ORNL/Sub-01-40000081030

Field Demonstration of Active Desiccant Modules Designed to Integrate with Standard Unitary Rooftop Package Equipment

Final Report: Phase 3

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FIELD DEMONSTRATION OF ACTIVE DESICCANT MODULES DESIGNED TO INTEGRATE WITH STANDARD UNITARY ROOFTOP PACKAGE EQUIPMENT

Final Report: Phase 3

J. Fischer
SEMCO, Inc.

J. Sand
Oak Ridge National Laboratory

February 2004

Prepared by
SEMCO, Inc.
for
OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee
managed by
UT-BATTELLE, LLC
for the
U.S. DEPARTMENT OF ENERGY

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ABSTRACT

This report summarizes the investigation of two active desiccant module (ADM) pilot site installations initiated in 2001. Both pilot installations were retrofits at existing facilities served by conventional heating, ventilating, and air-conditioning (HVAC) systems that had encountered frequent humidity control, indoor air quality (IAQ), and other operational problems. Each installation involved combining a SEMCO, Inc., ADM (as described in Fischer and Sand 2002) with a standard packaged rooftop unit built by the Trane Company. A direct digital control (DDC) system integral to the ADM performed the dual function of controlling the ADM/rooftop combination and facilitating data collection, trending, and remote performance monitoring.

The first installation involved providing preconditioned outdoor air to replace air exhausted from the large kitchen hood and bathrooms of a Hooters restaurant located in Rome, Georgia. This facility had previously added an additional rooftop unit in an attempt to achieve occupant comfort, without success.

The second involved conditioning the outdoor air delivered to each room of a wing of the Mountain Creek Inn at the Callaway Gardens resort. This hotel, designed in the “motor lodge” format with each room opening to the outdoors, is located in southwest Georgia. Controlling the space humidity always presented a serious challenge. Uncomfortable conditions and musty odors had caused many guests to request to move to other areas within the resort.

This is the first field demonstration performed by Oak Ridge National Laboratory where significant energy savings, operating cost savings, and dramatically improved indoor environmental conditions can all be claimed as the results of a retrofit desiccant equipment field installation. The ADM/rooftop combination installed at the restaurant resulted in a reduction of about 34% in the electricity used by the building’s air-conditioning system. This represents a reduction of approximately 15% in overall electrical energy consumption and a 12.5-kW reduction in peak demand. The cost of gas used for regeneration of the desiccant wheel over this period of time is estimated to be only $740, using a gas cost of $0.50 per therm—the summer rate in 2001. The estimated net savings is $5400 annually, resulting in a 1–2 year payback.

It is likely that similar energy/cost savings were realized at the Callaway Gardens hotel. In this installation, however, a central plant supplied the chilled water serving fan coil units in the hotel wing retrofitted with the ADM, so it was not metered separately. Consequently, the owner could not provide actual energy consumption data specific to the facility. The energy and operating cost savings at both sites are directly attributable to higher cooling-season thermostat settings and decreased conventional system run times.

These field installations were selected as an immediate and appropriate response to correct indoor humidity and fresh air ventilation problems being experienced by building occupants and owners, so no rigorous baseline-building vs. test-building energy use/operating cost savings results can be presented. The report presents several simulated comparisons between the ADM/roof HVAC approach and other equipment combinations, where both desiccant and conventional systems are modeled to provide comparable fresh air ventilation rates and indoor humidity levels. The results obtained from these simulations demonstrate convincingly the energy and operating cost savings obtainable with this hybrid desiccant/vapor-compression technology, verifying those actually seen at the pilot installations. The ADM approach is less expensive than conventional alternatives providing similar performance and indoor air quality and provides a very favorable payback (1 year or so) compared with oversized rooftop units that cannot be operated effectively with the necessary high outdoor air percentages.

The combined desiccant/rooftop HVAC equipment dramatically improved fresh air ventilation rates and controlled indoor humidity levels compared with the equipment previously installed. Before the retrofits, in both the Hooters and Callaway Gardens facilities, the only source of room ventilation air was infiltration through building envelope seams and open doors.

These pilot sites represent a continuation of previous U.S. Department of Energy–sponsored and Oak Ridge National Laboratory–sponsored active desiccant product development research (Fischer, Hallstrom, and Sand 2000; Fisher and Sand 2002; Fisher et al. 2002). The combined ADM/rooftop units installed at these two sites performed as anticipated, integrated well with the existing HVAC systems, operated reliably, and required minimal maintenance. Most important to the end users, the ADM retrofit resolved the humidity and IAQ problems that had been encountered at these facilities for years, using a cost-effective and energy-efficient equipment solution.

1. INTRODUCTION

1.1 Active Desiccant Module and Pilot Site Justification

As a result of previous work completed by SEMCO as part of an earlier research and development (R&D) program sponsored by the U.S. Department of Energy (DOE) (Fisher, Hallstrom, and Sand 2000), it was concluded that a significant market opportunity exists for cost-effective, compact, energy-efficient active desiccant modules (ADMs) that would process outdoor air and control humidity within facilities traditionally served by packaged heating, ventilating, and air-conditioning (HVAC) equipment. Successful integration with conventional packaged equipment was determined to be essential for gaining wide market acceptance, since approximately 90% of all commercial buildings use this type of air-conditioning hardware.

That same report includes market research that concluded that the availability of a cost-effective active desiccant system integrated with a packaged rooftop system would be especially beneficial to restaurant and hotel/motel facilities. The analysis concluded that these two market segments alone account for 22% of the projected $720 million per year that is the potential outdoor air-preconditioning market for new construction and renovation in the United States (Fisher 2000).

A more recent report (Fisher and Sand 2002) summarized the results of an R&D effort to design, prototype, and test an integrated ADM that sought to meet the criteria mentioned. That report describes both an add-on ADM that can be connected to a conventional packaged rooftop unit and a totally integrated system.

As a logical continuation of this earlier work, two pilot sites were chosen for field testing of the add-on version of the ADM. Since the initial market investigation had identified hotels and restaurants as having a particularly strong market-driven need for this technology, a pilot site was selected to represent each of these two applications. Since humidity control was the primary benefit offered by the ADM, both sites selected were in Georgia, which has a hot, humid climate.

1.2 Challenges to Packaged HVAC Units with High Outdoor Air Fractions

Fisher and Sand (2002) discussed the limitations of off-the-shelf packaged rooftop equipment with regard to conditioning high percentages of outdoor air. They discussed the challenge posed by part-load conditions with packaged units oversized to accommodate peak outdoor air loads, and how these units quickly cool the space to the desired setpoint temperature but then turn off the compressor. If the evaporator fan is run continuously, raw outdoor air is introduced to the space, and the indoor humidity level climbs until the thermostat once again calls for cooling. By this time, the return air entering the cooling coil is elevated in humidity. The result is an elevated dewpoint temperature leaving the cooling coil. Space temperature is maintained, but humidity control is lost, resulting in uncomfortable conditions.

Facilities such as restaurants and hotels often need makeup air or 100% outdoor air systems. Designers and end users often try to apply packaged rooftop units for this purpose because of their low cost and compact size. This can be particularly problematic.

For example, when a conventional rooftop system is applied to handle all outside air, the cooling capacity required at peak conditions is far greater than the cooling output available at the rated airflow (approximately 400 cfm/ton) of the conventional unit. For example, conditioning a 1500-cfm outdoor air stream from 85º and 130 grains to a 56º dewpoint requires 10 tons of net cooling capacity. However, the least amount of air that can be processed by a 10-ton unit without causing control problems, coil frosting, and compressor failure is approximately 3000 cfm (300 cfm/ton), twice the amount necessary.

Even at the reduced flow capacity of 300 cfm/ton, serious performance problems are often encountered at part-load conditions, during which the condenser side performance is extremely high at the very time that the amount of evaporator load is very low. Figure 1 shows the frozen evaporator coil observed at one of the pilot sites for a packaged system operated alone in an attempt to control space humidity at part-load conditions. The addition of the ADM at this location eliminated these problems.

Fig. 1. Frozen evaporator coil in an oversized packaged system.

1.3 The Active Desiccant Module Approach

The ADM add-on approach positions the ADM module downstream of the evaporator coil contained within the packaged unit (Fig. 2). The ADM includes an active desiccant wheel sized to handle approximately 33 to 45% of the airflow processed by the packaged unit (although this can vary). A bypass damper is included to maintain the desired flow through the desiccant wheel and allow bypass of the desiccant wheel during heating mode if desired. The ADM system tested at the pilot sites integrated a direct-fired burner and fan to process outdoor air for regeneration of the active desiccant wheel. This burner can easily be replaced with a hot water or steam coil if desired, as will typically be the case when this technology is applied indoors or as part of a combined cooling, heating, and power system.

Fig. 2. SEMCO active desiccant module connected to a 7.5-ton standard rooftop unit at a hotel at Calloway Gardens.

The ADM concept positions the active desiccant wheel "downstream" of the cooling coil to provide saturated air to the desiccant wheel, thereby maximizing its operating effectiveness and minimizing the required regeneration temperature (Fig. 3). As previously mentioned, typically about 33% to 45% of the air passing through the cooling coil is processed by the active desiccant wheel. This fraction of the air is dried to a very low dewpoint and heated by the energy released from the desiccant as heat of adsorption. This warm, very dry air is then mixed with the remainder of the cool, moderately dry air leaving the evaporator coil of a standard packaged rooftop unit to provide building ventilation air at the desired dewpoint and at a room-neutral temperature.

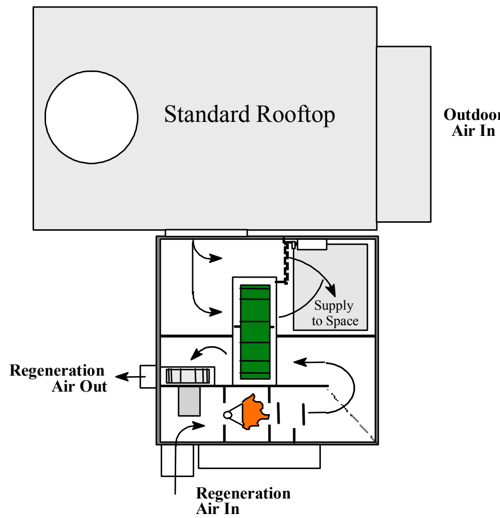

Fig. 3. In the ADM configuration, the active dehumidification wheel is positioned after the cooling coil located in the standard rooftop unit.

Fisher and Sand (2002) discussed the many advantages offered by this configuration, along with supporting test data. Later sections of this report discuss at length the specific advantages offered to the pilot sites at the Hooters restaurant and the Mountain Creek Inn at the Callaway Gardens resort.

One of the most important advantages offered by the ADM approach, which was required to remedy the humidity problems experienced at both the restaurant and Callaway Gardens, was the ability to vary the amount of latent and sensible cooling capacity delivered [i.e., achieve a variable sensible heat ratio (SHR)]. This capability is needed to balance the energy content in the outdoor air with the needs of the occupied space.

Variable SHR performance is one of the most fundamental differences between the ADM approach and a conventional or customized vapor compression system. The ADM can be controlled and operated to vary the dewpoint leaving the system while simultaneously varying the amount of reheat provided. This can be accomplished by modulating the rotational speed of the active desiccant wheel, the amount of regeneration energy used, the amount of air bypassed around the desiccant wheel, or the stage of vapor-compression cooling energized.

As shown in Fig. 4, the ADM can be operated to provide an SHR that can vary from 81 to 34% when processing 100% outdoor air at a typical dewpoint design condition. The conditions shown in dark green would occur when the system calls for maximum cooling. At this condition, the regeneration energy would be minimized or turned off, allowing the ADM to function more like a conventional packaged unit. The conditions shown in light green occur when maximum dehumidification is desired. This is accomplished with the least amount of bypass air around the dehumidification wheel and with the maximum regeneration energy input. The conditions shown as red and blue are obtained as the bypass air is increased and/or the regeneration energy is decreased.

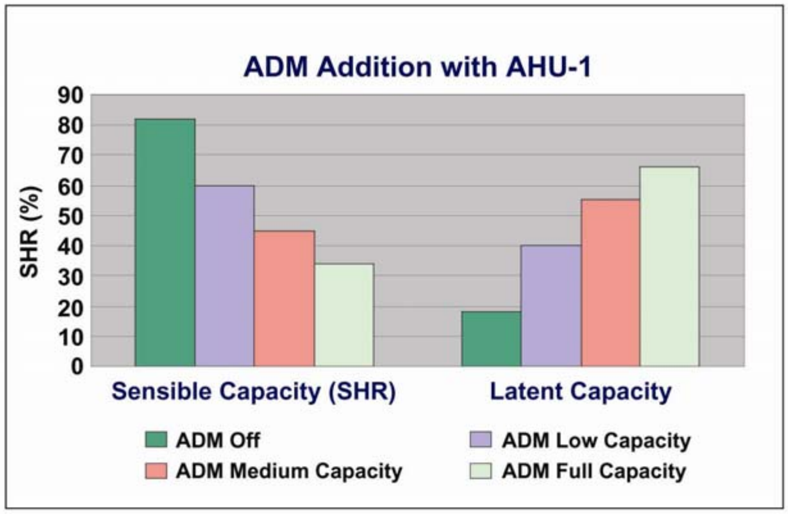

Fig. 4. The SHR and amount of latent capacity leaving the ADM can be varied by modulating either the amount of bypass air or the amount of regeneration energy used by the dehumidification wheel.

2. BACKGROUND: PILOT SITE SELECTION

2.1 Site Details

Restaurant Makeup Air—Hooters of Rome

The Hooters of Rome restaurant is a single-story structure approximately 10 years old located at the end of a traditional strip shopping mall in Rome, Georgia (Fig. 5). The facility contains a large dining area capable of seating approximately 120 people, a small food-preparation area located in a separate room, cold storage, and two restrooms. The kitchen is located within the dining area with the kitchen hood and cooking area in plain view of the customers.

Fig. 5. The Hooters restaurant building in Rome, Georgia.

Before the ADM retrofit, the facility was conditioned by four rooftop units. The dinning area was served by two 7.5-ton units and one 5-ton unit. The food preparation area was served by a separate 7.5-ton unit. Two kitchen exhaust hood fans were located on the roof of the facility, along with a makeup air fan that delivered approximately 3000 cfm of unconditioned outdoor air directly to the canopy located over the cooking area, in front of the 30-ft-long kitchen exhaust hood.

Hotel Room Pressurization Air—Callaway Gardens Resort

Callaway Gardens is a large resort complex located southwest of Atlanta in Pine Mountain, Georgia, that features a variety of accommodations such as cottages and villas. The 349-room Mountain Creek Inn consists of three mid-20th century-vintage two-story buildings built in the fashion of motor lodges, with each room accessed through doors opening to the outside (Fig. 6). The pilot site discussed by this report involved the installation of equipment on a portion of one of these three buildings.

Each building has a 12-in.-wide plumbing and electrical chase running down the middle of each wing, with two floors of rooms on either side. This chase also served as the air pathway for the bathroom exhaust air pulled from each room by two fans located on the roof of each building. Each room was cooled by a chilled-water fan-coil unit, served by a 600-ton central electric chiller plant. The makeup air for each room was pulled through the door leading to the outside when the door was opened or, for most of the time, through cracks under the doors or other pathways that existed within the building envelope.

Fig. 6. Callaway Gardens hotel building where the active desiccant module retrofit was installed.

2.2 Problems that Existed at the Pilot Sites before the ADM Retrofit

Restaurant Makeup Air—Hooters of Rome

The Hooters restaurant experienced the classic problem encountered by most restaurants, that of adequately conditioning the makeup air required to replace air exhausted from the kitchen hood. At this site, approximately 3000 cfm of air was being exhausted from the kitchen hood. The original design called for a makeup air fan to introduce unconditioned air directly to the canopy covering the kitchen area (Fig. 7). This fan was turned off soon after the building was occupied because it resulted in uncomfortable conditions for workers in the kitchen area (too hot during the summer and too cold during the winter). It also allowed a high percentage of the humidity in the outdoor air that was introduced to the kitchen area to migrate into the dining area, since the two areas were not physically separated.

Fig. 7. Photo of the dining area of Hooters, which is separated from the kitchen area by the bar and canopy.

Since all four rooftop units were operated in 100% recirculated air mode, no outdoor air was being provided to the restaurant to compensate for the kitchen exhaust. Therefore, the kitchen exhaust air fans were required to pull makeup air through the doors and other pathways within the building structure, resulting in excess static pressure. This resistance reduced the amount of exhaust airflow leaving the kitchen hoods, reducing the capture efficiency and impacting indoor air quality (IAQ).

Originally, the two 7.5-ton rooftop units serving the dining area were designed to bring in approximately 1000 cfm of outdoor air each to help make up for the kitchen exhaust. It was soon determined that the packaged units could not be effectively operated with what amounted to 50% outdoor air, especially in the relatively hot and humid environment of Rome, Georgia. Complaints of high space humidity were common and, on hot days, the space could not be adequately cooled. As a result, the owner closed the outdoor air intakes of all three packaged units.

Because the outdoor air entering the facility was unconditioned, the humidity in the space remained uncontrollable. The standard rooftop units, processing all return air, operated with a very high sensible heat ratio, thereby providing limited dehumidification capacity. These units ran long enough to meet the space temperature setpoint, then cycled off. During part-load conditions and partial occupancy (conditions that occurred most often) the humidity control was particularly bad.

In an attempt to achieve comfort, especially for the highly active wait staff, the thermostat setting was continually dropped to compensate for the high indoor humidity. This resulted in "cold and clammy" conditions within the facility much of the time. Because most of the customers were seated at rest, not highly active, they would quickly become uncomfortably cold under these conditions and ask the manager to turn off the air-conditioning. This presented a dilemma to the manager, who had to choose between making the customers happy or satisfying the employees. This common problem in restaurants has been termed "thermostat wars."

This facility was originally designed with only three packaged rooftop units. A fourth unit was added in an attempt to provide comfortable conditions. However, the manager found that adding the fourth unit provided little noticeable improvement in comfort.

The owner tried numerous operating strategies in addition to adding cooling capacity, none of which resulted in a comfortable facility. In addition to the complaints regarding comfort (temperature and humidity), the lack of effective ventilation caused the space to feel stuffy and allowed cigarette smoke to build up within the facility.

Hotel Room Pressurization Air—Callaway Gardens Resort

Callaway Gardens is a beautiful resort area that caters to many business retreats for Fortune 500 companies. The cost of the rooms in the Mountain Creek Inn is high, as is the level of service provided. Therefore, when important clients complained that the IAQ within the hotel was unacceptable, an effective solution was needed.

High humidity in the guest rooms was causing mold and mildew, dampness in the walls and bedding, and condensation on the windows, especially when guests were using the showers. This resulted in musty odors throughout the facility. Occupants complained of feeling wet and clammy. Wallpaper and beddings within the rooms were frequently replaced at a high cost to Callaway. Despite the cost, the frequent renovations failed to resolve the odor and comfort complaints.

The inn is a traditional motel-type structure with three two-story connected buildings or wings. Room access is from the outdoors, with no central hallways. The chilled-water system serving old fan-coil units simply did not deliver enough latent capacity to effectively maintain acceptable space humidity in each room.

In an earlier attempt to improve the space humidity control, a contractor had installed variable-speed drives and larger chilled-water pumps, but this attempt at capacity control created new problems. At part-load conditions, the humidity control within the rooms got even worse. Relative humidity (RH) levels in excess of 80% were recorded.

Despite the owner's best efforts, the conditions got so bad that the most important corporate customers (returning guests) began requesting rooms in one particular building where the dampness was somewhat less severe than in the other wings. Some threatened to relocate their conferences if improvements were not made.

3. RESULTS: HOOTERS PILOT INSTALLATION

3.1 Installation Description

The Hooters facility needed 3100 cfm of outdoor air to offset the air exhausted from a 30-ft-long kitchen exhaust hood and two large restrooms. An ADM manufactured by SEMCO was installed along with a standard-efficiency, two-stage 10-ton rooftop unit to effectively deliver 100% outdoor air for ventilation and pressurization. An older rooftop unit was removed, allowing the existing ductwork and the supply and return air roof penetrations to be reused.

As shown in Fig. 8, the rooftop unit is mounted with the return air entering the bottom of the unit and supply air being discharged from the side (standard horizontal airflow configuration). The air leaving the rooftop unit is delivered to the ADM via a short section of flanged ductwork, allowing the outdoor air leaving the cooling coil within the packaged unit to be pushed through the supply side of the active desiccant wheel or around the adjacent bypass damper.

Fig. 8. The ADM module and packaged rooftop unit being installed on the roof of the Hooters restaurant in Rome, Georgia.

The existing ductwork was used to distribute the preconditioned outdoor air to the space. One new supply grill was added to accommodate the increased airflow. This ductwork covers an area adjacent to a wall situated farthest from the kitchen area. This arrangement was desirable because it provided for the best possible IAQ, sweeping the conditioned outdoor air over the occupants before it exited from the kitchen area.

This arrangement presented a significant challenge to the ADM system. Since it is operated as a 100% outdoor air system and serves as the sole source of make-up air for the kitchen hoods and restrooms, it runs continuously. As a result, the condition of air leaving the ADM needs to be relatively consistent and comfortable. Otherwise the customers seated in the dining area would complain.

As discussed in Section 4.5, occupant comfort was significantly improved as a result of the addition of the ADM. New temperature and humidity sensors were located within the center of the restaurant. These sensors are used by the DDC system to modulate the two compressors in the rooftop unit serving the ADM, as well as the direct-fired gas burner control valve providing regeneration heat to the desiccant wheel. The space setpoints used are 75º dry bulb and 50% RH. The ADM is controlled to provide air dry enough to handle the latent load associated with both the outdoor air and the space (people plus cooking). As a result, the fans and coils for the two other rooftop units serving the dining area can be cycled since they are used primarily for maintaining space temperature and are not depended upon for dehumidification. The control logic used for this site is discussed in a later section.

The ADM system operates any time that the kitchen hood is in use, typically from 10:00 A.M. until 12:30 A.M., 7 days per week. During the off hours, a separate unoccupied mode is used that reduces the amount of outdoor air to a minimum and recirculates room air. In this mode, the fan located within the rooftop unit operates only when the unoccupied humidity setpoint is exceeded. The active desiccant wheel is energized, as necessary, to achieve the desired space dewpoint. The compressor-driven cooling within the rooftop is not typically used during the off hours.

3.2 System Description and Schematic

Approximately 1300 cfm (42%) of the 3100 cfm of preconditioned air is passed through the active dehumidification wheel. As shown in Fig. 9, the air that passes through the active dehumidification wheel leaves warm and extremely dry (typically 100°F and 20 grains of moisture). This dry, warm air mixes with the air bypassing the wheel (typically near 60°F and 74 grains) to deliver air to the occupied space that is at a room-neutral temperature and at a low dewpoint (typically in the range of 77°F and 50 grains, a 48°F dewpoint). The optimum temperature and humidity level delivered to the space at any given time is determined by the DDC system.

The only supply/outdoor air fan that is used by the combined ADM/rooftop system for the Hooters site is the fan located within the conventional packaged unit. This forward curve fan, typical of those supplied with package equipment, is not capable of overcoming high external static pressures. As a result, the active desiccant wheel used in the ADM must have a low pressure loss to avoid the need for a secondary fan.

The regeneration fan located within the ADM unit only needs to overcome the static pressure loss across the regeneration portion of the wheel and the direct-fired gas burner. Moreover, only about half of the airflow through the supply side of the wheel is used for regeneration. Thus, a small, fractional horsepower motor can be used, keeping any parasitic energy to a minimum.

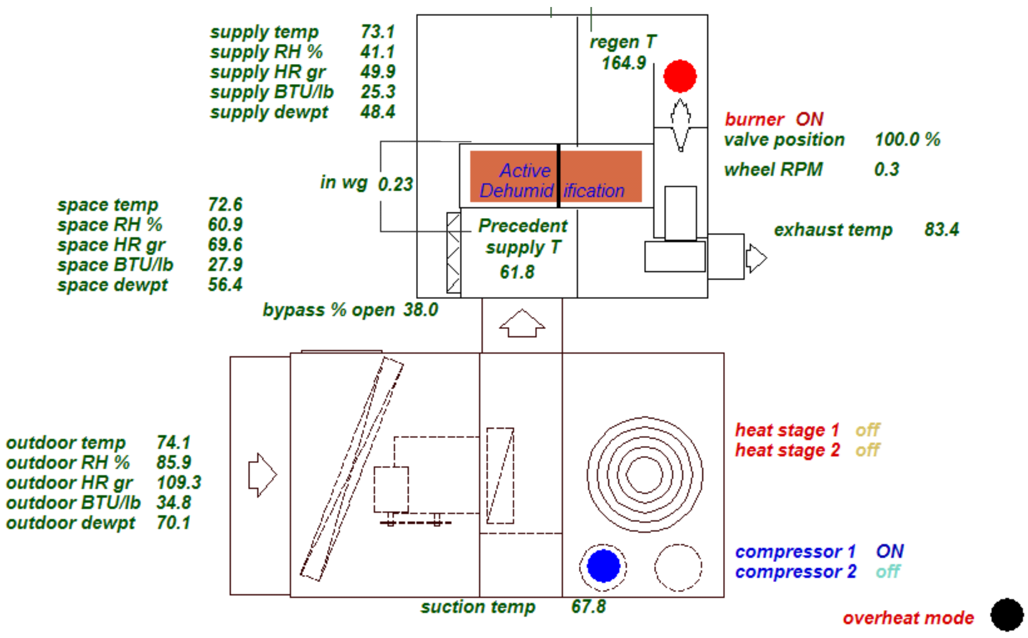

Fig. 9. Schematic showing the airflow-path through the ADM and some of the points monitored (all temperatures are in °F).

3.3 Controls, Instrumentation, and Data Acquisition

Controls

Based on the very positive results obtained with direct digital control (DDC) in previous pilot installations of similar desiccant-based equipment, a DDC system was used to modulate the active desiccant speed, the amount of bypass air, the regeneration temperature, and the stages of cooling. It also provided remote, real-time data acquisition and energy utilization monitoring. It provided a "virtual laboratory" for controlling unit operating parameters while observing and recording the overall performance of the ADM installed at the Hooters site.

As shown by Fig. 9, controlling and monitoring the ADM required seven precision RTD temperature sensors, three precision humidity sensors, various control valves, and start/stop signal inputs and outputs. The DDC controller was driven by a custom program module developed by SEMCO and based upon automated logic controls. It monitors the space temperature and humidity, compares them with user-specified inputs, then adjusts the modulating valve serving the regeneration burner, the stages of cooling, and other controlled components as needed to reach setpoint through a preprogrammed algorithm. System setpoints, control routines, and calculated energy savings were all programmable and changeable through the DDC system on site and remotely through a modem.

Instrumentation

The same sensors used to control the operation of the ADM were used for instrumentation and performance monitoring. A significant advantage offered by the DDC approach is its graphics capabilities. Customer feedback was extremely positive for this feature, because the function of the ADM, which is difficult to explain verbally or with sales literature, is easily understood by simply viewing this graphical presentation during actual operating conditions.

In addition to the computer graphics that could be accessed via modem, a liquid crystal display was installed within the restaurant that allows the manager to track the space temperature and humidity and change setpoints (which the manager elected not to do).

Data Acquisition

The performance data for the ADM system could be monitored and collected in three ways. Instantaneous data could be recorded by downloading the system schematic showing live data. The data for all of the selected state points could recorded and saved in the control module memory for up to a one-month period (depending upon the frequency of sample collection). The third and most effective option, used in preparing this report, is trending the stored performance data over time. Examples will be provided in the next section of live data and stored performance data.

3.4 System Performance Data

ADM as a Dedicated Outdoor Air System

The Hooters restaurant installed the ADM to serve as a dedicated outdoor air system (DOAS). A DOAS processes the entire outdoor air load (sensible and latent). Of equal importance, it is sized to handle at least a portion of the internal latent load associated with the building occupants and infiltration. During the engineering phase of the Hooters project, it was determined that since two existing rooftop units were in place, these units would be used to process most of the space sensible load and approximately half of the space latent load at peak design conditions (full occupancy and outdoor design). During the vast majority of the operating hours, the ADM would handle essentially all of the latent load, and the two existing rooftop units would operate as needed for sensible cooling only.

Table 1 shows the original load analysis calculations for this project. Note the design challenge, typical of restaurant facilities, posed by the high outdoor air load required for kitchen exhaust makeup and/or ventilation load as required by American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Standard 62, *Ventilation for Acceptable Indoor Air Quality* (ASHRAE 1999). The resulting SHR for the dining area of this building was determined to be only 0.38. With conventional equipment, typically operating at SHRs of more like 0.7 to 0.8, the anticipated results would be poor humidity control and cool, clammy operating conditions. This is precisely what was commonplace at the Hooters facility. Monitoring completed before the pilot installation confirmed space temperatures around 69°F and 75% RH.

Table 1. Original design load analysis for Hooters ADM retrofit (note the low sensible heat ratio)

Hooters Load Analysis: Peak Occupancy and 2% ASHRAE Dew-point Design (Space controlled to 75°F and 50% RH)			
Sensible Load			**Total BTU**
	People	120 people at 275 BTU/person	33,000
	Lights	40 lights at 40 watts	5,460
	Envelope		30,000
	Other		4,500
	Outdoor air		16,200
	Total Sensible		89,160
Latent Load			**Total BTU**
	Guests	110 people at 275 BTU/person	30,250
	Waitresses	10 people at 350 BTU/person	3,500
	Infiltration		2,500
	Outdoor air		112,200
	Total Latent		148,450
Total Load			237,610
Tons of cooling			20
SHR		(sensible heat ratio)	**0.38**

Table 2 shows the load distribution between the ADM and the two exisiting 7.5-ton rooftop units that were kept at the Hooters site. As shown, with the ADM operated to provide 3000 cfm of outdoor air continuously, conditioned to a room-neutral temperature (approximately 75ºF) and with a dewpoint of 51ºF (56 grains) during peak cooling conditons, the SHR is only 0.1 (90% latent cooling). The capability of the ADM to provide this level of effective dehumidification allows the exisiting rooftop units to operate with an ideal 0.8 SHR while processing only recirculated air.

In this way, the outdoor air load is “decoupled” from the space, allowing the ADM to maintain the desired humidity control while providing all of the ventilation/make-up air needed. The two separate rooftop units are cycled to maintain the space temperature.

Table 2. Load distribution example with all of the outdoor load and half the space latent load being handled by the ADM/rooftop combination.
Two of the original rooftop units are operating at the proper 0.8 SHR.

System Load Distribution at Design: ADM applied as a DOAS (ADM delivering air at 56 grains, room neutral temperature)		
ADM/Rooftop		**Btu**
Outdoor sensible load	all	16,200
Outdoor latent load	all	112,200
Indoor latent load	50%	18,125
Indoor sensible	none	0
Total Load Handled		146,525
Tons of cooling		12
SHR		**0.11**
Remaining Rooftops		
Outdoor sensible load	none	0
Outdoor latent load	none	0
Indoor latent load	50%	18,125
Indoor sensible	all	72,960
Total Load Handled		91,085
Tons of cooling		8
SHR		**0.80**

ADM vs a Custom Refrigeration-Based Dedicated Outdoor Air System

The supply conditions used for Table 2, 76°F and a 51°F dewpoint, are not easily attainable with customized outdoor air systems that are based on vapor compression refrigeration. The ADM offers numerous control advantages that are not available with refrigeration-based systems (Fisher and Sand 2002). The amount of installed condenser capacity, as well as the cost of operation, is significantly higher for the customized direct expansion (DX) system.

Table 3 shows a very simple comparison of the equipment capacity necessary to obtain the conditions listed in Table 2. What is interesting, and often ignored by designers, is that the "nominal" tons of cooling provided by a packaged system changes significantly depending upon the required air temperature leaving the coil and reflected by refrigeration suction pressure. Table 2 shows that the rooftop unit feeding the ADM produces approximately 12 tons of output with a 10-ton compressor. The output is a result of the relatively high temperature of the air leaving the cooling coil (high suction temperature). In contrast, the customized DX unit requires a capacity of more than 21 tons to deliver a net cooling capacity of 19.5 tons because the system has to cool the outdoor air to approximately 51°F to provide the necessary dehumidification.

The ADM's use of an active dehumidification wheel to attain very low dewpoints presents a significant performance advantage over alternative technologies. Figure 10 shows a summary of data from two major HVAC manufacturers that relate the temperature leaving the evaporator coil to the derating factor that determines the gross cooling capacity needed to achieve a desired net cooling output. As shown, the ADM approach will generally get a capacity credit (as experienced at Hooters and laboratory testing) because of the moderate evaporator temperatures. The conventional cooling approach, requiring very low evaporator temperatures, will generally be de-rated.

Table 3. Cooling capacity and energy comparison of the Hooters ADM and a custom direct expansion alternative

	ADM Rooftop Combination	Custom DX Rooftop (over-cool and reheat)
Installed Cooling Capacity Required (Gross Tons)	10 Tons	21 Tons
Cooling Capacity Output Delivered (Net Tons)	12 Tons	19.5 Tons
Reheat Energy Required (Btu/h) Regeneration Energy Required (Btu/h)	0 77,760[b]	45,360[a] N/A
Supply Dew Point Used for Analysis	51°F	51°F
Annual Cooling Energy Cost[c]	$1,760	$3,940

[a]Assumes that 3000 cfm of air leaving the cooling coil at 51ºF is reheated to 65ºF; colder air would over-cool the space at this airflow.
[b]Maximum regeneration energy reduced at off-peak conditions, direct-fired burner raising 600 cfm from 80 to 200ºF.
[c]Analysis assumes operation 7 days/week, 16 hours/day, electric cost $.07/kWh, $6.00 demand, gas cost at $0.50 per therm—summer rates during 2000–01 when test was conducted. Gas use of 1480 therms can be adjusted for current rates.

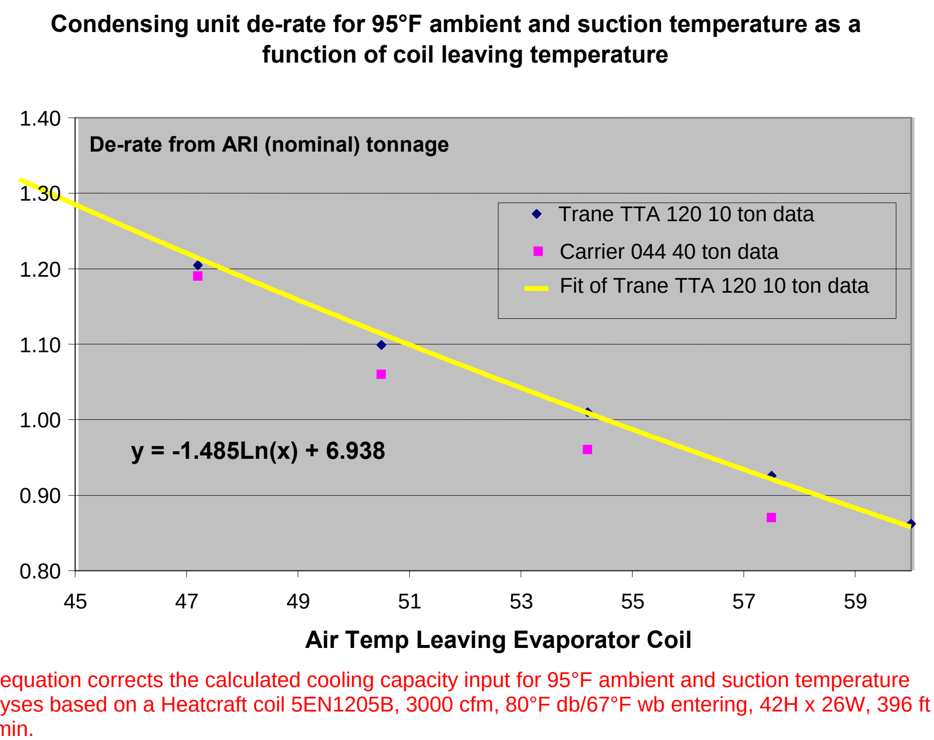

Fig. 10. Graph and algorithm for correcting the ARI nominal condenser capacity for low (or high) leaving coil temperatures. The graph combines suction temperature data with coil performance.

ADM Performance at the Hooters Site

The ADM/rooftop combination performed well at the Hooters installation. As shown by Figs. 11 and 12, the desired space conditions of 75°F and 50% RH were maintained with only minor fluctuations during extreme conditions. Before the installation of the ADM, the space was uncomfortably hot and humid during times of high outdoor humidity. During off-peak conditions, the space was consistently cold and clammy, typically less than 70°F with humidity levels exceeding 70% RH.

Figure 11 shows the outdoor air and space humidity content (grains) during a design day. The space is maintained at the desired humidity level even when the outdoor air reaches 140 grains. As the outdoor air drops to only 90 grains, the amount of regeneration energy and/or cooling energy input (stage of cooling) is reduced while the humidity set point is maintained. These part-load conditions, which provide a significant challenge to conventional cooling systems (as discussed in Section 2.1 and shown in Fig. 1), proved no problem for the ADM approach.

Figure 12 shows several consecutive days where the outdoor air temperature was over 90°F. The ADM/rooftop combination was controlled to maintain the space humidity while maintaining the space temperature below 75°F. The two existing rooftop units were primarily responsible for maintaining the space temperature. As shown, even when the outdoor air exceeded 90ºF, the space temperature setpoint was maintained.

Figure 13 shows a unique capability of the ADM system. Even during times of extreme outdoor humidity, when the latent load associated with the outdoor air and infiltration is greatest, the outdoor air supplied to the space can be dehumidified to a very low dewpoint (approximately 50ºF), allowing the space humidity to be maintained at the 50% RH setpoint. The fluctuation in the humidity level supplied by the ADM is in response to the variable humidity load that exists within the space, enabling a consistent indoor humidity level. The lower humidity is produced by increasing the regeneration energy and/or decreasing the amount of bypass air around the active desiccant wheel. Note that on design days, as reflected in Fig. 13, more internal latent load existed than anticipated during the initial design phase (Table 2). Therefore, the ADM was processing the latent load associated with the entire facility, allowing the existing rooftop units to provide the necessary sensible cooling only.

Note that air as dry as 47 grains (a 46.5ºF dew point) was required at times to maintain the desired space humidity level. For a conventional approach to provide this level of humidity, air leaving the coil would have to have a temperature of approximately 47ºF. As shown by Fig. 10, the gross cooling input would have to be increased by 20% over the net cooling output because of the low suction temperature. With entering conditions of approximately 76ºF and 125 grains, the conventional approach would require a cooling input of approximately 26 tons.

For the ADM, the condition leaving the coil is far more moderate, in this case approximately 62°F, and allows a 10-ton packaged system to provide approximately 12 tons of output. Since most of the dehumidification is accomplished by the active desiccant wheel contained within the ADM system, the 47-grain air was delivered using only a 10-ton input and approximately 77,760 Btu/h of direct-fired gas heat for regeneration energy. This reduction in required cooling capacity and the use of low-cost gas during the cooling season results in significant energy cost savings.

During unoccupied periods, typically between the hours of 12:30 A.M. and 10:00 A.M., the conventional rooftop systems were turned off and only the ADM was operated as necessary to maintain the desired unoccupied humidity setpoint. This arrangement allows the occupied periods to begin in an energy-efficient manner, without first removing the high humidity load typical of normal "shutdown" periods associated with conventional systems. It also contributes to additional overall energy cost savings.

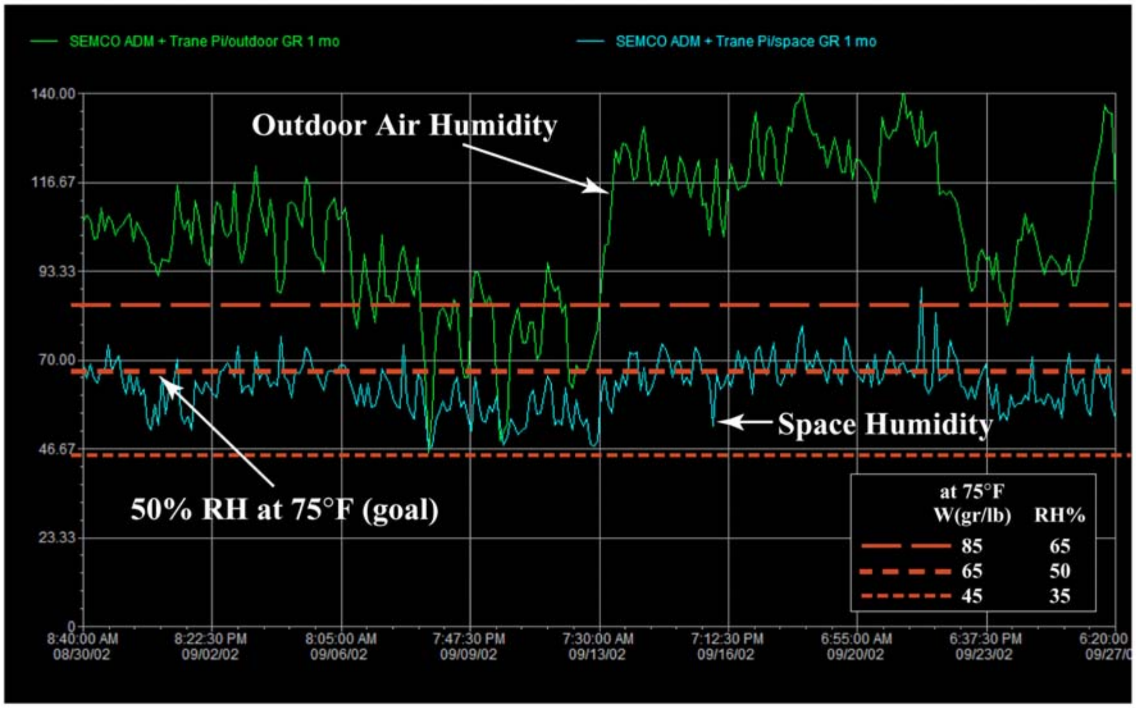

Fig. 11. Actual data comparing the outdoor air and space absolute humidity levels (grains) during a one-month period for the Hooters restaurant site.

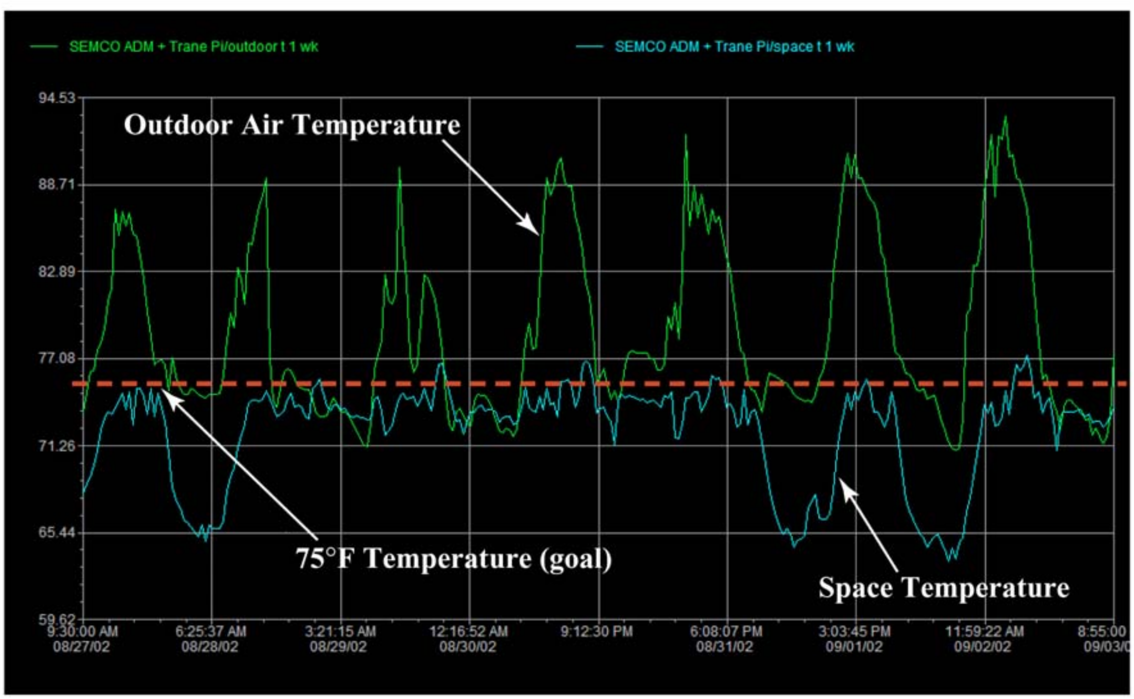

Fig. 12. Actual data comparing the outdoor air and space temperatures during a one-week period for the Hooters restaurant site.

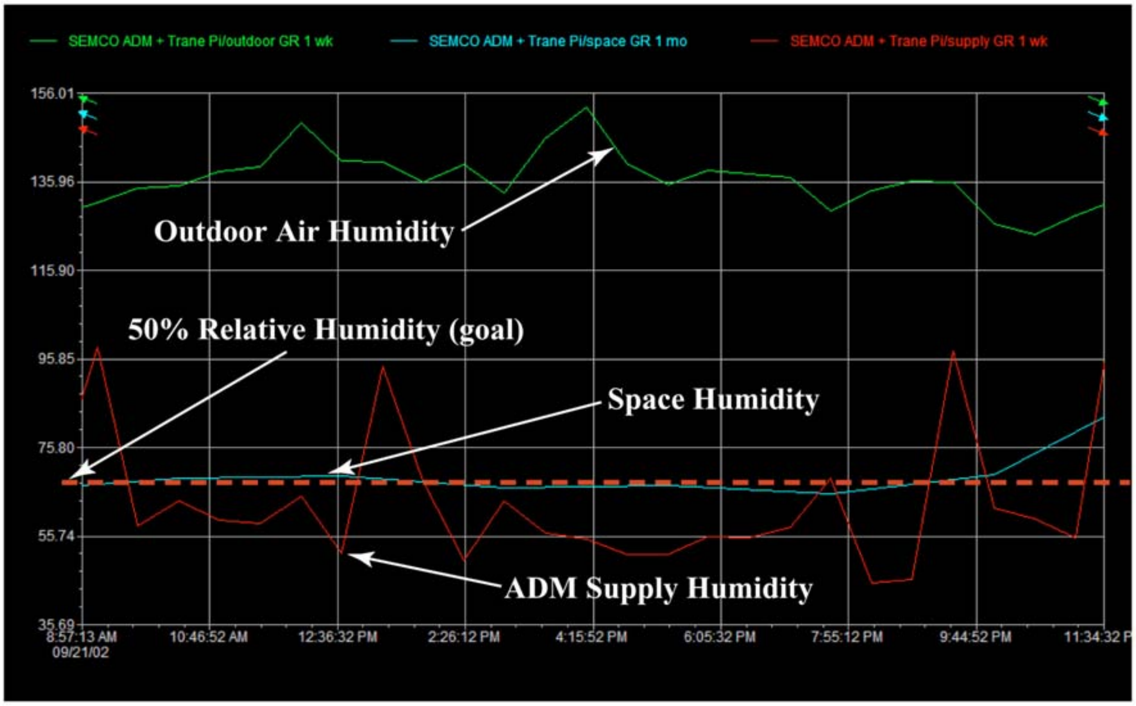

Fig. 13. Actual data showing the outdoor, space, and supply humidity levels leaving the ADM during extreme humidity conditions for the Hooters site.

Energy Benefit Resulting from Improved Humidity Control

Even though the ADM was installed to deliver an increased amount of ventilation air that would need conditioning, the installation at the Hooters restaurant resulted in a reduction of approximately 15% in overall electrical energy consumption and a reduction of approximately 12.5 kW in peak demand, representing a drop of about 34% in electricity consumption by the air-conditioning system (see Table 4). The restaurant was operated 16 hours per day, 7 days per week. Using an average electrical rate of $0.09 per kWh, the annual air-conditioning electrical energy can be estimated to be approximately $5400 per year. The cost of the gas used for regeneration of the desiccant wheel over the same period is about $740, assuming a gas cost of $0.50 per therm.

Table 4. Electrical breakdown for Hooters restaurant after ADM installation and 15% reduction in overall consumption

Source	Peak Power Consumption (KW)	Percent of Total
High Voltage Equipment (230V)(1)	23.7	28%
Low Voltage Miscellaneous (120V)(2)	9	11%
Kitchen Hood Exhaust Fans	5	6%
Lighting	9	11%
HVAC Equipment (3)	36.8	**44%**
Total	**83.5**	**100%**

(1) Includes coolers, ice machines, large toaster, steamer, etc.
(2) Includes dish washer, coffee machines, drink dispensers, small coolers and other miscellaneous items
(3) Original HVAC equipment included three 7 1/2 ton and one 5 ton conventional packaged unit

A primary factor contributing to the overall energy savings at the Hooters facility was the capability to effectively maintain humidity within the occupied space. Controlling humidity is important because the absolute humidity level (dewpoint) in a human environment impacts the perspiration evaporation rate, which helps regulate the body's energy balance, skin moisture levels, and thermal sensation. This is particularly important for restaurant facilities where the waiters are very active while the customers are seated at rest and often lightly dressed. (An excellent discussion of the interrelationship between human comfort and humidity can be found in Harriman 2001b, pp. 73–75).

As the dewpoint decreases, the rate of evaporation from the skin's surface increases, as does the associated energy loss. As a result, skin temperature drops, the body feels cooler, and a warmer space temperature is comfortable. The effect of high humidity is most pronounced during warm conditions (cooling season), especially at levels of increased activity (not seated at rest), since perspiration accounts for a much larger percentage of the body's overall energy balance. For these reasons, it is logical that as space dewpoint levels are reduced a desired comfort level can be maintained at warmer temperatures (higher thermostat settings). Conversely, at elevated dewpoints, building occupants will prefer much cooler space temperatures.

Before the ADM retrofit, the Hooters facility was monitored to benchmark typical indoor air conditions. This testing showed that the space temperature and humidity fluctuated widely, but typical conditions were found to be about 70°F and 70% RH. The wait staff was constantly adjusting the thermostat to a lower temperature, while the customers complained that the space was too cold and clammy. Adding a fourth packaged cooling unit did nothing to improve comfort. However, once the ADM was installed, the space temperature could be raised to 75°F and the space humidity dropped to 50% (Figs. 14 and 15).

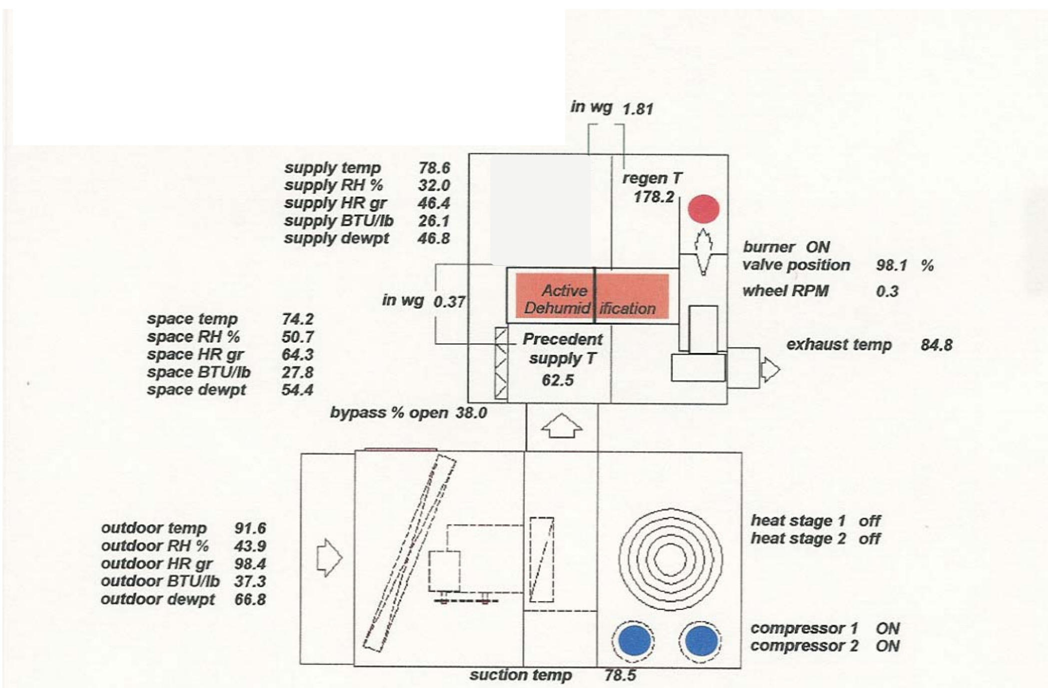

Fig. 14. Sample of the flow schematic used by the DDC control/monitoring system at Hooters. Shown are actual monitored cooling season conditions (all temperatures are in °F).

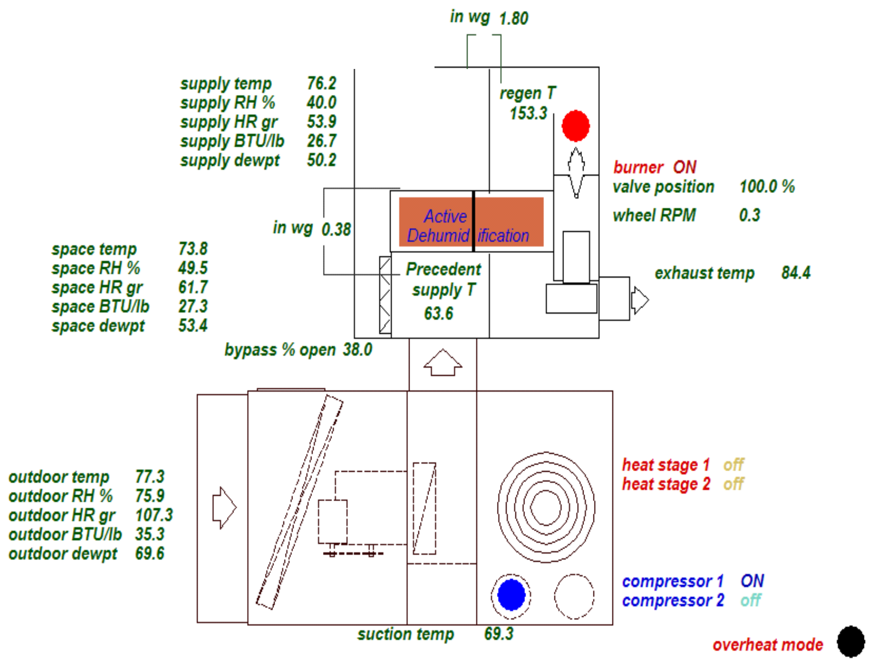

Fig. 15. Sample of the flow schematic used by the DDC control/monitoring system at Hooters. Actual data shown at part-load conditions (all temperatures are in °F).

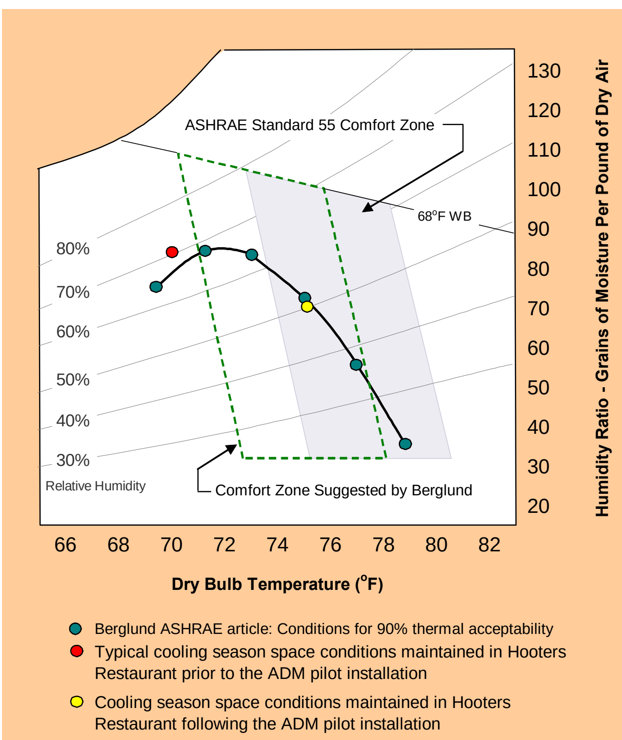

Fig. 16. Conditions within the Hooters facility before and after the ADM relative to the ASHRAE comfort zone.

Figure 16 is similar to one included in Berglund 1998 (referenced in the *ASHRAE Humidity Control Design Guide*). It presents test data reported by Berglund (shown as dark circles) that link humidity levels with a corresponding dry bulb temperature necessary to reach thermal acceptability for 90% of the adapted space occupants (10% dissatisfied) during the cooling season. A 90% criterion was used to determine the comfort zone for overall thermal acceptability shown in Fig. 16 and included in ASHRAE Standard 55, *Thermal Environmental Conditions for Human Occupancy* (ASHRAE 2001).

Shown in red and yellow are the space conditions measured before and after the ADM retrofit, respectively. Figure 16 makes clear why restaurant workers lowered the thermostat setting; they did so in response to high humidity. Likewise, the figure shows why a 5°F warmer space temperature could be more comfortable as the humidity level was reduced from 85 grains to 70 grains.

Controlling the space humidity provided a double benefit to the restaurant owner. First, both the waiters and the customers were more comfortable and, most important, they were both comfortable at the same space conditions. Second, because the thermostat setting could be raised by 5°F, significant cooling-season energy cost savings are achieved.

DOE 2.1 modeling runs for a restaurant like Hooters, located in northeastern Georgia, show the energy savings potential of higher cooling-season, thermostat set points made possible by lower space humidity conditions (Fig. 17). As shown in this figure, summer cooling-load energy savings ranging from 7 to 40% were predicted for a restaurant where the thermostat set point can be increased over a range of 1 to 5ºF, respectively, because more-comfortable space conditions are possible at higher temperatures at decreased humidity levels. These results agree remarkably well with energy savings actually seen at the field installation.

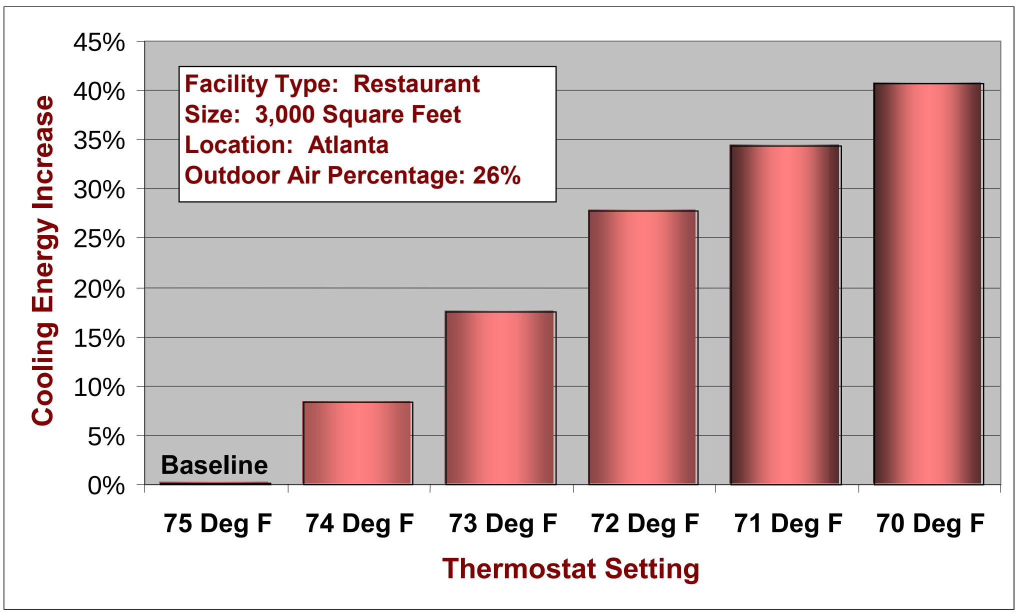

Fig. 17. Summer cooling energy increases resulting from lower thermostat set point temperatures—DOE 2.1 modeling results for an Atlanta restaurant like Hooters.

3.5 Interview with Building Owner

A follow-up interview with the restaurant owner confirmed his satisfaction with the system. One of the most telling comments was that he was surprised to see that "people could actually be cool at 75ºF degrees." He said that he always knew that humidity was important, but he had no idea that it could make such a difference in comfort. He is extremely pleased that both the waiters and the customers are now comfortable. In addition, the space is now much better ventilated, so it both feels more comfortable and better accommodates both smokers and non-smokers.

Previously, on hot, humid days, condensate would run down the outsides of the windows, making it impossible to see outdoors. The ADM installation has eliminated that problem.

Based on a comparison of energy bills, the cost of conditioning the facility has dropped by approximately 15% at the same time that more outdoor air is being delivered to the space and the humidity control has been improved.

4. RESULTS: MOUTAIN CREEK INN

4.1 Installation Description

The Mountain Creek Inn at Callaway Gardens required approximately 2100 cfm of outdoor air to offset the air exhausted from the bathrooms of approximately 42 guest rooms. An ADM manufactured by SEMCO was installed along with a standard-efficiency, two-stage 7.5-ton rooftop unit to satisfy this requirement.

As shown in Fig. 18, the rooftop unit is mounted with the return air entering the bottom of the unit and the supply air discharging from the side (standard horizontal airflow configuration). The air leaving the rooftop unit is delivered to the ADM via a short section of flanged ductwork, allowing the precooled, 100% outdoor air leaving the cooling coil of the packaged unit to be pushed through the supply side of the active desiccant wheel or around the adjacent bypass damper.

Fig. 18. The ADM and packaged rooftop unit installed on the roof of the Callaway Gardens facility.

New ductwork was installed to distribute the preconditioned outdoor air to the space. This presented the most significant challenge to this retrofit installation, since the existing piping chase would only accommodate 6-in. round ductwork. To overcome the high external static pressure resulting from this distribution system, a secondary supply fan was installed within the ADM module. Approximately 50 cfm of preconditioned outdoor air was delivered to each guest room though a small, conical diffuser that was easily adjustable. This facilitated the air-balancing process.

As in the Hooters project, the ADM system operates as a 100% outdoor air system and, since it serves as the sole source of make-up air for the hotel wing, it runs continuously. As a result, the ventilation air condition leaving the ADM needed to be very dry but at a room-neutral temperature. Otherwise, the guests could be overcooled at part-load conditions, requiring the use of heating during the cooling season. Noise was also an important factor, since the quiet surroundings at Callaway would allow guests to notice even a minor amount of equipment noise transferred to the room by the HVAC system. As discussed later in Section 4.5, occupant comfort was significantly improved with the addition of the ADM/rooftop combination.

A combination temperature and humidity sensor was located within the discharge duct leaving the ADM and served as the sole control point for this pilot site. Each room in the Callaway facility was conditioned by an existing fan-coil unit, served by chilled water from a large, central chiller. Serious humidity problems experienced at this facility were due, in part, to the fact that these units had little latent capacity. The lack of capacity was due to the age of the fan-coil units, an elevated chilled-water temperature, and low flow associated with a variable-speed pump installed as part of an energy savings program.

Fortunately, the individual fan-coil units were capable of controlling the sensible load within the space once the latent load (outdoor air and space components) was removed. To remove the latent load, the supply air delivered by the ADM needed to be maintained at approximately 47 to 60 grains, depending upon the desired space humidity level (50% or 60% RH at 75ºF). The dryness required to maintain a given space humidity level was determined by monitoring the absolute humidity level in the bathroom exhaust air ductwork. The supply air temperature and humidity sensors are used by the DDC system to modulate the two compressors in the rooftop unit serving the ADM as well as the direct-fired gas burner control valve providing regeneration heat to the desiccant wheel.

4.2 System Description and Schematic

In the Callaway installation, approximately 945 cfm (45%) of the 2100 cfm preconditioned air stream is passed through the active desiccant wheel. The air passed through the wheel leaves warm and extremely dry (typically 100°F and 20 gr/lb of moisture). This dry, warm air mixes with the air bypassing the wheel (typically about 62°F and 79 gr/lb) to deliver air to the occupied space that is at a room-neutral temperature and a low dewpoint (typically about 78°F and 51 gr/lb or 49°F dewpoint).

This condition was found to maintain the rooms at an RH level below approximately 55%. As mentioned previously, the supply conditions leaving the ADM were modulated to maintain the space at conditions ranging from 50% RH (requiring a 47°F dewpoint) to 60% RH (requiring a 53°F dew point). This level could be easily accommodated by the ADM under most conditions. A 49°F dew point was determined to be the best choice, since it maintained the RH in the rooms at 50% most of the time, allowing it to spike to slightly over 55% during times of high shower activity.

4.3 Controls, Instrumentation, and Data Acquisition

Controls

As in the Hooters project, a DDC system was used to modulate the active desiccant speed, the amount of bypass air, the regeneration temperature, and the stages of cooling. The system was also used to provide remote, real-time data acquisition and energy utilization monitoring. The instrumentation, as shown by Fig. 19, was similar to that used for the Hooters project.

Instrumentation

The sensors used to control the operation of the ADM also was used for instrumentation and performance monitoring. A significant advantage offered by the DDC control/instrumentation approach is the graphics capabilities. Figure 19 shows a custom schematic diagram that was developed by SEMCO to show the function of the ADM system at Callaway.

In addition to the computer graphics that could be accessed remotely via modem, a liquid crystal display located within the control panel mounted on the ADM could be used to track the space temperature and humidity and, if desired, change setpoints on site.

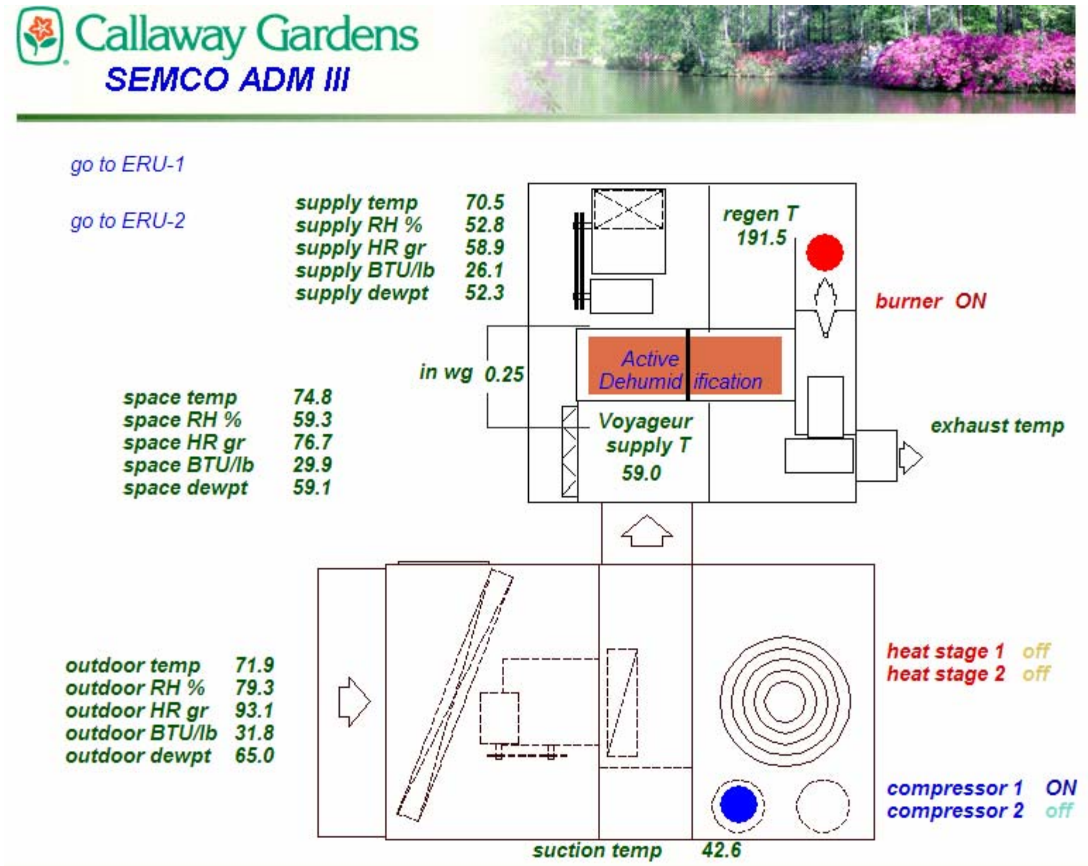

Fig. 19. Schematic showing the airflow path through the Callaway ADM and the points monitored. Note that one stage of the compressor is cycled off to minimize cooling input while delivering the desired humidity level (60 g/lb, required to maintain the humidity level at below 60% in the combined bathroom exhaust air stream). (All temperatures are in °F.)

Data Acquisition

The performance data for the ADM system were monitored and collected in the same manner as reported for the Hooters site.

4.4 System Performance Data

ADM as a Dedicated Outdoor Air System

Callaway Gardens also installed the ADM system to serve as a DOAS. The design concept applied to the Callaway site was based upon operating the ADM and standard 7.5-ton rooftop system to provide 100% outdoor air. This outdoor air stream was dehumidified to a level necessary to maintain the humidity level in air leaving the combined bathroom exhaust ductwork at a desired condition. Exhaust air RH conditions ranging between 50% and 60% were investigated. To achieve this level of humidity control, the outdoor air had to be dehumidified to dewpoint temperatures from 47°F to 53°F. Outdoor air dehumidified to a 49°F dewpoint (52 gr/lb) was determined to provide the best compromise between energy consumption and space comfort.

ADM vs a Custom Refrigeration-Based Dedicated Outdoor Air System

The supply air humidity condition shown in Table 5, 51 gr/lb (49°F dewpoint) is attainable with customized refrigeration-based outdoor air systems. However, the ADM offers numerous control advantages that are not available with refrigeration-based systems (Fisher and Sand 2002). The amount of installed condenser capacity, as well as the cost of operation, is significantly higher for a customized vapor compression system.

Table 5 shows a very simple comparison of the ADM/rooftop combination and the customized conventional equipment capacity necessary to deliver the dewpoint condition needed by the Callaway pilot. As shown in the Hooters analysis, the packaged system feeding the ADM produces more than its Air-conditioning and Refrigeration Institute (ARI) -rated cooling capacity, while the conventional approach, because of the low leaving air temperature required, has to be de-rated (see Fig. 10).

At both the Hooters and Callaway Gardens sites, the ADM/rooftop combination was found to significantly reduce the installed cooling capacity, cost less to operate and, most important, have the capability to deliver much drier air compared with the conventional approach.

Table 5. Cooling capacity and energy comparison of the Mountain Creek Inn ADM and a custom direct expansion alternative

	ADM rooftop combination	Custom DX rooftop (over-cool and reheat)
Installed cooling capacity required (gross tons)	**7.5 tons**	**16.6 tons**
Cooling capacity output delivered (net tons)	**8.8 tons**	**14.4 tons**
Reheat energy required (Btu/h)	**0**	**35,100[a]**
Regeneration energy required (Btu/h)	**75,600[b]**	**N/A**
Supply dewpoint used for analysis	**48.7°F**	**48.7°F**
Annual cooling energy cost[c]	**$2060**	**$4410**

[a]Assumes that 2100 cfm of air leaving the coil at 49.5°F is reheated to 65°F. Colder air would over-cool the guest room during part-load conditions.
[b]Maximum regeneration energy reduced at off-peak conditions, direct-fired burner raising 560 cfm from 85 to 210°F.
[c]Analysis assumes operation 7 days/week, 24 hours per day, electric cost of $.07/kWh, $6.00/kW demand, gas cost of $0.50 per therm—summer rates during 2000–01 when test was conducted. Gas use of 1950 therms can be adjusted for current rates.

Outdoor air condition used is 85°F and 115 grains/lb.

ADM Performance at the Callaway Site

As shown by Figs. 20 and 21, the exhaust air humidity setpoint could be controlled to anywhere between 45% RH (Fig. 20) and 60% (Fig. 19), as selected by the end user. Before the installation of the ADM, the space was uncomfortably hot and humid, with some guests refusing to stay in the rooms for more than one evening. During off-peak conditions, the guest rooms were consistently cold and clammy because the guests would lower the thermostat settings in an attempt to achieve comfort with the existing fan-coil units. But RH would rise and, consistent with the conditions shown in Fig. 16, would end up out of the comfort zone. According to the facility manager at Callaway, these attempts to control humidity by lowering the temperature frequently resulted in space temperatures below 70° and RH levels exceeding 80%. Aside from the issue of guest comfort, damage caused by the extended periods of high humidity necessitated frequent refurbishment of the guest rooms and furnishings.

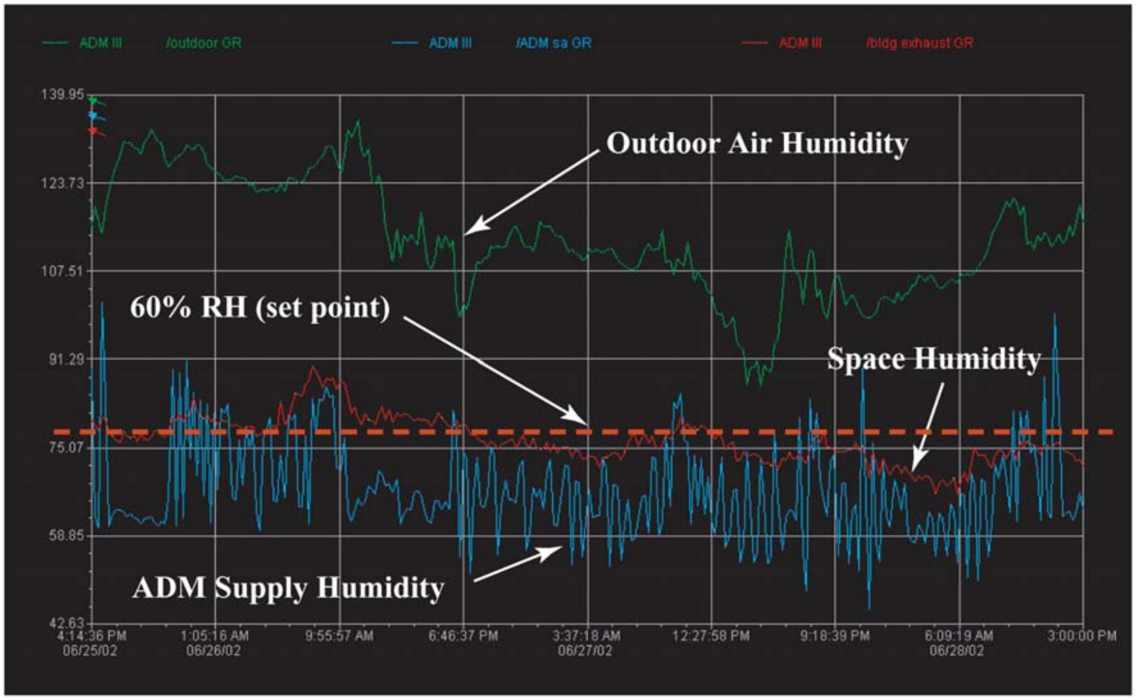

Fig. 20. Actual data comparing the outdoor air, the supply air, and the space absolute humidity levels (gr/lb) during a 4-day period at Callaway. Note the fluctuation in supply condition reflects the cycling of the compressor and, occasionally, the gas burner, while maintaining a 55% to 60% relative humidity exhaust air setpoint.

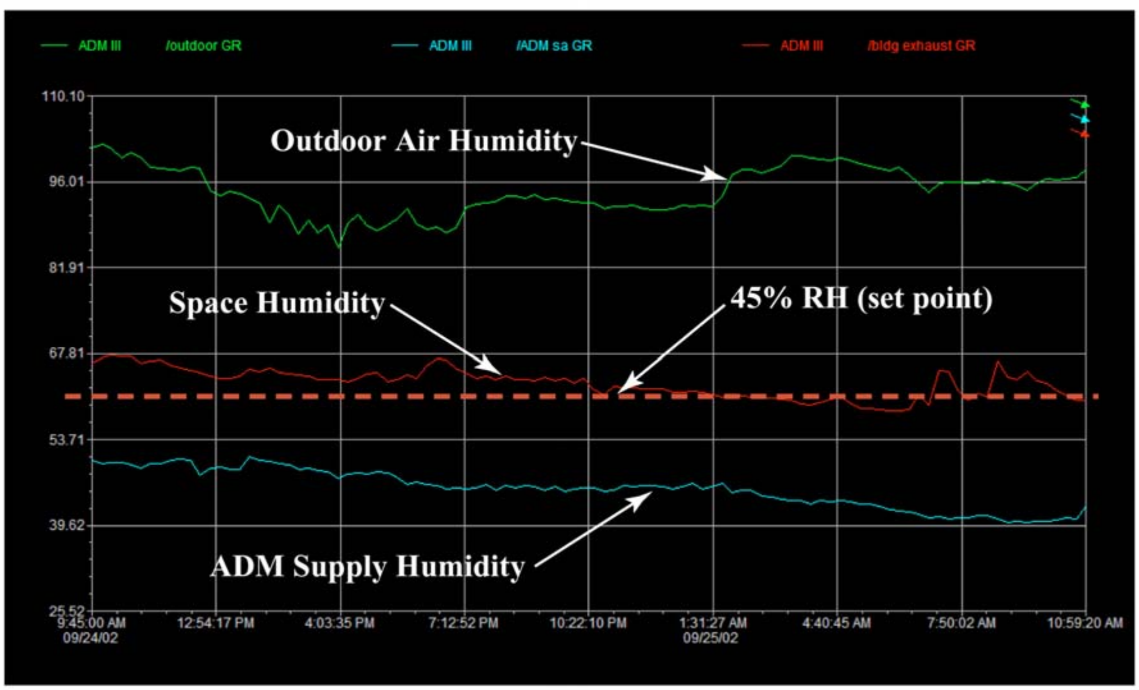

Fig. 21. Outdoor air, supply air, and space absolute humidity level data (gr/lb) during a 24-hour period at Callaway. Note lowering the exhaust air humidity set point to 45% relative humidity causes both stages of cooling to remain on, stabilizing the delivered humidity data. Note the humidity spike due to showers (8:00 A.M.).

Figure 20 shows the humidity content (gr/lb) during a design day of the outdoor air, the supply air, and the exhaust air that was connected to each bathroom in the facility. The exhaust air RH setpoint (in this case 60%) is maintained even when the outdoor air reaches 130 gr/lb (the guest rooms are maintained at lower humidity). As the outdoor air drops to only 90 gr/lb, the amount of regeneration energy and/or cooling energy input (stages of cooling) is reduced while the humidity setpoint is maintained. These part-load conditions, a significant challenge to conventional cooling systems (Section 1.2 and Fig. 1) proved to be no problem for the ADM approach.

Figure 21 shows data from a 24-hour period where the ADM was temporarily controlled to maintain the combined bathroom exhaust at 45% RH. At this setpoint, most rooms maintained an RH of about 40%, 5% below that of the exhaust air. The purpose for this change in setpoint was to demonstrate the capability of the ADM to respond to extreme indoor humidity loads if necessary.

Figure 22 shows the performance of the combined ADM/rooftop unit at part-load conditions. As shown, one stage of cooling is turned off and the active desiccant wheel is being regenerated to provide outdoor air at a dewpoint low enough to maintain the combined bathroom exhaust air at 50% RH. Note that even at these part-load conditions, the suction temperature entering the coil is maintained at a very favorable 42.5°F, eliminating any risk of frost formation.

Unlike the Hooters restaurant, the Callaway facility has no unoccupied periods. As a result, the system runs continuously. One obvious control alternative that would improve energy efficiency would be to project when guests are most likely to shower (early morning and early evening) and use these times to provide air at the lowest dewpoint. At all other times, the dewpoint of the ventilation air could be higher without compromising the comfort of the guest rooms. This control refinement is likely to be employed during the 2003 cooling season.

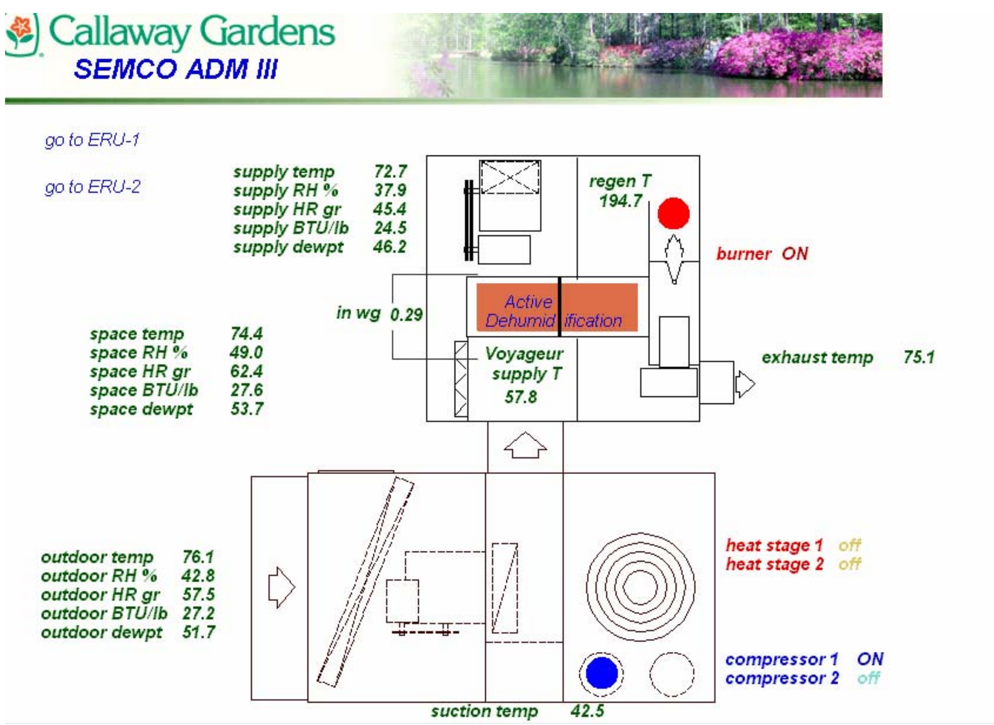

Fig. 22. Sample of Callaway performance data at part-load conditions. Such conditions make humidity control difficult for conventional packaged equipment, but they do not pose a problem for the ADM/rooftop unit combination (all temperatures are in °F).

Energy Benefit Resulting from Improved Humidity Control

Relative to the energy use with the previous direct expansion cooling units only, energy savings are being realized at the Callaway facility because guests feel more comfortable at higher cooling-season thermostat settings. Reasons why this is true at lower space humidity levels are discussed in detail in Section 3.4.

4.5 Interview with Building Owner

An interview with the restaurant building manager at Callaway confirmed his satisfaction with the system.

Callaway's assistant manager of facilities engineering commented that the SEMCO ADM "solved our humidity problems by providing energy savings and guest satisfaction." He also reported that "judging our customer reaction since the installation of the ADM has been simple … no calls, which means no complaints." He added that "guest satisfaction has been the primary goal from the start."

Callaway Gardens is located southwest of Atlanta in Pine Mountain, Georgia, and he noted that "in South Georgia, high humidity goes without saying."

Other specific benefits resulting from the ADM retrofit include energy savings, the elimination of water condensing on the outside of guest room windows, an end to complaints about musty odors in the rooms, and fewer condensate management problems associated with the fan-coil units.

5. CONCLUSIONS

Both the Hooters and the Callaway pilot installations resulted in effective solutions to the end users' IAQ and humidity problems. In each case, the indoor environment was significantly improved, serious humidity problems were resolved, and the occupants have experienced comfortable room conditions as a result of the ADM/rooftop unit installation.

At both pilot sites, the gross cooling input required by the ADM/rooftop combination was approximately half that needed by a conventional over-cooling, reheating approach applied to deliver outdoor air at the same dewpoint. The capability of the ADM/rooftop combination to deliver 100% outdoor air at low dewpoints, using moderate coil leaving air temperatures, results in a net cooling output that exceeds the ARI-rated capacity of the conventional rooftop system. Conversely, a conventional system, requiring very cold leaving coil temperatures (low suction temperatures) produces a net cooling output that is less than the nominal ARI-rated capacity.

The numerous control options available with the ADM/rooftop approach allowed control of the humidity conditions within the two pilot site spaces, as desired, despite wide fluctuations in the outdoor air humidity and temperature. The ADM approach easily responded to part-load conditions. Conventional systems often are faced with capacity control challenges resulting in short compressor run times, high space humidity, and the potential for freezing coils.

Energy cost savings and increased HVAC efficiency were clearly recognized at both sites. Savings resulted from reduced cooling capacity requirements, the use of low-cost gas available during the cooling season, and higher cooling-season thermostat settings that resulted from consistently lower space humidity conditions. The field demonstration found that replacing a conventional HVAC system with the ADM/rooftop combination unit at the Hooters restaurant reduced overall electricity consumption by approximately 15% and reduced peak demand by 12.5 kWh. These savings represent a reduction by approximately 34% in the electricity used by the restaurant's air-conditioning system.

The restaurant operated 16 hours per day, 7 days per week. Using an average electrical rate of $0.09 per kWh, the annual energy cost for the air-conditioning system would be reduced by $5400. The cost of gas used to regenerate the desiccant wheel during the same period is about $740, assuming a gas cost of $0.50 per therm.

Simulated results for the performance of ADM/rooftop units compared with the performance of HVAC systems similar to those used at the Hooters and Callaway Gardens sites also were presented. The results verify that the ADM approach manages a high latent load more energy-efficiently than do conventional HFAC approaches.

The space required to install the ADM/rooftop combination, as well as the routine maintenance required, was no more than that associated with a conventional cooling approach. The ADM/rooftop systems remain in operation at the sites, the instrumentation is still in place, and monitoring will continue.

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