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Open-Cycle Desiccant Air Conditioning as an Alternative to Vapor Compression Cooling in Residential Applications

The performance of open-cycle desiccant air conditioners for residential applications is evaluated. The performance of these systems is compared to that of vapor compression air conditioners on the basis of primary energy use and cost. Systems with improved dehumidifiers can achieve seasonal COP's on the order of 1.1. These systems, when coupled with a solar energy system to supply regeneration energy, are significantly better than conventional air conditioners on a primary energy basis, but are not presently cost-competitive.

Introduction

Desiccant air conditioners have been under active study and development since the mid 1960s. Prototypes of these systems have been built and tested [1-3]. Design-point test data are available for the systems developed by the Institute of Gas Technology and the AiResearch Manufacturing Company [4-6]. Despite this extensive development effort, the performance and utility of these systems have not been fully evaluated. The performance varies significantly as the operating conditions deviate from the design point, and either long-term tests or computer simulations must be used to evaluate the seasonal performance. Site testing of systems is an expensive and lengthly process. Computer simulations based on test data can be used to rapidly investigate seasonal performance.

Simulation studies of the IGT and AiResearch prototype systems indicate the performance of state-of-the-art systems, although the IGT study is limited by the lack of information concerning the system parameters and the modeling techniques [1, 2]. Parametric studies of desiccant cooling system performance to date have considerd a single system [7], single climate [8], or have used very simple load and system models [9]. These parametric studies have only considered systems with balanced air flow through a silica gel dehumidifier.

The performance of residential desiccant air conditioners using adiabatic, silica gel dehumidifiers is evaluated in this paper. Computer simulations of system operation in three U.S. locations (Miami, Fla., Washington, D.C., and Fort Worth, Texas) ranging in climates from warm and humid to hot and dry were performed. Both the ventilation cycle (Fig. 1) and the recirculation cycle (Fig. 2) are modeled using component parameters representative of those for state of the art as well as highly developed advanced generation systems. The performance of desiccant systems using gas, solar/gas, and solar energy for regeneration is compared to a current vapor compression air conditioner on both energy use and economic bases.

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Ventilation Cycle Desiccant Cooling System

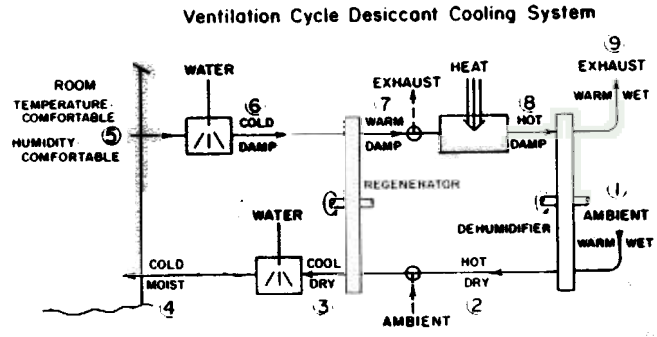

Fig. 1 Ventilation cycle desiccant air conditioner

Recirculation Cycle Desiccant Cooling System

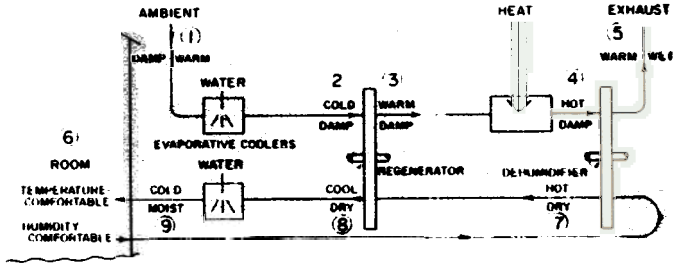

Fig. 2 Recirculation cycle desiccant air conditioner

Energy and Economic Criteria

To compare the total energy consumption of desiccant

Table 1 IGT Solar MEC III Design-Point Test Data

System	Ventilation	Recirculation
IGT Test No.	54	28
Room state (°C,g/kg)	(26.5,11.0)	(26.7,11.0)
Amb. state (°C,g/kg)	(35.0,14.0)	(35.2,13.9)
$\dot{m}_s$ (kg/sec)	0.45	0.45
Reg. temp. (°C)	95.39	103.0
COP_T (kW)	7.47	9.52
$\dot{E}_p$(kW)	1.16	1.18
COP_t	0.51	0.46
COP_e	6.45	8.06

cooling systems, the electrical and thermal energy use must be expressed on a common basis. The sum of the fossil fuel equivalent of the electrical energy use and corresponding equivalent of the thermal energy input is referred to as the primary energy use, which is defined as

$$P = \frac{Q_a}{\eta_{c,f}} + \frac{E_p}{\eta_{c,e}} \tag{1}$$

In this study, $\eta_{c,f}$ and $\eta_{c,e}$ were taken to be 1.0 and 0.29, respectively. In systems where solar energy is used to displace thermal energy provided by a fossil fuel, Q_a represents the purchased thermal energy input.

Coefficients of performance COP_a, $COP_{a,o}$, and COP_e are defined by the ratio of cooling output to thermal energy (Q_a), fossil fuel equivalent of thermal energy ($Q_a/\eta_{c,f}$), and electrical energy (E_p) inputs. For the desiccant cycle to use less resource energy than a conventional vapor compression machine with a coefficient of performance $COP_{e,vc}$, then [13]

$$COP_{a,o} = \eta_{c,f}\, COP_a > \left(\frac{1}{\eta_{c,e}}\left[\frac{1}{COP_{e,vc}} - \frac{1}{COP_e}\right]\right)^{-1} \tag{2}$$

Figure 3 illustrates this relationship using $COP_{e,vc}$ as a parameter. The curves represent the locus of points at which the primary energy use of the desiccant system just equals that of a vapor compression machine with a given value of $COP_{e,vc}$. A point that lies below or to the left of a curve for constant $COP_{e,vc}$ indicates a primary energy use greater than that of a vapor compression machine with $COP_{e,vc}$.

Operating energy cost instead of operating energy consumption can also be compared using equation (2) if $(1/\eta_{c,e})$ is replaced by ($\$e/\$f$), the ratio of the cost of electricity to the cost of auxiliary fuel. Figure 4 illustrates this application of equation (2) with ($\$e/\$f$) as a parameter for $COP_{e,vc}$ = 2.5. A point below or to the left of a given curve indicates a combined energy cost greater than that of a conventional vapor compression machine with $COP_{e,vc}$ = 2.5.

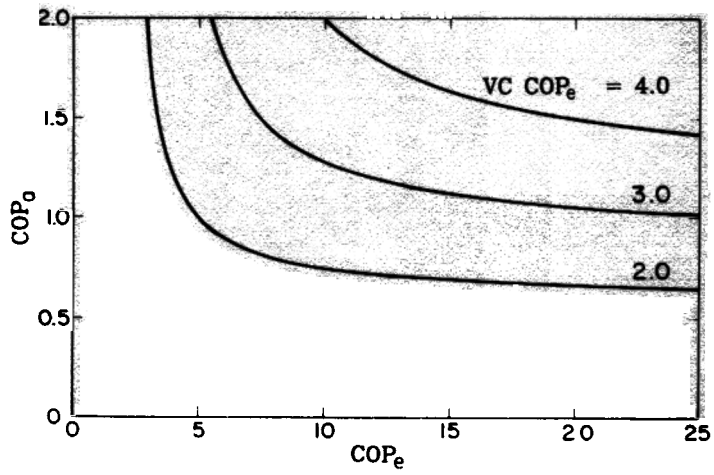

Fig. 3 Auxiliary COP for a desiccant air conditioner to use the same amount of primary resource energy as a vapor-compression air conditioner with $COP_{e,vc}$ of 2.0, 3.0, and 4.0

Nomenclature

CAP = cooling capacity
COP = coefficient of performance, ratio of cooling output to an energy input
COP_a = COP based on purchased thermal energy input; Q_c/Q_a
$COP_{a,o}$ = COP based on heating value of purchased fuel; $Q_c/(Q_a/\eta_{cf})$
COP_e = COP based on electrical energy input; Q_c/E_p
$COP_{e,s}$ = COP based on electrical energy use of both the desiccant cooling system and the solar energy system
$COP_{e,vc}$ = COP of an electrically driven vapor compression air conditioner
COP_t = COP based on total (purchased plus solar) thermal energy input
E_p = electrical energy input
P = primary resource energy
$\dot{m}_s$ = mass flow rate supplied to the cooling load
$\dot{m}_r$ = mass flow rate through the auxiliary furnace and regenerating period of the dehumidifier
Q_a = net thermal energy from a fossil fuel delivered to the desiccant cooling system
Q_c = cooling energy required
t_{reg} = dehumidifier regeneration temperature
$\$e/\$f$ = ratio of electrical energy cost to thermal energy cost in consistent units
$\eta_{c,e}$ = conversion efficiency of electricity from fuel at the point of generation to electricity at the point of use; assumed to be 0.29
$\eta_{c,f}$ = conversion efficiency from fuel to thermal energy, assumed to be unity
Λ_w = dimensionless mass transfer length of the dehumidifier [10, 12, 13]; $4h_wL/(\rho d_h v)$ where h_w is the mass transfer coefficient, ρ is the dry fluid density, L is the axial flow length, d_h is the matrix hydraulic diameter, and v is the air velocity.

Subscripts

a = auxiliary, or purchased thermal energy
b = balanced dehumidifier flows
e = electric
u = unbalanced dehumidifier flows
∞ = steady state

Superscripts

$\cdot$ = rate

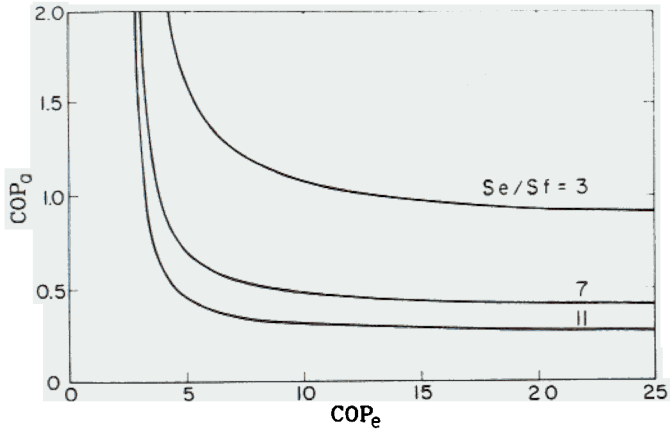

Fig. 4 Auxiliary *COP* for a desiccant air conditioner to have the same operating cost as a vapor-compression air conditioner with *COP* = 2.5 for energy cost ratios *Se/Sf* of 3, 7, and 11

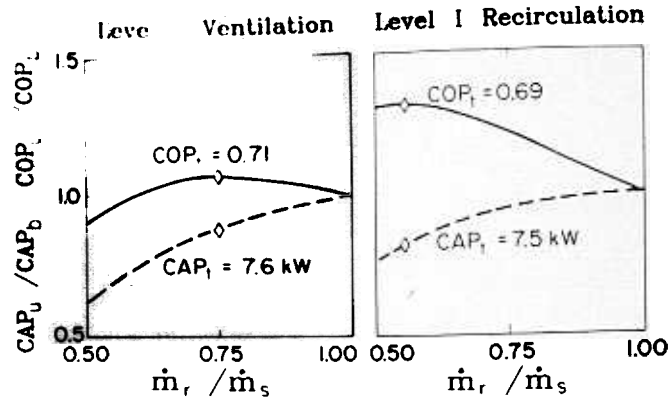

Fig. 5 Effect of dehumidifier flow unbalance on the cooling capacity and COP_t of the Level I ventilation and recirculation cycles at the standard design point

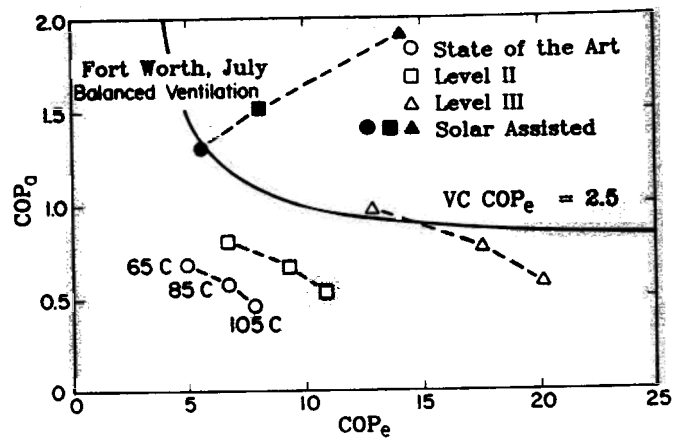

Fig. 6 Balanced flow ventilation cycle performance, Forth Worth, July

System Models

A single zone with thermal and moisture capacitance is used to model the transient cooling load in a two-story residence with a 5.67 kW (1.6 ton) Miami design cooling load [13]. The response of the desiccant system to the instantaneous cooling load is governed by a controller that senses both room temperature and relative humidity, and outputs two control functions to indicate whether the room requires sensible cooling. dehumidification, or both. The controller set points were chosen to closely correspond to the ASHRAE comfort zone [13], and the air conditioner is turned on if either the sensible or latent cooling functions output by the room controller are on.

The desiccant cooling system components are described using quasisteady-state models [13]. The analogy method [10, 12, 13] is used to model the silica gel dehumidifier. The supply air flow rate to the house is constant, though the air flow through the dehumidifier may be unbalanced. In addition to the external on/off control provided by the room state sensor, the desiccant air conditioner has an internal control strategy to maximize cooling capacity while minimizing auxiliary thermal energy input. The operating hierarchy followed is regenerative evaporative cooling, solar-regenerated desiccant cooling, solar and thermal auxiliary regenerated desiccant cooling, and finally, auxiliary-regenerated desiccant cooling. The output of the cooling machine is modulated to match the sensible and latent components of the total cooling load by varying the saturation effectiveness of the evaporative cooler at the load inlet. Neither bypass mixing of the room exhaust air with supply air nor derating of the sensible heat exchanger performance was employed [9].

The conventional vapor compression air conditioner is modeled using the steady-state performance data for a commercially available unit [14, 15], and has a 10.6 kw (3 ton) total cooling capacity and an overall electric *COP* of 2.34 at standard design conditions. The nominal capacity is about twice the Miami design cooling load, so the simulated unit is somewhat oversized. However, the sensible capacity and the power consumption scale with total cooling capacity, so that using this oversized unit in simulations does not affect estimates of the average $COP_{e,vc}$ [15].

Desiccant System Parameters

Three sets of desiccant component parameters were chosen to reflect current prototypes and two levels of improved desiccant air conditioners. The IGT Solar-MEC III ventilation and recirculation cycles are used to provide a baseline for current systems performance. Design-point test data for prototypes of the IGT ventilation and recirculation cycles from [1, 4, 11] are given in Table 1. The ventilation cycle has a cooling capacity of 7.5 kW (2.1 tons) at a regeneration temperature of 95.4°C (204°F) and the recirculation cycle has a capacity of 9.5 kW (2.7 tons) at a regeneration temperature of 103°C (217°F). Both systems have a nominal supply air flow rate of 0.45 kg/sec. The ventilation cycle has COP_t of 0.51 at the design point, while that of the recirculation cycle is 0.46. The total electric power draw in the ventilation cycle is 1.16 kW, and 1.18 kW in the recirculation cycle, resulting in COP_e values of 6.5 and 8.1, respectively.

The description of the Solar-MEC III system [1] indicates that the nominal sensible heat exchanger effectiveness is about 0.90. The flow through the sensible regenerator is unbalanced to increase the cooling capacity of the machine, though the level of flow unbalance is not specified. The gas burner is designed to introduce a nonuniform temperature in the regenerating stream. This has only a slight effect on dehumidifier performance [12]. System schematics indicate that the dehumidifier process outlet stream is purged, and the flow recirculated to the regenerating period, though the details are not discussed. This does not have a large effect on system performance, though the recirculation cycle benefits more from purging than does the ventilation cycle [13]. There is no indication in [1] that the flow through the dehumidifier is unbalanced in the Solar-MEC III system.

The state-of-the-art system parameters used in this study, denoted Level I, were obtained by roughly matching the balanced flow model predictions to the design-point data for the IGT recirculation cycle. The selected parameters result in a reasonable representation of the recirculation cycle per-

Table 2 Simulation results for the vapor-compression air conditioner

Location	Cooling Load (gJ)	Sensible Fraction	Electricity (gJ)	$COP_{e,uc}$
July				
Miami	8.88	0.67	3.54	2.57
Fort Worth	10.40	0.77	4.49	2.32
Washington, D.C	6.79	0.68	2.61	2.60
May–October				
Miami	49.64	0.67	19.15	2.59
Fort Worth	44.06	0.73	17.86	2.47
Washington, D.C.	36.24	0.66	9.87	2.66

Table 3 Balanced flow ventilation cycle, Forth Worth, July

	t_{reg}(°C)	COP_l	COP_a	COP_e	$COP_{e,s}$
Gas-Fired					
Level I	65	0.680		4.90	
	85	0.572		6.69	
	105	0.458		7.82	
Level II	65	0.800		6.67	
	85	0.667		9.32	
	105	0.527		10.87	
Level III	65	0.968		12.90	
	85	0.767		17.67	
	105	0.577		20.33	
Solar-Assisted					
Level I	s/g 85	0.594	1.315	6.08	5.57
Level II	s/g 85	0.673	1.528	9.13	8.09
Level III	s/g 85	0.762	1.922	17.61	14.24
Solar-Fired					
Level I	s 68	0.918	∞	5.93	5.34
Level II	s 70	0.951	∞	8.44	7.34
Level III	s 74	1.028	∞	16.65	12.94

Table 4 Balanced flow recirculation cycle, Forth Worth, July

	t_{reg}(°C)	COP_l	COP_a	COP_e	$COP_{e,s}$
Gas-Fired					
Level I	65	0.492		5.08	
	85	0.411		6.66	
	105	0.313		7.30	
Level II	65	0.533		6.83	
	85	0.440		8.96	
	105	0.334		9.80	
Level III	65	0.566		12.72	
	85	0.454		16.32	
	105	0.338		17.64	
Solar-Assisted					
Level I	s/g 85	0.444	0.930	5.19	4.79
Level II	s/g 85	0.480	1.060	7.45	6.67
Level III	s/g 85	0.570	1.178	13.98	11.49
Solar-Fired					
Level I	s 54	0.739	∞	4.71	4.30
Level II	s 55	0.761	∞	6.36	5.64
Level III	s 55	0.774	∞	11.53	9.40

formance, but lead to an overprediction of the performance of the ventilation cycle. Since the Munters dehumdifier used in the Solar-MEC III is composed of molecular sieve on a fiberglass core, and details of the system flow are not known, it is not surprising that a single set of parameters for a balanced flow silica gel dehumidifier system is unable to simultaneously match the performance of the IGT ventilation and recirculation cycles. However, the known specifications of the IGT systems, (e.g., heat exchanger effectiveness and total electric power consumption) are accurately reproduced.

The Level II system performance modeled in this study mirrors the change in the state-of-the-art system proposed by IGT [1]. The sensible heat exchanger effectiveness is increased to about 0.95 and the overall electric power consumption reduced through better flow passage design. The proportional increases in the total cooling capacity that result from the increase in heat exchanger effectiveness are similar to the IGT and the model predictions. However, the model system does not show the improvement in *COP*, indicated by IGT [1]. It is unlikely that this large discrepancy is due to differences in the dehumidifier characteristics alone. Rather, it would seem to indicate that there has been additional, unspecified alterations in the IGT system design or operating strategy.

A third level of system performance, Level III, is also modeled. The dehumidifier performance is greatly improved over the other two cases, and the overall electric power consumption of the system is dramatically reduced. The Level III parameters are representative of carefully designed parallel passage rotary components and highly efficient drive equipment.

The thermal energy input to the desiccant cooling cycles can be decreased if part of the regenerating stream flow is

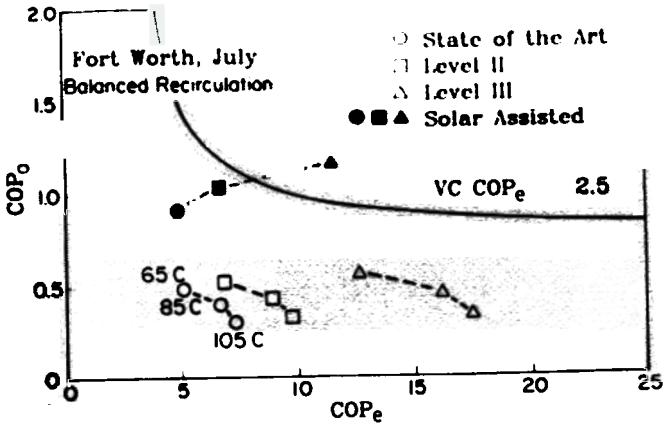

Fig. 7 Balanced flow recirculation cycle performance, Fort Worth, July

bypassed around the auxiliary furnace and the dehumidifier, unbalancing the dehumidifier flows. The effect of bypassing part of the regenerating flow on the design point cooling capacity and COP_t of the Level I ventilation and recirculation cycles is shown in Fig. 5. For the Level I ventilation cycle with a regenerating flow that is 70 percent of the process stream flow rate, the COP_t is about 9 percent greater than that for the balanced flow system, while the cooling capacity is 13 percent less. The maximum COP_t for the unbalanced flow recirculation cycle, which occurs at a flow rate ratio of about 0.60, is 33 percent greater than for balanced flow while the capacity drops 12 percent. The optimum unbalanced flow Level I ventilation and recirculation cycles have roughly equivalent capacity and COP_t.

The optimum flow rate ratio and the effect of flow unbalance on system performance depends both on the operating conditions and the level of system performance [13]. Generally the recirculation cycle benefits considerably more from unbalanced flow operation than the ventilation cycle. However, the maximum design point COP_t of the unbalanced Level III recirculation cycle is about 15 percent less than the corresponding maximum for the unbalanced ventilation cycle.

The TRNSYS [16] simulations of unbalanced flow operation are based on dehumidifier flow rate ratios of 0.80 for the ventilation system and 0.60 for the recirculation system. At these flow rate ratios, the Level I ventilation and recirculation cycles have considerably greater capacity and COP_t than do the AiResearch prototype [5, 6, 13], which bypass about 40 percent of the regenerating flow around the dehumidifier. Therefore, the lowest level of system performance modeled in this study can be considered to be an upper bound on that of state-of-art systems.

Results and Discussion

Desiccant cooling systems must perform better or cost less than conventional vapor compression air conditioners before they will gain commercial acceptance. Current generation vapor compression air conditioners thus provide a benchmark with which to evaluate desiccant cooling system performance. The simulation results for conventional vapor compression air conditioner operation during the month of July and the May–October cooling season are given in Table 2. The COP_e for the vapor compression machines range from a low of 2.3 in Fort Worth to a high of 2.6 in Washington, D.C. For the purposes of discussion, the average seasonal $COP_{e,vc}$ will be taken to be 2.5.

The performance of desiccant cooling systems for the balanced flow systems will be summarized using Fort Worth results. The July results for the ventilation and recirculation cycles are given in Tables 3 and 4 and illustrated in Figs. 6 and 7. For gas-fired operation, all three levels of systems are able to maintain room comfort at least 90 percent of the time. Regeneration temperatures were varied from 65°C–105°C. Below 65°C, the COP_t increases but the capacity drops very rapidly. The COP_t is greatest at the 65°C regeneration temperature, though the COP_e are greatest at the 105°C regeneration temperature. The variation in COP_t implies that the average thermal energy input increases more rapidly with increasing regeneration temperature than does the cooling capacity. The increase in COP_e with increasing regeneration temperature reflects the increases in cooling capacity at constant electric power draw.

The balanced flow ventilation cycle enjoys a considerable performance advantage over the recirculation cycle. The COP_t for the Level I ventilation cycle are 40 percent greater than those for the Level I recirculation cycle, though the average cooling capacities of the two systems are roughly equal. The ventilation cycle also benefits more from improved component performance than does the recirculation cycle. The thermal COP of the Level III ventilation cycle with an 85°C regeneration temperature is 35 percent greater than that of the corresponding Level I system, while the comparable increase in recirculation cycle COP_t is about 10 percent. The values of COP_t for the Level III ventilation cycle are about 70 percent greater than the corresponding values for the recirculation cycle.

Figures 6 and 7 show that only the Level III ventilation cycle system with a 65°C regeneration temperature has a lower total primary energy consumption than the vapor compression air conditioner. However, comparison with Fig. 4 shows that the Level I ventilation cycle becomes cost-competitive with the conventional vapor compression unit at an electrical to thermal energy cost ratio ($\$e/\$f$) of 7.

The primary energy consumption is considerably reduced by addition of a 40 m^2 collector array in series with an auxiliary furnace with an 85°C output temperature. The solar system provides at least 50 percent of the thermal energy required by desiccant cycles, and thus the COP_a of the solar-assisted systems are at least twice the values of COP_t. Because the solar-assisted systems can operate at a regeneration temperature that is less than the auxiliary temperature if the resultant cooling capacity is sufficient to meet the instantaneous cooling load, the average cooling capacity is less than for the gas-fired systems with a fixed regeneration temperature. This results in a decrease in the desiccant system COP_e and an increase in the operating time compared to the gas-fired systems. In addition, the electrical energy requirements of the solar system must be charged to the overall electricity consumption of the desiccant system, so that the $COP_{e,s}$ of the solar assisted desiccant systems are only about 80 percent of the COP_e for the gas-fired systems. Nonetheless, as shown in Figs. 6 and 7, the solar-assisted systems use less primary energy and have lower operating costs than the vapor compression air conditioner.

If the solar heating system with 40 m^2 of collector is the only source of the thermal energy for the cycle, the desiccant air conditioners cannot maintain room comfort. For the Fort Worth location, the room is uncomfortable at least 60 percent of the time with solar-fired operation. The minimum collector area required for 90 percent comfort in Fort Worth is estimated to be at least 60 m^2 for both the Level III ventilation and recirculation cycles. The $COP_{e,s}$ for exclusively solar-fired balanced Level III ventilation and recirculations systems with 40 m^2 of collector area are 12.9 and 9.4. Assuming that $COP_{e,s}$ is insensitive to increasing collector area, then on the basis of operating cost and primary energy use, an exclusively solar-fired Level III desiccant system with sufficient capacity to meet the entire cooling load is equivalent to an electric

Table 5 Unbalanced flow ventilation cycle, Forth Worth, July

	t_{reg}(°C)	COP_t	COP_a	COP_e	$COP_{e,s}$
Gas-Fired					
Level I	65	0.742		4.82	
	85	0.642		6.53	
	105	0.528		7.87	
Level II	65	0.865		6.62	
	85	0.732		9.17	
	105	0.601		10.85	
Level III	65	1.013		12.57	
	85	0.874		17.76	
	105	0.692		20.83	
Solar/Gas					
Level I	s/g 85	0.658	2.050	5.58	5.16
Level II	s/g 85	0.758	2.546	8.34	7.46
Level III	s/g 85	0.861	2.650	17.89	14.57
Solar-Fired					
Level I	s 63	0.896	∞	7.77	6.89
Level II	s 68	0.826	∞	5.15	4.72
Level III	s 75	1.015	∞	16.81	13.31

Table 6 Unbalanced flow recirculation cycle, Forth Worth, July

	t_{reg}(°C)	COP_t	COP_a	COP_e	$COP_{e,s}$
Gas-Fired					
Level I	65	0.681		4.96	
	85	0.604		6.63	
	105	0.496		7.68	
Level II	65	0.731		6.60	
	85	0.647		8.98	
	105	0.527		10.24	
Level III	65	0.777		12.17	
	85	0.714		17.09	
	105	0.561		19.06	
Solar/Gas					
Level I	s/g 85	0.625	2.287	5.34	4.95
Level II	s/g 85	0.684	3.164	7.11	6.44
Level III	s/g 85	0.756	4.146	13.84	11.54
Solar-Fired					
Level I	s 52	0.723	∞	3.61	3.40
Level II	s 54	0.751	∞	4.94	4.56
Level III	s 57	0.784	∞	9.91	8.55

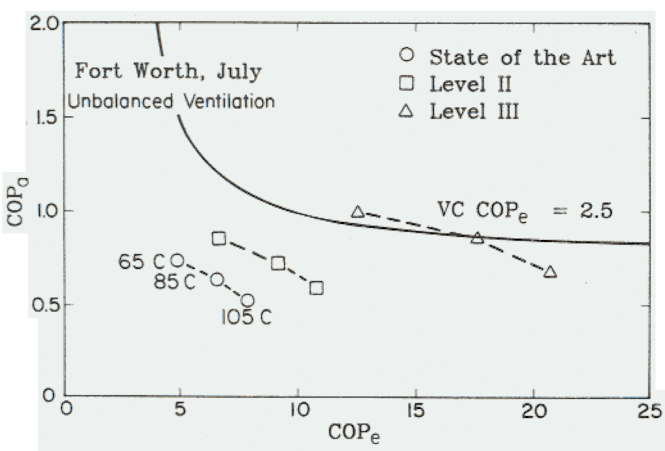

Fig. 8 Unbalanced flow ventilation cycle, Forth Worth, July

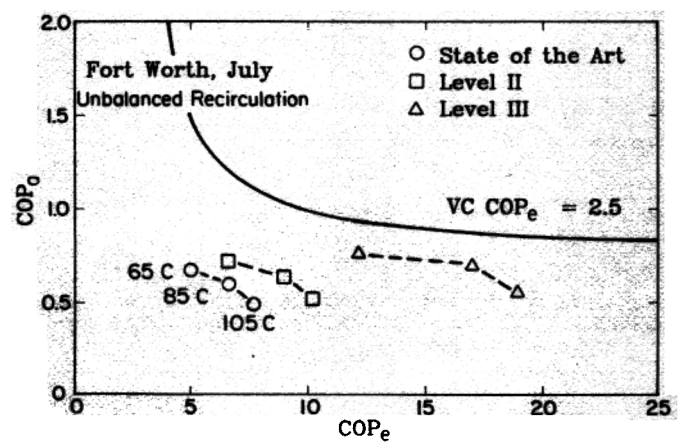

Fig. 9 Unbalanced flow recirculation cycle, Forth Worth, July

vapor compression air conditioner with a COP_e in the range of 9–13.

These results for Fort Worth are generally representative of the system performance in Miami and Washington, D.C. The COP_t for the balanced flow recirculation system are significantly less than those of the ventilation cycle, and the difference between the COP_t of the two systems increases as the level of performance of the system improves. Solar-assisted operation increases COP_a by a factor of at least two, but results in a decrease in the $COP_{e,s}$ of the system. Exclusively solar-fired operation with a 40 m^2 collector array is unable to maintain 90 percent room comfort in any of the locations considered, with values ranging from 10 percent comfort time in Miami to 60 percent comfort time in Washington, D.C. In terms of primary energy use, Level III solar-fired desiccant cooling systems with sufficient collector area to meet all of the cooling load are equivalent to electric air conditioners with COP_e of ~ 8 in Miami and ~ 11 in Washington, D.C.

The improvement in system performance with unbalanced

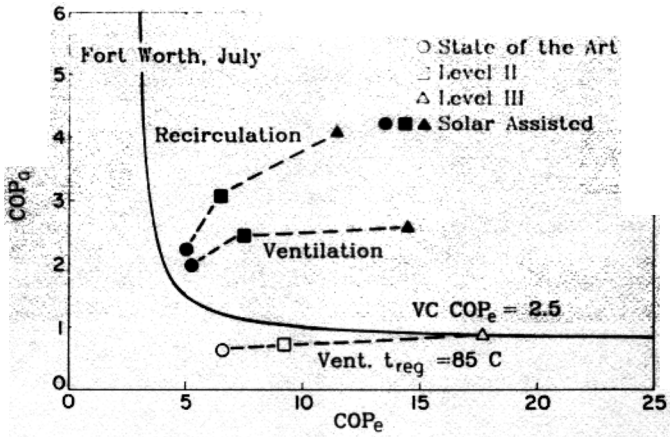

Fig. 10 Solar-assisted unbalanced flow desiccant cooling cycles, Forth Worth, July

flow operation is shown in Tables 5 and 6 and Figs. 8 and 9. The cooling capacity is adversely affected by unbalanced flow operation at low regeneration temperatures. However, for the 85°C and 105°C regeneration temperatures, the average cooling capacities of the unbalanced flow systems are nearly equal to those of the balanced flow systems. The COP_t are generally increased by the unbalanced flow operations. For the Fort Worth data shown, the COP_t of the ventilation cycle are increased about 12 percent by unbalanced flow operation, and the recirculation cycle COP_t are increased by about 50 percent. Unbalanced flow operation in Miami and Washington, D.C. results in similar increases in the COP_t of the desiccant cooling systems, except for the unbalanced flow ventilation cycle in Miami, which shows a slight decrease in COP_t. The unbalanced flow systems show a less pronounced decrease in COP_t with increasing regeneration temperatures than do the balanced flow systems.

The unbalanced flow Level I recirculation cycle performs at least as well as the Level I ventilation cycle in all three locations. However, as the performance level improves, the gas-fired ventilation cycle consistently performs better than the recirculation cycle. The COP_t for the Level III ventilation cycle are about 25 percent greater than those for the Level III recirculation cycle.

Despite the improvement in system performance obtained through unbalanced flow operation, Figs. 8 and 9 show that the gas-fired desiccant cooling systems still generally consume more primary resource energy than does a vapor compression machine with an average $COP_{e,vc}$ of 2.5. Only the Level III ventilation system with regeneration temperature of 65°C and 85°C in Forth Worth and Washington, D.C. uses less resource energy.

In all locations except Miami, the operating costs of the gas-fired Level I desiccant systems become competitive with the conventional air conditioner at electricity-to-gas cost ratios of 7.0. The improved systems are cost-competitive at lower cost ratios. It is the thermal *COP* rather than the electrical *COP* of the high-performance gas-fired systems that must be increased to significantly improve the resource energy conservation and cost competitiveness of the desiccant systems compared to the conventional vapor compression air conditioner.

The performance of solar-assisted desiccant cooling systems with unbalanced flow and an 85°C auxiliary regeneration temperature is shown in Fig. 10 for Fort Worth. In all three locations considered, the utilization of solar energy results in a marked improvement in COP_a, and in Forth Worth and Washington, D.C. even the Level I solar-assisted systems consume less primary energy than does the conventional air conditioner. In contrast to the gas-fired systems, the solar-assisted recirculation cycle has higher COP_a than does the ventilation cycle. This occurs because the amount of solar energy collected in the recirculation cycle is insensitive both to level of system performance and flow unbalance, while in the ventilation system, the amount of solar energy collected decreases as the level of component performance increases. On the average, the Level III recirculation cycle collects and utilizes about 40 percent more solar energy than the Level III ventilation cycle, and the free energy input more than compensates for the lower recirculation cycle COP_t.

The exclusively solar-fired unbalanced flow desiccant cooling systems with 40 m^2 of collector are able to maintain room comfort for significantly larger fractions of the time than their balanced flow counterparts. The solar-fired recirculation cycle with unbalanced flow more frequently achieves room comfort than does the ventilation cycle. The Level III recirculation cycle can maintain room comfort about 88 percent of the time in Washington, D.C.; 70 percent of the time in Fort Worth, but only about 10 percent of the time in Miami. The collector area must be at least 50 m^2 for the solar-fired recirculation cycle to approach 90 percent room comfort fraction in Washington and Fort Worth, and must be over 60 m^2 in Miami. The ventilation cycle requires about 20 percent more collector area than the recirculation cycle to give equivalent room comfort.

The simulation results may be used to estimate the annual operating costs of the cooling systems. The performance of the desiccant systems during July is assumed representative of the average seasonal performance. A limited number of seasonal simulations suggest that this is reasonable. The seasonal operating costs of the conventional air conditioner and the unbalanced flow Level III desiccant cooling systems are given in Table 7, based on electricity at \$0.10/kW-hr and natural gas at \$4.00/gJ (\$4.00/1000 ft^3), which gives an energy cost ratio of 7. The costs for the exclusively solar-fired systems reflect electricity use at reduced room comfort.

The gas-fired Level III ventilation cycles have lower operating costs than the corresponding Level III recirculation cycles and are about 60 percent of those for the conventional air conditioner. When assisted by a 40 m^2 collector array, there is little difference in the operating cost of the ventilation and recirculation cycle systems. The solar-assisted systems cost about 40 percent as much to operate as the vapor compression machine. However, the cost of the 40 m^2 of the collector area must be paid for by reduced operating cost of the solar-assisted desiccant system compared to that of the gas-fired desiccant system. The total annual electricity savings are about \$100 in Miami and Fort Worth, \$50 in Washington, D.C. This represents an annual net savings of \$1.25/m^2 to \$2.50/m^2 of collector. The differential savings for the exclusively solar-fired systems range from \$2.00/m^2 to \$4.00/m^2. Since air-based solar heating systems cost on the order of \$200/m^2 to install, these savings represent an annual return on investment of at most 2 percent. These results scale directly with electrical and thermal energy costs if the ratio of these costs is constant. Higher electricity-to-gas cost ratios will improve the economic viability of the gas-fired desiccant systems in comparison to conventional vapor compression air conditioners. However, this will decrease the advantage of the solar heating system, which uses electricity for fans to circulate air flow to offset natural gas for regenerating the desiccant. Conversely, lower cost ratios improve the economics of the solar heating system, but make it more difficult for the desiccant systems to be cost-competitive with the vapor compression air conditioner. Thus, even though the solar heating system can significantly reduce the primary energy consumption of the desiccant cooling system, the incremental cost savings do not seem to favor solar operation.

The operating cost data indicate that high-performance gas-

Table 7 Seasonal operating cost of Level III desiccant systems, based on July performance, $0.10/kW-hr electricity and $4.00/gJ gas

system	mode/t_{aux}	Miami	Fort Worth	Washington
		532	496	274
		319	271	151
		378	318	171
		229	151	99
		256	149	86
		156	92	62
		272	143	86

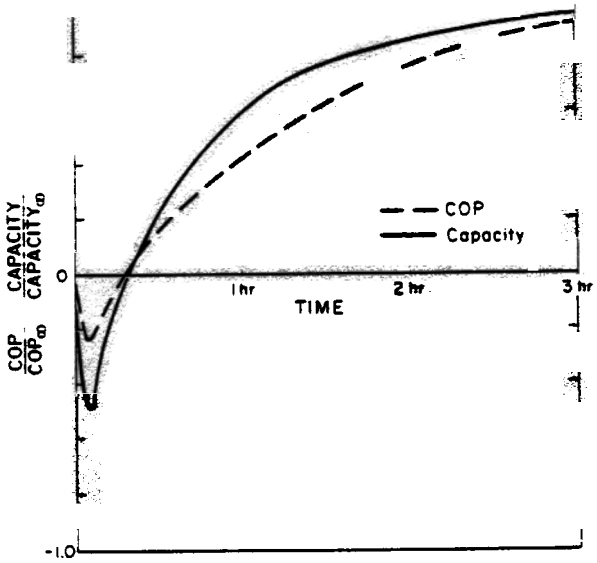

Fig. 11 Cold start-up transient of a ventilation cycle

fired desiccant cooling systems cost about 60 percent as much to operate as a state-of-the-art vapor compression air conditioner. However, considerable development effort separates the state-of-the-art Level I desiccant system prototypes currently available from the projected Level II and III systems. The performance of the vapor compression air conditioner could also be improved through development. For an energy cost ratio of 7, a vapor compression air conditioner with $COP_{e,vc}$ of 4.2 would break even in operating costs with the gas-fired Level III ventilation cycle air conditioner. This is within the range of projected improvements.

The average thermal *COP* of the gas-fired Level III ventilation cycle is approximately 0.9, and the electric *COP* is about 18. To become more competitive with the conventional air conditioner, the thermal *COP*, rather than the electric *COP*, must be increased. Two important dehumidifier parameters are the nondimensional heat and mass transfer length (Λ_w) and the matrix thermal capacitance. The Level III systems modeled have a nondimensional length of 30, and little improvement in capacity or thermal *COP* is obtained by larger values for dehumidifiers that are composed primarily of silica gel [13]. However, increasing Λ_w for a dehumidifier with large thermal capacitance does improve cycle performance. The COP_t of the Level III ventilation cycle can be increased by 20 percent by using a dehumidifier with Λ_w of 60 and a thermal capacity three times that of silica gel. The specific cooling capacity is concurrently decreased by 10 percent, implying that the physical size of the machine must be increased to achieve the same cooling effect. Also, the performance of the system with a large thermal capacitance dehumidifier is sensitive to operating conditions [13] and the seasonal improvement in performance may be less than that at the design point. If the same ratio between design point and seasonal average COP_t can be attained with the large thermal capacitance dehumidifier as for the nominal silica gel dehumidifier, it should be possible to realize a seasonal average COP_t on the order of 1.1 with these advanced systems.

On-off cycling of the desiccant system may adversely affect system performance because of the slow transient response of the rotary dehumidifier. A cold start-up response of a ventilation cycle system, predicted using a transient dehumidifier model [12], is illustrated in Fig. 11. In this worst case example, the silica gel dehumidifier is assumed to initially be in equilibrium with room state air. About 3 hr are required for the system to approach steady-state operating conditions. Most transients experienced in practice would be less severe, since during the cooling season it is unlikely that the dehumidifier will ever be off long enough to equilibrate both in temperature and humidity with room state air. The AiResearch prototypes have transients in system performance that persist for on the order of 1/2 hr [6] as operating points are changed. This response time is sufficiently long that development of hardware and control strategies to efficiently modulate cooling capacity (i.e., a multispeed blower) should be considered.

Conclusions

Ventilation and recirculation cycle designs have been evaluated for Miami, Forth Worth, and Washington, D.C. Three levels of component design representing state of the art to advanced technology were considered. The performance of balanced and unbalanced flow operation of gas-fired, solar-assisted, and solar-fired desiccant cooling systems were compared to a commercially available vapor compression air conditioner.

The thermal *COP* of the desiccant systems can be improved by unbalancing the air flow through the dehumidifier. Flow unbalancing can raise the thermal *COP* of the ventilation cycle by 10–15 percent, and can increase the COP_t for the recirculation cycle by 50 percent. Despite the enhancement of recirculation cycle COP_t by unbalancing the dehumidifier flows, the high-performance Level III ventilation cycle has a greater COP_t than the corresponding recirculation cycle.

The Level I, or state-of-the-art, unbalanced flow ventilation and recirculation cycles have nearly equivalent performance characteristics. However, the Level I systems use considerably more primary energy than the conventional air conditioner, and only break even in operating costs if the ratio of the cost of electricity to the cost of thermal energy is at least 7. Level III systems, with a high-performance heat exchanger and dehumidifier, are required for the gas-fired desiccant cooling cycles to use less primary energy and have lower operating costs than the conventional air conditioner. For an energy cost ratio of 7, the operating cost of the Level III ventilation cycle is about 60 percent of that of the nominal vapor compression machine.

An air-based solar heating system with 40 m^2 of flat plate collector is able to supply at least 50 percent of the thermal energy required by the Level III desiccant systems to meet the modest residential cooling load modeled in this study. This significantly reduces the primary energy consumption and operating costs of the cooling system. The solar-assisted Level III recirculation generally uses less auxiliary thermal energy than the ventilation cycle, although the operating costs of the

two systems are very nearly equal. The solar heating system alone cannot provide enough energy for the desiccant cooling systems to maintain room comfort. Analysis of the operating costs of the desiccant systems showed that if electricity costs \$0.10/(kW-hr) and thermal energy costs \$4.00/gJ, then addition of the solar heating system to the desiccant cooling system results in an annual return on investment of about 2 percent. This figure scales directly with increasing energy costs if the energy cost ratio is constant. Inceasing the cost of electricity relative to the cost of thermal energy has an adverse effect on the economics of the solar heating system.

The thermal *COP* of the high-performance desiccant systems must be improved to be competitive with conventional vapor-compression air conditioners. The design point COP_t of the high-performance ventilation cycle can be increased by about 20 percent by use of a very high heat and mass transfer unit dehumidifier with a large thermal capacitance matrix. This could result in seasonal average COP_t on the order of 1.1. The gas-fired system with the greatest development potential is a high COP_e ventilation cycle with a high-performance, large thermal capacitance dehumidifier.

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