FIELD EVALUATION OF DESICCANT-INTEGRATED HVAC SYSTEMS: A REVIEW OF CASE STUDIES IN MULTIPLE COMMERCIAL/INSTITUTIONAL BUILDING TYPES

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ABSTRACT

An independent field research effort co-funded by the Gas Research Institute and The U.S. Department of Energy (DOE) Oak Ridge National Laboratory is documenting the performance and energy usage characteristics of active desiccant-integrated HVAC systems at a variety of commercial and institutional facilities. The tests comprise the instrumentation and one-year-plus monitoring of two or more nearly identical sites, one serving as the test site and the others as base-case or control sites.

While the research program is ongoing, work completed in two market sectors, retail and lodging, indicates that there are significant comfort control, energy usage and equipment efficiency benefits to be derived from integrating desiccant units into HVAC system design to handle latent and sensible loads independently. In some cases, installed first costs associated with including desiccant units may be lower if the HVAC system is optimized to take advantage of reduced conventional cooling equipment requirements and downsized ductwork. In most cases, lower energy consumption and/or reduced energy costs may provide reasonable payback of first cost premiums.

INTRODUCTION

Ongoing field research is proving that active desiccant dehumidification equipment is well suited to outside air treatment strategies in commercial and institutional (C/I) facilities. As with any emerging equipment category, current desiccant equipment market penetration levels in the C/I sector are below their full potential as familiarity with the technology, its application and benefits is still relatively low. A demonstration program underwritten by the Gas Research Institute (GRI) and the U.S. Department of Energy (DOE) is documenting the performance of desiccant-integrated HVAC systems in a variety of C/I facility types, thus helping to overcome the market penetration barriers often associated with new and emerging equipment categories.

For over forty years, desiccant equipment has been used successfully in the industrial sector in moisture-sensitive manufacturing and hydroscopic storage applications where there are process improvement economics associated with drier air. During the last decade, desiccant sales and marketing efforts have expanded into niche C/I sector applications such as supermarkets, ice arenas and cold warehouses where similar process improvement economics are evident. Acceptance and adoption of desiccant-integrated designs in these sectors is growing steadily and represents a growth market. The greatest opportunity for desiccant systems may still be emerging, however, as C/I facilities managers become more aware of the impact of comfort and indoor air quality (IAQ) on building occupants.

While IAQ is a topic of continued debates, i.e. its causes, its impact on health, how it's measured and control and abatement issues, one common "given" has arisen: increased outside air quantities and ventilation rates alleviate poor IAQ. In 1989, in response to the alarming increase of "sick building syndrome" incidents purportedly caused by tighter building envelopes, elevated internal pollutant loads and higher occupancy levels per square foot, the American Society of Heating Refrigerating and Air-conditioning Engineers (ASHRAE) released a revised Standard 62, "Ventilation for Acceptable Indoor Air Quality." The new standard, which has been incorporated in-whole or in-part by all three major model code bodies, nominally trebled outside air quantities for most C/I facilities. This has created an even greater humidity control challenge than before as most conventional HVAC systems are sized to meet peak sensible loads and are ill-suited to handling large quantities of moisture-laden outside air streams.

Further highlighting the need for HVAC designs that can respond to latent and sensible loads independently is new weather-design data, as published in the 1997 ASHRAE Handbook of Fundamentals (HOF) – Chapter 26, "Climatic Design Data." Among the many important new tabular data are extreme humidity ratios (dew point), and dry-bulb temperatures coincident to peak wet-bulb conditions. The new tables illustrate that, in many cities, the highest enthalpy occurs at the peak dew point, <u>not</u> the peak dry-bulb condition. Using this updated information, engineers are better able to size equipment to meet both sensible and latent loads more effectively.

The primary task of a building's HVAC system is to provide "comfort," a subjective term that encompasses not only good IAQ but an expectation of consistent conditions within a range of temperatures and humidity levels. ASHRAE's HOF defines the "Comfort Zone" as "the acceptable ranges of temperature and humidity for people in typical summer and winter clothing during primarily sedimentary activity." ASHRAE further defines the zone as conditions that eighty percent (80%) of people would find suitable.

The HVAC equipment performance and energy consumption associated with delivering different conditions within the ASHRAE "comfort zone" are important. While a majority of C/I facilities operators might agree that delivering indoor summer conditions of 73-75F at 50-55% relative humidity (RH) is ideal, most control temperature only. Some implement humidity control measures if relative RH exceeds 60-65%, but temperature is the primary control determinant, not RH. If RH of 45-50% is consistently maintained, temperatures of 77-79F are well within the ASHRAE comfort zone. As the GRI/DOE research illustrates, the energy implications of this shift in delivered space conditions can be quite significant.

The unique application and performance characteristics of desiccant equipment makes them ideally suited to humidity control strategies, particularly in facilities where there are large percentages of outside air to be treated. The goal of the GRI/DOE research program outlined below is to provide independent documentation of desiccant-integrated HVAC system performance in C/I buildings where outside air loads are high and humidity control is desired.

GRI/DOE NATIONAL ACCOUNTS DESICCANT DEMONSTRATION PROGRAM

To facilitate broader market awareness and acceptance of the findings, the GRI/DOE program's demonstration activities are being conducted with prominent multi-site "opinion-leader" accounts in seven market sectors: restaurants, theaters, hotels/motels, hospitals (operating rooms), nursing homes, schools and retail facilities. In addition, leading engineering societies and trade associations have been included in the process to identify sector-specific concerns and needs and to secure venues for sharing the research findings.

Key Program Elements

1. Identify two identical or nearly identical sites within the same geographic area to serve as the base-case and test locations. Working with the host's headquarters development, facilities management and engineering consultant teams, new construction and prospective retrofit sites are identified taking into account comparative factors such as building size, construction materials, age, general HVAC system design, geographic proximity and other operational factors that might affect building performance.

2. Install desiccant equipment at the test site. In new construction projects, HVAC designs are optimized to take full advantage of the desiccant equipment performance. This may include downsizing of conventional cooling equipment and ducts. In retrofit projects, desiccant equipment applications are optimized as much as possible given the existing facility's design, construction and operating constraints.

3. Normalize operating characteristics of the two sites to minimize "apples-to-oranges" comparisons. At both the base-case and test locations, HVAC systems are given a complete review at the beginning of the test including air balance (and adjustments, if necessary), controls calibration and operational sequences. As much as is possible, HVAC operational schedules are also established with site managers, i.e. setting consistent on/off times and default set-points. Variances in connected equipment and other factors that affect HVAC load are noted. Given the fact that the tests are conducted in "real-life" settings, some internal load factors will vary (e.g., number of meals served, surgeries performed, theater tickets sold, etc.). These factors are periodically reviewed with the host's management to assess their possible impact on data and, if necessary, normalized.

4. Install monitoring equipment at both sites. While each project has its own unique monitoring needs, instrumentation usually includes multiple temperature and humidity sensors in the spaces to be conditioned, the supply and return ducts, pre- and post-cooling coils and outside air intake manifolds (redundant sensors are installed to assure critical ambient data). Current transformers, Watts transducers and other energy meters are used to capture fan and compressor run-times, and electric and gas usage (individual HVAC equipment and total site). Both internal and external CO_2 sensors are installed as one measure of IAQ and to provide a benchmark for ventilation effectiveness. Where necessary, flow meters are installed on chiller lines

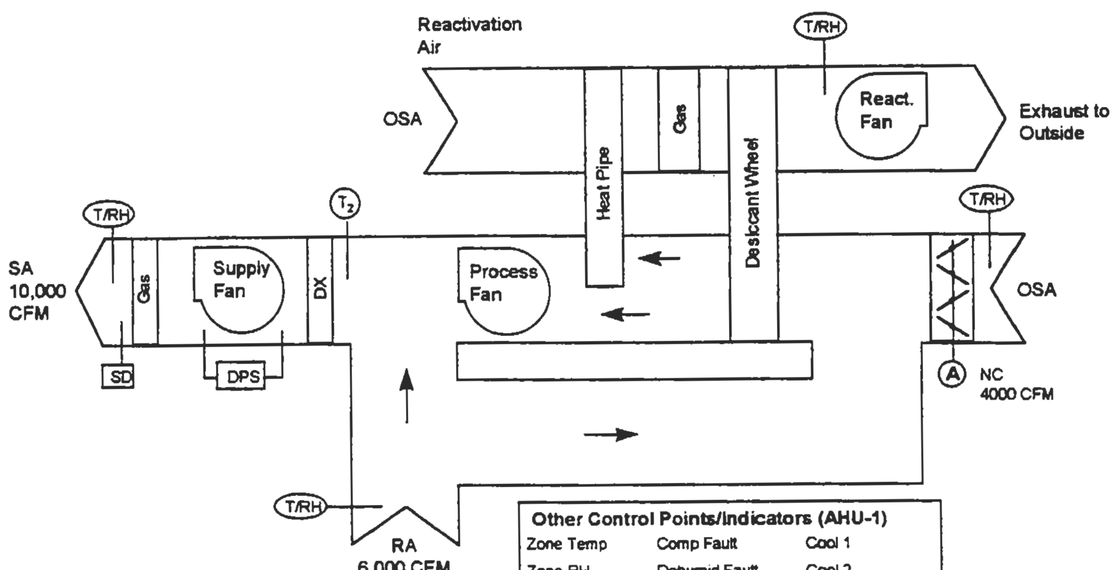
Treatment of Outside Air

Until building the Norfolk store, Wal-Mart's store engineering team had not designed a store to meet ASHRAE Standard 62-1989. The impact of the new revised standard was considerable, effectively doubling the amount of outside ventilation air. The previous revision of the standard required 5 cfm fresh air supply per person, based on an occupancy of thirty persons per one thousand square feet of floor area. This was equivalent to about .15 cfm per square foot. The 1989 revision states the requirement as .3 cfm per square foot. Of particular concern to their engineering team was the impact of latent loads present in outside air. Wal-Mart's standard approach to humidity control in the retail portion of their Supercenters has been to use the dehumidification capabilities of their individual RTUs. When sensible loads are low, RTUs satisfy the cooling requirement quickly and cycle "off," prior to handling the latent load. When store humidity levels exceed 60%, a setback control strategy is employed whereby RTU temperature set points are lowered temporarily from 75F to 71F and, when necessary, reheat is employed. Unfortunately, this can lead to inconsistent space conditions and inefficient operation of the RTUs. The increased outside air requirements of ASHRAE 62-1989 would further exacerbate the problem, especially during times of high latent-to-sensible load ratios.

Wal-Mart's engineering team had successfully controlled humidity in the grocery portion of many of their stores using gas-fired desiccant systems. At Norfolk, they decided to employ a new approach to outside air treatment i.e. bringing in all outside air via a desiccant system and shutting the RTU s' dampers, using them for cooling and heating re-circulated air only. A central indoor-mounted humidistat controls operation of the desiccant dehumidification units while zone thermostats control the RTUs and DX portions of the desiccant units.

Wal-Mart approached the local code officials concerning overall outside air quantities necessary for their dual-use facility (grocery/retail) as ASHRAE Standard 62-1989 did not address multi-use facilities such as Supercenters. They noted that ASHRAE Standard 62-1989 recognizes CO_2 concentration as a key indicator of indoor air quality. The standard states in section 6.1.3, "Comfort (odor) criteria are likely to be satisfied if the ventilation rate (fresh air supply) is set so that 1,000 PPM CO_2 is not exceeded." Wal-Mart engineers proposed a design that included a base ventilation rate of 12,000 cfm with capability to increase to 24,000 cfm if CO_2 exceeded 1,000 PPM. The design included three CO_2 sensors located in the retail merchandise area.

Figure 1. AHU 1 – Grocery Unit (both stores)



Other Control Points/Indicators (AHU-1)		
Zone Temp	Comp Fault	Cool 1
Zone RH	Dehumid Fault	Cool 2
CO2	Supply AFS	Heat 1
Gas MCF		Heat 2
Elect kW		Dehumid
		Supply Fan
		Damper A

and hydronic coils. When available, the customer's EMS is used for data collection. Usually, however, a separate data-logging system with dedicated phone line is installed to ensure that critical data is not lost.

5. Monitor both sites for a full operating year. In order to assess the full range of operating characteristics including ambient conditions and internal load variations, both sites are monitored for at least one full year. After a representative baseline of data is established, some operational parameters are deliberately manipulated to measure HVAC system performance capabilities and energy usage. Examples include adjusting temperature or humidity set-points and controls sequences. Again, these tests are conducted simultaneously at both sites for set periods of time to ensure comparability of data.

6. Provide periodic reports to project stakeholders and, upon completion, to general public. Throughout the monitoring period, the host's management and their engineering team are given periodic updates on building and equipment performance. This also provides them with the opportunity to become more familiar with the desiccant equipment, maintenance/operational issues and future design options. At the conclusion of the test, findings are also presented to interested engineering and trade groups via symposia, conferences, trade press and other communications channels.

As of this writing, tests have been completed in two market sectors, lodging and retail. Additional tests are underway at quick-service and full-serve restaurants, theaters and hospital operating rooms. Tests at nursing homes and schools are slated to begin in the next six months. This paper will focus primarily on the results of the retail test conducted in cooperation with Wal-Mart Stores and briefly review some of the key findings from the lodging test, published in October 1994.

Test: Retail - Wal-Mart Supercenters, Nebraska

A 14-month study provides conclusive proof that desiccant-integrated HVAC systems can help Wal-Mart comply with ASHRAE 62-1989 while providing superior comfort control, reduced operating costs and lower first cost.

The test, which ran from July 1995 through August 1996, was conducted at two nearly identical stores in Nebraska. Both stores are 188,000 square foot Wal-Mart Supercenters, single-story, slab-on-grade, concrete block construction facilities. Both are open 24hours and contain a grocery, a mini-McDonalds module, a bakery, an auto repair shop, a pharmacy, photo lab, hair salon, video store and large general merchandise area.

Description of Test Site HVAC System.

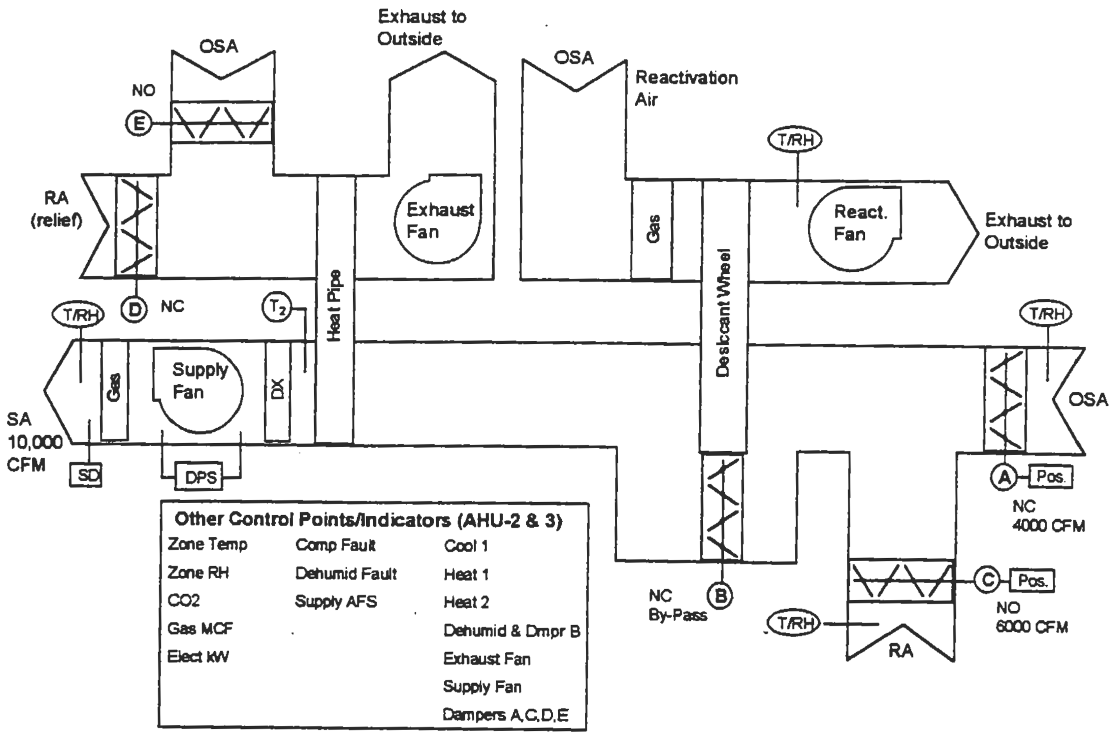
At the test store in Norfolk, Nebraska, a desiccant make-up air system handles all outside air requirements. Norfolk code officials required that the store be designed to meet ASHRAE Standard 62-1989 with the capacity to deliver 24,000 cfm of outside air. It comprises three desiccant units, two serving the retail area and one serving the grocery side of the store. The amount of outside air is controlled by CO_2 sensors. The base ventilation rate is a continuous 12,000 cfm of outside air. If the indoor CO_2 level exceeds 1000 parts per million (PPM), the system operates at the full 24,000 cfm until CO_2 drops below 1000 PPM. However, during the test period, CO_2 levels never exceeded 1000 PPM and, therefore, the system never operated above the normal 12,000 cfm.

At Norfolk, the desiccant unit serving the grocery area (AHU 1) has an additional DX cooling capacity of 32 tons and gas heat while the desiccant units serving the retail area (AHUs 2 and 3) have an additional 18 tons DX cooling capacity each. All three AHUs deliver 4000 cfm of outside air during "normal" operations and AHUs 2 and 3 have the ability to deliver 10,000 cfm outside air at "high" settings when CO_2 levels exceed 1000 PPM. . Figures 1 and 2 show the configuration and operation of AHUs 1 (grocery side, both stores) and AHUs 2 and 3 (Norfolk store only). In addition to the three desiccant units, the Norfolk store's HVAC system comprises 33 rooftop units (RTUs) with electric cooling and gas heat. There are twelve 20-ton units, seven 10-ton units, six 5-ton units and eight 3-ton units.

Description of Base-Case Site HVAC System.

The base-case store is a "standard" Wal-Mart Supercenter in Fremont, Nebraska, approximately 60 miles west of Norfolk. Its outside supply air is provided via a combination of RTUs and one desiccant system serving the grocery area (AHU 1) which is identical to the one serving the grocery area at the Norfolk store. When built approximately one year prior to the Norfolk site, local code officials did not require this store to meet ASHRAE Standard 62-1989. It's outside air delivery varies between 6000-9000 cfm depending on the operation of the RTUs. In addition to the desiccant unit, the Fremont store's HVAC system comprises 36 RTUs with electric cooling and gas heat. There are fourteen 20-ton units, eight 10-ton units, six 5-ton units and eight 3-ton units.

Figure 2. AHUs 2 and 3 (Sales Floor -Norfolk only)



Unit capacities shown in Table 1 are from the manufacturer's literature. Capacities are noted at their respective design entering and leaving air conditions. Some component capacities are impacted by entering air conditions.

Table 1

Item	Units	AHU-1 (Norfolk & Fremont)	AHU-2 & -3 (Norfolk only)
Delivered Air	CFM	10,000	10,000
Fresh Air	CFM	4,000	4,000 - 10,000
Dehumidification	LB/HR	216	276
Cooling Capacity			
Sensible	BtuH	381,000	300,000
Latent	Btu	226,800	289,800
Heat Pipe Capacity (cooling)	Btu	49,700	334,800
Heat Pipe Capacity (heating)	Btu	-	442,800
Heating Capacity (output)	Btu	560,000	320,000

After providing historical data about store occupancy levels, and general merchandise retail versus store-room square footage, they received approval from the code official for this two-tiered approach.

At the Fremont store, outside air quantities varied rather significantly. AHU 1 ran continuously delivering 4,000 cfm of outside air while the RTUs cycled on and off in response to sensible loads, delivering an average of an additional 3,500 cfm of fresh air based on run times and outside air damper settings of 10 percent. CO_2 was not monitored at this site.

Data Collection

All data collected for this field study was provided through Wal-Mart's central control and monitoring department which uses a NOVAR Controls system. While most of the data points were part of Wal-Mart's standard monitoring system, additional sensors were installed to more closely evaluate the desiccant units' performance and to isolate HVAC system energy usage. The NOVAR system allowed for regular logging of functions such as average, high, low and cumulative. Data sampling rates were as frequent as every 10 seconds and logged hourly (average, sum, etc.) Data was downloaded periodically via phone modem and archived. In all, over 100 data points in each store were monitored and over 1.75 million data sets were collected.

Summary of Wal-Mart Study Findings

The results of this study are conclusive and dramatic. By using a gas-fired desiccant system to pre-treat ventilation make-up air, Wal-Mart is able to build and operate their stores in compliance with the increased outside air requirements of ASHRAE Standard 62-1989. They also realize three additional benefits: improved indoor comfort control; lower first cost; and reduced operating costs through lower energy expenses and decreased HVAC maintenance costs.

Improved Comfort Control: At the Norfolk store where a constant supply of 12,000 cfm of outside air was delivered into the space, the desiccant make-up air system provided excellent control of indoor humidity and comfort, generally maintaining relative humidity within 5% of the 45% set point. The system maintained this space condition even when ambient conditions were in the 85-90F range with 80+%RH. At Fremont where outside air supply averaged only between 6000-9000 cfm, average relative humidity was 45% but typical relative humidity levels fluctuated widely between the set point of 45% and 60% (see Figures 3 and 4). By maintaining a consistent relative humidity in the 40-45% range, the Norfolk store's temperature set point was able to be raised from 75F to 79F without getting outside the ASHRAE comfort zone; upon receiving one complaint from an employee, the temperature set point was subsequently lowered to 77F where it stayed the remainder of the cooling season.

Figure 3. Norfolk Indoor Relative Humidity, 7/96 High, Low and Average

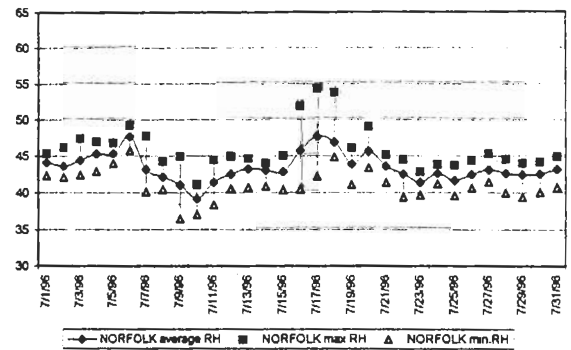


Figure 4. Fremont Indoor Relative Humidity, 7/96 High, Low and Average

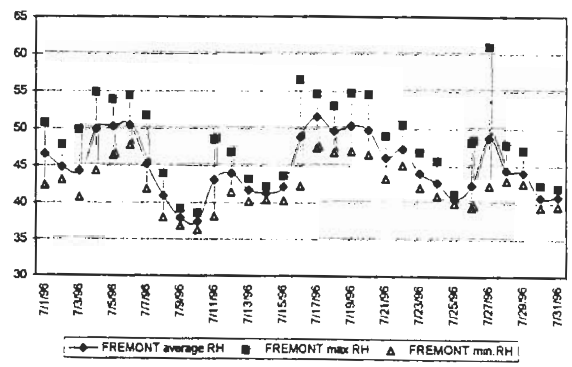


Figure 5. AHU-2 Dehumidifier On-Time and Delivered Dew Point July 15, 1995

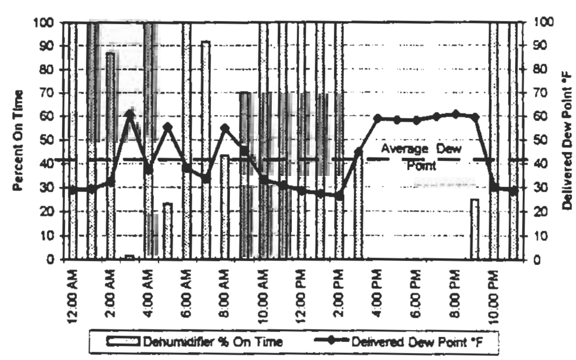


Figure 5 shows the measured dew point hourly averages during a typical day early in the test period. Since the desiccant wheel is called on by the need for dehumidification, this graph can be viewed as a picture of the dehumidification needs of the space during this period. The average delivered dew point for this 24 hour period was approximately 42F. A commonly accepted value for the delivered dew point achievable from a conventional unit is 52-55F. Considering this and the space dehumidification needs depicted by the graph, it is easy to understand why wide fluctuations in space relative humidity were recorded at the Fremont site where conventional systems were relied upon for both cooling and dehumidification. Clearly, a cooling system with a minimum delivered dew point of 52-55F would clearly fall short of providing adequate dehumidification.

Reduced Energy Costs: The increased outside air ventilation rates required ASHRAE 62-1989 impose a much greater sensible and latent load on store HVAC systems. Gas-fired desiccant systems shift the latent load from electricity to natural gas which is significantly less expensive, especially during summer months. This shift to lower cost gas for a large portion of the cooling load greatly offsets the increased ventilation load cost incurred with conventional systems. Figure 6 summarizes the energy costs per month for both stores. During summer, the Norfolk store shows slightly increased energy cost as a result of the increased ventilation load. During the swing seasons (September through November, and April and May), Norfolk's energy costs were lower due to the combination of the work of the desiccant systems and some additional "free cooling" from the increased outside air ventilation. During winter, the increased ventilation rate at Norfolk increased heating costs.

Despite handling a 74% higher ventilation rate with better humidity control, the Norfolk store cost only 2.6% more to operate than the Fremont store during the year of monitoring (see Figure 7). This difference is statistically insignificant and could be considered a normal variation between two similar stores. These results indicate that the Norfolk store met the increased ventilation rate of ASHRAE Standard 62-1989 and controlled indoor humidity with no significant additional energy cost. When the full benefits of drier air are translated into store operating procedure, as they were during the April/May 1996 period when Norfolk's store temperature set point was raised from 75F to 78F, additional energy savings may be realized. Figure 8 shows a comparison of the two stores' energy costs during the April/May period; Norfolk realized 13% savings compared to Fremont.

Figure 6. Monthly Gas Plus Electric Cost

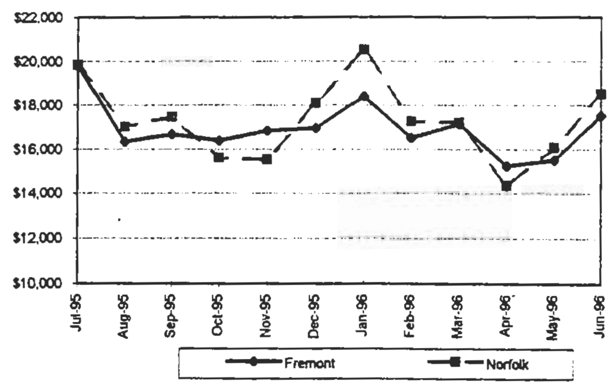


Figure 7. Annual Energy Cost

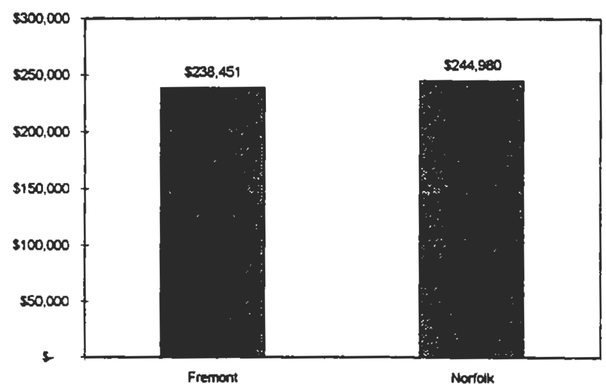
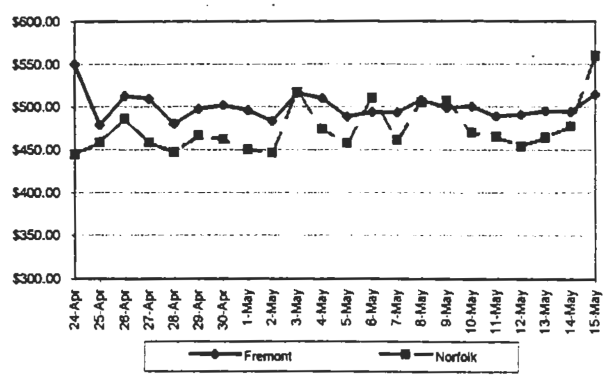
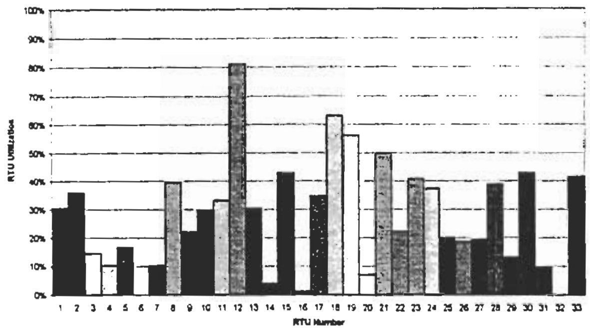


Figure 8. Daily Energy Cost Comparison April/ May 1996 (temperature set-up period)



Lower First Cost: One of the most interesting findings of the study at the Wal-Mart stores was the impact of transferring latent load from conventional equipment to the desiccant units. Throughout the test, each RTU's run time was measured at both stores. During a 3-day hot spell in July 1996 when daytime temperatures reached 100F on the roof and nighttime temperatures were near 70F, RTU run times at both stores fell significantly short of the expected 50-75% level. Figure 9 charts the run times of all 33 RTUs at Norfolk during this 3-day period.

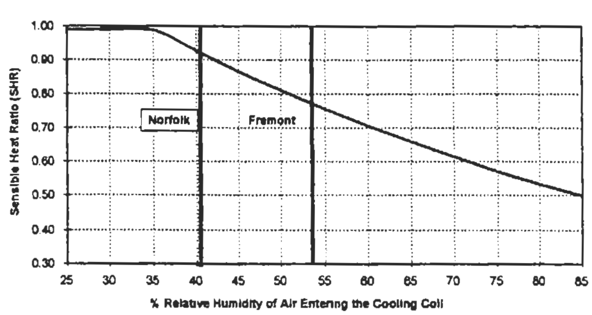
Figure 9. Percent of Cooling Capacity Used During Peak Period (7/15-7/18/1996) - Norfolk



Based on the run times of the cooling compressors during this peak period, the average cooling load at Norfolk was 132 tons. This is an average load and does not reflect peak demand. Using a peak demand factor of 2.11 as based on the oversize factor as measured at Fremont during the same 3-day period, Norfolk's total cooling demand was estimated at 279 tons or 119 tons less than what is installed there. Another approach used as a comparison benchmark was to sum the RTU tonnage on all Norfolk store units that ran less than 25% of the time during the peak period; this tallied 108 tons. While it is impossible to determine the exact amount of tonnage needed at this site based on these methods of measurement, it is clear that there is excess cooling capacity of approximately 110 tons.

A portion of the excess RTU capacity may be attributed to the inside conditions achieved at the Norfolk site using desiccant systems. With Norfolk's RTUs' outside air inlets closed and store conditions warmer and drier, the Sensible Heat Ratio (SHR) of the RTU coils was increased by 14%. SHR is simply

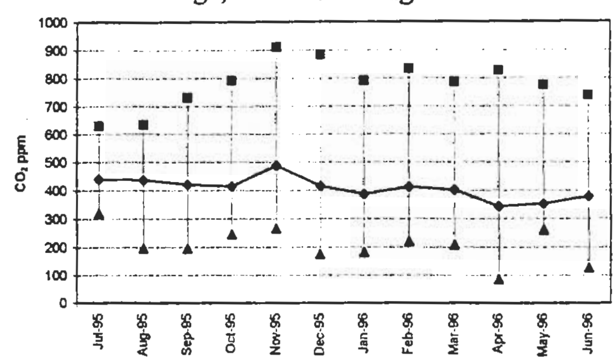
Figure 10. RTU Sensible Heat Ratio vs Relative Humidity of Air Entering the Cooling Coil



the ratio of sensible cooling to total cooling accomplished. When a cooling coil conditions incoming air that is warmer and drier, a greater percentage of the cooling accomplished is sensible load and a lesser portion is latent load. Figure 10 illustrates the relationship between the cooling coil's entering air stream relative humidity and the coil's SHR, based on the manufacturer's data. When the conditions of air entering the coils were plotted for both stores, the Norfolk store RTUs' coil efficiencies were 14% greater. This equates to approximately 25 refrigerant tons more sensible capacity from the Norfolk RTUs due to the desiccants.

The Norfolk store's installed make-up air capacity of 24,000 cfm was never called for because in-store CO_2 levels never exceeded 1000 PPM with constant 12,000 cfm outside air supply (see Figure 11). Data verifies that the ventilation requirement can be met with just one 10,000 cfm desiccant make-up air unit in addition to the one standard desiccant serving the grocery area (total of two units). If additional make-up air is required by the local code official to comply with ASHRAE Standard 62-1989, a limited number of RTUs can be fitted with motorized outside air dampers to be activated only if CO_2 levels exceed 1000 PPM. Since data verifies that this will occur rarely (if ever) and have a short duration time, energy and comfort cost will amount to only a small fraction of the avoided first cost.

Figure 11. Norfolk Interior CO_2 Levels High, Low and Average Values



Using a conservatively low installed-cost estimate of \$600-1000 per ton (based on a chain's buying power), an RTU first cost reduction of between \$66,000-110,000 is achievable:

110 tons x \$600-1000/ton = \$66,000-110,000 savings

If the store uses one 10,000 cfm desiccant make-up air unit at an installed cost of \$63,000, the net first cost savings would range from \$3,000 - \$48,000 per store.

Test: Lodging – Marriott Courtyard, Swan Hotel, FL

The results of the Wal-Mart test confirmed that using desiccant-integrated HVAC systems to treat latent and sensible loads independently provides for better comfort control and energy cost savings. Findings of research studies conducted at two Florida hotels also illustrate the effectiveness of this strategy. In addition to monitoring comfort levels as represented by temperature and relative humidity, these studies also included measurements of mold, mildew and bacterial growth. The lodging industry estimates that in excess of $75 million per year is spent by U.S. lodging companies to repair damage to interior surfaces, furnishings and some structural members caused by mold and mildew damage. In addition to these costs are the "customer dissatisfaction" costs associated with uncomfortable space conditions and stale odors.

The first test involved a two-wing, three story, 150-room Marriott Courtyard located in West Palm Beach, Florida. In October 1990, the North Wing's 25-ton conventional vapor compression cooling system, which serves the guestroom corridors, was retrofitted with a 6000 scfm desiccant air handler that pre-treated outside air. The property's South Wing, which is nearly identical to the North Wing, was used as the comparison. While the HVAC system in the South Wing was identical, minor control revisions were made to the corridor air handler/cooling system in order to employ a cool-reheat dehumidification strategy. The site was fully instrumented; over 300 data points were monitored and logged for a twelve-month period. In addition, periodic inspections of both wings were made to sample and record mold, mildew and bacterial growth.

The results of this test were impressive. Lower humidity levels maintained by the desiccant system in the North Wing lead to 25% drier wallboard, 75% lower levels of fungus growth, reduced moisture damage and a higher level of occupant comfort. Based on humidity measurements of ventilation air entering the building, 500,000 more pounds of water entered the South Wing over the course of the year than the North Wing. This equates to emptying two one-gallon buckets of water into each of the South Wing's 75 guestrooms each and every day of the year.

A second field-monitoring test was conducted from June 1992 to August 1993 at The Walt Disney World Swan Hotel in Orlando, Florida. At this 700 room, 12-story resort hotel, two large air handling systems delivery a combined 47,500 scfm of conditioned fresh air to the guestroom corridors. As was the case at the Marriott Courtyard, the Swan Hotel's guestrooms relied on individual room PTACs for temperature control. Fresh air is drwan into the rooms by bathroom exhaust fans through a small space beneath the room door. Two gas-fired desiccant units – one rated at 20,000 scfm and the other at 27,500 scfm - were retrofitted to the existing air handlers. The combined moisture removal capacity of these two units at design conditions was 1380 pounds of water per hour (1,461,000 Btu/h latent cooling).

While this application did not present the opportunity to compare two areas (one with desiccant air treatment and the other, without it), it did illustrate several significant benefits. Prior to the installation of the desiccant units, the Swan's engineering and facilities management staff had observed a pronounced temperature and humidity gradient from the top floors to the ground level. The more expensive uppermost floors were uncomfortably warm and humid. In addition, humidity levels throughout the hotel 's guest corridors often exceeded 70% RH. Refurbishing of guestrooms and some corridor areas damaged by mold and mildew from excess humidity was a regular occurrence.

After employing the desiccant systems, the staff noticed a much more consistent comfort level on all floors. Corridor RH levels dropped between 15-20%, even during the highest humidity periods. Bacterial growth, as measured at fifty points before installation of the desiccant systems and three times afterward, declined over 90% due to drier conditions.

Additional Commercial/Institutional Sites in Testing

Similar comparative monitoring studies are underway at other commercial/institutional facilities where desiccant-integrated HVAC designs appear to be well suited. Current monitoring activities include two movie theaters, several fast-food locations, three full-serve restaurants, and two suites of operating rooms at a major metropolitan hospital. Field studies are scheduled to start within the next six months at nursing homes and schools. GRI and DOE plan to publish findings from these research studies as they are completed; the first several are due to be released in late 1998 and first quarter 1999.

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