An Hourly Building Simulation Tool to Evaluate Hybrid Desiccant System Configuration Options

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ABSTRACT

This paper develops a building simulation tool to evaluate hybrid desiccant systems. The software tool is used to evaluate the impact of various desiccant equipment configurations and control approaches on annual operating costs and the ability to control space humidity levels. The simulation model is unique in that it properly accounts for many of the building/HVAC system interactions that must be considered to properly predict space humidity levels. The software tool is used to assess the impact that alternative equipment configurations would have had on the desiccant system installed at a test school (Henderson et al. 2002). The simulation results predicted that the desiccant system as installed had an annual operating cost premium of 61%, compared to a base cooling unit without humidity control. If the system had been installed to recirculate air during unoccupied periods, humidity control in summer and on weekends would have been greatly improved with only a modest increase in operating costs. The results also show that configuring the same desiccant unit to operate in the recirculation mode year-round (instead of pretreating ventilation air) would have significantly reduced operating costs, primarily due to lower process fan power. This somewhat surprising result shows that fan power issues can sometimes overshadow the thermal performance of the desiccant wheel in terms of overall operating costs. The operating cost benefit of common desiccant system components, such as the regeneration-side evaporative cooler, was also shown to be modest, bringing into question the cost-effectiveness of this component. Overall, these results show that configuration and control issues can have a big impact on the cost-effectiveness of desiccant technologies. Manufacturers must consider these details in order to minimize equipment costs and operating costs so that desiccant technology can meet its full potential.

INTRODUCTION

It is generally accepted that one of the best ways to apply desiccant technology in commercial buildings is as a ventilation air pretreatment system. Collier et al. (1982) and others have shown that desiccant technology is most promising in this application. Allowing a desiccant wheel to directly treat ambient air increases both the latent capacity and efficiency of the overall HVAC system. Since most of a building's latent load is associated with incoming ventilation air (Harriman et al. 1997), this load can be most effectively met by reducing the humidity and temperature of that airstream. The concept of pretreating ventilation air and providing it to the space or HVAC system at "space neutral" conditions can also minimize the need for additional sensible cooling components.

Several field tests of desiccant systems have recently been completed (Henderson et al. 2002; Miller et al. 2002; Yborra 2002). These field tests included detailed monitoring to quantify the operating costs of these systems and to demonstrate their ability to improve humidity control in the various commercial building applications. In each of these field tests, one HVAC system configuration was ultimately selected, installed at the test site, and monitored. In some cases, the measured field test results implied that performance and operating costs might have been improved if the desiccant unit had been configured or controlled differently within the HVAC system.

Several detailed computer models of desiccant equipment and auxiliary components have been developed (Collier 1997) to simulate the performance of built-up desiccant systems. These tools have generally focused on detailed equipment performance and the interaction between components such as desiccant wheels, sensible heat exchangers, evaporative cool-

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ers, etc. In some cases, the equipment models have been coupled into simple building models.

Traditional hourly building simulation programs are an effective way to quickly evaluate the impact of different HVAC system options for given application. Dozens of general building simulation tools are available (DOE 2002) to predict the performance of the integrated building/HVAC system. While numerous generalized tools are available, very few can simulate the performance of desiccant equipment. DesiCalc, a screening tool based on the DOE-2 calculation engine (Buhl et al. 1993), is one few building simulation programs to include modern desiccant systems.

However, the DOE-2-based DesiCalc tool has some limitations. First, the building model in DOE-2 does not accurately predict space humidity levels in all situations since it uses a quasi-steady-state moisture balance for each hour that neglects any moisture capacitance in the space and building furnishings. This quasi-steady moisture balance approach provides poor predictions of space humidity during hours when the cooling loads are modest. Second, the DX cooling coil in DOE-2 does not accurately predict the mix of latent and sensible cooling capacity at off-design, entering-coil conditions. Models that provide realistic predictions of latent coil performance are especially important with hybrid systems that combine desiccant and conventional HVAC components, since the dry supply air from the desiccant wheel can skew coil performance. Finally, since DesiCalc was intended to be a screening tool, it primarily focuses on the most common desiccant system configurations. Therefore, it offers very little flexibility in terms of component arrangements and control options.

This paper develops a flexible building/HVAC system simulation model that can be used to access the impact of various desiccant, conventional, and hybrid equipment configurations. The single-zone hourly building model is based on TRNSYS (Klein et al. 2000), a highly flexible program intended to simulate the performance of complex thermal systems. The computer tool includes a simple user interface that allows non-expert users to compare various equipment configuration and control options. The tool is unique in that it includes detailed component models and also considers many of the detailed building-equipment interactions necessary to accurately predict space humidity levels.

MODEL DEVELOPMENT AND APPROACH

Conceptual Framework

The conceptual layout of the hybrid desiccant model is shown in Figure 1. The model includes three main components:

1. A dehumidification/ventilation pretreatment module that can be a gas-fired desiccant wheel, an enthalpy wheel, a liquid desiccant unit, or any other specialized ventilation pretreatment or dehumidification system.
2. A DX cooling coil that cools and dehumidifies a mix of air from the space, ambient, or the ventilation pretreatment module.
3. A heating section capable of maintaining space temperatures in the winter or providing reheat in conjunction with cooling operation in the summer.

Both the ventilation pretreatment module and the AC cooling coil each have a dedicated fan that can be controlled to cycle with the component, run continuously, or be a mix of these two strategies, depending on building occupancy. The system also includes an optional economizer that can use outdoor air for free cooling when ambient conditions are appropriate. The airflow to and from each component can be

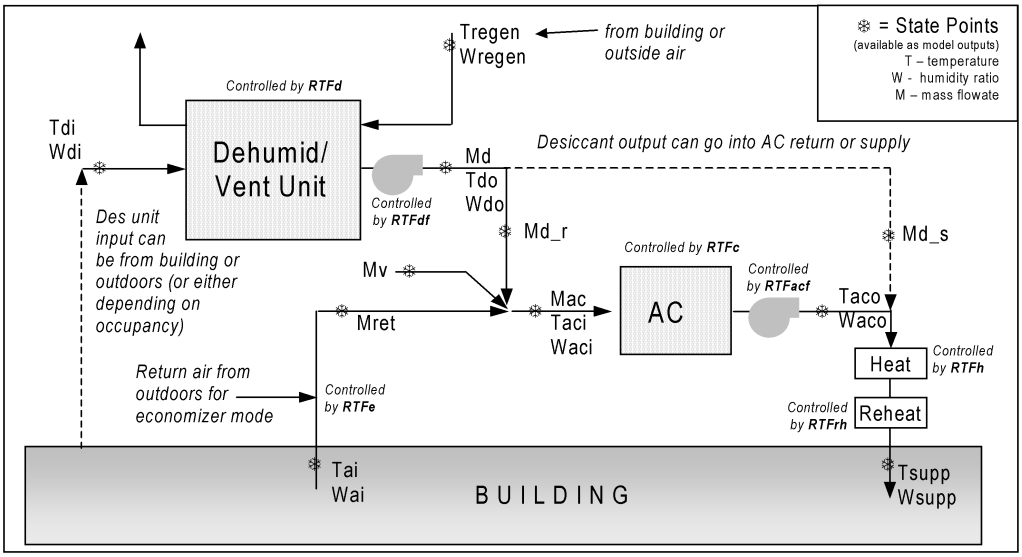

Figure 1 *Conceptual schematic of hybrid desiccant system model.*

specified to simulate a wide range of equipment configurations. In some cases, the configuration can be controlled based on operating conditions or occupancy schedules.

The overall system is simulated using a state point approach, where the leaving pyschrometric conditions from each component are tracked and passed on as the entering conditions to the next component. At points where multiple airflows are mixed, energy and mass balances are used to predict the resulting conditions. Temperature, absolute humidity, and CO_2 concentration are all tracked throughout the system for each hour. The building model considers internal heat, moisture, and CO_2 generation rates, as well as the conditions of the supply air entering the building.

The TRNSYS solver is used to determine the runtime fraction (RTF) of each component that maintains the desired space setpoints. Iterations are often necessary to find the combination of component runtimes that satisfy the control requirements for each hour. Other control points can vary with occupancy, ambient conditions, or the state of other components in the system.

The modeling approach for the main subsections of the model are described below.

Building

The building envelope and internal gains are modeled with the TYPE 19 single-zone model in TRNSYS. This model uses the transfer functions for predefined wall, roof, and floor assemblies from the *1977 ASHRAE Handbook—Fundamentals* (ASHRAE 1977). Several dozen predefined assemblies with multiple layers are available. The building is modeled as a single, well-mixed zone with a slab-on-grade floor. The building geometry is assumed to be a simple rectilinear box oriented along the north-south axis. The bottom of the slab is thermally coupled to the earth temperature, which varies seasonally. The earth temperatures are predicted with the data from the IGSHPA Installation Manual (1985). Windows can be added to each of four walls. Direct and diffuse solar gains through fenestration are added to floor surface or the zone air mass as required. An internal radiation network determines the radiative heat transfer between the walls, ceiling, and floor based on surface temperatures. The thermal capacitance of building furnishings are modeled with a factor of 50 lbs/ft^2 (244 kg/m^2) based on floor area. The moisture capacitance of the building volume and its furnishings is considered by using the moisture-holding capacity of the zone air mass multiplied by a simple lumped capacitance factor of 20. This factor is consistent with the recommendations of the IHAT software reference manual (EPA 2001) for a school with a medium-to-high amount of interior furnishings. The IHAT analysis determined the best factors for the lumped capacitance approach by comparing it to more detailed moisture storage models. The volume of zone air is also used in a first-order mass balance to predict CO_2 concentration levels for each hour.

Scheduled internal gains due to lighting and equipment can be specified daily, weekly, and seasonally. Occupant-generated heat, moisture, and CO_2 can also be scheduled. Infiltration is assumed to be zero when mechanical equipment operates and increases to a constant value for hours when the building is unoccupied.

Cooling Coil

The AC cooling coil is based on the DX coil model from the ASHRAE secondary toolkit (Brandemuehl et al. 1993). This coil model has also been included in EnergyPlus (2001). The model uses nominal efficiency and sensible heat ratio (SHR) to develop a rational performance map of capacity and efficiency that applies over a broad range of ambient and entering coil conditions. The model uses the apparatus dew point and bypass factor method to predict the mix of latent and sensible capacity over a broad range of entering temperature and humidity conditions. The model is also unique in that it accurately predicts the capacity and efficiency of the coil near dry-out conditions—a common condition when the supply air from a desiccant wheel enters a DX coil.

The part-load performance of DX system is modeled using the capacity degradation factor (C_D) from the SEER test procedure (ARI 1994). This linear part-load function (PLF) uses C_D to predict the efficiency degradation as a function of the runtime fraction for each hour. C_D for the AC unit is assumed to be 0.2. No efficiency degradation is assumed for the desiccant unit.

Another important part-load effect is the degradation of latent capacity at cyclic conditions. Research by Khattar et al. (1985) has shown that when the supply fan operates continuously, as in a commercial building, moisture on the fins of the cooling coil is evaporated back into the airstream when the cooling coil is deactivated. Henderson and Rengarajan (1996) developed a model that predicts the part-load SHR under constant fan operation as a function of runtime fraction. That model is shown in Figure 2 using the model parameters based on the observed performance of commercial AC systems. The

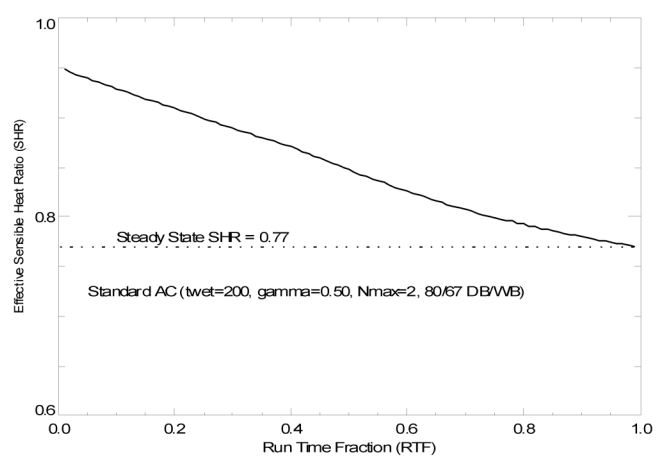

Figure 2 *Part-load function determining the variation of SHR with runtime fraction.*

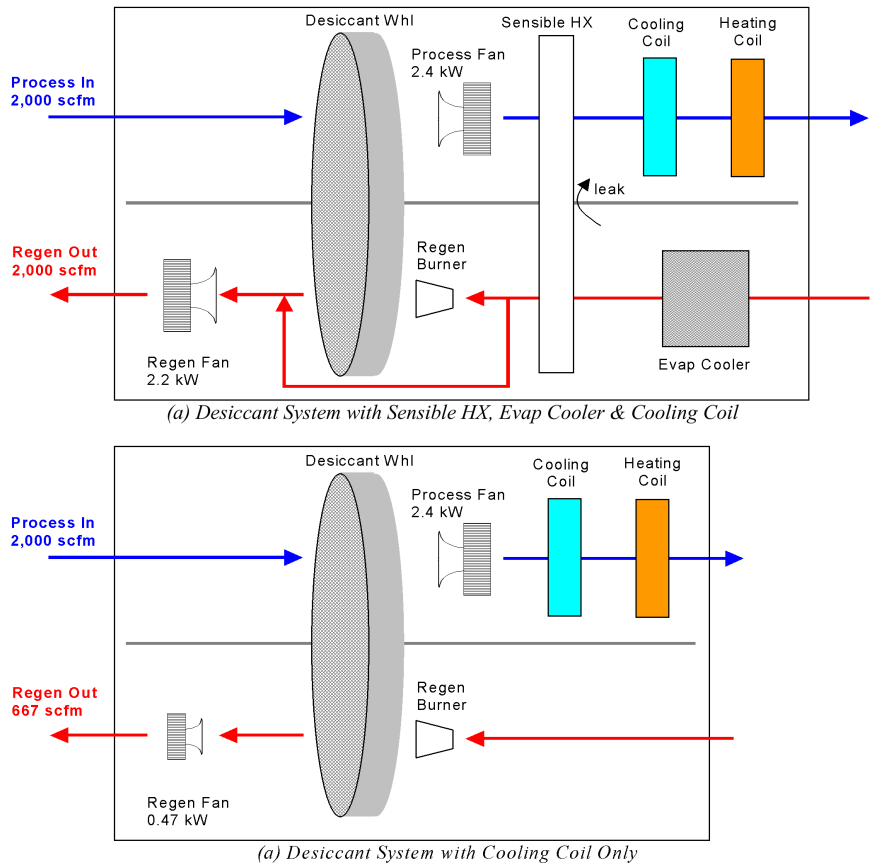

Figure 3 *Desiccant system with and without sensible HX and evaporative cooler.*

figure shows the part-load latent function at entering conditions of 80°F DB / 67°F WB (26.7°C / 19.4°C), or 50% RH. In the model, the parameters t_{wet} and γ are calculated for each hour based on the current temperature and humidity entering the cooling coil. The latent capacity of the air conditioner during the "on" cycle is calculated with the steady-state performance map while the evaporative capacity of cooling coil during the "off" cycle is calculated, assuming the coil is an evaporative cooler.

Desiccant Unit

The model of the desiccant unit can optionally contain several components, including a desiccant wheel, regeneration burner, sensible heat exchanger (HX), fans, and a DX post-cooling coil. Figure 3a schematically shows a desiccant system that includes all of these components. The performance model of the desiccant wheel was based on field measurements (Henderson et al. 2002) as well as the algorithms from the DesiCalc tool. Figure 4 compares the resulting trends for the measured data and the DesiCalc model. The regeneration energy input is the heat required to regenerate the wheel per mass of moisture removed from the airstream. The line on the plot is a best fit regression model to the measured data. The desiccant model based on measured data should only be applied in applications with a similar equipment configuration as the test site and with similar entering conditions. Because of these limitations, the DesiCalc algorithm was used for the analysis presented in this paper. The desiccant subsystem also includes a sensible heat exchanger model based on a fixed effectiveness. Thermal and moisture leakage across the sensible heat exchanger is also simulated by a leakage factor. The factor indicates the fraction of the total process airflow that leaks from the regeneration side to the process side of the HX. The total heat and moisture transfer rate depends on the leakage flow and the conditions on the process outlet and regeneration inlet sides of the HX. The saturation effectiveness of the evaporative cooler is specified as constant value. The psychrometric calculations and iterations necessary to solve for the state of the system are completed at each time step. The gas consumption is reduced or increased in proportion to the calculated temperature entering the regeneration burner (the DesiCalc algorithm for gas consumption assumes the regeneration burner inlet temperature is the same as the process inlet). The post-cooling coil uses the DX coil model described above. The post-cooling coil can be controlled to (1) operate whenever the desiccant wheel operates, (2) operate when the desiccant wheel is on and the ambient temperature is above a specified setpoint, or (3) operate to maintain a desired supply air temperature.

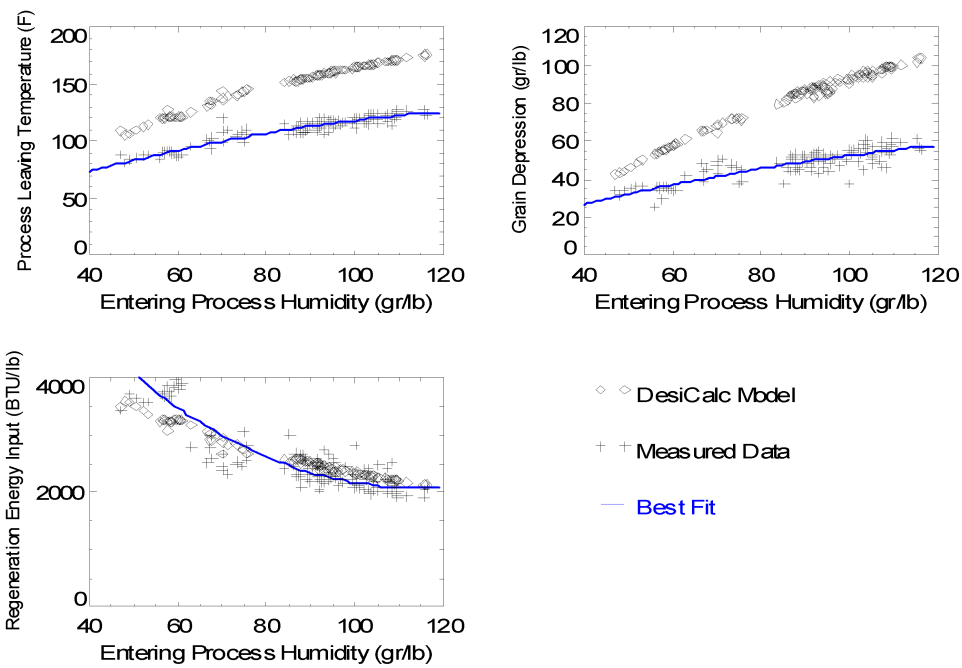

Figure 4 *Performance maps used for desiccant wheel (measured data and DesiCalc).*

Desiccant unit power and gas use are separately accounted for in order to consider various control options. For instance, regeneration fan energy and gas use for each hour are proportional to the dehumidification runtime fraction (RTF_d). Power use of the post-cooling DX coil is proportional to its own runtime fraction, which is controlled based on the criteria described above. RTF_{df} similarly determines the process fan energy use.

The dehumidification/ventilation pretreatment unit can also be one or more of the following:

1. A "null unit" that provides ventilation airflow with no fan power (used for the Base AC).
2. An enthalpy wheel unit with separately controlled process and exhaust fans.
3. A fresh air pretreatment unit based on mechanical refrigeration.
4. Any other electric, gas-fired, or hybrid unit that provides ventilation pretreatment and/or dehumidification in the recirculation mode.

For each option, fan power can be assigned to cycle based on a call for dehumidification or to operate continuously based on occupancy or other factors.

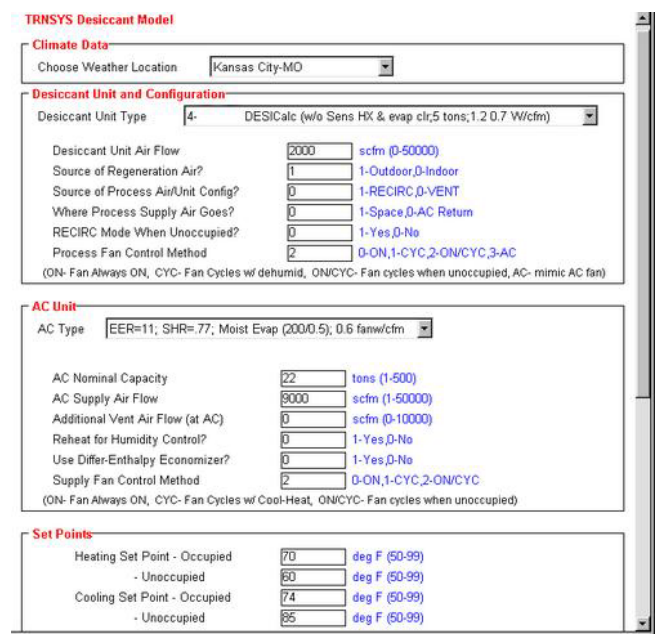

Figure 5 *Portion of user interface for hybrid desiccant simulation tool.*

User Interface

Figure 5 shows a portion of the user interface for the resulting simulation tool (Henderson and Ciolkosz 2002). The available user inputs are focused on the issues associated with HVAC system configuration shown in Figure 1. The user inputs associated with the building envelope (not shown on Figure 5) are more basic parameters such as building area, fraction of each wall with glazing, and building application type (school, office, retail establishment). The details associated with each building application also include internal loads, occupancy and setpoint schedules, as well as other key characteristics.

TABLE 1
Summary of School Building Characteristics Used in the Model

Building	
Gross Floor Area:	5,841 ft^2 [543 m^2]
Building Dim:	60 ft x 97 ft x 10 ft (high) [18 m x 30 m x 3 m]
Exterior Walls:	8 inch [20 cm] light weight concrete block (w/ 1 inch [2.5 cm] of insulation), ASHRAE Wall #65[1]
Roof:	1 inch [2.5 cm] wood deck with 2 inches of insulation, Light Color, ASHRAE Roof #4[1]
Floor:	4 inches [10 cm] of heavy weight concrete, ASHRAE Floor #16[1]
Glazing:	10% of West Wall Area (97 ft^2 [9 m^2]) Single Pane Glass, U=1.04 Btu/h-ft^2-°F [0.2 W/m^2-°C]
Occupied Period:	8 am to 3 pm each weekday of school year (summer vacation May 26 to August 16)
Occupancy:	70 students during day, none at night (190 Btu/h sensible [55 W], 190 Btu/h latent [55 W], 0.2 liter/min CO_2 generation)
Infiltration:	0.1 ACH with supply fans off
Lights:	2 Watts/ft^2 [0.2 W/m^2] occupied, 0.5 Watts/ft^2 [0.05 W/m^2] unoccupied
Equipment:	1 Watts/ft^2 [0.1 W/m^2] occupied, 1.0 Watts/ft^2 unoccupied
HVAC	
Cooling Capacity: Supply Airflow:	22 tons [77 kW] (7 HPs lumped together) 9,000 cfm [4,250 l/s]
AC Fan Power	0.6 Watts/cfm [1.2 W-s/l]
AC Coil:	11.0 Btu/Wh [3.2 W/W] & 0.77 SHR at nominal conditions
Desiccant Wheel:	DesiCalc Algorithm
Desiccant Airflow:	2,000 cfm [943 l/s] ventilation
Sensible HX:	Heat pipe, 60% effectiveness
Evap Cooler:	92% saturation effectiveness
Cool Setpoints:	74°F [32°C] occupied, 85°F [29°C] unoccupied
Heat Setpoints:	70°F [21°C] occupied, 60°F [16°C] unoccupied

Notes:1 - Numbers refer to transfer functions listed in Chapter 25 of 1977 ASHRAE Fundamentals, Table 26 for roofs, Table 27 for walls, and Table 29 for floors.

SIMULATION RESULTS

Model Verification

The simulation model was verified by comparing its output to measured results from the desiccant test site at an Olathe, Kansas, school (Henderson et al. 2002). The measured and observed characteristics of the building are summarized in Table 1. The test site included identical areas of the school that used conventional AC systems as well as hybrid desiccant systems. The heating and cooling load lines for the conventional AC predicted by the model were found to compare well to the measured trends. The trend of space humidity with ambient for both the model and the measured data were shown to be similar. For the conventional AC, the model predicted humidity levels of 60% to 65% RH, which were in good agreement with measured results for the base case pod. Overall, the model was able to accurately predict the energy use and space humidity trends observed and measured at the site.

Comparing Conventional AC and Desiccant Options

Table 2 and Figure 6 compare the simulation runs for the base AC system and desiccant systems as they were configured at the test school (with a heat pipe sensible HX). The DesiCalc model was used for all of the desiccant simulation results in this section because it is more flexible than the model based on measured data from the test school. The base AC (System #1) is assumed to provide cooling to the space during the occupied periods from 7:00 a.m. to 3:00 p.m. However, space humidity levels are not directly controlled, so the space RH reaches as high as 65%, with more than 200 occupied hours over 50% RH. The desiccant-based ventilation pretreatment system with an occupied setpoint of 40% RH (System

TABLE 2
Comparing Simulated Results for Base Case AC and Desiccant Systems

	System #1	System #2	System #3	System #4	System #5
	Base AC	Desiccant System (Vent Mode)	Desiccant System (recirc when UNOCC)	Desiccant System (Recirc Mode)	50% Smaller Desiccant System (Recirc Mode)
Supply Fan Use (kWh)	10,021	10,022	10,302	10,045	10,046
AC Comp Use (kWh)	6,123	5,384	5,271	5,128	5,480
Des Unit Gas Use (therms) [m^3]	-	474 [16,740]	520 [18,360]	530 [18,720]	490 [17,300]
Des Unit Electric Use (kWh)	-	1,879	2,165	2,341	1,663
Des Fan Electric Use (kWh)	-	4,342	4,579	792	562
Annual HVAC Costs	$1,130	$1,822	$1,900	$1,626	$1,561
AC Comp Runtime (h)	306.1	279.6	274.7	257.8	274.3
Economizer Runtime (h)	-	-	-	-	-
Heating Runtime (h)	149.9	140.2	140.3	151.0	150.9
ReHeat Runtime (h)	-	-	-	-	-
Supply Fan Runtime (h)	1,855.7	1,856.0	1,907.8	1,860.3	1,860.3
Dehumid Runtime (h)	-	257.2	292.2	330.0	468.6
Des Fan Runtime (h)	1,809.0	1,809.0	1,907.8	330.0	468.6
Hours Above 50% - All Periods	1,397	281	-	300	562
Hours Above 50% - OCC Periods	210	-	-	-	33
Max Humidity - All Periods (%)	65.8	57.4	42.1	57.4	57.4
Max Humidity - OCC Periods (%)	65.5	43.2	42.1	46.6	54.2

Notes: 1 – Annual HVAC costs based on $0.07/kWh and $0.65/therm. Energy costs for space heating are not included.
2 – 'AC Comp Use' includes compressor and condenser fans
3 – 'Des Unit Electric Use' includes regeneration/exhaust fans, post-cooling compressors, and any other power that cycles on and off with a call for dehumidification.
4 – 'Des Fan Electric Use' includes process/vent supply fan power for the module.

#2) significantly reduces the humidity level, at least during the occupied period. However, just as was observed at the test site, humidity levels on nights and weekends were uncontrolled.

Total energy costs are 61% higher for the desiccant system in the vent mode compared to the base AC system. The comparison of measured data from different sections of the test school for a brief period in the summer had implied a cost premium of 40% for the same desiccant system arrangement. The cost breakdown in Figure 6 shows that the AC energy use actually decreases with desiccant system since the post-cooling DX coil picks up a portion of the sensible cooling load. The higher costs are due to the gas use of the desiccant dehumidifier, the added electricity use for the regeneration fan and post-cooling DX coil, and the additional process fan power. A key portion of the added costs is the process fan power required to bring ventilation air through the desiccant system year-round. While the desiccant wheel was required to operate about 257 hours per year, the 2.4 kW process fan motor was required whenever the space was occupied (about 1,810 hours per year in this application). If the humidity setpoint increases to 50% RH for the desiccant system, then the cost premium drops to 38% and dehumidification runtime drops to only 70 hours.

As demonstrated from the field-measured data (Henderson et al. 2002), one main concern with the ventilation pretreatment approach in a school was the inability to control space humidity levels during the weekends and extended summer break. One solution would have been to add return ductwork and dampers to allow the desiccant system to pull process air from the space during the unoccupied periods, as shown by the center diagram in Figure 7. This configuration would allow the desiccant unit to dehumidify the space for those periods without drawing in unneeded fresh air. The simulation results for this option are shown as System #3 in Figure 7 and the third bar in Figure 6. Comparing Systems #2 and #3, the desiccant unit was only required to operate an addi-

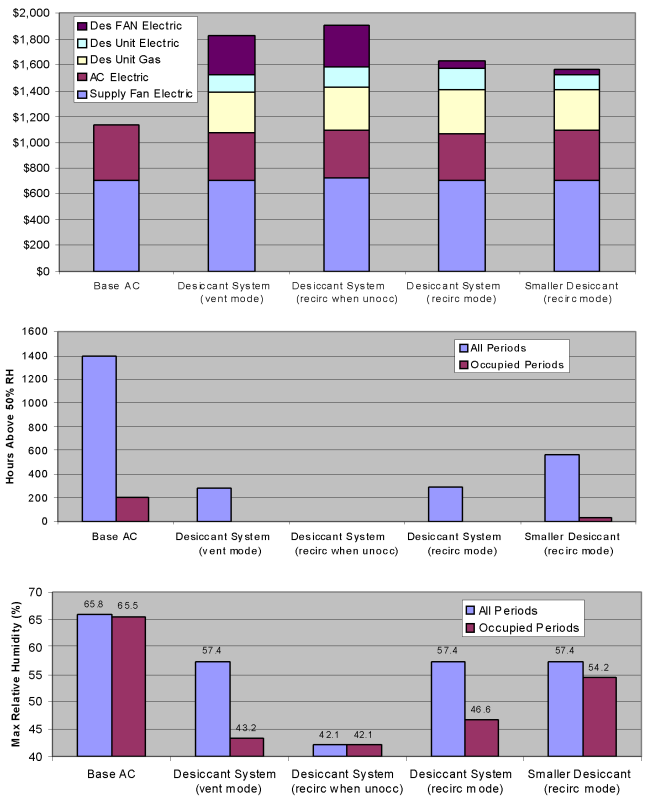

Figure 6 *Comparing simulation results for base AC and desiccant systems.*

tional 35 hours per year to maintain the space at 40% RH for weekends and throughout the prolonged summer break. The assumed infiltration rate of 0.1 ACH for the unoccupied period was determined using measured CO_2 decay data from the field-tested school. Overall, the added cost to fully dehumidify the space during the unoccupied periods was about $78 per year (or 4% of operating costs).

As discussed above, desiccant systems have consistently been shown to have the highest efficiency and greatest capacity in the ventilation mode. The moisture removal capacity is highest because the entering process airstream is most humid. However, other system issues, such as desiccant unit fan power, might overwhelm the performance benefit of using the desiccant wheel to pretreat ventilation air. Systems #4 and #5 in Table 2 and Figure 6 show that, in spite of poorer desiccant wheel performance, a desiccant system in the recirculation mode can have lower operating costs than the desiccant unit in the ventilation mode. This equipment configuration is shown in the bottom diagram in Figure 7. Even though the dehumidification runtime of the desiccant wheel increases by 28% and regeneration gas use increases by 12% in the recirculation mode, the overall annual operating costs of this approach are lower by $196 (compared to System #2). The most significant change is that the desiccant unit process fan can now cycle with the desiccant wheel. This reduces the annual desiccant fan power by a factor of 5 and makes this "thermally inefficient" desiccant system configuration more attractive.

System #5 shows the results for a 50% smaller desiccant system in the recirculation mode (i.e., with a 50% smaller wheel, process flow rate, regeneration flow rate, and post-cooling coil). The annual operating costs for the smaller unit are $65 lower. The humidity trends show that the smaller desiccant unit did not maintain the same level of humidity control at peak conditions though the number of hours that the system lost control of humidity were fairly modest (i.e., only 33 occupied hours over 50%). While the humidity control results for the smaller desiccant unit were not as good, it is still interesting that a 50% smaller (and presumably 50% lower cost) desiccant unit could provide some degree of humidity control. Precise humidity control in this type of commercial application is not a requirement. It may be sufficient to simply provide better control than can be offered by the base case AC. The main goal from a school's point of view would be to consistently maintain the space below 60% so that mold and mildew growth are controlled.

The Impact of Desiccant System Components and Control Options

Desiccant equipment manufacturers usually include sensible heat exchangers and regeneration-side evaporative coolers to cool the process air to partially negate the heat added by the desiccant wheel. These components are usually very beneficial at design conditions, using ambient air to provide significant amounts of "free cooling." Figure 3a shows this arrangement of components in a desiccant system. The sensible heat exchanger also preheats the regeneration air before it enters the gas burner, thus reducing the amount of fuel the burner must consume to hold the nominal 250°F regeneration temperature.

While these components clearly reduce energy use and operating costs at design conditions, it is not clear if there is a net annual benefit to adding these components. Figure 3b shows the desiccant system without the evaporative cooler or the sensible heat exchanger. Eliminating the sensible heat exchanger eliminates the need for balanced airflow on the process and regeneration sides of the unit and reduces the required regeneration airflow to about one-third of the process airflow. Not only does the regeneration fan power drop, but the fan power per scfm drops as well since air must be pushed through fewer components. In this case, we assumed that the fan power dropped from 1.1 W/cfm to 0.7 W/cfm (2.2 to 1.4 W-s/l) (based on our experience with field measurements from other desiccant test sites). As a result, the total regeneration fan power dropped from 2.2 kW to 0.47 kW.

Table 3 compares the energy use and operating costs of the various desiccant system configurations. Comparing system configurations A and C in Table 3 shows the impact of using the heat pipe heat exchanger compared to the more effective (but leakier) sensible heat wheel. The dehumidification runtime of the desiccant wheel with the heat pipe (config-

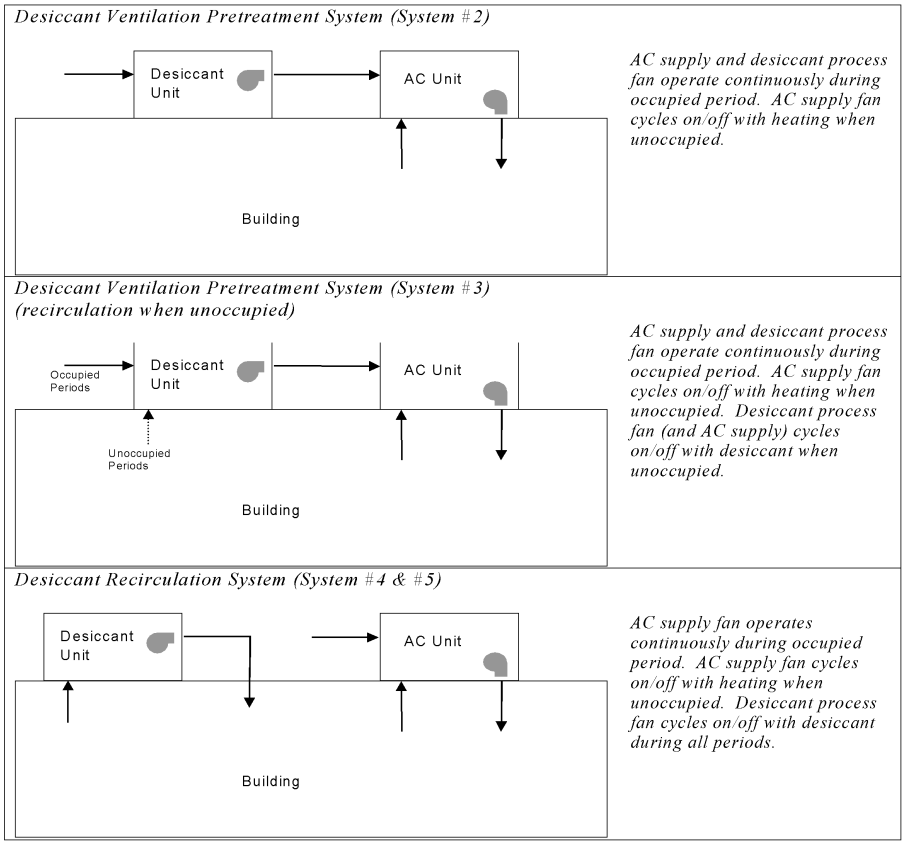

Figure 7 *Schematic of desiccant system configuration options.*

TABLE 3
Impact of Desiccant System Components

	Description	AC Electric (kWh)	Desiccant Unit			Des Fan Electric (kWh)	Annual Cost ($)
			Runtime (h)	Gas Use (therms) [m^3]	Electric (kWh)		
A	Desiccant & SH Whl (w/ SH Whl, Evap Clr & Post Cool)	4,941	273.8	436 [15,400]	1,898	4,342	$1,767
B	Desiccant & SH Whl—NO Evap Cooler (w/ SH Whl & Post Cool)	5,114	272.0	428 [15,110]	1,929	4,342	$1,776
C	Desiccant & Heat Pipe ***[same as System #2]*** (w/ HP, Evap Clr & Post Cool)	5,384	257.2	474 [16,740]	1,879	4,342	$1,822
D	Desiccant & Heat Pipe—NO Evap Cooler (w/ HP & Post Cool)	5,504	257.6	464 [16,400]	1,908	4,342	$1,826
E	Desiccant—Post Cooling ONLY (Run #4)	6,655	263.2	658 [23,240]	1,551	4,342	$2,007

Notes: SH Whl: Sensible Heat Wheel is 80% effective with leakage rate of 8% (heat and moisture).
HP: Heat Pipe is 60% effective with no leakage.
Evap Clr: Evaporative cooler with 92% saturation effectiveness
Post Cool: cooling DX coil is 5 tons [17 kW] with nominal EER of 10.5 Btu/Wh [3.1 W/W] and SHR of 0.80.
Costs based on $0.07/kWh and $0.65/therm. Costs do not include heating costs
Desiccant Wheel: DesiCalc algorithm for wheel with 75%/25% process/regen spilt

uration C) is 15 hours shorter since the moisture leakage of the sensible wheel is eliminated and the system provides more net latent capacity. However, the less effective heat pipe also provides less preheating of the regeneration inlet air, so more gas use is required to maintain the desired regeneration temperature. Overall, the heat pipe heat exchanger option increased operating costs by more than $55 per year compared to the sensible heat wheel.

The regeneration-side evaporative cooler provides only a modest benefit across the year (less than $10 in annual operating costs). Removing the evaporative cooler did increase post-cooling energy use and AC energy use for both the heat pipe and the sensible wheel systems. Removing the evaporative cooler also reduced gas use since less gas heating was required because the entering temperature to the burner was higher. For the sensible heat wheel, the elimination of the evaporative cooler also reduced the amount of moisture leakage from the regeneration inlet to the process output. Reducing this latent capacity degradation also slightly decreased the dehumidification runtime of the desiccant unit, which further reduced the gas use.

One way to make the desiccant system much simpler (and potentially lower cost) is to remove the sensible heat exchanger and the evaporative cooler. This not only removes the components but also significantly downsizes the regeneration fan size and its energy use. Configuration E in Table 2 shows the performance of a desiccant system with only a DX post-cooling coil. As would be expected, more cooling is required by the conventional AC in the space, though the energy use of the desiccant unit itself goes down because the regeneration fan power is substantially lower. The biggest change is the increase in gas use that occurs because the preheating of regeneration air by the sensible heat exchanger is no longer provided. Overall, annual operating costs increase by $180 to $240 when the sensible heat exchange components are eliminated.

Other HVAC Technologies

Other dehumidification and ventilation pretreatment technologies can also provide better humidity control than the base AC system. Table 4 and Figure 8 compare some of these

TABLE 4
Comparing Simulated Results for Basecase AC and Alternative Systems

	System #1	System #2	System #6	System #7	System #8
	Base AC	Desiccant System (vent mode)	AC with Reheat (cond heat)	Enthalpy Wheel (0.3 W/cfm)	Liquid Desiccant (0.6 W/cfm)
Supply Fan Use (kWh)	10,021	10,022	10,024	10,025	10,020
AC Comp Use (kWh)	6,123	5,384	10,180	5,758	4,802
Des Unit Gas Use (therms) [m^3]	-	474 [16,740]	-	-	-
Des Unit Electric Use (kWh)	-	1,879	-	766	3,195
Des Fan Electric Use (kWh)	-	4,342	-	1,085	2,171
Annual HVAC Costs	$1,130	$1,822	$1,414	$1,234	$1,413
AC Comp Runtime (h)	306.1	279.6	523.2	295.7	248.4
Economizer Runtime (h)	-	-	-	-	-
Heating Runtime (h)	149.9	140.2	149.6	83.0	145.9
ReHeat Runtime (h)	-	-	44.4	-	-
Supply Fan Runtime (h)	1,855.7	1,856.0	1,856.3	1,856.5	1,855.6
Dehumid Runtime (h)	-	257.2	-	1,276.0	373.4
Des Fan Runtime (h)	1,809.0	1,809.0	1,809.0	1,809.0	1,809.0
Hours Above 50% - All Periods	1,397	281	591	704	280
Hours Above 50% - OCC Periods	210	-	32	70	-
Max Humidity - All Periods (%)	65.8	57.4	57.4	57.5	57.4
Max Humidity - OCC Periods (%)	65.5	43.2	56.8	56.9	47.7

Notes: 1 – Annual HVAC costs based on $0.07/kWh and $0.65/therm. Energy costs for space heating are not included.
2 – 'AC Comp Use' includes compressor and condenser fans
3 – 'Des Unit Electric Use' includes regeneration/exhaust fans, post-cooling compressors, and any other power that cycles on and off with a call for dehumidification.
4 – 'Des Fan Electric Use' includes process/vent supply fan power for the module

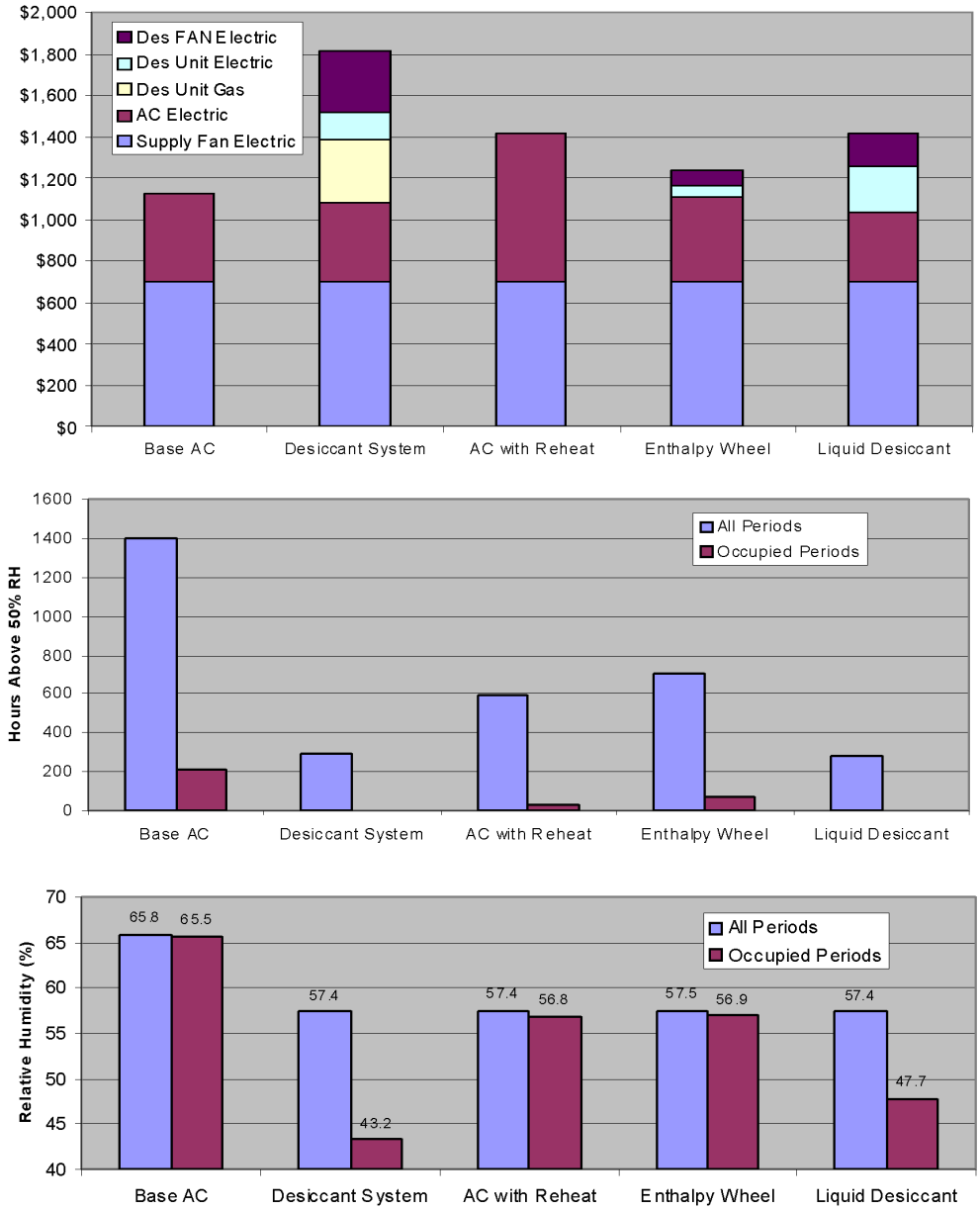

Figure 8 *Comparing simulation results for base AC and alternative systems.*

systems to the base AC and the desiccant system in the ventilation mode. The alternative systems include:

- A reheat system that uses "free" condenser reheat. A humidistat enables the compressor if the space humidity is above a dew point of 48°F (8.9°C) (equivalent to 74°F [23°C] and 40% RH). To prevent overcooling, reheat is enabled to hold the space above 72°F (22°C). No humidity control during unoccupied periods.
- An enthalpy wheel heat exchanger system. This passive desiccant wheel is 85% effective in terms of temperature and humidity. Flow is balanced and the supply and exhaust fans each use 0.3 W per cfm (0.6 W-s/l). The supply fan runs continuously with the AC supply fan when the space is occupied. The regeneration/exhaust fan (and the enthalpy wheel) are controlled to only operate when there is a net benefit (i.e., the exhaust stream enthalpy is higher than ambient in heating mode; space humidity is lower than 45% RH in heating; exhaust enthalpy is lower than ambient in cooling mode).
- A compressor-based liquid desiccant system that uses condenser heat to regenerate the weak desiccant solution. The unit is assumed to pretreat ventilation air. The additional process/ventilation fan power to pull air through the unit is 0.6 W per cfm (1.2 W-s/l). No humidity control during unoccupied periods.

The performance results for each of these systems are discussed below.

The reheat system (System #6)—which is often identified as the standard humidity control system—increases the compressor runtime by 70% compared to the base AC. However, very little reheat is used (only 44 hours) since the space must be overcooled by 2°F (0.6°C) before the heat is activated. The operating costs of this option are much lower than the gas-fired desiccant unit, though the degree of humidity control is also much poorer.

The enthalpy wheel (System #7) is the passive humidity control option that reduces some of the latent load from the fresh airstream before it enters the AC coil. This system reduces the space humidity levels to about the same degree as the reheat system but has the lowest operating costs of all but the base case system. This system also provides the additional benefit of reducing heating costs (though the cost data shown in the table do not reflect this). A key assumption is the added fan power required for the supply and exhaust fans in the enthalpy wheel unit. In this case we assumed that fan power was 0.3 W per cfm (0.6 W-s/l). The intelligent control of the enthalpy wheel reduces the operation of the exhaust fan and decreased its runtime by more than 500 hours compared to the process fan. Continuous operation of the exhaust fan during the occupied period in this case would have increased costs by about $20 per year.

The new compressor-based liquid desiccant system (System #8) has much lower operating costs than the gas-fired desiccant system. The operating cost premium of the liquid desiccant system over the base AC is only 25%, compared to 61% for the gas-fired desiccant (System #2). The unit does not include a supply fan so additional fan power of 0.6 W per cfm (1.2 W-s/l) was added into the system. This unit provides a similar level of humidity control as the gas-fired unit but with a smaller operating cost premium.

The Future of Desiccant Systems

New desiccant units are now being developed and introduced to the market by some manufacturers. Many of these new concepts acknowledge many of the operating cost issues identified in this paper and attempt to overcome many of these issues of unit size and parasitic power. One example is an add-on desiccant system (Fischer and Sand 2002) that has been developed to treat a portion of the supply air from a traditional cooling coil. This desiccant wheel is sized to only treat part of the supply airstream, so it is much smaller than a traditional gas-fired desiccant unit. Because it treats supply air with a relative humidity near 90%, the wheel can be highly effective even at modest regeneration temperatures. It also requires no sensible heat exchangers to increase the cost and size of the equipment, since the entering air is already cool. Another concept recently presented by Harriman and Judge (2002) is a ventilation pretreatment system that uses condenser heat from a traditional compressor-based cooling system to regenerate the desiccant wheel. It is also unique in that the traditional cooling coil provides the first stage of ventilation pretreatment. Then the cool, humid supply air from the coil is adiabatically dried and heated by the desiccant wheel without the need for additional sensible heat exchangers. Both of these new approaches have potential to make desiccant technology more cost-effective in commercial space conditioning applications.

CONCLUSIONS

A user-friendly building simulation model has been developed to evaluate a wide range of conventional, desiccant, and hybrid HVAC systems in a single-zone building application. The software tool focuses on comparisons of various system configurations and control options that have a significant impact on annual operating costs. The model combines detailed performance models of desiccant equipment with accurate models of the conventional HVAC components as well as realistic algorithms to account for the interactions between the systems and building. The result is a computer model that can quickly and accurately predict hourly space humidity levels and annual operating costs for a wide range of dehumidification and ventilation pretreatment options. The model is also unique in that it can track the impact of HVAC control and configuration on space CO_2 levels.

The simulation model was used to evaluate several configuration options based on the application details of a test school in Olathe, Kansas. The desiccant unit had been installed at the test site to pretreat ventilation air. The simulation results showed the following:

- The desiccant unit as installed increased annual operating cost by 61% compared to the base AC without humidity control.
- If the desiccant unit had been configured to control humidity without providing ventilation during unoccupied periods, the increase in operating costs would have been modest (increasing the operating cost premium by 7%).
- Operating the same desiccant unit in the recirculation mode separate from the AC unit reduced operating costs by nearly $196 per year (decreasing the cost premium to 44%). While desiccant capacity and efficiency was poorer in this mode, the large fan power savings negated the desiccant wheel performance penalty.
- Traditional system components such as regeneration-side evaporative coolers provide very little savings and are not to be warranted given their high installation and maintenance costs. Eliminating the sensible heat exchanger did increase operating costs by about $180 to $240 per year, primarily due to higher gas use. However, the high cost of including and packaging this component into a desiccant system may not make it cost-effective in every application.
- Alternative dehumidification and ventilation pretreatment technologies are also available, including con-

denser reheat systems, enthalpy wheels, and compressor-based liquid desiccant systems. These approaches generally have lower operating costs than the traditional desiccant pretreatment approach but do not always provide the same degree of humidity control. The compressor-based liquid desiccant system looks especially promising in applications where precise humidity control is required.

The simulation results show that performance of desiccant systems can vary considerably depending on how the unit is combined into the HVAC system. Basic component and fan control details can also have significant impact on operating costs and the quality of humidity control provided. The potential of each configuration and control approach is expected to vary with the details of each building application. The modeling approach described in this paper provides the means to evaluate various desiccant system configurations and fairly compare them to other established and emerging dehumidification and ventilation pretreatment technologies. This tool can also be used evaluate emerging desiccant technologies to determine how they work compared to other dehumidification technologies.

ACKNOWLEDGMENTS

The authors would like to acknowledge the staff at Briarwood Elementary School and the Olathe School District for their support of this field test. We would especially like to thank Mr. Robert Courtney, the district's energy manager, for the generous support he provided throughout the project. This work was sponsored by the U.S. Department of Energy, Office of Power Technologies, Distributed Energy Resources, under contract DE-AC05-00OR22725 with UT-Battelle, LLC.

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