ORNL/SUB/94-SV044/3b

Active Desiccant Dehumidification Module Integration with Rooftop Packaged HVAC Units

Phase 3B

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Active Desiccant Dehumidification Module Integration with Rooftop Packaged HVAC Units

Final Report: Phase 3B

Prepared by

J. Fischer
SEMCO, Inc.

J. Sand
Oak Ridge National Laboratory

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OAK RIDGE NATIONAL LABORATORY
Oak Ridge Tennessee 37831-6285
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ABSTRACT

This report summarizes a research and development program that produced a stand-alone active desiccant module (ADM) that can be easily integrated with new or existing packaged cooling equipment. The program also produced a fully integrated hybrid system, combining the active desiccant section with a conventional direct expansion air-conditioning unit, that resulted in a compact, low-cost, energy-efficient end product. Based upon the results of this investigation, both systems were determined to be highly viable products for commercialization.

Major challenges—including wheel development, compact packaging, regeneration burner development, control optimization, and low-cost design—were all successfully addressed by the final prototypes produced and tested as part of this program.

Extensive laboratory testing was completed in the SEMCO laboratory for each of the two ADM system approaches. This testing confirmed the performance of the ADM systems to be attractive compared with that of alternate approaches currently used to precondition outdoor air, where a return air path is not readily available for passive desiccant recovery or where first cost is the primary design criterion.

Photographs, schematics, and performance maps are provided for the ADM systems that were developed; and many of the control advantages are discussed. Based upon the positive results of this research and development program, field tests are under way for fully instrumented pilot installations of ADM systems in both a hotel/motel and a restaurant.

1. INTRODUCTION: ACTIVE DESICCANT MODULE DEVELOPMENT PROGRAM

As a result of an earlier effort completed by SEMCO as part of the first phase of this DOE-sponsored research and development (R&D) program, it was concluded that a significant market opportunity exists for a cost-effective, energy-efficient active desiccant module (ADM) that would process outdoor air and control humidity within facilities traditionally served by packaged heating, ventilating, and air-conditioning (HVAC) equipment.

The phase 1 report, *Desiccant-Based Preconditioning Market Analysis* (Fischer, Hallstrom, and Sand 2000), provides a desiccant preconditioning market analysis, which concludes, among other things, that restaurant and hotel/motel facilities could benefit significantly from the use of cost-effective active desiccant systems. This analysis concluded that that these two market segments alone account for 22% of the projected $720,000,000 annual market for outdoor air preconditioning that exists for new construction and renovation in the United States.

The phase 2 report, *Active Desiccant-Based Preconditioning Market Analysis and Product Development* (Fischer 2000), recognized that the vast majority of HVAC systems sold each year in the United States are packaged units using compressor-driven cooling cycles. The report therefore concluded that, for widespread market acceptance of active desiccant systems, an ADM designed to augment packaged cooling equipment and to eventually be integrated into future hybrid systems is essential.

As Fig. 1 indicates, the size of the packaged HVAC market is so dominant that serving only the large commercial (chilled water) markets would be extremely restrictive. Applications such as restaurants, hotels/motels, retail stores, and schools all have a strong preference for packaged HVAC equipment. At the same time, they all need relatively large amounts of outdoor air to comply with the building codes reflecting the ASHRAE Standard 62 recommendations (ASHRAE 1989).

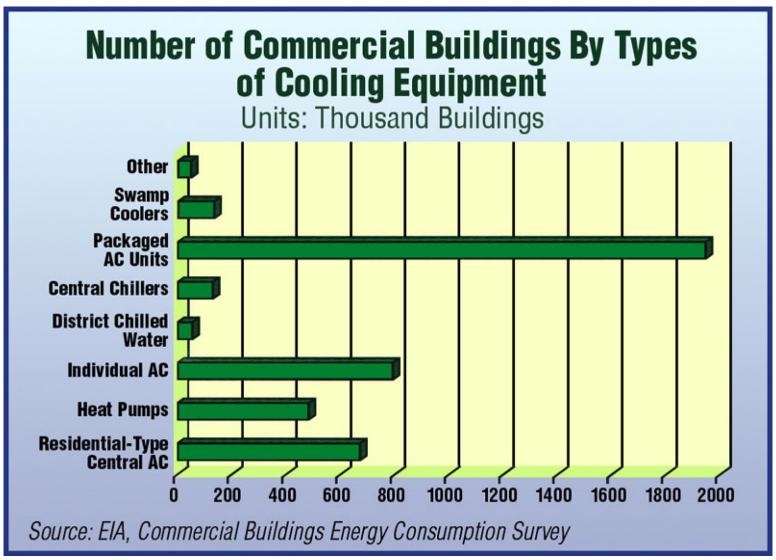

Fig. 1. Packaged air-conditioning equipment dominates the cooling equipment market.

For example, restaurants need large quantities of outdoor air to replace the air exhausted by the kitchen hoods and bathrooms. Currently, this makeup air is supplied via conventional packaged HVAC units that are sized to provide high percentages of outdoor air. The outdoor air limitations inherent in this type of equipment often result in poor humidity control and wide temperature fluctuations. Thus many restaurants would benefit significantly from the improved humidity control made possible by the higher latent-to-sensible heat ratios associated with an active desiccant approach.

Like restaurants, hotel/motel facilities need increased volumes of outdoor air to replace air exhausted from the bathroom areas of the guest rooms. Mold and mildew problems are common and costly in these facilities as a result of variable cooling loads, intermittent peak latent loads associated with the showers, and the inability of low-end packaged HVAC equipment to control these latent loads effectively.

In addition to restaurants and hotels/motels, the Phase 1 and Phase 2 reports concluded that many retail stores and schools offer excellent market opportunities for a compact, low-cost, easy-to-apply packaged HVAC unit that combines a conventional vapor-compression unit with an active desiccant wheel to precondition the outdoor air delivered to occupied spaces. The Phase 1 market analysis projected that the market potential for a well-designed ADM would be approximately $45,000,000 annually in the United States.

This Phase 3B report summarizes an R&D program completed by SEMCO in which it developed a compact, cost-effective ADM—with an integrated direct-fired gas burner for regeneration—that can be used with a conventional rooftop air-conditioning unit. Two modules were designed, constructed, and tested. The first was a stand-alone ADM coupled with a standard Trane Voyager® 4-ton rooftop unit (Fig. 2). The second was a complete hybrid system integrating the ADM with the complete refrigeration section, through extensive modification of a standard 5-ton Voyager unit.

Fig. 2. An add-on active desiccant module and standard rooftop unit in the SEMCO test loop.

2. BACKGROUND: MARKET-DRIVEN NEED FOR THE ACTIVE DESICCANT MODULE

A large body of research supports the need for continuous ventilation—in accordance with the American Society of Refrigerating, Heating and Air-conditioning Engineers (ASHRAE) Standard 62-1989—while relative humidity in the space is maintained at between 30% and 60%. Indoor air quality (IAQ) problems ranging from unacceptable odors to microbial infestation can occur if HVAC systems are not able to meet these design criteria. The Phase 1 summary of a DOE-sponsored research project entitled *Causes of Indoor Air Quality Problems in Schools* (Bayer, Crowe, and Fischer 2000) is an excellent resource for understanding potential IAQ problems.

Most facilities use packaged heating and cooling equipment designed to provide inexpensive, efficient heating and cooling with *minimal* outdoor air. This type of equipment was not designed to handle the increased continuous supply of outdoor air necessary to comply with ASHRAE 62-1989 guidelines. This limitation is especially true of applications that require 100% outdoor air, such as makeup air to restaurants or hotel/motel facilities.

For many reasons, engineers would prefer to use off-the-shelf packaged rooftop equipment to process these outdoor air streams. But because of a wide range of well-known limitations, they are currently unable to do so. The most common attempts to solve this problem include the use of either oversized packaged equipment that provides more airflow than required and sub-par performance, or expensive, customized overcooling/reheating systems. These systems are not only costly to buy, but also expensive to operate. The complicated refrigeration circuits they require may be difficult to troubleshoot and expensive to keep operational. The ADM approach allows for the use of a standard rooftop system and combines the strength of an active desiccant system (a supply of very dry air) with the many strengths of the conventional rooftop unit (low first cost, simplicity, familiarity, and compact design).

Currently, active desiccant dehumidification systems are routinely specified for supermarkets, ice arenas, and cold storage warehouses. All three of these applications base their success on the economic benefits associated with the improved refrigeration operation resulting from the introduction of very dry air. The ADM approach presented in this report applies an active desiccant wheel in a system configuration designed to take the best advantage of its ability to efficiently produce very dry air while minimizing the size and the cost of the final product.

2.1 Outdoor Air Limitations of Packaged HVAC Units

Much of the discussion concerning the impact of the ASHRAE 62-1989 standard has been focused on the cfm/person recommendation for ventilation air. The need to provide the outdoor air *continuously* provides an even bigger challenge to packaged HVAC equipment than does the increased amount of outdoor air.

On mild, humid days (part-load conditions), an oversized packaged unit will quickly cool the space to the set point temperature and 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 that time, the return air entering the cooling coil is elevated in humidity. The result is that when the rooftop unit's compressor does restart, the air leaving the cooling coil has an elevated dewpoint temperature. The temperature of the space is maintained, but humidity control is lost, resulting in uncomfortable conditions. We all have been in hotel conference rooms and restaurants and experienced cold, clammy conditions. In an attempt to feel comfortable, occupants lower the thermostat setting, further increasing the relative humidity of the space. If this high humidity persists, as is often the case in hotel rooms and corridors, conditions promote microbial growth and other moisture-related IAQ problems.

Packaged equipment has been further challenged by the fact that the design weather data used by many engineers have not accurately reflected the design latent load (i.e., humidity content) of the outdoor air. ASHRAE has addressed this problem by publishing new latent design data that are now included in the ASHRAE 2001 *Fundamentals Handbook* (ASHRAE 2001). These more-accurate latent design data were badly needed by the design community. For example, a design engineer in Detroit would underestimate the load associated with the outdoor air by approximately 20% if the old dry bulb design conditions were used in lieu of the ASHRAE latent (dewpoint) design data.

Another problem characteristic of oversized packaged equipment selected to process outdoor air on a continuous basis is the re-evaporation of moisture that has condensed on the evaporator coil. Henderson et al. (1996) and Khattar et al. (1985) both have confirmed the phenomenon, often observed in the field, whereby the actual amount of moisture removed by a packaged HVAC unit is significantly less than the amount anticipated based on published performance data. Their research shows how moisture condensed on a direct-expansion (DX) coil evaporates back into the supply airstream when the compressor is cycled off and the supply fan continues to run. Henderson (1996) has shown that the actual latent removal can be reduced to less than 50% of the unit's rated capacity at part-load conditions.

These and other limitations present real problems when packaged rooftop systems are forced to handle high percentages of outdoor air, especially for 100% outdoor air systems. For example, when a conventional rooftop unit is used to handle all outside air, the tons of capacity required at the peak load condition are far greater than the cooling output available at the rated airflow of the unit. For example, conditioning a 1500-cfm outdoor air stream from 85° dry bulb temperature and a 71°F dewpoint to a 56° dewpoint requires a 10-ton unit. However, the smallest amount of air that a 10-ton unit can process without causing control problems, coil frosting, and compressor failure is approximately 3000 cfm (300 cfm/ton), twice the amount necessary.

If the unit is set up to provide 50% outdoor air (1500 cfm), the higher 3000-cfm supply airflow quantity may overcool the space and require expensive parasitic reheating, especially at part-load conditions. Most of these problems are well known and understood by HVAC designers and do not merit further explanation in this report.

2.2 Previous Active Desiccant System Approaches

Active desiccants have been applied in conjunction with rooftop packaged equipment in the past. These systems used the traditional approach: process outdoor air with the active desiccant wheel first, in an attempt to handle most or all of the latent load with the desiccant wheel, and then post-cool the air with the rooftop unit as necessary. This approach did not find market acceptance for several reasons, including, most notably, first cost, operational cost, and energy efficiency.

When an active desiccant wheel removes moisture from an airstream, heat is released as a byproduct of the adsorption process. The more moisture that is adsorbed by the desiccant wheel, the more heat is released. As a result, if an active desiccant wheel is asked to remove, say, 60 grains of moisture (from 130 grains to 70 grains), the temperature of the air it releases to be used for building ventilation increases significantly. When the wheel is attempting to remove large quantities of moisture (e.g., 60 grains), a high regeneration temperature must be used. This further raises the temperature of the air that the system processes for the building. When the active desiccant wheel is applied upstream of the cooling coil (as in Fig. 3), that heated air will add to the sensible cooling load for the building.

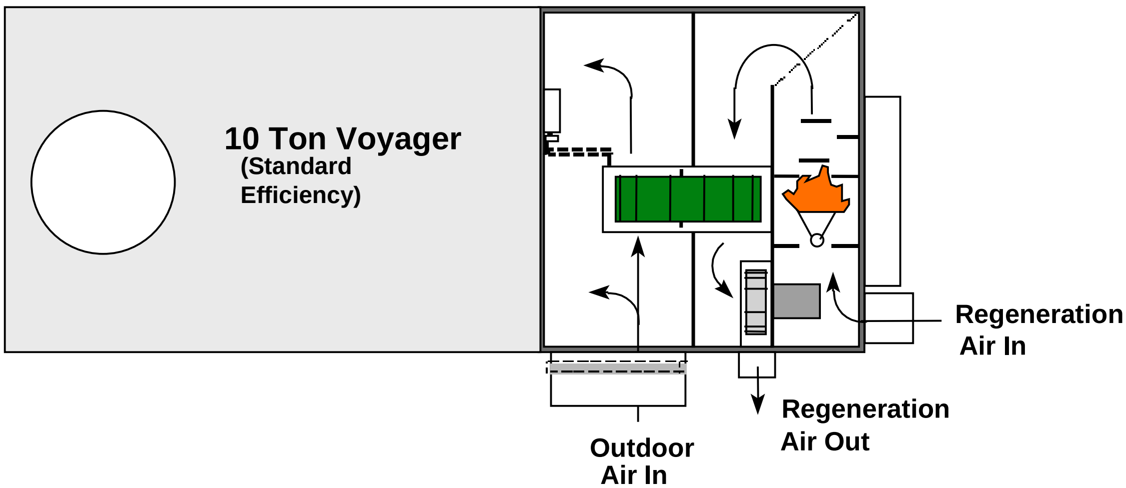

Fig. 3. Schematic of an active desiccant system preconditioning the outdoor air entering the rooftop unit.

Based upon the literature for the best-performing active desiccant wheels currently available, this 60-grain reduction would increase the temperature of the outdoor air entering the cooling coil from approximately 85° to 138°F. Herein lies a significant problem with this "active desiccant preconditioning" approach. If the active desiccant wheel is expected to handle all or most of the outdoor air latent load, the amount of post-cooling required to remove the additional sensible heat associated with the dehumidification process will be similar to the amount that would be required to remove the humidity without the desiccant system. As a result, this approach does not reduce energy consumption; it actually increases it (see Tables 1 and 2 in Section 3).

The other major problem with positioning the desiccant wheel before the evaporator coil is that it must then be sized to process the *entire* outdoor airstream. Since it is being asked to remove 60 grains of moisture, the velocity of the wheel face must be very low. Consequently, the active desiccant preconditioning module may have to be twice the size of the rooftop unit it is serving and thus be extremely costly (see Table 2 in Section 3). A third disadvantage associated with this approach is the need for very high regeneration temperatures to remove the high moisture load entering the desiccant wheel.

The ADM investigated as part of this program addresses these issues by applying a novel system configuration, positioning the active desiccant wheel module after the evaporator coil located in the packaged rooftop system, not before.

3. THE ACTIVE DESICCANT MODULE APPROACH

The ADM approach positions the ADM downstream of the evaporator coil contained within the packaged unit (Fig. 4). This configuration offers numerous advantages over having the desiccant wheel in front of the cooling coil that are not obvious to the casual observer. (These advantages are summarized in Sect. 3.1)

The ADM includes an active desiccant wheel sized to handle approximately 33% of the airflow processed by the packaged unit (the amount can vary). A bypass damper is included to maintain the desired flow through the desiccant wheel and allow bypass of the desiccant wheel during the heating mode if desired. The ADM also integrates 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 (CHP) system, or in other applications where waste heat is available for regeneration of the active desiccant wheel.

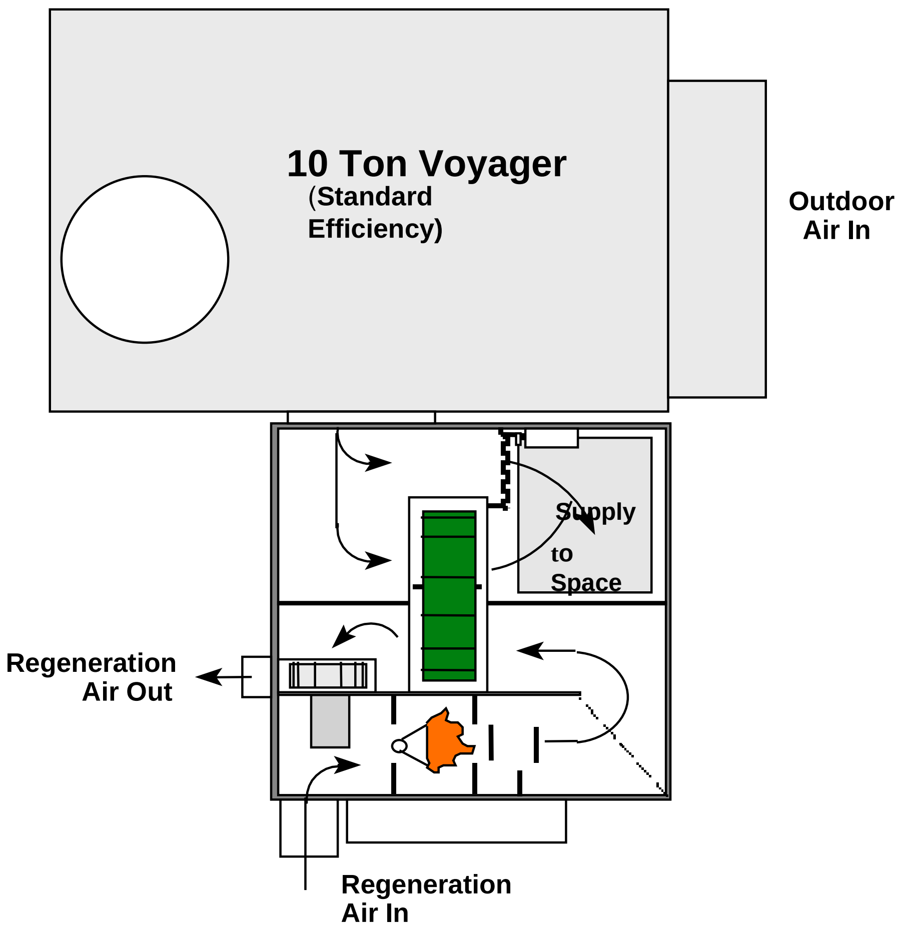

Fig. 4. Schematic of an active desiccant module conditioning outdoor air downstream of the rooftop unit.

3.1 Advantages Offered by the ADM Approach

The advantages of the ADM approach can be best summarized through comparisons with the conventional packaged cooling approach and previously marketed active desiccant systems.

ADM advantages relative to conventional packaged systems

The ADM can be applied in a variety of ways, but to aid in a simplified comparison, it is assumed here that the ADM will process only outdoor air and that it will be coupled with a standard, single-stage packaged rooftop unit.

Table 1 shows the results of a simple comparison made between the ADM–rooftop packaged system combination and a customized 10-ton packaged unit designed to handle 100% outdoor air. The comparison assumes that each system will process 1500 cfm of outdoor air from cooling season design conditions of 85°F dry bulb temperature and a 71°F dewpoint to a 56° dewpoint. It also assumes that to avoid overcooling the space, the outdoor air will be reheated to 68° prior to its introduction to the space. The energy analyses assume continuous operation and use utility costs of $.07/kWh for electricity and $4.50/million Btu for gas.

As shown in Table 1, the first obvious advantage is that the mechanical cooling capacity (tons) required for the ADM approach is only half that required by the customized package unit. Aside from the obvious advantage of reduced electrical demand and electrical service requirements, the amount of compressor cycling is reduced because the smaller rooftop unit is fully loaded far more frequently. This advantage also minimizes the problem of condensate re-evaporation from the cooling coil, which was mentioned earlier.

Table 1. Comparison of the active desiccant module (ADM) approach with a customized packaged rooftop unit approach

	ADM rooftop combination	Custom DX rooftop (over-cool and reheat)
Cooling capacity required (tons)	5 tons	10 tons
Reheating energy required (Btu/hour)	0	32,400
Regeneration energy required (Btu/hour)	33,500	N/A
Supply dewpoint used for analysis	56°F	56°F
Annual cooling energy cost	$1360	$2480
Unit approximated size (H × W × L)	31 × 46 × 46 in.	33.5 × 46.5 × 83 in.

Table 1 also compares the estimated energy consumption for both approaches. This analysis projects the annual operating cost of the ADM–rooftop combination to be 45% less than that of the customized packaged unit. This differential in energy savings will be even more pronounced in markets where gas rates are seasonally low during the cooling season, where incentives are offered for gas cooling, and where electrical demand charges are high.

Unlike the customized rooftop unit, the ADM requires a secondary energy input for desiccant regeneration. However, as shown in Table 1, this regeneration energy at peak load conditions is often on a par with the energy that the customized rooftop unit requires for reheating. As the outdoor air loads become less extreme, the amount of regeneration energy the ADM requires can be reduced while the desired supply air dewpoint is maintained. For the customized package unit, on the other hand, the required amount of reheating energy remains constant. The customized packaged unit can be designed to use the heat of rejection from the refrigeration circuit to provide “free” reheating. However, overall cooling efficiency (kW/ton) is often decreased in order to meet the reheating requirements. Additionally, the reheating temperature delivered from a condensing coil of a refrigeration system is not easily controlled.

Another significant benefit of the ADM is realized when the outdoor air is at cool and humid part-load conditions. The ADM allows the packaged rooftop compressor to be cycled off because all of the dehumidification needed can be provided by the active desiccant wheel. At these conditions, the customized packaged unit requires the addition of hot gas bypass or multiple staging with sophisticated controls to avoid frosting the cooling coil and potentially damaging the compressor. During cool ambient conditions, the condensing section of the customized packaged unit provides excess capacity at the very time that reduced capacity is required at the evaporator coil. Without proper design considerations, this results in unacceptably low suction temperatures (frozen coils). The ADM approach resolves this problem.

The ADM approach provides other significant control options that are not possible with conventional systems. (Many of these are discussed in Section 4.3.) One of the more important options is the ability of the ADM to provide air at much lower dewpoints than are possible with the DX cooling cycle alone. During times when the building is not occupied, the 100% outdoor air system can be operated as a recirculated air system, allowing very dry air to be introduced to the space to provide dehumidification without overcooling the space.

ADM advantages relative to previous active desiccant systems

Table 2 provides an analysis summary similar to that in Table 1, but it evaluates the ADM–packaged rooftop unit approach relative to two previously marketed active desiccant system configurations. The first one places the active desiccant wheel ahead of the packaged rooftop system (see Fig. 3). The second one is the traditional "desiccant-based cooling system" (DBC). It has also been installed upstream of the packaged rooftop system, but in addition to the active desiccant wheel, it includes a sensible-only recovery wheel and evaporative cooling section. These components were added to remove much of the heat of adsorption from the outdoor airstream before it is delivered to the cooling system, and to preheat the regeneration airstream. This DBC approach is discussed extensively in the Phase 2 final report (Fischer 2000).

Table 2. Comparison of the active desiccant module– (ADM) rooftop combination with other active desiccant system approaches previously marketed for preconditioning outdoor air for conventional cooling systems

	ADM rooftop combination	Active desiccant preconditioning	Traditional preconditioning
Cooling capacity required (tons)	5 tons	8.4 tons	2.5 tons
Air processed by active wheel (cfm)	540	1,500	1,500
Regeneration energy required	33,500	82,620	61,480
Supply dewpoint used for analysis	56°F	56°F	56°F
Annual cooling energy cost	$1360	$2620	$1560
Approximate unit size (H × W × L)	31 × 46 × 46	52 × 66 × 66	52 × 66 × 106
Relative cost of manufacturing	1	2.2	3

The data presented in Table 2 highlight the benefits offered by the ADM approach. With respect to performance, the ADM provides the desired dehumidification capacity using only 5 tons of mechanical cooing capacity compared with the 8.4 tons required by the active desiccant preconditioning approach (installed upstream of the cooling coil). It also uses only 54% of the regeneration energy required by the traditional preconditioning approach and 41% of that required by active desiccant preconditioning.

The cost of operating a system based on the ADM approach is approximately one-half that required to operate a system using the active desiccant preconditioning approach. Just as important, the size of the system is only 30% of the size required by the active desiccant preconditioning approach.

Many of these advantages stem from the fact that only 33% of the outdoor airflow is processed by the active desiccant wheel in the ADM, while the other systems need to process the total amount. (The reason for this is discussed in Section 4.2.) The corresponding reduction in the diameter of the active desiccant wheel results in a much smaller final product. Maintaining a module size compatible with that of the packaged cooling equipment is a significant advantage. The combined ADM-rooftop system approach also uses far less fan horsepower in proportion to the amount of air handled.

In comparing the ADM approach with the DBC system, it is clear that although the energy input and operating cost associated with the two approaches are similar, the size and first-cost premium associated with the DBC approach make the ADM the obvious choice.

3.2 Description and Schematic of the Add-On ADM

The novel thinking behind the development of the ADM approach was driven by a number of key assessments derived from technological and market research analyses. The message from the marketplace is clear: An active desiccant product will have to be compact and cost-effective first, energy-efficient second. A growing trend in the marketplace is that engineers want more control options, especially as they relate to humidity control and outdoor air delivery.

We also know from research that the performance of any active desiccant wheel is significantly enhanced as the relative humidity of the air entering it increases (the converse is also true). We also know from experience that the cost of manufacturing the product, the maintenance involved, and the number of regeneration source options available all can be improved as the regeneration temperature required becomes more moderate (less hot).

The market history has shown that acceptance of active desiccants will occur only when the technology is applied in its area of strength—producing drier air than that obtainable with standard cooling equipment. Therefore, market acceptance of active desiccants has been limited to ice rinks, supermarkets, cold-storage warehouses, and industrial dehumidification.

The ADM

The ADM includes an active desiccant wheel sized to process approximately 33% of the air that passes across the cooling coil of the packaged rooftop unit. However, the amount of bypass air can change from application to application, or within a given application, to meet the latent and sensible load requirements. The concept provides saturated air to the desiccant wheel, thereby maximizing its operating effectiveness and minimizing the required regeneration temperature. This fraction of the air is dried to a very low dewpoint and is heated by the energy released from the desiccant as heat of adsorption. This warm, very dry air is then mixed with 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.

The ideal active desiccant wheel for this application is one that has a very low pressure loss, since it is advantageous to use the supply fan located in the packaged rooftop unit as the sole means of delivering air to the space. Given that all packaged units use forward-curved fans, the external static capability is limited. Also, the desiccant wheel should be optimized for the best performance at moderate regeneration temperatures and with saturated inlet conditions.

In addition to the active desiccant wheel, the module includes a modulating bypass damper, a regeneration source, a regeneration fan, and controls. Given that the primary market is for integration with rooftop equipment, hot water and steam will seldom be available as a thermal regeneration source. Electric regeneration is not economically viable in most cases. As a result, direct-fired gas provides the best regeneration solution. One major hurdle presented by this R&D program was the development of a compact, low-cost, effective direct-fired gas regeneration section and controls.

Figure 5 shows an early schematic of the ADM layout. The final design was modified significantly (see Fig. 7), but this initial layout provides an accurate description of the components used and their relative placement within the module.

Figure 6 is a photograph of the ADM showing the active desiccant wheel, controls, and various compartments. Figure 2 in Sect. 1 shows how this ADM is typically connected to the rooftop unit. As Fig. 2 shows, a standard horizontal configuration is chosen for the packaged unit, and a short transitional duct connects it with the ADM.

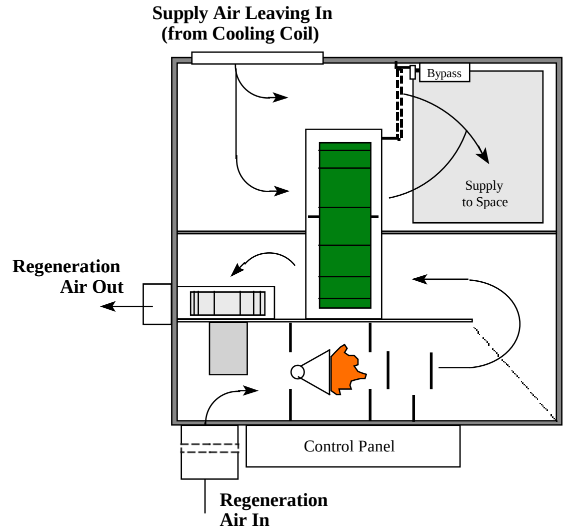

Fig. 5. Simplified schematic of the add-on active desiccant module showing key subcomponents.

3.3 Description and Schematic of the Integrated ADM Packaged Unit

Following the development and testing of the add-on ADM, an integrated hybrid system was designed, constructed and tested. This combined system resembled a standard rooftop unit when completed. One key advantage of this "integrated system" is that it involves no more effort to select, specify, or install than would a conventional rooftop unit. Once commercialized, this integrated system would also be more compact and cost-effective than a conventional rooftop unit–ADM combination.

As shown schematically in Fig. 7, the integrated ADM combines the filtration, supply air fan, and DX evaporator/condenser sections typically found in the standard rooftop unit with the components contained within the add-on ADM.

Figure 8 is a photograph of the integrated ADM during assembly. The photograph shows clearly how the active wheel section is positioned within the standard rooftop unit.

Figure 9 is a photograph of the completed integrated ADM system produced from a standard Trane Voyager® 5-ton rooftop unit. The side panel covered with glass shows the gas-fired burner section. With the exception of the added 3 feet in length, there is no way to visually distinguish this active desiccant hybrid system from a conventional rooftop unit.

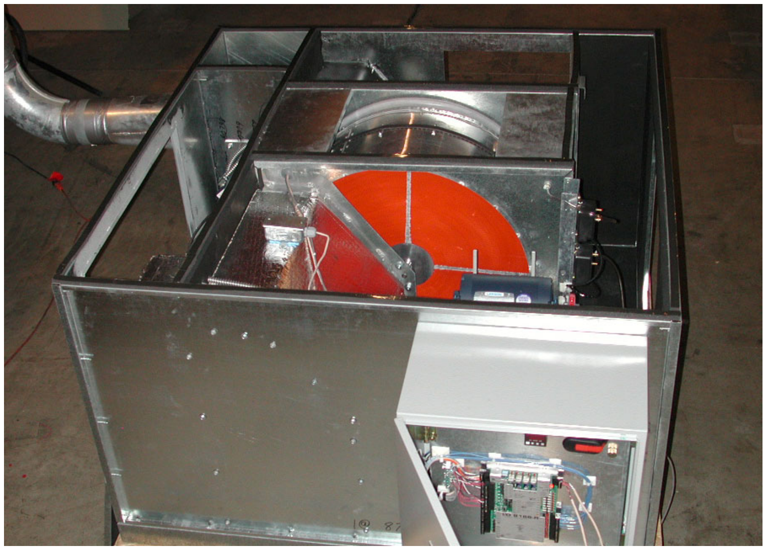

Fig. 6. Photograph of an active desiccant module showing the active desiccant wheel.

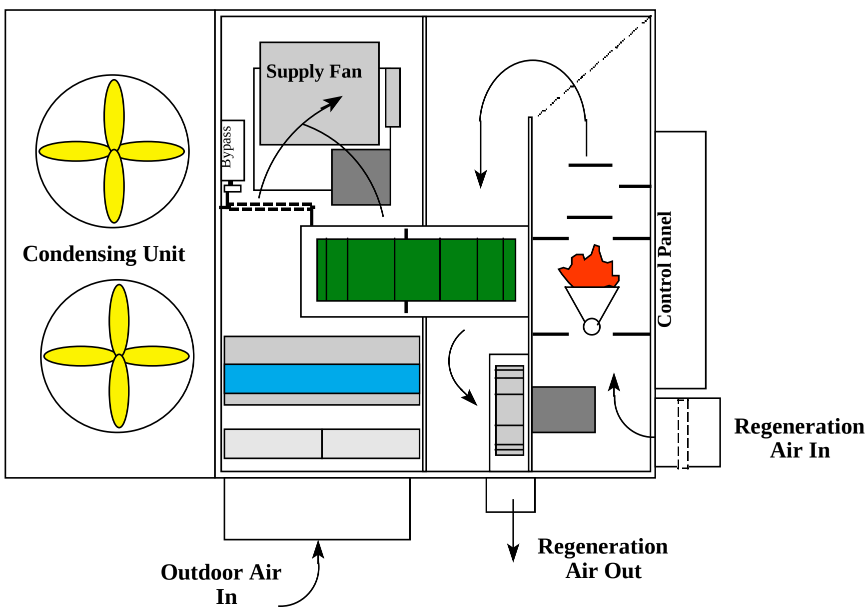

Fig. 7. Simplified schematic of the integrated active desiccant module system approach.

Fig. 8. A photograph of the integrated active desiccant module during assembly, showing how the active wheel section is positioned within the standard rooftop unit.

Fig. 9. A photograph of a fully assembled integrated active desiccant module system with the direct-fired regeneration section shown through the glass panel.

4. LABORATORY TESTING OF THE ACTIVE DESICCANT MODULE AND INTEGRATED SYSTEM

The ADM itself, the ADM unit added as a module to the rooftop unit, and the integrated ADM system were tested as part of this program. The ADM as a unit was tested first to optimize operational parameters such as wheel and burner performance. The final cooling/dehumidification performance data obtained from testing both the ADM piggybacked onto a rooftop unit and the integrated desiccant/rooftop systems were, as expected, essentially the same. As a result, the performance testing procedures discussed in this section pertain to both systems. Likewise, the performance data presented in the section pertain to both systems.

4.1 Test Facility and Procedures Used

Both prototype systems, the ADM and the integrated ADM, were tested thoroughly in the SEMCO air test lab located near the SEMCO headquarters facility in Columbia, Missouri. Figure 10 is a photograph of the SEMCO test loop and instrumentation station.

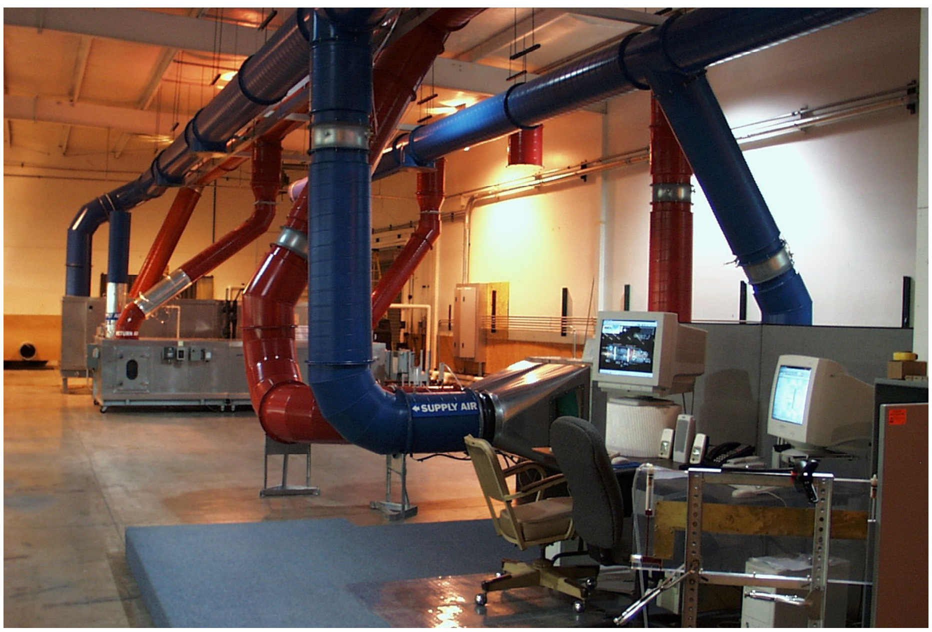

Fig. 10. A photograph of the test loop and instrumentation station in the SEMCO air test laboratory.

This test facility allows two airstreams to be carefully conditioned to simulate indoor and outdoor conditions. The ADM systems were connected to this test facility and fully instrumented. The instrumentation was connected to the central data acquisition system and monitoring station shown in Fig. 10, allowing real time data to be reviewed and collected. The outdoor air conditions created by the facility's preconditioning system were also controlled via a direct digital control panel that is also an integral part of the monitoring station.

As shown in Figs. 2 and 11, a duct connection was made directly to the outdoor air intake section of the Voyager unit (in the case of the ADM) and of the integrated ADM system. As noted in the performance data presented in the following section, the condensing section ambient conditions and the regeneration air inlet conditions were pulled from the test lab, which was maintained at

Fig.11. The instrumented integrated active desiccant module system in the SEMCO test loop.

approximately 80°F and 90 grains per pound of dry air throughout most of the testing. Figure 12 shows the approximate location and number of sensors used to instrument the ADM module during testing.

4.2 Performance Map for the ADM Systems

Tables 3 and 4 summarize the key performance parameters that were obtained from testing the ADM systems. Most of the data contained in Tables 3 and 4 were taken from runs with the integrated ADM system (one outer shell). However, as previously stated, the cooling and dehumidification performance data obtained from the “piggyback” ADM/rooftop combination and the totally integrated units were essentially the same. Therefore, only one set of tables is presented in this report.

To simplify the presentation of the data, the systems were operated as 100% outdoor air preconditioning systems. The units were optimized to provide dehumidified outdoor air at a room-neutral condition. For this testing, room neutral was determined to be no greater than 78°F and no less than 68°F dry bulb temperature. The dehumidification upper limit was chosen to be a 57°F dewpoint (70 grains of moisture per pound of dry air). To highlight the dehumidification capability of the ADM approach and to simplify the data presentation, a constant regeneration temperature of 200°F was used to produce the data presented in Tables 3 and 4.

Parameters that were optimized in these studies include the bypass air fraction (the fraction of the intake air that bypasses the desiccant wheel), active desiccant wheel speed, and cfm/ton processed by the cooing coil. Decreasing the bypass air fraction would allow the system to deliver drier air, but the result would be an increase in the delivered air temperature. By decreasing the rotational speed of the active desiccant wheel, a cooler delivered air temperature could be obtained at the cost of some dehumidification capacity. Reducing the cfm/ton of air processed by the cooling coil would allow the system to deliver colder, drier air at a reduced airflow capacity. In short, there are many parameters that can be adjusted for this ADM approach, which is an advantage as it relates to control options (which are discussed in Sect. 4.3) but a disadvantage as it relates to

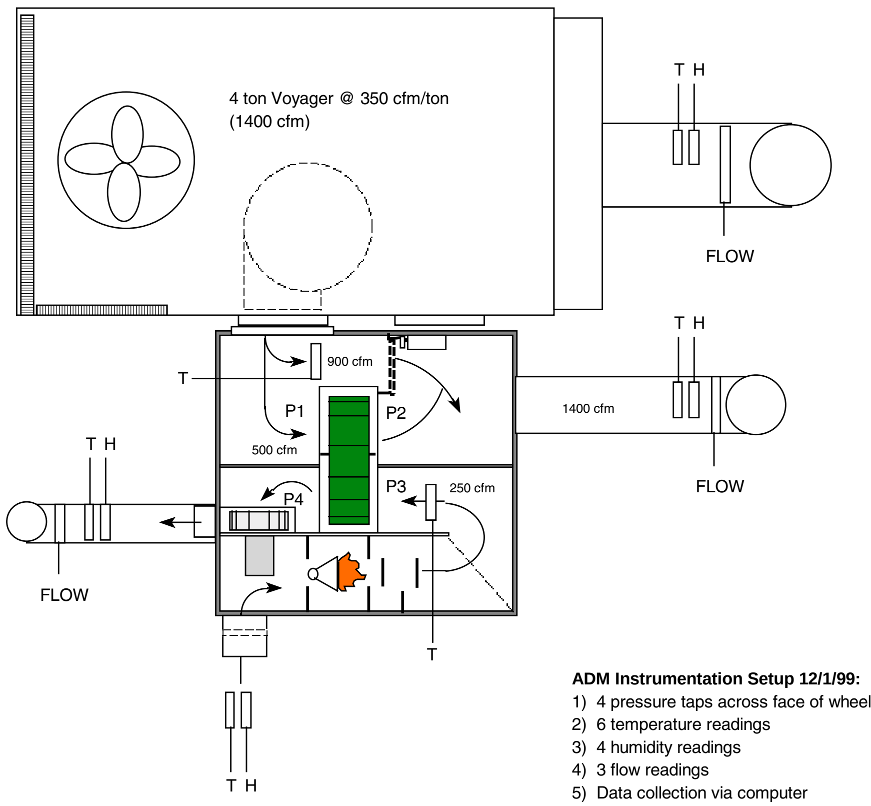

Fig. 12. Schematic of key instrumentation used for active desiccant module testing.

presentation of the performance data in a concise manner. Therefore, numerous parameters had to be fixed for the purposes of this report.

Table 3 shows a wide range of ambient outdoor air conditions. For each of these outdoor air test conditions, the predicted values for the condition of air leaving the coil (based upon an interpolation of manufacturer's data) in addition to the actual data measured during testing are shown. Good agreement was found between the anticipated leaving air conditions and those recorded during laboratory testing. In addition to the conditions leaving the coil, the conditions leaving the active desiccant wheel and those supplied by the system are also presented.

With these data, it is easy to see how the ADM technology allows the conventional rooftop unit to provide dry, room-neutral temperature ventilation air in a simple, energy-efficient manner. The very dry, warm air leaving the active desiccant wheel mixes with the cooler, more humid bypass air leaving the evaporator coil to reach the temperature and humidity condition desired.

Note that two tests were conducted at the 65°F dry bulb temperature, 63°F dewpoint outdoor air condition to highlight the ability of the ADM to handle outdoor air conditions that are cool yet humid, without the use of the compressor section. As discussed previously, this arrangement avoids potential coil frosting, compressor failure, and costly control mechanisms and reduces energy consumption. The first 65°F dry bulb/63°F dewpoint (86 grains/lb dry air) test point shows the temperature that would be delivered by the ADM/rooftop unit combination if both were in operation. As can be seen, air as dry as 43 grains (a 41° dewpoint) can be delivered at this

Table 3. Performance map for the active desiccant module (ADM) systems tested

Test conditions (°F/grains/lb)	Outdoor air test conditions	Estimated cooling coil leaving conditions	Actual cooling coil leaving conditions	Actual active desiccant wheel leaving conditions	Actual conditions leaving the ADM piggyback or ADM integrated system	Regeneration temperature
	(Dry bulb temperature/grains of moisture)			(Dry bulb temperature/grains of moisture		(°F)
Temperature	95	66	66.4	98.2	78	200
Grains	115	92	92	32	70	
Temperature	85	65.5	64.6	96.1	76	200
Grains	130	89	87.2	31	67	
Temperature	85	61	63.8	95.9	75	200
Grains	110	76	84.5	30	65	
Temperature	95	63.5	64.2	96	76	200
Grains	100	83	85	30	65	
Temperature	75	62.5	63.3	95.6	75	200
Grains	130	80	83.3	28	64	
Temperature	85	60	58.4	92.5	71	200
Grains	90	73	69.7	23	53	
Temperature	75	56	58.5	85	68	200
Grains	100	63	70	23	53	
Temperature	70	53	54.5	88.5	67	200
Grains	90	57	60.4	20	46	
Temperature	65	53	52.5	84.2	64	200
Grains	85	57	56.1	19	43	
Temperature	65	Coil	65	98	77	200
Grains	85	off	85	29	65	
Temperature	90	57	57.9	88.8	69	200
Grains	70	65	68.3	22	52	

Notes:

1. ADM coupled with a standard, single-stage 5-ton rooftop packaged unit.
2. Packaged unit operated at 300 cfm/ton (1500 scfm).
3. Bypass air around the active desiccant wheel was 64%.
4. Test data reflect a constant regeneration temperature for simplicity.
5. Bypass fraction and regeneration temperature selected to achieve the delivery of preconditioned outdoor air at a space-neutral temperature (between 68 and 78°F) and at or below a 57°F dewpoint (70 grains/lb).
6. Condenser temperature maintained at 80°F.
7. Regeneration inlet humidity conditions maintained at 90 grains/lb.

Table 4. Latent performance summary for the active desiccant module (ADM) systems tested

Test conditions (°F/grains/lb)	Outdoor air test conditions	Actual cooling coil leaving conditions	Actual conditions leaving the ADM module	Latent load processed by 5-ton roof/top (vapor-compression system only)		Latent load processed by ADM and 5-ton roof/top combination	
	(Dry bulb temperature/grains of moisture)			Latent tons	Delivered dewpoint	Latent tons	Delivered dewpoint
Temperature	95	66.4	78	2.0 tons	65.6°F	3.8 tons	57.0°F
Grains	115	92	70				
Temperature	85	64.6	76	3.6	63.8°F	5.4 tons	56.0°F
Grains	130	87	67				
Temperature	85	63.8	75	2.2	63.0°F	3.8 tons	55.0°F
Grains	110	85	65				
Temperature	95	64.2	76	1.3	63.4°F	3.0 tons	55.0°F
Grains	100	85	65				
Temperature	75	63.3	75	4.0	62.5°F	5.7 tons	54.5°F
Grains	130	83	64				
Temperature	85	58.4	71	1.7	57.6°F	3.2 tons	50.0°F
Grains	90	70	53				
Temperature	75	58.5	68	2.6	57.7°F	4.0 tons	50.0°F
Grains	100	70	53				
Temperature	70	54.5	67	2.5	53.7°F	3.8 tons	46.0°F
Grains	90	60	46				
Temperature	65	52.5	64	2.5	51.7°F	3.6 tons	44.0°F
Grains	85	56	43				
Temperature	65	65	77	0.0	64.2°F	1.7 tons	55.0°F
Grains	85	85	65				
Temperature	90	57.9	69	0.1	57.1°F	1.6 tons	49.0°F
Grains	70	68	52				

Notes:

1. ADM coupled with a standard, single-stage 5-ton rooftop packaged unit.
2. Packaged unit operated at 300 cfm/ton (1500 scfm).
3. Bypass air around the active desiccant wheel was 64%.
4. Test data reflect a constant regeneration temperature for simplicity.
5. Bypass fraction and regeneration temperature selected to achieve the delivery of preconditioned outdoor air at a space-neutral temperature (between 68 and 78°F) and at or below a 57°F dewpoint (70 grains/lb).
6. Condenser temperature maintained at 80°F.
7. Regeneration inlet humidity conditions maintained at 90 grains/lb.

condition. The second test point shows how the targeted supply air conditions are met by operating only the ADM section.

Table 4 provides the test data formulated in a different way to highlight the increased latent capacity made possible by the ADM. As is shown, the ADM increased the latent capacity of the conventional 5-ton rooftop unit by between 50 and 130% without increasing the airflow delivered or the amount of conventional cooling capacity used.

The ADM can be controlled and operated to deliver a variable dewpoint while the regeneration input remains constant. This control method would be the most basic (least costly) control scheme. It would also most closely resemble how packaged rooftop units typically are controlled in the market today.
Just as the compressor is cycled on and off as the space-temperature or supply-air-temperature conditions are satisfied, the ADM can be configured to simply cycle the regeneration burner to deliver additional dehumidification until a pre-set humidity level is achieved. This control approach is the basis for the data presented in Table 3.

Likewise, the ADM regeneration burner or other thermal regeneration source can be modulated and operated to establish a desired dewpoint in the space. If the system produces drier air than is desired, the amount of heat delivered to the regeneration airstream can be reduced until the desired supply air condition is met. Test data were also collected using this variable regeneration energy approach, but they are not presented as part of this report.

4.3 Control Options and Advantages Offered by the ADM Approach

As discussed previously, one of the key advantages offered by the ADM approach is the number of control options available. These options range from very simplified logic, much like that applied to the vast majority of buildings today, all the way to sophisticated control logics required to optimize energy efficiency and space comfort conditions. Some of these control options are presented in this section.

Regeneration energy modulation
The regeneration energy input can be modulated by a control valve serving the direct-fired gas burner to provide only the amount of heat necessary to reach a desired dewpoint. A typical application for this approach would be conditioning a school facility where the desire is to provide a constant supply of dehumidified outdoor air to each classroom at a specified dewpoint. As the outdoor conditions change, the regeneration temperature would vary until the desired delivered outdoor air condition was achieved.

The regeneration energy input can also remain constant, eliminating the added cost of the modulating valve and necessary control components. In this case, the burner would be cycled in much the same way that a standard rooftop unit cycles both the cooling coil and heating source. A typical use for this approach might be positioning a dewpoint sensor in the occupied space and using it to turn the active desiccant regeneration heater on and off. Once the space humidity requirement is satisfied, either the DX section continues to run to provide further sensible cooling, if needed, or both the desiccant regeneration section and the conventional air-conditioning sytem are cycled off.

Bypass air modulation
Another control option offered by the ADM approach is modulation of the amount of air bypassed around the active desiccant wheel or, conversely, processed through the active desiccant

wheel. This control option offers several advantages. If more air is moved through the desiccant wheel, dryer, warmer air will be delivered by the ADM system. If more air is bypassed, cooler, less dry air will be provided. The capability to modulate the bypass air fraction allows the unit to cost-effectively respond to changing space sensible/latent load conditions, especially when the regeneration energy is fixed. Another advantage offered by bypass air modulation is that during a true "economizer" period, the active desiccant wheel can be bypassed to reduce the system's internal static pressure and provide more outdoor air to the space.

Wheel speed modulation
By modulating the rotational speed of the desiccant wheel, the amount of reheat provided can be increased or decreased to help match the sensible load conditions of the space without significantly impacting the dewpoint of the supply air delivered to the space. For example, on cool, humid days, the space will likely benefit from air that is dry but reheated to a room-neutral condition. If the active wheel speed is increased from, say 0.25 to 0.5 rpm, this condition is met. During conditions where the outdoor air is hot and humid, the space will likely be best served by a supply air condition that is as dry and as cool as possible. This condition will be met when the wheel is operated at the slower wheel speed (say 0.25 rpm). This control scheme allows the regeneration energy to serve a dual purpose, both dehumidification and reheating as needed.

Unoccupied mode
Although 100% outdoor air systems are the focus of this report, the ADM approach can be effectively used in applications requiring recirculated air as well. Even 100% outdoor air systems often benefit from the incorporation of a "unoccupied" mode. For example, in a case where the ADM is used to precondition outdoor air going to school classrooms, the classrooms are not occupied for a high percentage of the time. During the summer months, the school facility may not require comfort cooling, but it nevertheless requires humidity control to avoid microbial infestation. The ADM can easily be configured to allow the recirculation of indoor air, reducing or eliminating the delivery of outdoor air. By passing the recirculated air through the ADM, the system can dehumidify the space to a very low dewpoint with minimal running time. Just as important, the dehumidified air can be efficiently delivered at a room-neutral temperature to avoid overcooling spaces with a minimal sensible load. The space dewpoint is easily monitored, and the ADM is activated only when the space calls for dehumidification.

Using condenser heat for reducing regeneration energy
A detailed investigation was conducted into the feasibility of using hot gas or condenser heat to augment the ADM's regeneration energy requirement. This analysis concluded that when the packaged rooftop unit is designed for optimum efficiency, the temperature of the condenser heat is typically in the range of 125°F. Although there are many "free" Btus available for regeneration heat, they exist as "low-grade" energy. It was determined that the added cost and pressure drop associated with incorporating a second condenser coil before the gas-fired burner would not prove cost-effective.

However, a simple yet effective solution was identified that provided a 20% to 30% reduction in regeneration energy consumption by the integrated ADM system. It was determined that simply pulling the regeneration inlet air from the compartment that housed the condenser fan preheated the air entering the direct-fired burner by approximately 25°F. Given that the average regeneration inlet temperature during the cooling season is approximately 75°F, and that the average regeneration temperature is approximately 180°F, the reduction in energy used for regeneration is significant. In this simplified approach, the energy efficiency is realized without increasing the equipment cost or adding significant pressure loss. This simple design feature could easily be integrated into any commercialized integrated ADM system.

5. CONCLUSIONS

This R&D program produced a stand-alone ADM that is easily integrated with new or existing packaged cooling equipment. The program also produced a fully integrated hybrid system that combines the active desiccant section with a conventional DX air-conditioning unit, resulting in a compact, low-cost, and energy-efficient system. Both products were determined to be highly viable products for commercialization.

Several major challenges, including wheel development, packaging, regeneration burner development, control optimization, and low-cost design, were all addressed by the final prototypes produced and tested as part of this program.

Testing confirmed that ADM systems perform effectively, compared with alternate approaches currently used to precondition outdoor air, in applications where a return air path is not readily available for passive desiccant recovery or where funds are not available for the added cost of the necessary return air ductwork.

A simplified energy analysis concluded that the cost of operating the ADM hybrid system would be 45% less than the current operating cost of the overcooling/reheating packaged equipment. Based on actual test data, the ADM approach required only half the installed condensing capacity required by the conventional approach, and it was able to deliver outdoor air at a much lower dewpoint without overcooling the occupied space.

Compared with previous active desiccant system approaches, the ADM was determined to be far more energy-efficient, compact, and cost-effective. The ADM approach described in this report was specifically designed for integration with packaged rooftop cooling equipment, which accounts for the vast majority of all cooling systems sold in the U.S. market.

The ADM approach offers unprecedented control flexibility, which is increasingly desired by the design community. More important, it allows for a cost-effective way to apply conventional, off-the-shelf rooftop units to process 100% outdoor air streams. There is a growing market driven by the need to decouple the outdoor air latent loads from the sensible building loads as a result of ASHRAE 62 requirements. Indeed, a growing segment of the engineering community is beginning to specify outdoor air preconditioning equipment because of the many benefits that this approach provides (Mumma 2001; Fischer 1996))

At the same time, deregulation of the electric and gas industry is providing opportunities to cost-effectively generate electricity on-site for building power. The generation of power on-site is made cost-effective by utilizing the waste heat streams from gas-fired power generators (reciprocating engines, microturbines, and, in the future, fuel cells) to power thermally driven cooling, heating, and dehumidification systems in the building. CHP systems can provide overall on-site efficiencies of 70% for buildings, compared with conventional gas-fired power plant efficiencies of 30%. Many buildings in the commercial sector can benefit from the integration of CHP with desiccant dehumidification. The compact, cost-effective ADM approach identified as part of this research program provides an excellent product for integration into CHP systems.

Future Work

A market investigation completed during the early phase of this R&D program (Fischer, Hallstrom, and Sand 2000) provides a desiccant preconditioning market analysis, which concludes, among other things, that restaurant and hotel/motel facilities could benefit significantly from cost-effective active desiccant systems. Based upon the success of this development phase, two field pilot projects are already under way for add-on ADMs combined with standard packaged rooftop units. The first site, which is currently under construction, involves a renovation to a hotel/motel facility. The second pilot site, scheduled for completion in early 2002, will be a restaurant. The results of this field investigation will be made available after the completion of the monitoring phase at both facilities.

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