case (7), an air stream of 3.4 kg/s is dehumidified to a humidity ratio of 0.067 kg/kg with 8°C chilled water and then mixed with building return air. In case (b), an air stream of 11.2 kg/

s is dehumidified to a humidity ratio of 0.095 kg/kg with 13.3°C cooling water with no mixing. Case (1) compares more closely to the conditions provided by the liquid desiccant system.

4. RESULTS AND CONCLUSIONS

Tables 2 to 4 give the total gas and electricity energy use for the April–October period as a function of gas cogenerator capacity for the chiller, heat pump, and solar modes. In these tables, Gas Energy refers to the amount of purchased gas required by the cogenerator and boiler. Chiller Electricity refers to the electrical energy use by the vapor compression units that provide 12.8°C water for the conditioner. Parasitic Electricity is the electrical use of the pumps and blowers listed in Table 1.

Since both gas and electricity are required, the operational costs were compared assuming a gas price of \$0.03 \$/kW-hr and an electricity price of 0.07 \$/kW-hr. If excess electricity were produced by the gas cogenerator, it was assumed to be resold to the power plant at 0.04 \$/kW-hr. Figures 3 and 4 show the minimum average operational costs of all operation modes in Cape Hatteras, NC, and Sterling, VA, respectively. The following conclusions can be drawn from the simulation results:

1. The required total energy input into an air conditioning system increases using the liquid desiccant configuration investigated in this article. The operating cost of such a liquid desiccant system is nearly as high or higher than that of a conventional chiller system for electricity to gas price ratios of 7:3 and less. The larger this price ratio, the greater is the cost advantage of the liquid desiccant system over a conventional system. For example, if the electricity rate changes from 0.07 \$/kW-hr to 0.09 \$/kW · h in the heat pump mode for Cape Hatteras conditions, the minimum operational costs increase from 8.37 \$/h to 10.10 \$/h, while the costs for a conventional cooling system increase from 7.97 \$/h to 10.25 \$/h.
2. The chiller mode is an unattractive method of operation. Its minimum average hourly costs are even

Table 2. Simulation results for April–October in the chiller operations mode

Cogenerator capacity (kW)	Regen. water temp (°C)	Cape Hatteras, NC			Sterling, VA		
		Gas energy (GJ)	Chiller elec. (GJ)	Parasitic elec. (GJ)	Gas energy (GJ)	Chiller elec. (GJ)	Parasitic elec. (GJ)
0	60	2087	381	1251	1198	316	1159
70	60	2893	57	1208	2129	33	1077
140	60	3700	0	901	3063	0	746
280	60	5312	0	0	4928	0	149
560	60	8790	0	0	8773	0	0
0	70	2071	417	1238	1197	353	1153
70	70	2956	87	1205	2172	47	1092
140	70	3858	0	926	3151	0	775
280	70	5628	0	255	5107	0	160
560	70	9170	0	0	9012	0	0

Table 3. Simulation results for April–October in the heat pump operations mode

Cogenerator capacity (kW)	Regen. water temp (°C)	Cape Hatteras, NC			Sterling, VA		
		Gas energy (GJ)	Chiller elec. (GJ)	Parasitic elec. (GJ)	Gas energy (GJ)	Chiller elec. (GJ)	Parasitic elec. (GJ)
0	60	0	639	1251	0	476	1159
70	60	1107	258	1222	1096	149	1093
140	60	2198	14	1062	2193	3	848
280	60	4395	0	335	4386	0	211
560	60	8775	0	0	8773	0	0
0	70	0	765	1238	0	565	1153
70	70	1107	378	1213	1096	222	1103
140	70	2198	130	1055	2193	73	860
280	70	4395	0	438	4386	0	270
560	70	8775	0	0	8773	0	0

Table 4. Simulation results for April–October in the solar operations mode

Cogenerator capacity (kW)	Collector area (m ²)	Cape Hatteras, NC			Sterling, VA		
		Gas energy (GJ)	Chiller elec. (GJ)	Parasitic elec. (GJ)	Gas energy (GJ)	Chiller elec. (GJ)	Parasitic elec. (GJ)
0	250	1427	380	1250	665	316	1159
70	250	1977	57	1208	1216	33	1077
140	250	2527	0	901	2198	0	747
280	250	4396	0	235	4396	0	150
560	250	8792	0	0	8792	0	0
0	500	761	380	1250	120	316	1159
70	500	1312	57	1208	1099	33	1077
140	500	2198	0	901	2198	0	747
280	500	4396	0	235	4396	0	150
560	500	8792	0	0	8792	0	0
0	750	95	380	1250	0	316	1159
70	750	1099	57	1208	1099	33	1077
140	750	2198	0	901	2198	0	747
280	750	4396	0	235	4396	0	150
560	750	8792	0	0	8792	0	0

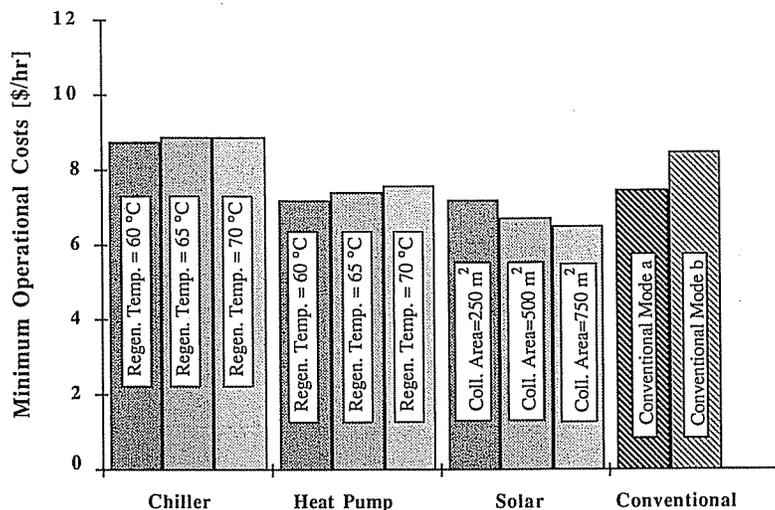


Fig. 3. Simulation results for Sterling, VA.

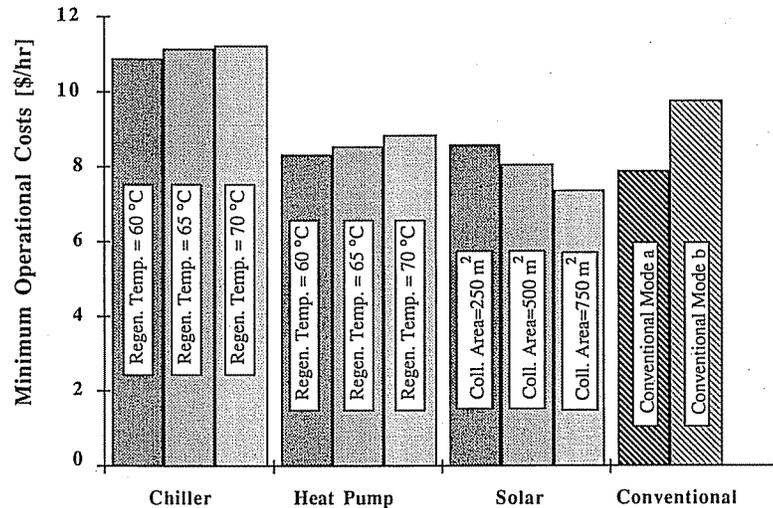


Fig. 4. Simulation results for Cape Hatteras, NC.

higher than those of inefficiently run conventional cooling systems. The supply of the required energy demands completely from purchased sources without use of "free" heat is the reason for the poor performance. The improved thermodynamic process is negated by these additional energy needs.

3. The heat pump mode can be comparable to conventional cooling systems in the cost of operation, because regeneration heat and part of the cooling energy is supplied efficiently. For the conditions investigated in this study, the cost ratio of the optimum heat pump mode to the comparable conventional mode is 0.97 for Sterling and 1.05 for Cape Hatteras.
4. If first costs are not considered, lowest operational costs are obtained by the solar mode where flat-plate collectors contribute to the regeneration heat. Still, the estimated operational costs are close to those of a conventional system. The cost ratio of the solar mode to the conventional mode for 500 m² collector area is, in this study, 0.90 for Sterling and 1.02 for Cape Hatteras conditions.
5. The optimum cogenerator capacity is dependent on economic as well as load parameters. High electricity to gas price ratios and high latent loads fractions (compared to the sensible loads) favor the installation of a cogenerator. The operational costs are quite dependent on the cogenerator size, especially in the Solar Mode.
6. Relatively low regeneration temperatures of 60° C or less result in lower operating costs. Although increasing the regeneration temperature reduces the number of hours the regenerator needs to operate, the higher hot water temperature heats the desiccant in the entire system. Consequently, more solution cooling is required to obtain the conditioned air set temperature.

The results obtained in the investigation of this liquid desiccant system suggest that other system

configurations should be studied. Promising alternatives to the presented configuration are configurations analogous to those used in solid desiccant air conditioning systems[17] in which the liquid desiccant system would operate by over-drying the air using a hot-water temperature that is optimal for the regeneration. The dehumidified and heated air can then be blown through heat exchangers where it is cooled by outside and/or building return air to a temperature close to the initial air state. Direct or indirect evaporative coolers would allow the air to be cooled to the desired set point. The major advantage of this system is that it eliminates the need to supply mechanical cooling. The use of evaporative coolers with a hybrid liquid desiccant system has been proposed in the preliminary analytical study of Howell and Peterson[18].

REFERENCES

1. G. Meckler, Designing energy integrated HVAC systems based on thermodynamic efficiency (Science Museum, Richmond, VA). *Proc. Designing and Managing Energy Conscious Commercial Buildings Workshop*, Denver, CO (1982).
2. T. K. Buschulte, Analysis of hybrid liquid desiccant cooling systems. M.S. Thesis in Chemical Engineering, University of Wisconsin—Madison (1984).
3. H. M. Factor and G. Grossman, A packed bed dehumidifier/regenerator for solar air conditioning with liquid desiccants. *Solar Energy*, **24**, 541–550 (1980).
4. C. S. P. Peng and J. R. Howell, The performance of various types of regenerators for liquid desiccants. ASME Paper 81-WA/Sol-31 (1981).
5. G. Meckler, Data collection and model development of liquid desiccant integrated HVAC system. Science Museum of Virginia, Richmond, VA (1984, 1985).
6. F. Sick, Analysis of the seasonal performance of hybrid liquid desiccant cooling systems. M.S. Thesis in Chemical Engineering, University of Wisconsin—Madison, (1986).
7. A. W. Westerberg, H. P. Hutchison, R. L. Motard and P. Winter, *Process Flowsheeting*, Cambridge University Press, New York (1979).

8. R. W. H. Sargent and A. W. Westerberg, "SPEED-UP" in chemical engineering design. *Trans. Inst. Chem. Eng. (London)*, **42**, 190-197 (1964).
9. S. A. Klein, et al., TRNSYS—A transient simulation program. Version 12.1, Solar Energy Laboratory, University of Wisconsin—Madison (1983).
10. *ASHRAE Handbook, 1981 Fundamentals*. Chapter 5 (Psychrometrics) and 17 (Refrigerant Tables and Charts). American Society of Heating, Refrigeration and Air Conditioning Engineers, Atlanta, GA (1981).
11. E. F. Johnson and M. C. Molstad, Thermodynamic properties of aqueous lithium chloride solutions. *J. Phys. Colloid Chem.*, **44**, 257-281 (1951).
12. *Handbook of Chemistry and Physics*, 61st edition, Chemical Rubber Company (1980).
13. *Gmehlin's Handbuch der anorganischen Chemie*, Vol. Lithium, Verlag Chemie (1960).
14. Landolt-Bornstein: *Zahlenwerte und Funktionen aus Physik, Chemie, . . .*, New Series, Vols. IV/2 and II/4, Springer Verlag, New York (19xx).
15. T. Uemura, Studies on the lithium chloride—water absorption refrigerating machine. *Technology Reports of the Kansai University*, Osaka, Japan, Vol. 9, pp. 71-88 (1967).
16. I. L. Maclaine-cross, A theory of combined heat and mass transfer in regenerators. Ph.D. Thesis, Department of Mechanical Engineering, Monash University, Clayton, Victoria, Australia (1974).
17. D. R. Crum, Open cycle desiccant air conditioning systems. M.S. Thesis in Mechanical Engineering, University of Wisconsin—Madison (1986).
18. J. R. Howell and J. L. Peterson, Preliminary performance evaluation of a hybrid vapor-compression/liquid desiccant air conditioning system. paper 86-WA/SOL-9, ASME Winter Annual Meeting, Anaheim, California (December 7-12, 1986).

