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FIELD TEST OF A DESICCANT-BASED HVAC SYSTEM FOR HOTELS

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ABSTRACT

A manufacturer of desiccant dehumidification systems is working with the Gas Research Institute, hotel owners, and the American Hotel and Motel Association Executive Engineers Committee to solve the problem of mold, mildew, and other humidity-related problems in hotels. Buildings and furnishings can be kept dry by using a desiccant dehumidifier to remove excess humidity from the air. Dry air, in turn, removes excess moisture from the building materials and furnishings, which prevents the growth of mold and mildew.

A gas-fired desiccant dehumidification system that incorporates a heat pipe is being field tested at a low-rise hotel in West Palm Beach, Florida. The system is dehumidifying the outside air since it is the source of 90% of the moisture that enters a typical hotel.

The hotel has two wings. One wing is using a desiccant-based dehumidification system, while the other wing is using a direct-expansion cooling and reheating system. The preliminary results of this field test demonstrate the desiccant-based system maintains lower humidity levels and also uses less energy than a properly sized cool/reheat system.

This paper describes the test site, instrumentation, dehumidification systems, and the results obtained for the period from November 1990 through July 1991. The test ended in November 1991.

INTRODUCTION

According to a 1990 survey of hotel, motel, and resort general managers, mold and mildew cost members of the American Hotel and Motel Association approximately $68 million each year in lost revenues and damage repair.

In a second survey assessing problems identified by guests, 70% of respondents complained that rooms smelled stale—a condition often created or aggravated by musty odors emitted by mold and mildew. Other frequent problems include rust, wall covering with black, reddish, yellow, or purple stains, and guests who demand alternative accommodations.

Mold and mildew are forms of fungus. Its growth and reproduction create the familiar musty odors we smell in damp rooms and humid climates. There are three ways to eliminate fungus growth: kill it, remove its food, or remove its water.

Killing mold and mildew is generally effective only for limited periods. If food and water are available, fungus will grow back. If the dead microorganisms are not entirely removed, they become food for another of the 65,000 forms of mold and mildew.

Starving the organism is also difficult, since virtually any substance that contains carbon-based molecules will sustain one or more kinds of fungus. Even products that seem sterile, such as plastics, gasoline, and paint, contain material that can be metabolized by fungus.

Removing water necessary for fungal growth is often the most practical means of controlling mold and mildew—particularly when the problem spreads to walls, carpets, and other parts of a building structure.

Dr. James Kimbrough and Dr. Virginia Peart of a national university suggest the "mildew square" (see Figure 1) as a means of understanding the necessary conditions for fungal growth. The four sides of the square are spores, temperature, food, and moisture. Stopping mildew growth requires elimination of one or more of these essential elements. Unfortunately, however, in a comfortable human environment, the temperature is ideal for most fungi and it is virtually impossible to eliminate spores and food.

According to recent research reports (see references), keeping excess moisture out of materials or removing it if materials are already moist should be a primary focus of design and management professionals who are concerned with mold and mildew problems.

Engineers often rely on the air-conditioning units in each guest room to dehumidify the make-up air. Generally, however, the cooling unit is controlled by temperature rather than humidity. If such a unit is not operating, it

The "Mildew Square" provides a visual aid to understanding the four elements needed to support fungal growth.

Removing excessive moisture from materials is usually the most practical means of stopping mold and mildew.

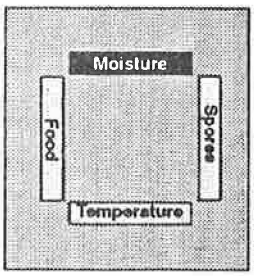

Figure 1 The mildew square.

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will not dehumidify. Typically the units are oversized and do not have a low enough sensible heat ratio to remove the large latent load from moist air infiltrating into the building.

The engineer can minimize the resulting problems by ensuring that outside air is adequately dried before it is allowed into the building. Also, by operating the building under slightly positive pressure with dehumidified air, infiltration of humid outside air into wall cavities is reduced. This practice can help compensate for less than perfect installation of air/vapor barriers and minor building water leaks.

The engineer can reduce the likelihood of excess humidity in the wall by specifying that ventilation air be dried to approximately a 53°F dew point before it enters the building. This ensures that air will not exceed 60% relative humidity, even near surfaces that are at 68°F (i.e., when guest rooms are overcooled). By keeping the air in the building structure below 60% RH, the HVAC system can prevent moisture absorption by building materials and therefore prevent the growth of mold and mildew behind walls and in building furnishings.

A certified air balance contractor should be engaged to demonstrate that the volume of dry make-up air exceeds the volume of exhaust air before the building is accepted as complete by the owner. As the building ages, it is important for the owner to ensure the system operates as designed so that it maintains this slight positive pressure inside the building. Otherwise, humid air infiltration into building cavities will allow moisture absorption regardless of how dry the rooms are maintained.

TEST SITE

The test site is a hotel in West Palm Beach, Florida, which opened in January 1989. The property is three stories high and has 150 rooms. There are north and south wings separated by a courtyard. The wings are connected on the first floor by a lobby/restaurant area on the west side and a hot-water spa on the east side. The guest rooms have individual packaged terminal air conditioner (PTAC) units (11,700 Btu/h cooling capacity) to condition the rooms.

Existing System

Building pressure relationships are established primarily by the balance between the exhaust and make-up air quantities. This hotel was designed without a careful check of the make-up and exhaust air balance, which results in a negative pressure balance relative to the supply. The relatively standard industry practice of closing all PTAC outside air dampers was implemented. The make-up air to each room exhaust would then be theoretically supplied under the corridor door into the guest room. However, the negative air balance resulted in untreated moisture-laden outside air entering the property.

Additionally, the guest wing corridor systems have problems associated with the original design. Each wing was served by a 25-ton split system supplying air through a ceiling plenum. The north side was designed for 5,330 scfm of outside air while exhausting 5,400 scfm. The south side was designed for 4,775 scfm of outside air while exhausting 5,000 scfm. Field testing revealed that the plenum was ineffective. The outside air distribution is ducted from the rooftop units to each floor, where it is delivered to the corridor ceiling plenum. The air ducting was improperly designed from a sizing and construction standpoint. The ductwork in several cases does not extend into the corridor plenum, allowing for leakage into the building construction. The erratic sizing made it very difficult to balance. The certified air balance report showed that the required air per floor was not being delivered. The supply air diffusers did not have balancing dampers in the neck. Furthermore, a check of the original system capacity demonstrated that the make-up air systems could only cool to 65°F and 92 gr/lb. This would result in relative humidity levels in the corridors of 70%.

The outside air corridor ventilation system was controlled by a hallway thermostat. Once the hallway thermostat was satisfied (76°F), the cooling would shut off but continue to supply outside air to the corridor. Using ASHRAE weather data for Miami, it was calculated that 25 tons of standard packaged refrigeration controlled by a thermostat would add 140,201 gallons of water per year to the building per wing. If the system was controlled by a humidistat, each system would still add 39,725 gallons of water (see Figure 2).

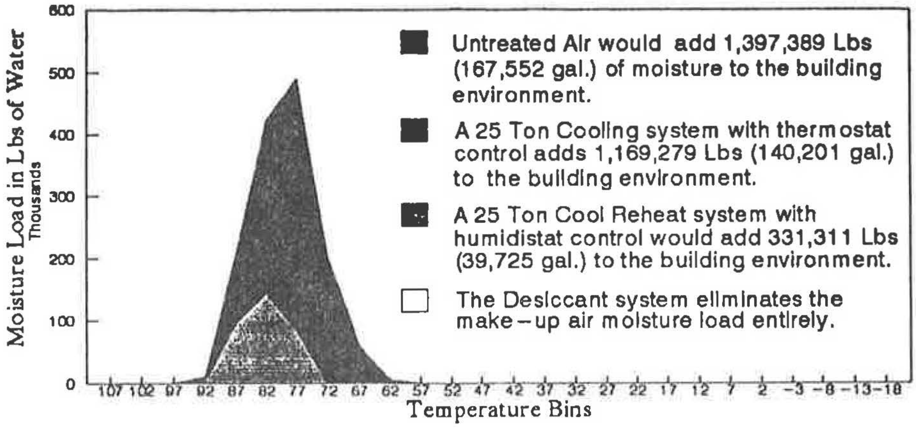
The initial site visit in November 1989 revealed a major mold and mildew problem at the property. More than 100 of the 150 one-year-old rooms were experiencing the effects of mold and mildew and either undergoing repairs or had already under gone major repairs. The relative humidity in the corridors and rooms ranged from 60% to 80% RH. The majority of the rooms were humid and musty. The corridor ceiling grid work was also rusting.

TEST PLAN

The main objectives of the test were to evaluate the effectiveness of positively pressurizing a building with dry air to eliminate mold and mildew and improve comfort and to establish the technical and cost-effectiveness of a desiccant-based HVAC system in a hotel/motel. The research was intended to prove the relationship between space humidity level and mold and mildew problems. The test also monitored and established energy consumption characteristics of the desiccant and cool/reheat dehumidification strategies.

The Desiccant System

The desiccant module was added in front of the existing air handler on the north wing. The supply air



Sample Calculation: Cooling to 68 degrees F and 96 grains/lb

$$\frac{6000\ \frac{Ft^3}{min} \times 60\ \frac{min}{hr} \times .075\ \frac{lb}{Ft^3} \times (96\ \frac{gr}{lb} - 60\ \frac{gr}{lb})}{7000\ \frac{gr}{lb} \times 8.34\ \frac{lb}{gal}} = 16.6\ \frac{gal}{hr}$$

Cooling systems with conventional thermostat control do little to minimize the moisture entering the building via the make-up air.

Controlling a cool reheat system with a humidistat shows an improvement, however a substantial moisture load is still introduced via the make-up air.

The desiccant system eliminates the make-up air moisture load entirely.

Figure 2 Annual make-up air moisture load per wing.

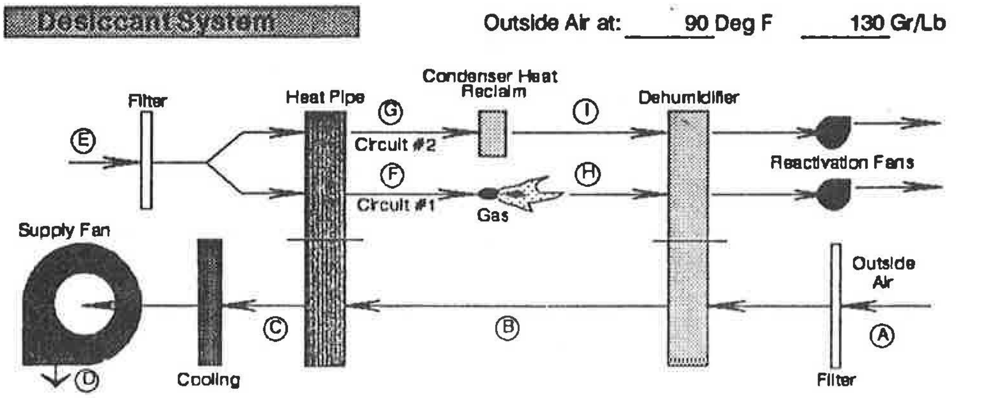
volume was increased to 6,000 scfm to achieve positive pressure. The process side of the desiccant system conditions 6,000 scfm beginning at a design condition of 90°F and 130 gr/lb. The system passes the air through the desiccant wheel, which removes 270 lbs of water per hour. The air then passes through the evaporator portion of a heat pipe, which cools the air from 150°F to 113°F, providing 239,760 Btu/h of sensible cooling. The air is then passed across the existing air handler, which cools the air to 74.5°F, providing 249,480 Btu/h of cooling before the cooled, dry air enters the corridor. The sensible load in the corridors is 9,720 Btu/h due primarily to lights. The relatively large volume of air (6,000 scfm) results in only a 1.5°F differential to maintain 76°F in the corridor. The heat pipe is an important part of the desiccant system. Heat transfer fluid inside the pipe evaporates and condenses, transferring the heat from one airstream to another. In this application, the excess sensible heat from the warm process airstream is transferred to the air entering the reactivation sector. A heat pipe greatly enhances the economics of the desiccant system by saving in both post-cooling and reactivation energy.

The reactivation circuit of the desiccant system has two stages. First it heats air using the heat pipe and condenser heat reclaim, then it adds additional heat by burning natural gas. The condenser heat reclaim is a desuperheating coil, which utilizes the superheated gas from the compressor on the direct-expansion system used for post-cooling. The refrigerant is then passed through the existing condensing unit.

The gas COP of the system is .83 compared to industrial desiccant systems that have .5 COP. The gas COP was calculated by dividing the measured latent work done into the measured gas used (see Figure 3). The psychrometric path is shown in Figure 4.

The Cool/Reheat System

Changes to the system on the south side of the building included increasing the volume of outside air from 4,775 scfm to 6,000 scfm and modifying the controls to allow the system to provide dehumidification and reheat when required. The system was originally installed with a gas burner downstream of the air handler, although the system had never been operated in a cool/reheat strategy. Like many outside air systems, the original cooling system was sized at 200 scfm per ton. This did not satisfy the humidity removal requirement, even for the original airflow. The increase in outside air volume required for positive pressure results in conditions leaving the coil of 68°F and 96 gr/lb, (155,520 Btu/h sensible cooling and 137,700 Btu/h for latent cooling). To meet the design conditions, the installed refrigeration capacity should be 45 tons rather than the 25 tons actually installed. The psychrometric path is shown in Figure 5. The owners felt that changing the operational strategy to humidistat control vs. strictly thermostat control would solve the majority of the humidity problems. They elected not to increase the cool/reheat system to 45 tons, as shown in Figure 6.



Desiccant System	A	B	C	D	E	F	G	H	I
SCFM	6000	6000	6000	6000	5800	2600	3200	2600	3200
Temperature	90	150	113	74.5	90	138	120	250	150
Moisture	130	60	60	60	130	130	130	130	130
Required Cooling Capacity (Tons)			20.5	DH Op	Required Reclaim Output (MBH)				104
Available Cooling Capacity (Tons)			25.0	100%	Required Gas Htr Output (MBH)				315

Figure 3

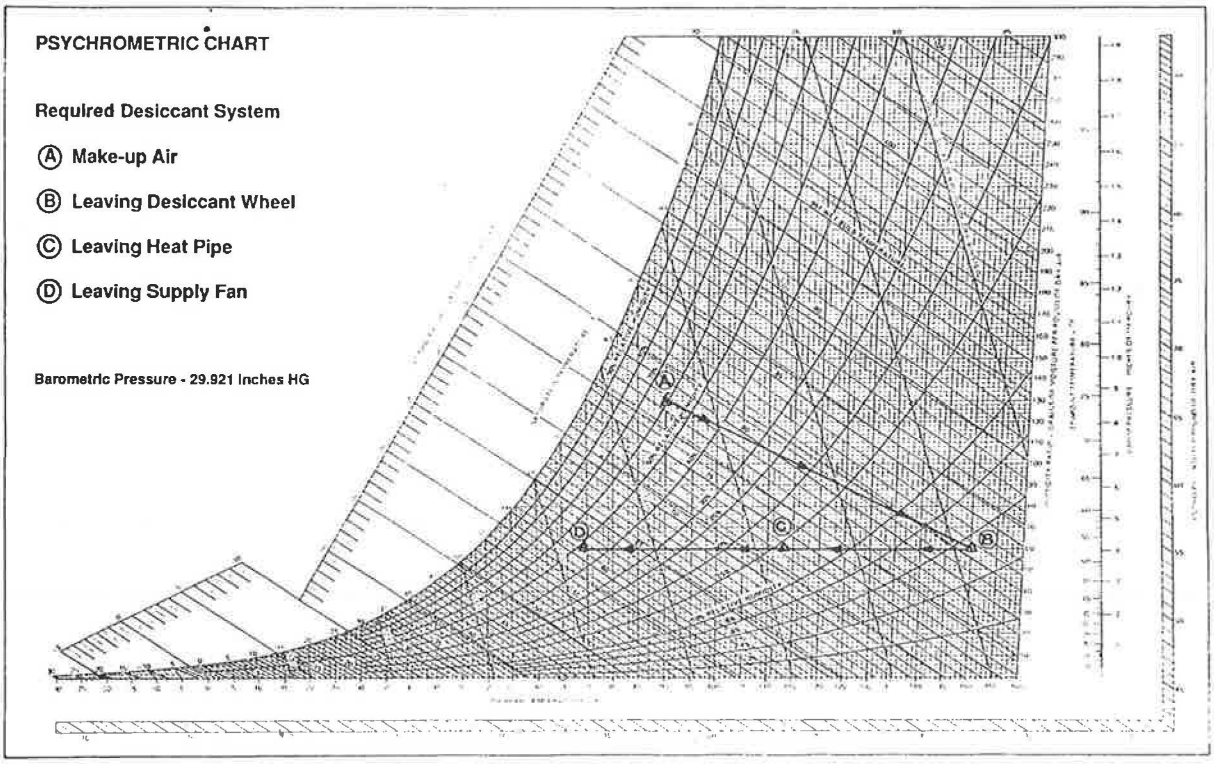


Figure 4

Air Distribution Improvements

Several other changes were made to the property before the test began. The first challenge was to rebalance the building in order to maintain a positive pressure. This was accomplished by first reducing excess exhaust (i.e., the vending area exhaust is now cycled based on temperature, and flow from storage closet exhaust systems was reduced). The outside air volume also was increased by approximately 20% on both wings, providing the building with 10% more fresh air than total exhaust to maintain a positive pressure relationship.

Another major change was made to the air distribution system in the corridors. Ducting air directly to each room is the best way to ensure all rooms have an adequate supply of fresh, dry air. The third floor on both

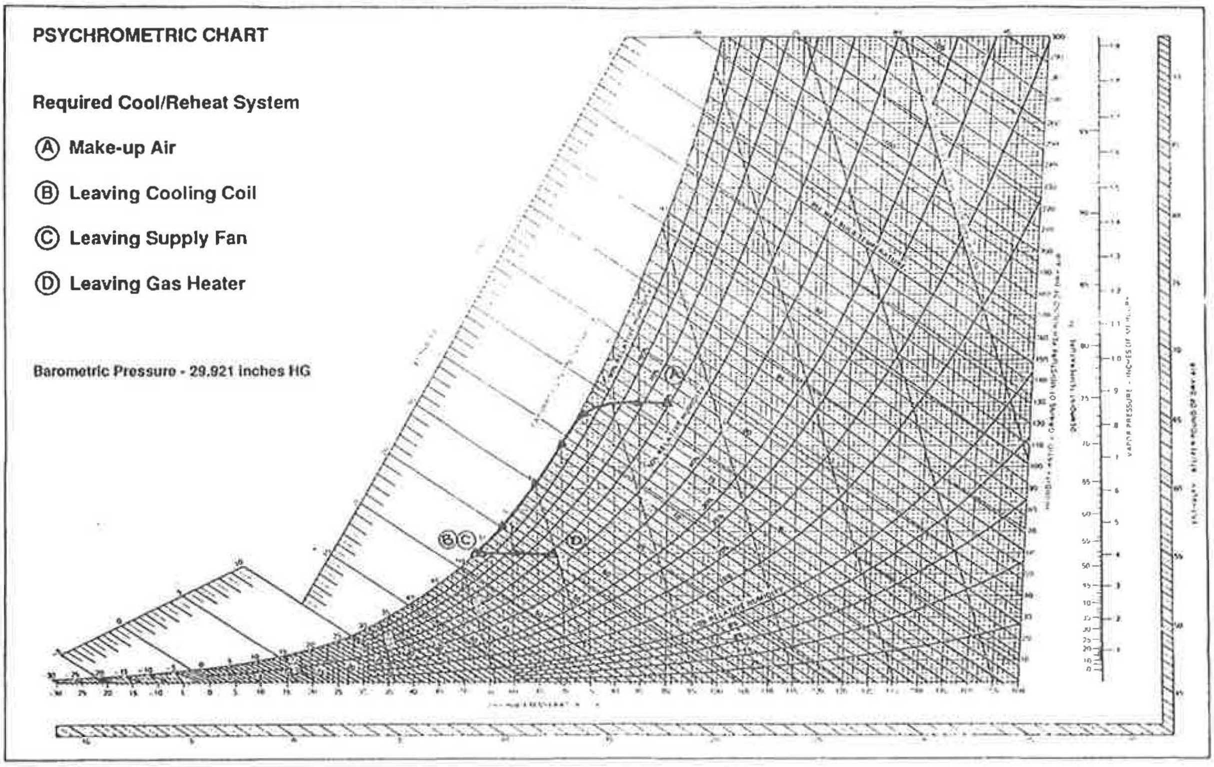
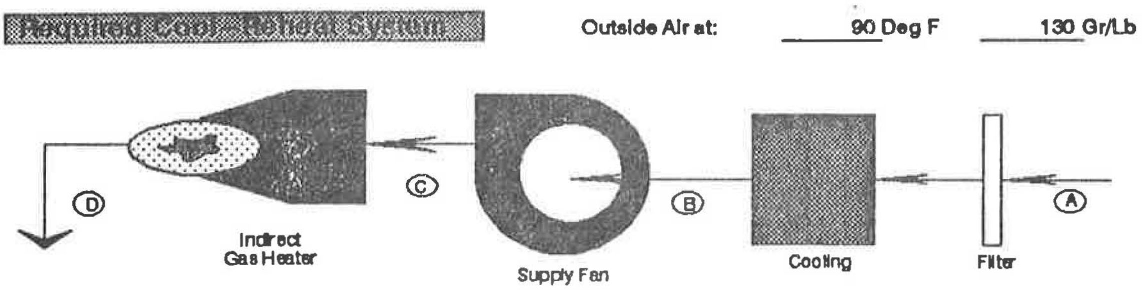


Figure 5



Cool-Reheat	A	B	C	D		
SCFM	6000	6000	6000	6000	Required Cooling Capacity (Tons)	45.0
Temperature	90	53	54	68	Available Cooling Capacity (Tons)	0.0
Moisture	130	60	60	60	Required Heating Capacity (MBH)	91
					Available Heating Capacity (MBH)	220
		Coil Cap	Htr Cap	Fan Bhp		
		100%	41%	3.3		

Figure 6

the north and south sides was retrofitted, with ductwork being installed above the corridor ceiling plenum and ducted to each room. The next best strategy would be to ensure air is distributed correctly within the corridor. To accomplish that alternative, the first-floor ceiling plenums on both sides were modified with four in-line fans at each diffuser to ensure even distribution through the entire length of the corridor. The least expensive strategy would be to simply turn the air down into the side wall and distribute the air at a single point at the end of the corridor. The second floor on both sides was modified with a side wall diffuser. Although this strategy was not ideal with regard to distribution, it did ensure that the air was delivered to the corridor in the correct quantity.

PTAC Sizing

Another significant change was to reduce the size of the room PTACs. It was believed that the original 11,700-Btu/h systems were too large, which means they run less and accomplish very little dehumidification. Twenty-four rooms were modified with smaller PTACs ranging from 5,800 to 6,800 Btu/h.

Instrumentation

The building was instrumented with more than 300 sensors to monitor temperature, indoor and outdoor humidity, wall moisture content, occupancy, patio door positions, pressure differentials, electricity consumption, and gas consumption. The monitoring began in November 1990 and continued for 12 months.

Preliminary Results

At the time of this writing, nine months of data have been collected and are discussed here.

Humidity Levels

The desiccant system maintained an average relative humidity in the hallway of 42% at an average temperature of 76°F over the first seven months of the test period. The cool/reheat system maintained an average relative humidity of 54% at 72°F. (The control level in both corridors is 45% RH and 76°F.) The rooms are approximately 4% to 9% higher than the corridor, as shown in Figure 7, and the humidity levels behind the walls in the cavities are approximately 15% to 18% higher than the corridor, as shown in Figure 8.

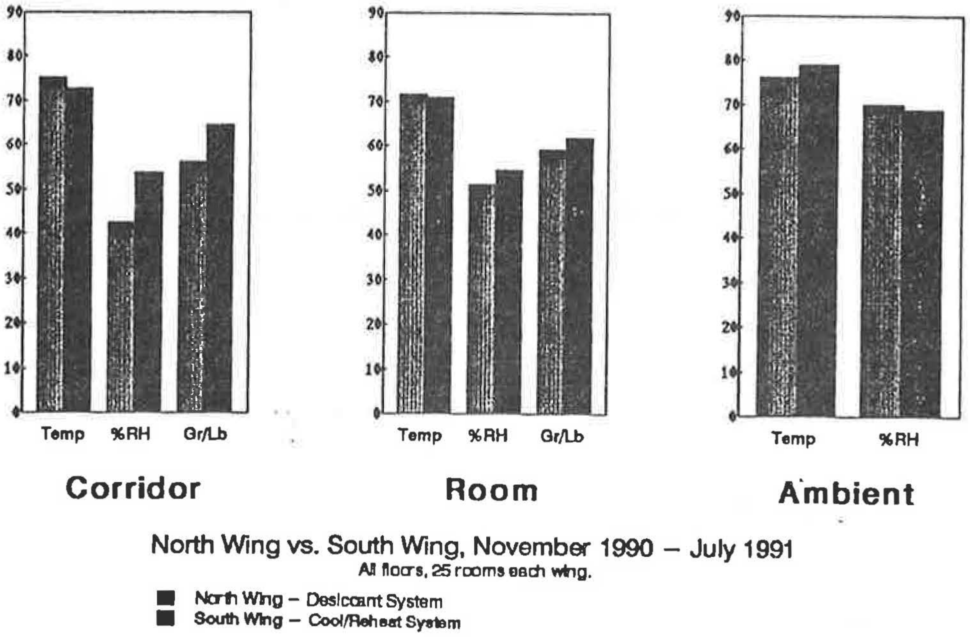


Figure 7 *Space conditions.*

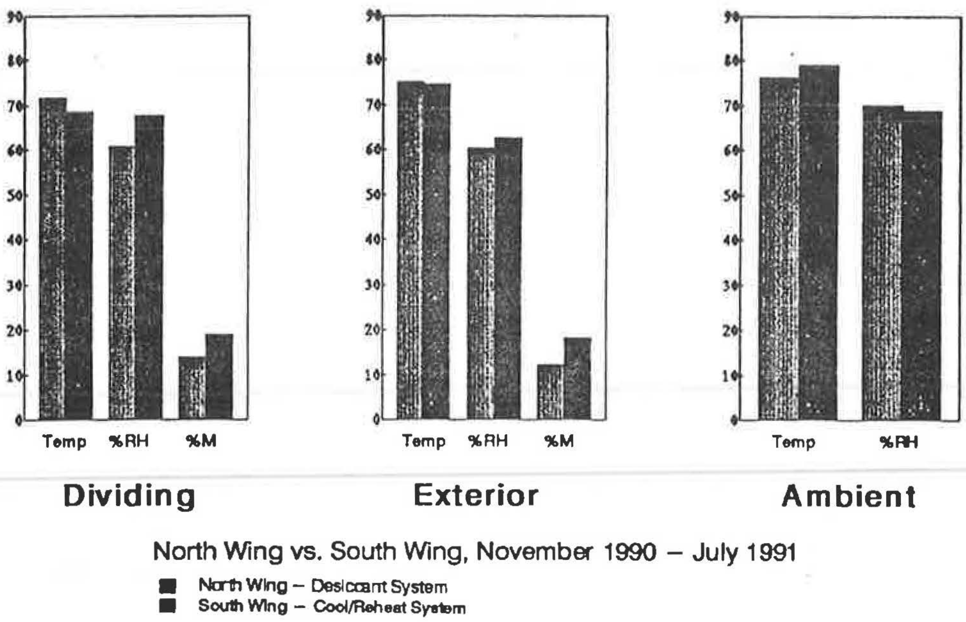


Figure 8 *Wall cavity conditions.*

The higher moisture levels in the wall cavities suggest vapor transmission through the outside of the building. One of the main areas of concern is the humidity level behind the guest room wall. The mold/mildew that shows through the vinyl wall covering as pink/red spots is a function of the humidity level in the wall cavity. The average humidity level in the wall cavity is 60% RH on data collected to date on the desiccant-based system, with the moisture content of the wallboards averaging 13%. The cool/reheat wall cavities are averaging 69% RH and the moisture content of the wall board is averaging 20%. Microbiologists suggest that significant mold/mildew problems can occur when humidity levels exceed 65% RH because material can absorb enough moisture to sustain fungal growth.

Air Distribution Results

The three different air distribution systems appear to have little impact on the specific humidity levels. The temperature gradients along the length of the second-floor hallways are significant—9°F on the desiccant side and 13°F on the cool/reheat side. On the third floor, with air ducted directly to each room, rooms and wall cavities are the driest.

PTAC Sizing Impact

To date, the smaller PTAC units are maintaining room conditions similar to those with the larger PTAC units. The 6,800-Btu/h units are using the least amount of energy, 18% less than the 11,700-Btu/h units. This suggests the 11,700-Btu/h units were, in fact, oversized for the sensible loads they must remove.

Energy Use

To date, the desiccant portion of the system has run 69% of the time at an average of 60% of its capacity. The cooling coil portion of the desiccant system has run 81% of the time at an average of 57% of capacity.

The cool/reheat system has run cooling 81% of the time at an average of 99% of capacity. Its reheat has run 10% of the time at an average of 20% of capacity, which indicates the cooling system is, in fact, undersized for the sensible and latent loads it must remove from the make-up air.

The field data have been used to develop a computer model to project energy costs. The model was used to project yearly energy use for the two installed systems and the 45-ton system that would be required to handle the full load as a cool/reheat system.

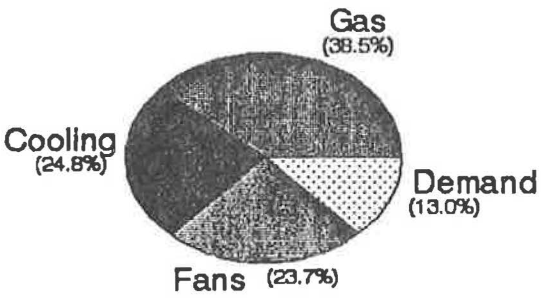
The prototype desiccant system will cost $2,704 more to operate per year than the existing, undersized cool/reheat system (approximately a 10% increase). If the desiccant system improves comfort and minimizes the mold and mildew problem, the benefits will outweigh the modest increase in energy costs. When the installed desiccant system is compared to the required 45-ton system, the desiccant system costs $3,807 less to operate (see Figure 9).

CONCLUSIONS

A desiccant system has been retrofitted onto an existing make-up air system to reduce the space relative humidity by 15% to 30% without adding any additional direct-expansion cooling capacity. This lower humidity on the desiccant-equipped wing of the building has resulted in significant increases in comfort and elimination of

Desiccant System		
Energy and Annual Operating Costs		$22,082
Gas	22359 Therms	$8,496
Cooling	78234 Kwh	$5,476
Fan	74889 Kwh	$5,242
Peak	32 Kwh	$2,867

Cool - Reheat System 45 Ton		
Energy and Annual Operating Costs		$25,828
Gas	9932 Therms	$3,774
Cooling	230585 Kwh	$16,141
Fan	18248 Kwh	$1,277
Peak	52 Kwh	$4,636



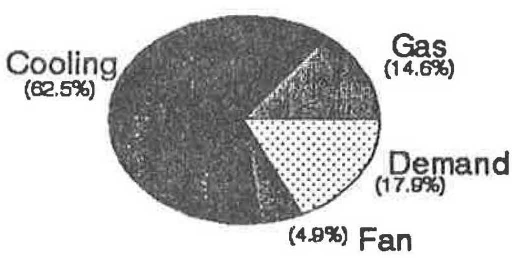


Figure 9

musty odors. Rust on the ceiling tile grid has also been eliminated.

The cool/reheat-equipped south wing has been improved by operating under humidistat control, but the odor still exists and the ceiling tile grid rust has come back.

There had not been any significant mold and mildew growth on either side at last inspection. The months of August, September, and October are critical because 90% of the time the humidity is above 110 gr/lb. The 25-ton system will not have sufficient capacity to maintain the control level. The desiccant system has the capacity to dry much deeper than the cool/reheat system. The cool/reheat-based system is limited, not only by capacity, but also by dew point.

According to the computer model developed from the test data obtained to date, a properly sized cool/reheat system will cost more to operate than a gas-fired desiccant system at this location. In this building, the total installed direct-expansion tonnage for conditioning outside air could be reduced by 40 to 50 tons when using a desiccant module (more than 50% reduction).

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