THE FIELD OPERATION OF A THERMALLY DRIVEN LIQUID-DESICCANT AIR CONDITIONER

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ABSTRACT

Recent advances in liquid desiccant technology have created new opportunities for solar cooling. The high flooding rate systems that characterize industrial liquid-desiccant equipment have been replaced by low-flow technology that reduces desiccant recirculation rates by a factor of 10 to 50. Liquid-desiccant air conditioners (LDAC) that use the low-flow technology have superior performance, lower pressure drops and greatly reduced maintenance requirements. With the ability to overdry ventilation air and control indoor humidity, a solar liquid-desiccant air conditioner provides the building owner an exceptional value proposition.

During the summer of 2007, a 6,000-cfm (10,200 m^3/h) beta prototype of a LDAC processed a fraction of the ventilation air to 25,000 ft^2 (2,323 m^2) machine shop in Wrightsville, PA. The primary energy source to the LDAC was hot water at between 160 F (71 C) and 200 F (93 C). Although the source for this hot water could be solar thermal collectors, a gas-fired water heater was used throughout the 2007 test.

At typical summer operating conditions—89 F (32 C), 0.01815 lb/lb—the LDAC delivered 20 tons (70 kW) of cooling, most of which was latent. The three major desiccant components in the LDAC (i.e., the conditioner, the regenerator and the interchange heat exchanger) all came close to or exceeded their design performance. With hot water supplied at 185 F (85 C), the thermal Coefficient of Performance (COP) for water removal was 0.74.

A parametric design study estimated a payback of 9.5 years for a solar cooling system that uses a 6,000-cfm (10,200 m^3/h) LDAC and 2,000 ft^2 (186 m^2) of evacuated-tube collectors (based on absorber area).

Keywords: solar cooling, solar air conditioner, liquid-desiccant air conditioner

1. INTRODUCTION

In the 100 years since Willis Carrier built the first practical air conditioner, the technology has evolved from a curiosity to a luxury, and now to a necessity. In the U.S. alone, air conditioning is a $10 billion industry that uses over 4.3 quads (4.54 billion GJ) of primary energy. With almost all of this primary energy coming from non-renewable sources, air conditioners are one of the major contributors to worldwide climate change.

As the HVAC industry meets the rapidly expanding demand for comfort conditioning, it faces other challenges equally as critical as climate change. Most air conditioners run on electricity. In many parts of the U.S., a utility's peak demand for electricity will occur on hot, summer afternoons when the air conditioning loads are greatest. Brownouts and blackouts caused by overburdened electrical grids, high demand for natural gas to run peaking turbines and summertime air pollution are all problems exacerbated by electric air conditioners powered by fossil-fuel power plants.

The HVAC industry is also struggling with the conflicting needs to increase building ventilation to purge indoor pollutants, but at the same time, keep indoor humidity at comfortable and healthy levels. The problem is most critical in humid climates where the ventilation air brings into the building more humidity than a conventional air conditioner can remove. Mold, which becomes a danger whenever indoor relative humidity consistently exceeds 70%, is frequently cited in news stories as the source of health problems, property damage and litigation.

The preceding challenges present a unique opportunity for solar cooling to gain entry into the HVAC market. By fuel switching to a renewable resource, solar cooling can be an essential part of a strategy to conserve fossil resources and reverse global climate change. At the same time, the summertime stress on regional electric systems can be relieved.

Furthermore, if the solar cooling system can more effectively provide latent cooling (i.e., dehumidification) then it can help improve building ventilation and address humidity related IAQ problems.

2. LOW-FLOW LIQUID DESICCANT TECHNOLOGY

Desiccants are materials that have a high affinity for water vapor. They are unique in that they can dry air without first cooling the air below its dewpoint. In applications that require mostly latent cooling, desiccants can avoid the inefficient operation of vapor-compression and absorption systems, which have to overcool the air to condense the water vapor and then reheat the air so that the conditioned space can be kept at a comfortable temperature.

The value that a LDAC provides is illustrated by the following simple example. A 300 ft^2 (28 m^2) meeting room with 10 people and 600 W of lighting and plug loads will have internal sensible and latent loads of 4,550 Btu/h (1,333 W) and 2,000 Btu/h (586 W). If this room is supplied with 300 cfm (510 m^3/h) of air saturated at 55 F (12.8 C), mixed air conditions within the room will be 69 F (20.6 C) and 70% rh. These conditions are well outside the ASHRAE comfort zone. If the supply air is first reheated before it is delivered to the room with the goal of bringing the room to 75 F (23.9 C), the amount of reheat will equal 29% of the room's internal load. This reheat is a double penalty on the building's energy use: not only is energy used to supply the reheat, but the reheat is an additional load that must be handled by the cooling system. Room temperature can be increased to 77 F (25.0 C) and relative humidity reduced to 61% by reducing the air flow to 150 cfm (255 m^3/h). However, at this low air volume, it is unlikely that the room will be adequately ventilated.

The LDAC can eliminate the preceding need to reheat by overdrying the ventilation air. Under typical summer conditions, the LDAC can condition the ventilation air to 88 F (31.1 C) and 30% rh. If the ventilation air is 25% of the recirculated air flow and the central air conditioner again supplies air saturated at 55 F (12.8 C), then the mixed air conditions within the meeting room will be 77 F (25 C) and 52% rh—conditions that are close to the center of the ASHRAE comfort zone. Comfort is maintained, as is adequate ventilation, without the need to overcool and reheat the supply air.

Liquid desiccants have been used for industrial drying of air since the 1930s. These industrial systems typically use beds of porous contact media that are flooded with a liquid desiccant that has been cooled in a separate heat exchanger. The process air is drawn through the bed and is dried as it contacts the liquid desiccant. Heat is released as the desiccant absorbs the water vapor. The desiccant's ability to dry air decreases exponentially with temperature, and since the bed is typically adiabatic (i.e., there are no cooling coils within the bed), flooding rates must be sufficiently high to prevent more than a few degrees rise in the desiccant's temperature.

A fundamental characteristic of industrial liquid desiccant systems is that the process air flowing through the bed will entrain droplets. Droplet filters must be employed to limit the loss of desiccant from the system.

Although the technology of industrial liquid-desiccant systems has been used in HVAC applications, it has not been widely accepted. Several characteristics of the technology can be blamed for this including: (1) the high cost for industrial-grade equipment, (2) high fan power for moving air through the flooded beds, (3) high operating costs associated with industrial regenerators, (4) high maintenance requirements for the droplet filters, and (5) potential damage from desiccant droplet carryover past the filters.

A new generation of liquid-desiccant conditioners and regenerators that meets the needs of HVAC applications has been developed and proven. The two most important improvements are (1) desiccant flooding rates have been decreased by a factor of 10 to 50, and (2) contact surfaces are no longer adiabatic, being continually cooled in the conditioner and continually heated in the regenerator. These two changes are related in that when the desiccant flooding rate is decreased, the thermal capacitance of the flow is proportionately decreased. If the contact surface were adiabatic, the desiccant's temperature would either rapidly increase in the conditioner or rapidly decrease in the regenerator and the driving potential for the exchange of water vapor would be lost.

The preceding two improvements in liquid-desiccant technology lead to a much more competitive cooling system. Compared to the industrial technology now in use, a low-flow liquid-desiccant air conditioner (LDAC) will:

- have much lower pressure drops
- be more compact
- produce a greater cooling effect (e.g., lower cfm/ton)
- more deeply dry the process air, and
- have a higher COP

Perhaps most importantly, both the low-flow conditioner and regenerator will operate without the entrainment of desiccant droplets by the air streams, i.e., zero desiccant carryover.

As shown in Figure 1, a LDAC that uses the low-flow technology has three main components: (1) the conditioner, (2) the regenerator, and (3) the interchange heat exchanger. The conditioner is a parallel-plate heat exchanger in which the plates are water-cooled. Films of desiccant flow in thin wicks on the outer surfaces of the plates. The process air (horizontal arrows) flows through the gaps between the plates and comes in contact with the desiccant. The desiccant absorbs water vapor from the air, and the heat that is released is transferred to the cooling water. The air leaves the conditioner drier and at a lower enthalpy (i.e., cooling occurs, although most of the cooling may be latent rather than sensible).

The water absorbed by the desiccant in the conditioner is desorbed in the regenerator. This component is again a parallel-plate heat exchanger, but now hot water (or other heat transfer fluid) flows within the plates. The hot desiccant films that flow on the outer surfaces of the plates desorb water to a flow of scavenging air (horizontal arrows) that rejects the water to ambient.

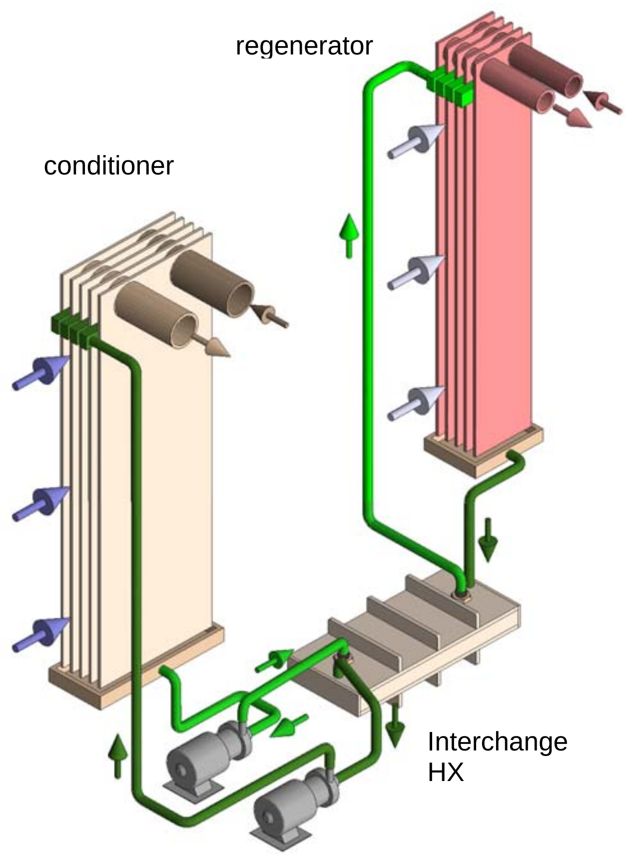

Figure 1 – Operation of a Low-Flow Liquid-Desiccant Air Conditioner

The interchange heat exchanger, which transfers heat from the hot, strong desiccant leaving the regenerator to the cool, weak desiccant flowing to the regenerator, performs a dual function. It improves the efficiency of the regenerator by preheating the weak desiccant. It also increases the cooling provided by the conditioner by reducing the heat load imposed by the strong desiccant.

3.0 THE FIELD OPERATION OF A 6,000-CFM LDAC

At the start of the summer of 2007, a beta-prototype LDAC was installed as a Dedicated Outdoor Air System (DOAS) on a machine shop in Wrightsville, PA. The LDAC, shown installed at the machine shop in Figure 2, cooled and dried approximately 6,000-cfm (10,200 m3/h) of ventilation air for the building.

Figure 2 – LDAC Installed as DOAS

The beta prototype, although fundamentally the same as the alpha prototype that is described by Lowenstein (2006), differed from the earlier unit in that,

- Its regenerator had the capacity to remove over 250 lb/h (114 kg/h) of water from the desiccant (as opposed to 140 lb/h (64 kg/h) for the alpha prototype's regenerator), and

- It used a second generation interchange heat exchanger that had a higher effectiveness.

The beta prototype's higher capacity regenerator complicated the start up of the unit. An important element in the control of a LDAC is modulating the regenerator so that the desiccant never approaches a concentration at which it crystallizes. For the alpha prototype that was field tested in 2005, control of the unit was simple: the under-sized regenerator could run at 100% of its capacity without over concentrating the desiccant. With a regenerator that had twice the water-removal capacity, a means of modulating the regenerator was imperative.

The value of a LDAC is its ability to keep indoor humidity at safe and comfortable levels when latent loads on the building are high. In applications where the LDAC might overdry the indoor space, a control scheme that adjusted the desiccant concentration to the lowest level that kept the humidity at setpoint would be preferable to one that cycled the LDAC on and off. Continuous operation of the LDAC at a reduced desiccant concentration has the advantages of avoiding cycling losses and improving regeneration COP (since it takes less thermal energy to remove water from a weaker desiccant).

Overdrying of the indoor space was not an issue for the 2007 field test: only about 20% of the outdoor air brought into the machine shop was processed by the LDAC. The control algorithm, therefore, tried to keep the desiccant concentration at a maximum value that avoided crystallization (i.e., a value between 41% and 43%).

Industrial liquid-desiccant systems commonly control desiccant concentration by monitoring the level of the desiccant in the sump. At commissioning, the concentration and level of the desiccant in the sump are set. As long as no desiccant is added or removed from the unit, the level of the desiccant in the sump can then be used to calculate its concentration. Although simple, this approach to control has three important limitations: (1) improper or unauthorized servicing of the LDAC could change the amount of desiccant in the sump thereby changing the relationship between liquid level and desiccant concentration, (2) the level responds relatively slowly to changes in the desiccant concentration, and (3) in LDACs that use a single, stratified sump (i.e., weak and strong desiccant are stored in the same sump with the more dense, strong desiccant stored below the less dense, weak desiccant), liquid level measures the average desiccant concentration of the weak/strong mixture.

The preceding limitations of level control more greatly affect a low-flow LDAC than a high-flow industrial unit. The concentration of the desiccant may increase by four to six points in one pass through a low-flow regenerator. For a high-flow regenerator, the increase is typically a few tenths of a point. With the greater change in concentration across the regenerator, problems of crystallization can happen much faster in a low flow system.

An in-line sensor that measures the desiccant's refractive index and temperature was installed in the desiccant outlet line from the regenerator. By measuring both the refractive index and temperature of the desiccant, its concentration could be calculated. With the desiccant concentration at the outlet from the regenerator continuously monitored, in principle, the rate of water removal could be modulated to keep this parameter at a setpoint value.

The two most practical methods for modulating the regenerator are (1) control the temperature of its hot-water supply, and (2) control the flow of scavenging air (i.e., control the regenerator fan). Under most conditions, a regenerator's water removal capacity will decrease as either the hot-water temperature or scavenging-air flow decrease. However, the regenerator's efficiency (i.e., COP) will also decrease with decreasing hot-water temperature, but it will increase with decreasing air flow. Given this behavior, the first attempt at controlling the LDAC in the 2007 field test modulated the regenerator's blower to control air flow while simultaneously modulating the gas to the hot-water boiler to maintain a fixed supply temperature to the regenerator.

Perhaps the most important lesson learned in the 2007 field test was that the preceding approach to controlling the LDAC is very difficult to implement. Although past comparisons between the measured performance of the regenerator and the predictions of AILR's computer models have shown good agreement, these comparisons were made at air flow rates close to design values. (The lowest air flow for the NREL test reported by Lowenstein (2006) corresponds to a 1,500 cfm (2,550 m^3/h) regenerator flow in the beta prototype. This flow is close to the nominal design value for the beta prototype.) The computer simulation suggested that a reduction in air flow by 75% would decrease the water removal rate of the regenerator by 40% and increase COP by 8%.

The preceding 75% reduction in air flow is far outside the range where the regenerator's performance had been measured. Although it is difficult to quantitatively determine the deviation (i.e., the regenerator air flow is not directly measured), the regenerator's water removal capacity was much less dependent on air flow than the computer simulation implied. Although it was difficult to trace the source of the deviation during field operation, it is possible that natural circulation of the air flow between the hot plates of the regenerator is becoming important at very low flows. The secondary flows induced by natural circulation are not included in the computer simulation of the regenerator.

A second control problem that was encountered in the 2007 field test was the strong influence that temperature transients had on the output signal from the refractive index sensor. The desiccant's refractive index is a function of both concentration and temperature. The refractive index sensor was a prototypical instrument that was developed by AILR for this application. Prior to installing the sensor in the LDAC, it was calibrated over a range of desiccant concentrations and a range of temperatures. However, this calibration did not account for the possibility of rapid changes in the desiccant temperature. In the field test, the refractive index sensor frequently gave erroneous readings during transient conditions that then led to short cycling of the hot-water supply to the regenerator.

TABLE 1: LDAC PERFORMANCE

		24-Aug	model	delta
Ambient T	F	89.0	(1)	
Ambient w	lb/lb	0.01815	(1)	
Supply Air T	F	92.0	89.4	
Supply Air w	lb/lb	0.01079	0.01110	
Air Enthalpy Change	Btu/lb			
C Des Conditioner Inlet		0.372	(1)	
Conditioner Des Flow	gpm	3.15	3.15	
Conditioner Air Flow	scfm	6,352 (2)	(1)	
Supply CW T	F	84.0	(1)	
Return CW T	F	91.8	92.0	
Cooling Water Flow	gpm	60.7	(1)	
Q Air Total	Btu/h	210,576	217,184	3.1%
Q Air Latent	Btu/h	222,922	212,596	-4.6%
Q Cooling Water	Btu/h	234,194	239,674	2.3%
Water Removal	lb/h	210.5	200.8	
Supply HW T	F	184.6	(1)	
Regenerator Des Flow	gpm	3.20	(1)	
Regenerator Thermal COP		0.742	0.685	-7.7%
IHX Effectiveness		0.855		
Total Fan/Pump Power	kW	5.80		

CW - cooling water for conditioner
HW - hot water for regenerator
(1) - value set to match data
(2) - inferred from energy balance between air, water and desiccant
assumes that heat dump from desiccant is 10% of total heat gained by water

During the month of August, the beta prototype operated for approximately 76 hours without short cycling and with the hot-water supply temperature to the regenerator close to its setpoint value. Table 1 presents a one-hour average of performance on the afternoon of August 24. Latent loads are high during this period with the beta prototype operating at 84% of its design water-removal capacity of 250 lb/h (114 kg/h). For this period of relatively steady operation, the beta prototype's performance closely matches the predictions of the computer model that was used to design the unit.

4.0 DESIGN OF A SOLAR COOLING SYSTEM

The thermal energy that is needed to regenerate the desiccant for a LDAC can be effectively provided by solar thermal collectors. Perhaps the most important decisions to be made when designing a solar regenerator are (1) the type of solar thermal collectors to be used, (2) the hot-water (or glycol) supply temperature, and (3) the size of the array.

Desiccant regeneration is a mass transfer process that is driven by the desiccant's equilibrium water-vapor pressure. This pressure exponentially increases with increasing temperature, and so the efficiency of a regenerator will increase with increasing supply hot-water temperature. This behavior is shown in Figure 3 by the curve marked with triangles. (The regenerator performance assumes calcium chloride is concentrated from 39% to 43%.)

The efficiency of the collector, however, decreases as it supplies hotter water. As shown in Figure 3, this decrease is more significant for flat-plate collectors (curve marked with diamonds) than it is for evacuated-tube collector (curve marked with squares).

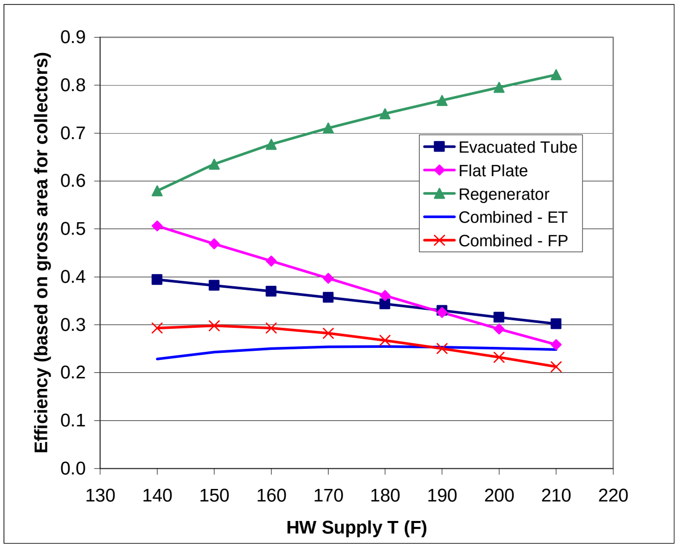

Figure 3 – Collector, Regenerator and Overall Efficiencies

(The efficiency curves in Figure 3 are for collectors in Tampa, FL that face south and are tilted at an angle equal to latititude. They are seasonal averages: the annual thermal energy supplied as hot water divided by the solar radiation that intercepts the gross area of the collector. The collector performance was modeled using TRYNSYS with the TESS Type 538 for the collector's performance (SCCL 2008) and TMY2 weather data. The efficiency curves are for a single-glazed flat-panel collector with selective surface and a dewar-type evacuated-tube collector.)

At each hot water supply temperature, the regenerator's efficiency can be multiplied by the collector's efficiency to yield the overall efficiency for the solar regeneration process. As shown in Figure 3, the opposite slopes for the collectors' and regenerator's efficiency curves lead to relatively flat overall efficiency curves for each.

Although the overall efficiency of a collector/regenerator pair may only weakly depend on the hot-water supply temperature, a second consideration will steer the design towards hotter supply temperatures: the water-removal capacity of a scavenging-air regenerator increases exponentially with supply temperature. Over the temperature range shown in Figure 3, the water-removal capacity of a regenerator increases by a factor of approximately 2 for each 20 F (11 C) increase in supply hot-water temperature. Although the array area needed to meet a latent load may not change significantly as the supply hot-water temperature increases, the size of the regenerator will decrease.

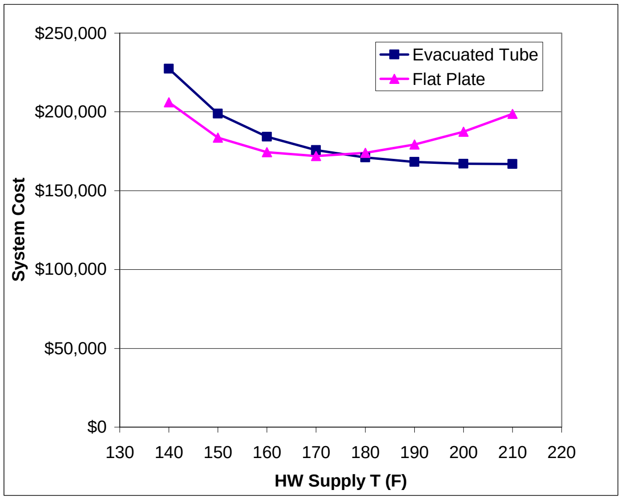

Figure 4 – “Partial” System Cost as a Function of HW Supply Temperature

In many applications, the designer is trying to reduce the installed cost of the solar cooling system. Figure 4 shows the “partial” cost for a solar thermal array and regenerator that have a constant seasonal water removal capacity when the incremental cost for adding area of flat-plate and evacuated-tube collectors are $25 and $40 per square foot, respectively and the incremental cost of increasing the size of the regenerator is $17 per square foot of contact area. (The cost in Figure 4 is referred to as “partial” since it does not include the cost of storage and other “balance of system” components.)

As shown in Figure 4, the solar cooling system that uses the flat-panel collectors has its lowest installed cost when operating at a 170 F (77 C) supply hot-water temperature. For the system with evacuated-tube collectors, the minimum cost occurs at a temperature above the 210 F (99 C) limit for the graph. At their minimums, both systems have close to the same “partial” installed cost: $172,000 for the flat-plate system and $167,000 for the evacuated-tube system.

An important advantage of a solar cooling system that uses a LDAC is the capability to store energy as concentrated desiccant. This allows the LDAC to provide cooling throughout the day and night.

In solar applications, concentrated desiccant would be stored in an uninsulated plastic tank. The amount of desiccant would be significant: 12 hours of full-load operation of a 6,000-cfm (10,200 m^3/h) LDAC would require 3,200 gallons (12.1 m^3) of desiccant (assuming that the desiccant was calcium chloride and its weak and strong concentrations were 39% and 43%). This amount of storage is equivalent to 272 ton-hours (957 kWh) of latent cooling. This is slightly more volume than would be required to store an equivalent amount of cooling with ice and much less than if hot water was stored for an absorption chiller, e.g., a 0.6 COP single-effect absorption chiller working over a 20 F (11 C) differential in hot water supply and return would require about ten times the storage volume. Furthermore, both ice storage and hot water storage need more expensive insulated storage tanks.

A parametric study was performed to determine the impact that the sizes of the solar array and desiccant storage have on the competitiveness of a solar LDAC. For this study, the following assumptions were made:

Location	Tampa, FL
Collector type	evacuated tube
LDAC capacity	6,000 cfm (10,200 m^3/h)
LDAC EER	45
Desiccant	39%/43% calcium chloride
Application	Dedicated Outdoor Air System
Operating hours	6 AM to 6 PM, 365 days per year
Collector cost	$40 per square foot absorber
LDAC cost	$48,000 (with 320 lb/h regen)
Regenerator cost	$23 per lb/h additional capacity
Storage cost	$0.70 per lb calcium chloride
Balance of system	$7,000
Incentives	30% investment tax credit
Cost of gas	$10 per million Btu
Cost of electricity	$0.10 per kWh

This application is based on a design study of a library in the Tampa area that placed a high priority on controlling indoor humidity. For many hours throughout the year the air that is recirculated within the library is over-cooled to remove moisture and then reheated to an acceptable temperature. Based on meter readings, the library used 853 million Btu in 2006 for reheating. As part of this study, it is assumed that the LDAC avoids 60% of this reheating. Also, by reducing the need for reheat, the LDAC reduces the cooling load on the building.

The LDAC cools and dries the 6,000 cfm (10,200 m^3/h) of ventilation air to the building. A central air-cooled chiller handles the remaining load on the building. This central chiller is assumed to have a 10 EER. Since this is a retrofit application, no credit is given to the solar LDAC for reducing the load on this chiller.

Figure 5 shows the effect that array size and storage capacity have on the utilization of solar thermal energy from the array. ("Utilization" is the percent of the array's full potential that is used to regenerate the desiccant). Figure 6 shows the effect of array size on the percentage of the LDAC's thermal requirement that is met by solar (i.e., the "solar fraction"), the balance being supplied by an 80% efficient gas-fired water heater. The behaviors shown in these figures are expected: (1) utilization increases both as the array size decreases and storage increases, (2) the solar percentage for regeneration increase both as the array size increases and storage increases.

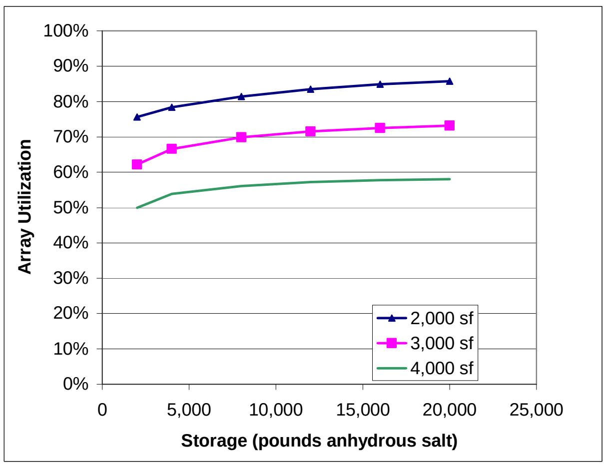

Figure 5 – Array Utilization as a Function of Array Size and Storage Capacity

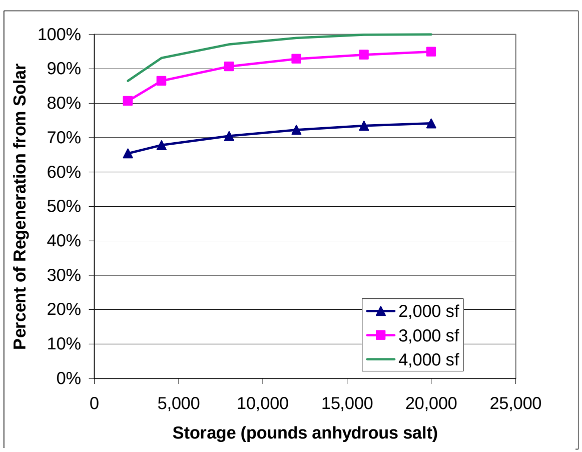

Figure 6 – Solar Fraction as a Function of Array Size and Storage Capacity

For the solar LDAC to be competitive both the utilization of the array and the solar fraction of energy for regeneration must be high. (The second requirement minimizes the amount of expensive gas that is used to back up the LDAC when solar is not available.) Using a simple payback as a measure of competitiveness, Figure 7 shows that the solar LDAC with the 2,000 square foot array (absorber area) has the shortest payback: 9.5 years when 12,000 pounds of salt storage is part of the system. This system uses 84% of the array's full potential for regeneration. It also meets 72% of its thermal requirements for regeneration from solar.

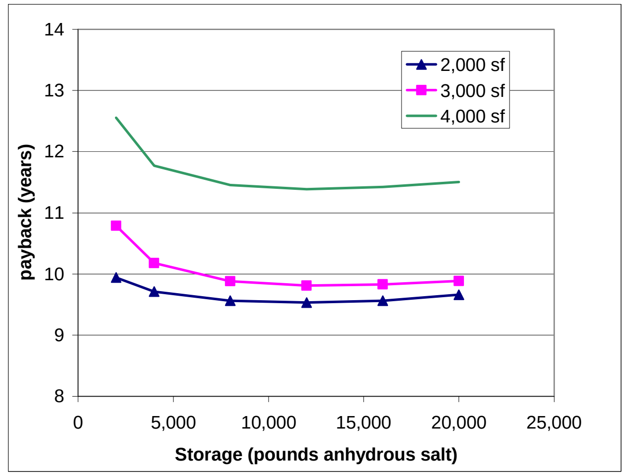

Figure 7 – Simple Payback for Solar LDAC

The LDAC with a 3,000 square foot array has only a slightly longer payback: 9.8 years when the storage is again 12,000 lb. With the larger array, utilization is slightly less, 72%, but solar now meets a much larger fraction of the regenerator's thermal requirements: 93%.

The LDAC with a 3,000 square foot array and 12,000 lb of storage has an installed cost of $187,000 before the tax credit is applied and $131,000 after. The LDAC provides up to 23 tons of cooling, almost all of which is latent. Therefore, on a per-ton basis, the cost for the solar LDAC is $8,130 before the tax credit and $5,695 after.

5. CONCLUSION

Air conditioning is one of the more energy intensive end-uses demanded by a rapidly developing world. In many applications, solar thermal collectors coupled with a liquid-desiccant air conditioner will be the most competitive approach to switching this need to a sustainable resource.

The conditioning of ventilation air to commercial buildings in humid climates is the "low hanging fruit" in the competition between sustainable and fossil-based cooling technologies. Although it is a niche of the total air conditioning market, cooling and drying ventilation air is a very large niche.

Air conditioners are sized based on the cooling demand during peak periods. On a summer design-day in a humid climate, an office building with a relatively low 15% ventilation rate (i.e., 15% of the air recirculated in the building is exhausted and replaced with fresh outdoor air), will require about one ton of cooling for the ventilation air for every two tons of cooling to meet internal and envelope loads. Thus, one-third of this building's cooling capacity is sized to meet the ventilation load. In many applications, the ventilation load will be a much larger fraction of the total: for schools, the ventilation rate may be 50%, theaters, 60%, laboratories and hospital surgical suites, 100% (Kosar 1998).

Furthermore, several important trends within the HVAC industry are increasing the importance of cooling and dehumidifying ventilation air:

- Ventilation rates are increasing as a way to improve indoor air quality; a building can earn LEEDs points for increasing it ventilation rate above minimum standards.
- Conservation activities such as more efficient lighting, advanced insulation and glazing, and low-energy equipment reduce a building's internal and envelope loads, but leave its ventilation loads unchanged.
- Advanced HVAC designs, such as displacement ventilation, that can decrease the overall energy needed to air condition a building increase the fraction of total cooling that serves the ventilation loads.

Processing ventilation air in humid climates is the "low-hanging fruit" because conventional DX air conditioners and chillers have trouble serving the latent component of this load. As previously discussed, a conventional air conditioner can meet high latent loads only by overcooling the air and then reheating it back to acceptable temperatures. The energy for both overcooling and reheating are a cost born by the conventional air conditioner, but not the solar LDAC.

The preceding analysis for the economics of owning a solar LDAC showed a 9 to 10 year simple payback with a 30% investment tax credit. This analysis is intended only as a rough indication of the competitiveness of the technology. It shows the challenges that this renewable technology faces as it begins to enter the market, (i.e., many purchasing decisions require simple paybacks that are less than five years).

However, one can expect the economic incentives for owning a solar LDAC to improve significantly in the future. At $0.10 per kWh for electricity and $10 per million Btu for natural gas—the assumed prices in the preceding analysis—energy is still relatively inexpensive. In many parts of the world, energy prices are already several times these costs. A doubling of energy prices is certainly possible in the next ten years if there is a serious worldwide effort to curtail carbon dioxide emissions and mitigate climate change.

The competitiveness of the solar LDAC will also improve as the technology develops and manufacturing becomes more efficient. The thermal collectors are the most expensive component in the solar LDAC, accounting for about two-thirds of the installed cost. The preceding analysis assumed that evacuated-tube collectors could be installed for $40 per square foot of absorber. This cost is at the very low end of the cost now to install these collectors in the U.S. (The individual dewar-type tubes used in the evacuated-tube collectors that were modeled can be purchased in container quantities for $5 per square foot of absorber, and complete assemblies, again in container quantities, for $22 per square foot. However, solar thermal collectors are a novel product in the U.S., and the distribution chain has significant mark ups to give dealers large incentives to accept the risk of selling them. An informal survey of a few possible installations for the same dewar-type collector gave installed costs that ranged from $40 to $120 per square foot.)

The HVAC industry is conservative and new technologies are accepted slowly only after their performance, maintenance and reliability have been proven. The field test presented here is a first step toward this acceptance

6. ACKNOWLEDGEMENTS

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