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The Impact of Desiccant Dehumidification on Classroom Humidity Levels

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ABSTRACT

Desiccant-based dehumidification technologies offer the potential to provide improved IAQ in schools by properly controlling space humidity levels while also providing the 15 cfm (7 L/s) of ventilation air per student required by ASHRAE Standard 62. This paper reports field test results from a school near Kansas City, Kansas, that was retrofitted with a desiccant system. Monitoring equipment was installed to monitor energy use and space conditions in the desiccant-treated classrooms as well as two other similar areas that used conventional HVAC equipment. Measured space humidity levels were shown to be 15 to 20 gr/lb (2 to 3 g/kg) lower in the desiccant area than in other areas with the same ventilation rates. Classroom areas that provided only 5 cfm (2.3 L/s) of ventilation air per student were found to maintain acceptable humidity levels in the space. However, areas that used conventional HVAC equipment to provide 15 cfm (7 L/s) per person were shown to have much higher humidity levels. All studied areas, including the desiccant-treated area, had unacceptably high humidity levels during the unoccupied periods (nights, weekends, and summer break). In order to provide good IAQ in classrooms, humidity control must be provided continuously to minimize the risk of biological contamination. The desiccant unit installed in this test was configured as a ventilation pretreatment system, so dehumidification could not be provided independently of ventilation. For desiccant technology to realize its full potential in schools, packaged systems must be configured and applied to allow for dehumidification during both occupied and unoccupied periods.

INTRODUCTION

Desiccant-based dehumidification technologies offer the potential to cost-effectively control humidity levels in commercial buildings with significant fresh air requirements. This paper presents measured results from a field test of a desiccant system installed to pretreat ventilation air in a school to control classroom humidity levels and improve indoor air quality. The test approach was to monitor the performance of a desiccant system as well as two base-case HVAC systems. This side-by-side test allowed us to assess the impact of desiccant technology on space conditions and total HVAC system energy use.

BACKGROUND

The adoption of *ASHRAE Standard 62-1989, Ventilation for Acceptable Indoor Air Quality* resulted in a three-to-four-fold increase in the amount of ventilation air required in many building applications. These increased fresh air requirements have dramatically increased the moisture, or latent, loads imposed on HVAC systems.

In schools —where occupancy densities are very high and ventilation needs are substantial—the increased ventilation requirements have raised concerns that indoor humidity conditions are not adequately maintained to provide good indoor air quality (IAQ). High humidity conditions can negatively impact human health by increasing the risk of mold and mildew growth. A recent study of Georgia schools (Bayer et al. 2001) demonstrated a link between high humidity in classrooms and elevated levels of metabolic volatile organic compounds (MVOCs), which are linked fungal and microbial growth.

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It is generally accepted that indoor relative humidity levels in the 40% to 60% range are optimal for human comfort and health. At relative humidity (RH) levels above 60%, molds, fungi, and other microorganisms start to have a negative impact (Sterling et al. 1985). In order to address this IAQ concern, humidity should always be maintained below 60% RH.

Desiccant technology offers one of the most cost-effective means to actively control space humidity levels. While conventional cooling technologies do provide some dehumidification, the amount of moisture removal is often less than is necessary to maintain humidity levels below 60% RH. Because conventional equipment provides dehumidification as a consequence of satisfying sensible cooling loads, the space humidity level is not directly controlled. As a result, the space humidity level floats across the day and cooling season and can often drift out of the acceptable range—especially at times when sensible loads are low. In contrast, desiccant dehumidifiers can directly meet latent loads and ensure that the desired humidity setpoint is maintained.

It is generally thought 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 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 also minimizes the need for additional sensible cooling components. Standard desiccant unit components such as sensible heat exchangers and evaporative coolers can often be sufficient to meet the sensible cooling requirements with minimal postcooling required from conventional equipment.

TEST APPROACH

The goal of this project was to field test a desiccant unit in a school application to evaluate its ability to maintain adequate humidity levels. An elementary school in Olathe, Kansas, was selected as the test site. This single-story, 114,000 ft^2 (10,591 m^2) school is divided into multiple teaching areas, or pods. Each pod includes four to six classrooms, a common area, and restrooms. Cooling, heating, and ventilation are provided separately to each pod area. This school was selected because of its location in a moderately humid climate and because the school was physically configured in way that facilitated side-by-side comparisons of desiccant and base-case systems.

A desiccant unit was installed to provide 15 cfm (7 L/s) per student of ventilation air to one of the six classroom pods at the school. A second similar pod was used as the base-case or "control" area for this study. A third pod was also included in the field test because it used an alternate ventilation system (i.e., a fresh air heat pump) to provide 15 cfm [7 L/s] per student.

Site Description

The 3,000 cfm (1416 L/s) desiccant unit was installed at the elementary school in a suburb of Kansas City, Kansas. The single-story, 114,000-ft^2 (10,591 m^2) school was built in 1988. It is divided into four main teaching areas, or pods, as shown in Figure 1. Each pod includes four to six classrooms, a common area, and restrooms. A media center and library are located between the three original pods in the center of the building. The gymnasium is located at the north end of the school. Administrative offices are located near the media center by the front entrance. The fourth pod was added to the south end of the school in 1995. The areas included in this study were Pod #1, Pod#2, and Pod#4. These areas are highlighted on Figure 1.

The school uses a water loop heat pump system to provide space conditioning. Each zone had its own heat pump that independently provides heating or cooling for that space. The heat pumps are mounted above the ceiling near each zone. Conventional thermostats control each heat pump. Heat is rejected from the loop through a closed cooling tower in the summer. In the winter, heat is added to the loop by a natural gas boiler. The water loop plant (tower, boiler, and pumps) is operated by a simple control system that keeps the loop temperature within the specified limits of 60°F to 90°F (16°C to 32°C).

In the original system, fresh air was provided to the pods by three "passive" intake grilles on the roof of the building. Fresh air is ducted from the roof-mounted intake grille to the return side of each heat pump, as shown in the top of Figure 2, to nominally provide about 5 cfm (2.4 l/s) per student (as required by code when the school was built in 1988). Ventilation airflow is passively induced through the oversized ductwork by the supply fan on each heat pump as well as by exhaust fans in the zones. The fresh air system is configured so that one intake grille serves each of the three original pods, as shown in Figure 1.

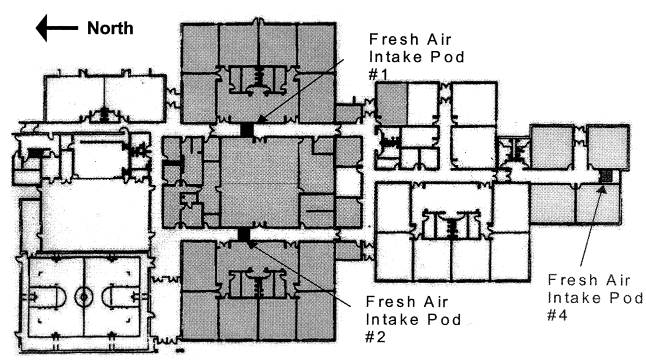


Figure 1 *Elementary school floor plan and ventilation system layout.*

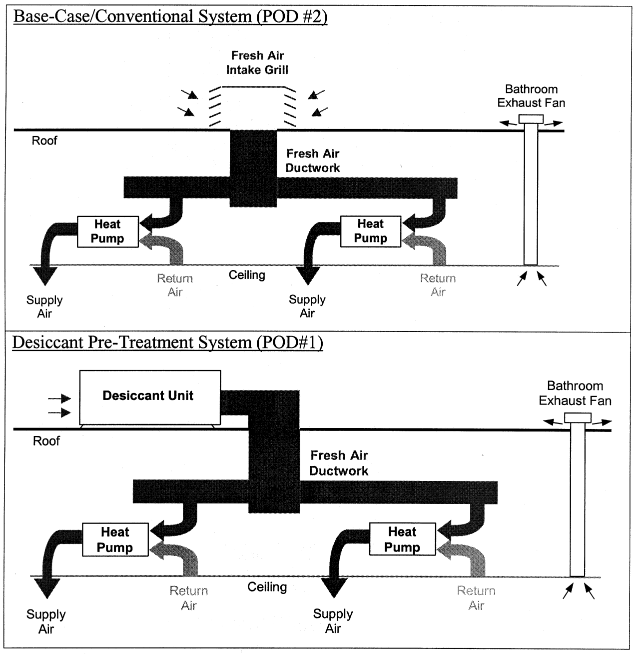


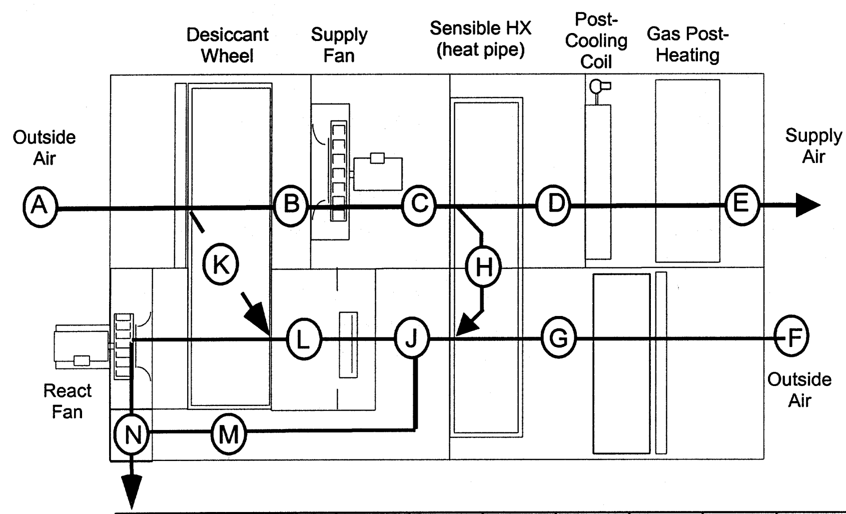
Figure 2 *Schematic of base-case and desiccant HVAC systems.*

A desiccant unit was installed on Pod #1 at the east side of the school in April 1999. The passive air intake grille for that pod was replaced with desiccant unit ductwork, as shown on the bottom of Figure 2. The desiccant system pretreats the fresh air for Pod #1 and the neighboring areas. As shown in the unit schematic in Figure 3, the desiccant unit includes a 5-ton DX cooling coil and condensing unit to provide post-cooling after the sensible heat exchanger on the process side of the unit. The regeneration side of the unit also includes an evaporative cooler to precool air entering the sensible heat exchanger.

The performance of the desiccant unit is given in the flow schematic of Figure 3. In the dehumidification mode at design conditions (87°F and 118 gr/lb [31°C and 17 g/kg] ambient), the desiccant unit can supply 3,000 scfm (1416 sl/s) of temperature-neutral dry air 75°F and 57 gr/lb [24°C and 8 g/kg], or 44% RH.

Pod #4 used a conventional water-source heat pump as ventilation pretreatment system (configured with the heat pump in the same position as the desiccant unit in Figure 2). This alternative ventilation pretreatment system was designed to provide 15 cfm (7 L/s) per student (since it was built in 1995). The controls were set up to operate when the ambient temperature was above 85°F (29°C). Since the heat pump's cooling coil treated 100% of outdoor air, the coil's moisture removal capacity was very high at design conditions (i.e., a sensible heat ratio near 0.5). Electric resistance heat elements were used to temper ventilation air in the winter.

The heat pumps and ventilation system were controlled by a central time clock. The time clock controller enabled each heat pump during the occupied period from 6 a.m. to 3 p.m. each weekday. The time clock controls were also tied into ventilation dampers that sealed off the ventilation ductwork during the unoccupied periods. When the desiccant unit was installed, it was tied into the time clock as well so that venti-



SUMMER	A	B	C	D	E	F	G	H	J	K	L	M	N
SCFM	3307	3107	3107	3000	3000	3000	3000	107	3107	200	1320	1787	3307
DEGREES DRY-BULB	87	135	139	92	75	87	77	139	125	213	256	125	130
GR / LB	118	57	57	57	57	118	132	57	129	103	150	129	194

WINTER													
SCFM	2390	2390	2390	2390	2390	0	0	0	0	0	0	0	0
DEGREES DRY-BULB	42	42	46	46	123	-	-	-	-	-	-	-	-

Figure 3 *Schematic desiccant unit.*

lation was only supplied during the occupied period. As a result, dehumidification could only be provided during the occupied period.

Instrumentation

The desiccant and base-case areas of the school were fully instrumented to measure system performance, energy use, and indoor air quality. Three dataloggers were installed to collect the required data. One-time measurements and other tests were taken to complement the continuously monitored data (collected at five-minute intervals). The measured data from the desiccant, base-case, and fresh air heat pump systems were directly compared to determine the energy impact and indoor air quality benefits associated with the desiccant system. More than 100 data points were collected throughout the monitoring period from March 1999 to October 2000. The data points included

- space temperature, humidity, and CO_2 level in the classroom and common areas in each pod;
- ventilation flow rates for each pod or system;
- ambient temperature and humidity;
- power and operating (heating/cooling) status for all heat pumps in each pod;
- desiccant unit power and gas use, operating status, flow rates, and operating temperature/humidity; and
- power use, operating status, flow rate, and operating temperature and humidity for the base-case ventilation systems.

RESULTS AND DISCUSSION

The initial months of testing revealed various problems with the desiccant unit that were subsequently analyzed and corrected. The most significant change was the replacement of the sensible heat wheel with a heat pipe. The sensible heat wheel was found to degrade the latent capacity of the system by transferring moisture from the regeneration to the process side of the system (i.e., acting as a moderately effective enthalpy wheel). In April 2000, the sensible heat wheel was replaced by a heat pipe heat exchanger. The heat pipe had a slightly lower heat transfer effectiveness but did not allow any moisture transfer from the regeneration to the process side of the unit. The results reported in this paper are based on data collected after April 2000 when the desiccant unit was operating as expected (with the heat pipe).

Ventilation Measurements

The ventilation airflow rate into each classroom area was determined by a number of means, including the following:

1. multiple-point velocity traverses in system ductwork
2. energy balance calculations on systems in the heating mode
3. tracer gas decay tests using artificially introduced CO_2

A complicating factor in Pods #1 and #2 was that only a portion of the fresh air entering through the ventilation system was supplied to actual pod area (the balance of the air went to other areas, as shown in Figure 1). Table 1 lists the ventilation rate determined for each system and area. The measured fraction of the ventilation air from each system that was supplied to the respective pod areas was generally in line with the fractions given in the design drawings. The desiccant unit airflow was 1,000 cfm (472 L/s) lower than the design value of 3,000 cfm (1416 L/s) for the summer of 2000 after the heat pipe was installed.[1] This also lowered the per person ventilation rate from 15 cfm (7 L/s) to 10 cfm (5 L/s).

The ventilation flow rate into each pod was measured using CO_2 as a tracer gas in a decay test. The techniques for using CO_2 as a tracer gas from Persily (1997) and the tracer gas test standard (ASTM 1995) were used to estimate the effective ventilation rates. CO_2 was artificially introduced into each zone to achieve levels well in excess of 2,000 ppm. Then the exponential decay in CO_2 levels was recorded at short time steps with the installed sensors. All tracer gas tests took place overnight with the space unoccupied but with all fans and equipment operating as during the occupied mode. The tracer gas test approach was also used in Pod #4, where the ventilation rate to the classroom area was already known, to confirm the accuracy of the method. The tracer gas measurements were within 8% of velocity-based readings, confirming the validity of the method.

Comparing Pod Humidity Levels

The desiccant unit, heat pumps, and other ventilation equipment at the school all operated based on a time clock for

TABLE 1
Comparing Ventilation Airflow Rates Determined for Each Area

	Total SYSTEM Ventilation Airflow (scfm) [sl/s]	Ventilation Airflow Into POD AREA (scfm) [sl/s]	Ventilation Provided Per Student (scfm/p) [sl/s]
Pod #1—Desiccant unit	~2,000 [944]	1,157 [546]	9.6 [4.5]
Pod #2—Base-case	1,062 [501]	471 [222]	3.9 [1.8]
Pod #4—Fresh air HP	1,510 [713]	1,510 [713]	15.1 [7.1]

Notes: Pod #1 and #2 occupancy averaged 120 students in six classrooms.
Pod #4 occupancy average 100 students in four classrooms.

[1] The lower ventilation rate was selected after April 2000 to ensure that the modified system had sufficient dehumidification capacity to hold the 40% RH setpoint. Subsequent testing did imply the desiccant system with the heat pipe could have also held conditions at 3,000 cfm (1416 L/s).

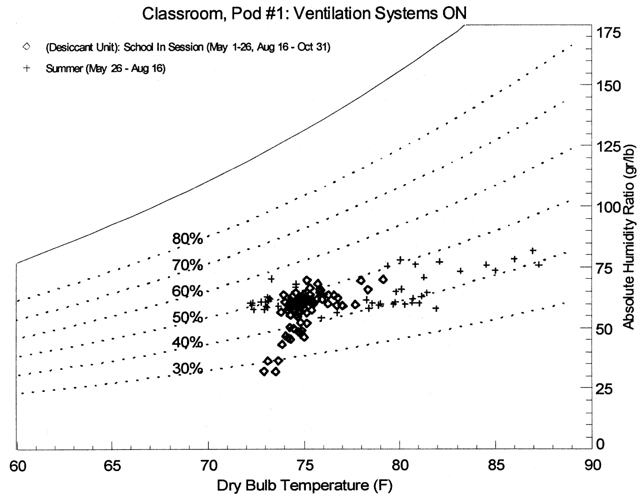


Figure 4 *Daily space conditions in Pod #1 (desiccant unit with 10 cfm [5 L/s] per person).*

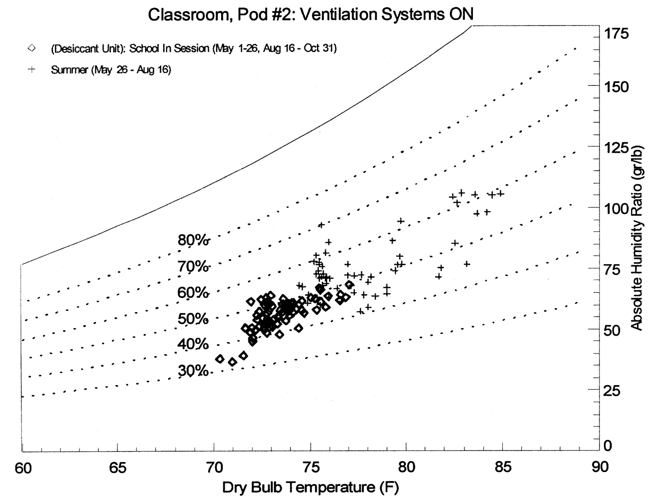


Figure 5 *Daily space conditions in Pod #2 (base-case with 4 cfm [2 L/s] per person).*

nine hours each weekday (6 a.m. to 3 p.m.). During this period, the desiccant unit in Pod #1 cycled on and off to maintain the space near the 40% RH humidistat setpoint. The heat pumps in both pods operated to maintain a temperature setpoint of 75°F (24°C). In Pod #2, the only dehumidification was passively provided by the heat pumps. The fresh air heat pump in Pod #4 provided some precooling and dehumidification of ventilation before it was introduced into the space.

Figures 4, 5, and 6 display the observed space conditions on a psychrometric chart for all three areas during the 2000 cooling season. The data are the daily average values during the 6 a.m. to 3 p.m. period. The points are shown with different symbols to distinguish between summer break and school year operation. The summer period included days when space conditions were controlled to the same temperature as during the school year as well as some days when the heat pumps were set up to a higher temperature.

During the school year, the desiccant unit operated maintained relative humidity levels in the area near the setpoint of 40% RH (see Figure 4). Even during the summer period when space temperatures reached as high as 87°F (31°C), the relative humidity in the classroom seldom exceeded 50%.

During the school year, the base-case pod maintained only slightly higher humidity levels compared to the desiccant area, as shown in Figure 5. The low humidity levels were due to the much lower ventilation rate in this area (4 cfm [2 L/s] per student). The main difference between the two systems was their ability to maintain space humidity levels in the summer. During the summer break when the heat pumps were often set up to a high setpoint, space conditions approached 70% RH and absolute humidity levels reached 125 gr/lb (18 g/kg) in Pod #2. The lack of sensible cooling load caused space absolute humidity levels to approach ambient conditions.

In Pod #4, where ventilation rates were confirmed to be 15 cfm (7 L/s) per person, space humidity levels were noticeably

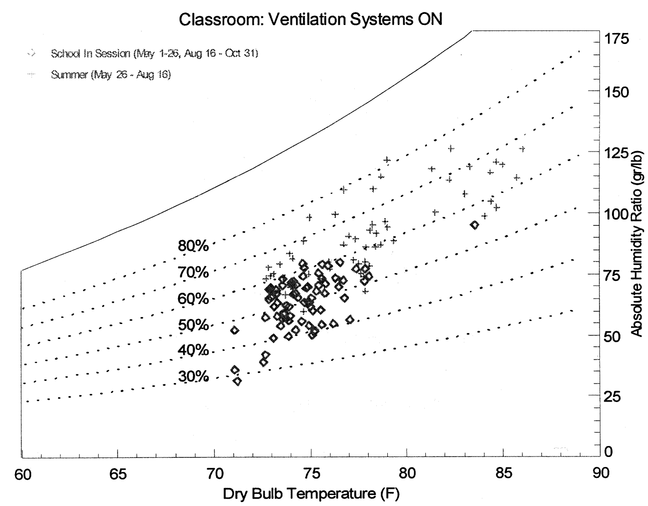


Figure 6 *Daily space conditions in Pod #4 (fresh air HP with 15 cfm [7 L/s] per person).*

higher than in the other areas (see Figure 6). During the school year, the classroom humidity levels often exceeded 60% RH. During the summer break, when the classroom heat pumps were off but the ventilation system remained on, space humidity levels reached as high as 80% RH. This demonstrates the IAQ risks associated with using conventional equipment with ventilation rates of 15 cfm (7 L/s) per student.

Another useful way to understand and compare the space humidity trends of the different areas is to plot the space and ambient absolute humidity against each other. Figure 7 shows this plot for the classroom in Pod #1. The desiccant unit was able to maintain 60 gr/lb (9 g/kg) in the space during the school year. The trend of space humidity versus ambient shows a well defined trend. Below 50 gr/lb (7 g/kg) ambient, no dehumidification is required so space conditions tend to track ambient conditions. At lower ambient humidity levels, the impact of occupant-generated moisture on space humidity levels becomes more apparent (i.e., the space is 10-15 gr/lb [2-3 g/kg] higher than ambient on the driest days).

During the summer break, several other factors come into play in Pod #1. Space temperatures were high for a few summer days because of no cooling operation, though the desiccant unit continued to maintain the space relative humidity level near their setpoint of 40% RH. At the higher space temperature, the same relative humidity translated into a higher absolute humidity. Days with the heat pumps (HPs) off correspond to the dotted line in Figure 7. When the heat pumps did operate normally during summer break to control temperature, the space humidity was held below 65 gr/lb (9 g/kg) on the most humid days.

Figure 8 shows the same plot for Pod #2, where the base-case system provided only 4 cfm (2 L/s) of ventilation. At these low ventilation rates, a very similar humidity trend is apparent during the school year when using just the conventional HPs for dehumidification. The main difference with this system is the lack of humidity control during summer break on days when cooling was disabled. The space humidity trend shows a significant degree of scatter during these days with the heat pumps off. When the heat pumps did run, the humidity was maintained near 60 to 70 gr/lb (9 to 10 g/kg) (or around 50% RH).

Figure 9 shows the trend for Pod #4, where the dedicated fresh air HP was used to precondition the 15 cfm (7 L/s) per person of ventilation air. At the higher ventilation rate, this conventional system was not able to provide the same degree of humidity of control during either the school year or summer break. During the school year, the space humidity reached 80 gr/lb (11 g/kg), or about 15 to 20 gr/lb (2 to 3 g/kg) higher than was maintained in the desiccant pod. During the summer break, when the HPs were off but the ventilation system continued to run, space humidity levels often approached ambient conditions (and reached as high as 130 gr/lb [19 g/kg]). For days during the summer when the HPs did operate to maintain the 75°F (24°C) setpoint, the space humidity was held below 80 gr/lb (11 g/kg) (60% to 62% RH).

Unoccupied Humidity Levels

Applying a desiccant unit to pretreat ventilation air is desirable because it allows the unit to provide the highest latent capacity at the best efficiency. However, this configuration makes it difficult to maintain humidity levels during the unoccupied periods when ventilation is not required. Figure 10 shows what happened to space conditions over a hot, humid

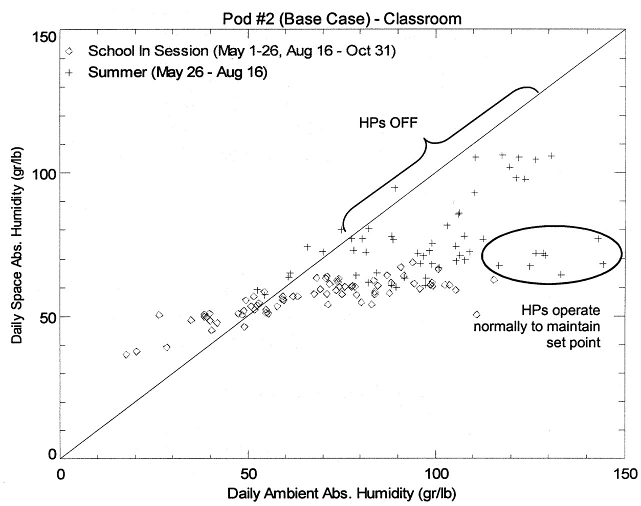


Figure 8 *Trend of space and ambient humidity in Pod #2 (base-case with 4 cfm [2 L/s] per person).*

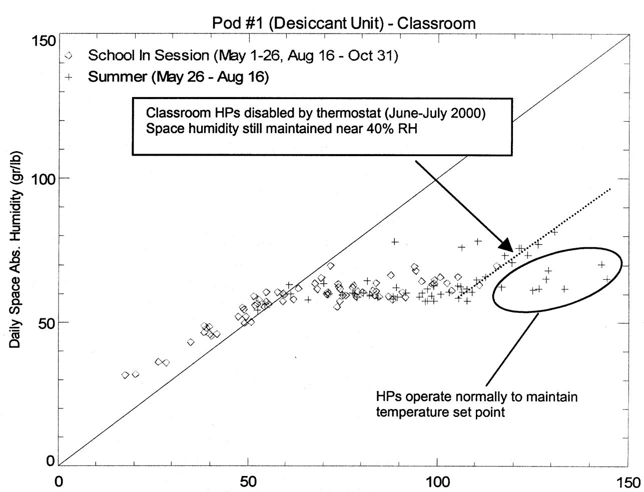


Figure 7 *Trend of space and ambient humidity in Pod #1 (desiccant with 10 cfm [5 L/s] per person).*

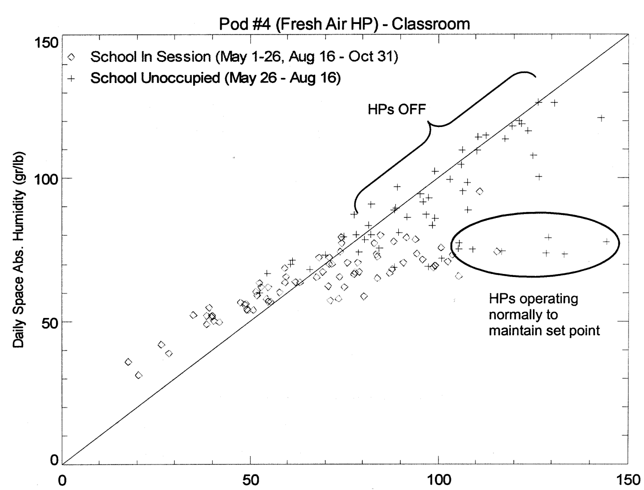


Figure 9 *Trend of space and ambient humidity in Pod #4 (fresh air HP with 15 cfm [7 L/s] per person).*

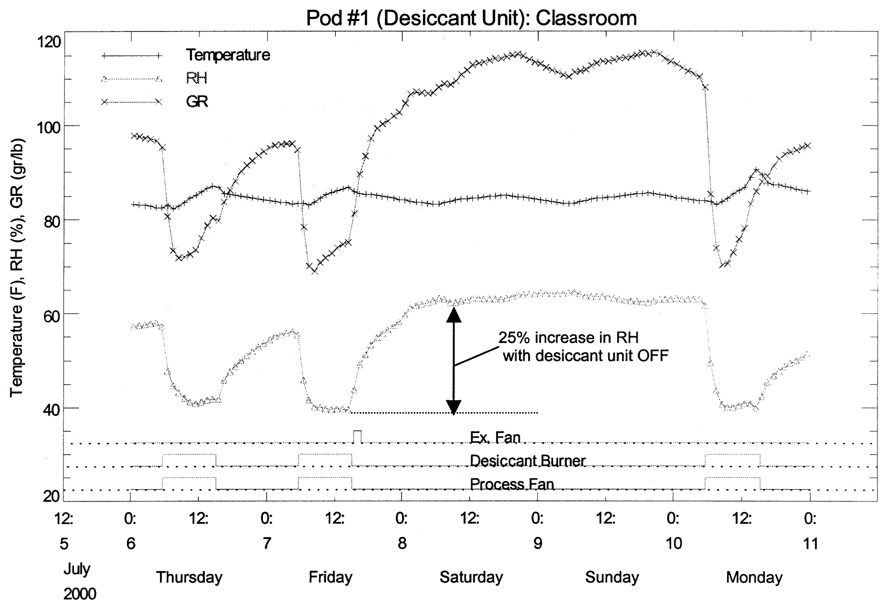


Figure 10 *Drift in space humidity level during unoccupied periods.*

weekend during summer break in Pod #1. During the eight-hour occupied period, the desiccant unit process fan operated to supply ventilation air and the desiccant burner operated continuously to maintain the humidity setpoint. After the ventilation/dehumidification system shut down at 3 p.m. each weekday, space humidity levels quickly rose in the space due to the infiltration of ambient air. Over the weekend, the space humidity increased by 25% RH or 40 gr/lb (6 g/kg). Humidity levels remained high until the desiccant system was restarted on Monday morning.

While all systems provided excellent to moderately good humidity control during the occupied period, the lack of any humidity control during unoccupied periods still had a negative impact on IAQ. Maintaining adequate humidity levels (i.e., below 60% RH) during the occupied period is not the only concern. There is no direct human comfort issue associated with high humidity levels identified in ASHRAE Standard 55-1992 (ASHRAE 1992). However, the risk of high space humidity levels is the potential for the onset of mold and mildew growth. Allowing unacceptably high humidity levels at night, over weekends, and during the summer break increases the risk of biological contamination of the classroom. In schools—which are only occupied for 25% of the week during the school year and even less often on an annual basis—it is important to provide dehumidification during unoccupied periods.

Comparing Total HVAC Energy Use

The measured energy use data for the three pods (in Table 2) show that total HVAC energy use was highest in the desiccant pod. While the water loop heat pumps (WLHPs) in the desiccant pod used less energy than their counterparts in the base-case pod, the electric use of the desiccant unit and heat pumps combined was much greater than in the base pod system. The additional gas use of the desiccant unit was due to dehumidification in the summer months and space heating in the winter months. The electric use of the fresh air heat pump (FAHP) system in Pod #4 was slightly less than the desiccant pod (though space humidity levels were typically much higher). The electric duct heater in Pod #4 accounted for a significant portion of the annual energy use for that system.

This study focused on the dehumidification and cooling performance of the three systems. Figure 11 compares the normalized HVAC energy costs for the three systems during the cooling season (May to October). Costs were determined using local electric and gas costs of \$0.08/kWh and \$0.70/therm, respectively. The costs in each area were normalized using the gross floor area (i.e., 5,840 ft^2 [542 m^2] in Pods #1 and #2; 2,350 ft^2 [218 m^2] in Pod #4). June and July are excluded from the plot since the heat pumps did not operate to maintain the temperature setpoint for portions of these months, making comparisons difficult.

Pod #2 consistently had the lowest operating costs during the cooling season, due to the low ventilation rates. Pod #4, the area with 15 cfm (7 L/s) per student, had the next highest energy costs. Monthly energy costs were from 11% to 64% higher than in the base pod. August was the month with the highest energy cost since the impact of the added ventilation loads was greatest at this time. Pod #1 had the highest energy costs, ranging from 46% to 122% of the base pod costs. Again, August was the month with the highest energy costs due to the impact of increased ventilation. Overall, the normalized energy costs for the cooling season (excluding June and July) were 36% greater for Pod #4 and 86% greater in Pod #1 compared to the base pod. Table 3 summarizes the cooling season costs for the three HVAC systems. Electric costs are the same for the desiccant system in Pod #1 and the fresh air HP

TABLE 2
Monthly Energy Use Summary for the Three Pod Areas

	POD #1				POD #2	POD #4			
Month	Des Unit (therms)	Des Unit (kWh)	WLHPs (kWh)	Total (kWh)	WLHPs (kWh)	FAHP (kWh)	Duct Heater (kWh)	WLHPs (kWh)	Total (kWh)
Nov-99	323	810	1,250	2,060	1,068	167	167	268	602
Dec-99	371	847	738	1,585	545	175	909	360	1,445
Jan-00	352	755	724	1,479	477	162	1,207	507	1,875
Feb-00	208	708	489	1,197	512	160	581	341	1,082
Mar-00	208	796	266	1,062	664	161	308	276	746
Apr-00	-	806	434	1,240	935	209	133	466	808
May-00	90	1,119	1,182	2,301	1,841	164	-	744	908
Jun-00	152	1,355	782	2,137	597	162	-	155	317
Jul-00	239	1,886	1,044	2,931	808	175	-	239	413
Aug-00	227	2,008	2,448	4,457	2,899	210	-	1,703	1,913
Sep-00	65	1,022	1,550	2,572	2,144	168	0	908	1,077
Oct-00	102	966	983	1,948	1,435	158	19	463	640
Annual	**2,337**	**13,078** *(52%)*	**11,890** *(48%)*	**24,968** *(100%)*	**13,923** *(100%)*	**2,070** *(18%)*	**3,324** *(28%)*	**6,430** *(54%)*	**11,824** *(100%)*

Notes: Desiccant unit gas use includes dehumidification and vent pre-heating. Desiccant unit electric use includes ventilation/process fan, regeneration fan, and AC condensing unit for post-cooling coil. WLHPs – water loop heat pumps—in each Pod. FAHP is fresh air heat pump in Pod #4.

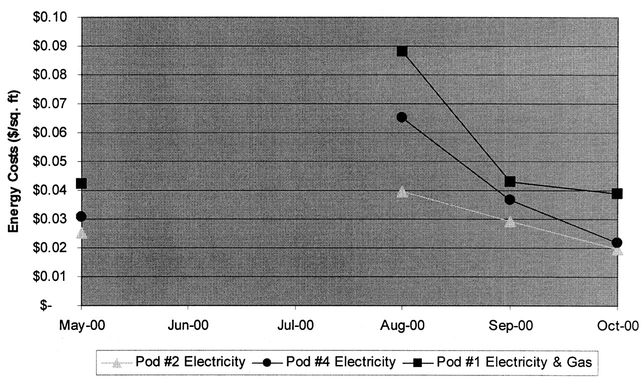


Figure 11 *Normalized monthly energy costs for the three HVAC systems.*

TABLE 3
Monthly Energy Use Summary for the Three Pod Areas (May, August-October)

System	Electric Costs ($/ft^2)	Gas Costs ($/ft^2)	Total Costs ($/ft^2)	Increased Energy Costs
Pod #2 (base-case, 4 cfm [2 L/s])	0.11	-	0.11	-
Pod #4 (fresh air HP, 15 cfm [7 L/s])	0.15	-	0.15	+36%
Pod #1 (desiccant, 10 cfm [5 L/s])	0.15	0.06	0.21	+86%

Notes: $0.08/kWh and $0.70/therms

system in Pod #4. Gas use accounts for the difference between these systems.

The higher energy costs of the desiccant system were in part due to its less than optimal configuration. Bayer et al. (2001) point out that the "active desiccant preconditioning" approach, as used in this field test, may not be the most cost-effective or energy-efficient way to apply desiccant technology. This configuration requires a large desiccant system to treat the entire ventilation airstream, which increases fan power requirements. The pretreatment configuration also requires the desiccant wheel to be regenerated at a high temperature to provide the required grain depression. Bayer et al. proposed a smaller desiccant module applied to treat a portion of the supply airstream after the cooling coil as a more effective approach. The measured energy costs from this field test appear to confirm the higher costs of applying desiccants for ventilation air pretreatment. Configurations that minimize equipment size and reduce fan power requirements would clearly lower overall electrical use, which, in this case, accounted for nearly 75% of the total cooling season operating costs.

CONCLUSIONS

Using the desiccant unit to pretreat ventilation air in this school application provided more stable humidity levels during the occupied period. The desiccant unit maintained space humidity levels that were typically 10 to 15 gr/lb (2 to 3 g/kg) lower than a conventional system that provided 10 to 15 cfm (5 to 7 L/s) per person. Humidity conditions in the desiccant pod were similar to those maintained in a classroom area that used a conventional system to provide only 4 cfm (2 L/s) per student. While conventional HVAC equipment can provide adequate humidity control at 4 cfm (2 L/s) per student, humidity levels are not properly maintained at ventilation rates of 15 cfm (7 L/s) per student. Desiccant technology offers the means to maintain adequate humidity levels while also providing the ventilation required under ASHRAE 62-1999.

Proper humidity control with the desiccant system was provided at an additional cost. The area with the desiccant system had the highest HVAC energy costs for the cooling season. The gas use required to meet the added latent loads as well as the extra fan power to push air through the desiccant system gave this system the highest operating costs. The pod that provided 15 cfm (7 L/s) per student had 36% higher energy costs than the base-case pod at 4 cfm (2 L/s) per student. Providing adequate dehumidification at the 10 cfm (5 L/s) per student ventilation rate with a desiccant system further increased cooling season energy costs (by 86% over the base-case system).

One important observation from this field test was the need to provide dehumidification during the unoccupied periods. In a school, the unoccupied periods account for 84% of the hours in the year. While applying desiccant technology as a ventilation pretreatment system offers several performance advantages, it also limits the ability of the system to provide dehumidification during unoccupied hours. The results from this field test have demonstrated that maintaining proper humidity control during unoccupied periods is critical in classroom applications. Desiccant products for school applications need to be configured so that dehumidification can be provided independently of ventilation. Further study is necessary to determine the best size and configuration of desiccant systems for this application.

For desiccant technology to be more cost-effective in school applications, new packaged system configurations must be developed that minimize equipment costs, allow for lower fan power, and offer the flexibility to provide dehumidification independently of ventilation.

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