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Active desiccant integration with packaged rooftop HVAC equipment

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Abstract

Current research indicates a direct correlation between indoor air quality and fresh air ventilation rates which supports requirements for building ventilation standards calling for continuous supply and increased amounts of ventilation to help assure safe and healthy interior air environments [O. Seppänen, W.J. Fisk, M.J. Mendell, Ventilation rates and health, ASHRAE Journal (August) (2002) 56–58; C.C. Downing, C.W. Bayer, Classroom indoor air quality vs. ventilation rate, ASHRAE Trans., 1993, Vol. 99, Part 2, Paper Number DE-93-19-1, pp. 1099–1103. [1,2]]. Off-the-shelf, packaged rooftop equipment used to air condition most facilities is not designed to handle the increased or continuous supply of outdoor air necessary to comply with building ventilation codes written to this new standard [American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc. (ASHRAE), Ventilation for acceptable indoor air quality, Standard 62-1989, 1791 Tullie Circle, NE, Atlanta, GA 30329. [3]].

Integration of a rooftop, unitary air conditioner with an active desiccant module (ADM) allows the use of a standard rooftop air conditioner with a thermally regenerated active desiccant component to provide a compact, cost-effective, and simple-to-use packaged system for efficiently pre-treating and supplying ventilation air adequate to ensure healthy indoor environments. By designing a combined vapor-compression/active desiccant system with the desiccant component positioned after a conventional cooling coil, the dehumidification effectiveness of the desiccant is significantly enhanced because it operates on cold, saturated, or nearly saturated, air leaving the evaporator. "Post-coil" rather than the normally used, "pre-coil" desiccant arrangement also minimizes the regeneration temperature required for the active desiccant, allows

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for partial bypass and post-cooling of the desiccated air after recombination, and dramatic decreases in the overall size for the pre-conditioning unit.

Keywords: Desiccants; Humidity control; Ventilation; Packaged systems

1. Introduction

Most commercial and public buildings utilize packaged heating and cooling equipment, designed to provide inexpensive, efficient heating and cooling, with minimal outdoor air [4]. This type of equipment was not designed to handle the continuous supply or increased volume of outdoor air necessary to comply with minimum ventilation standards currently being adopted by commercial building codes. Attempts to meet these ventilation and makeup air recommendations using conventional packaged HVAC units often result in poor indoor humidity control and wide temperature fluctuations.

The most common answer to this problem is specifying oversized packaged equipment which leads to providing more airflow than required and sub-par performance. On mild, humid days (part load conditions) an oversized packaged unit will quickly cool the space to the set point temperature then cycle off the compressor. If the supply fan is run continuously, raw outdoor air is introduced to the space and the indoor humidity level climbs until the space thermostat once again calls for cooling. By this time, the mixed air condition supplied to the coil is elevated in humidity. The net result is a high dew point temperature leaving the cooling coil. Space temperature is still maintained but humidity control is lost, resulting in uncomfortable conditions for building occupants. We have all been in hotel conference rooms and motel rooms where we experienced cold, clammy conditions. In an attempt to feel comfortable, occupants often lower the thermostat setting further, once again increasing the space relative humidity.

Another problem experienced by oversized packaged equipment is the re-evaporation of moisture from the evaporator coil. Henderson et al. [5] and Khattar et al. [6] have both confirmed this phenomenon often observed in the field where the actual moisture removed by a packaged HVAC unit is less than that anticipated based on published performance data. The research shows how moisture condensed on a DX coil evaporates back into the supply air stream when the compressor and cooling system is cycled off and the supply fan continues to run. Henderson [7] documented that the actual latent removal can be reduced to less than 50% of the unit's rated capacity under part load operating conditions.

These and other limitations present real problems to packaged rooftop systems applied to handle high percentages of outdoor air, especially for 100% outdoor air systems. For example, when applying a conventional rooftop system to handle all outside air, the refrigeration capacity required at peak condition is far greater than the cooling output available at the rated airflow of the conventional unit. For example, conditioning a 0.71 m^3/s outdoor air stream from 29.4 °C and a 23.6 °C dew point to a 13.3 °C dew point requires a 35.2 kW (10-ton) unit. However, the least amount of air that can be processed by a 35.2 kW unit without potential control problems, coil frosting and compressor failure is approximately 1.42 m^3/s, twice that necessary.

If the unit is operated at 50% outdoor air to provide the required 0.71 m^3/s of fresh air ventilation with a total of 1.42 m^3/s made up of outside + return airflow, it may well overcool the space and require expensive parasitic reheat, especially at part-load conditions. Most of these problems are well known to designers and do not merit further explanation here [8].

There is a growing market driven by the need to de-couple the outdoor air loads from the sensible building loads as a result of ASHRAE 62 requirements. A growing segment of the engineering community is beginning to apply a dual system employing outdoor air preconditioning equipment for ventilation and humidity control and a separate system to control indoor temperatures due to the many benefits that this approach provides [9,10].

Active desiccants have been applied in conjunction with rooftop packaged equipment in the past to address increased ventilation air pre-conditioning concerns in other building applications like restaurants and theaters. Often these systems utilized a traditional approach of processing all of the outdoor air with the active desiccant wheel first, in an attempt to handle most or the entire latent load with the active desiccant wheel, then post cooling as necessary. This approach did not find market acceptance for several reasons, including first cost, operational cost, and energy efficiency [11].

When an active desiccant wheel removes moisture from an air stream, heat is released as a byproduct of the adsorption process. The more moisture that is removed, the more heat released. As a result, if an active desiccant wheel is asked to change the moisture content of incoming air from a 23.6 °C dew point to a 14 °C dew point, a very significant increase in the processed air temperature results. This is exacerbated by the fact that a high regeneration temperature must be used to remove such a large quantity of moisture from the solid desiccant wheel in the regeneration step of the process. This higher regeneration temperature results in more heat carry-over by the wheel from the hot regeneration air stream to the outdoor air stream being processed prior to use as building ventilation air, further raising the air temperature.

Based upon literature values for active desiccant wheels currently available, this 9–10 °C reduction in process air dew point would increase the ventilation air temperature entering the cooling coil of the packaged rooftop unit in this traditional desiccant/rooftop approach from approximately 29° to 59 °C. The amount of post cooling required to remove the additional sensible heat associated with the solid desiccant dehumidification process will be similar to that required to remove the humidity without the desiccant system. As a result, this approach does not reduce energy consumption; it actually increases it.

The other major problem with using an active desiccant system to process all of the outside air before it enters a conventional rooftop unit is that the desiccant wheel in this configuration must be sized to process the entire outdoor air stream. In order to remove 8.6 g of moisture per kilogram of dry air, the process air stream face velocity at the desiccant wheel's surface must be very low. This results in a large, extremely costly active desiccant wheel and preconditioning module that may be two times the size of the rooftop it is serving.

The active desiccant module (ADM)/rooftop combination investigated in this research work resolves these issues by applying a system configuration which positions the desiccant module after the packaged rooftop cooling coil, not before. As shown in Fig. 1, the ADM approach used the desiccant to dehumidify only a portion of the incoming air after it is cooled and possibly

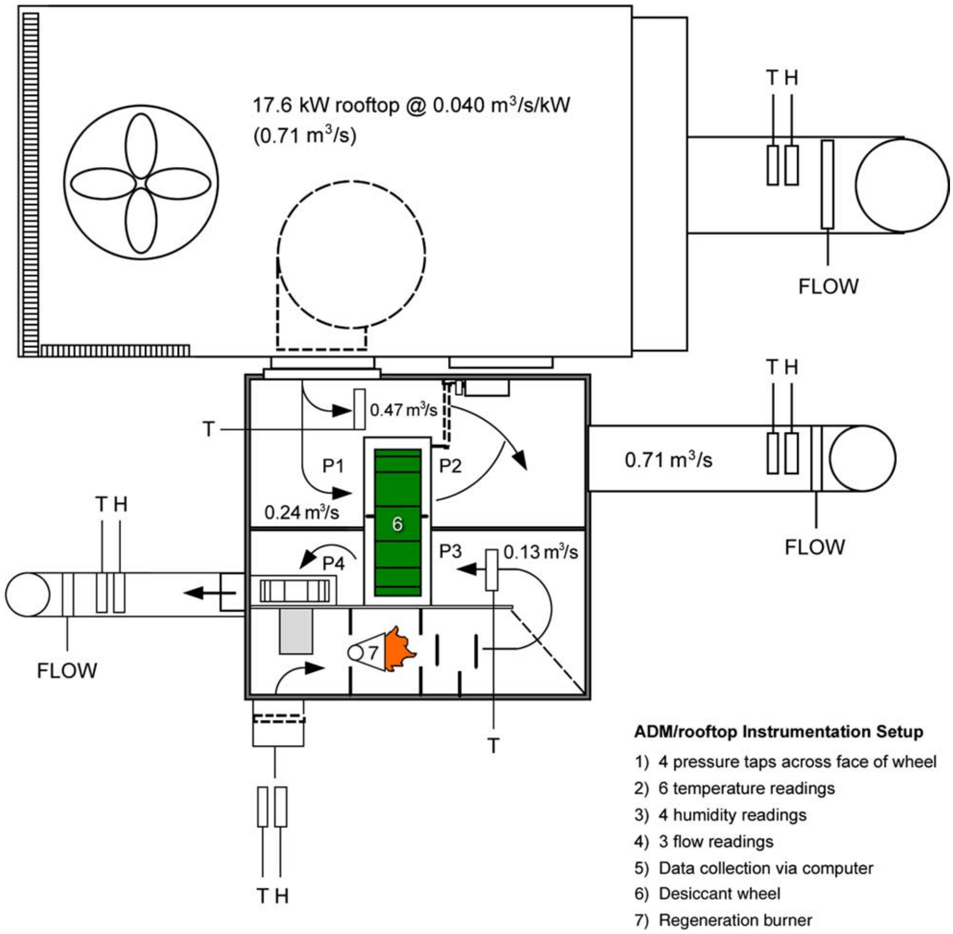

Fig. 1. Schematic of ADM/rooftop outdoor air preconditioning unit with airflow paths/flow rates and monitoring instrumentation.

dehumidified by the evaporator of the packaged unit. Fig. 2 illustrates the difference between ‘‘pre-coil’’ and ‘‘post-coil’’ desiccant dehumidification.

In Fig. 2a, relatively hot and humid outside air is passed through a desiccant wheel, as it would be in a ‘‘pre-coil’’ arrangement. This results in a dry, hot air stream, point B, that must be extensively cooled before it can be delivered to the building as comfortable ventilation air. Fig. 2b, which illustrates ‘‘post-coil’’ desiccant dehumidification, shows that outside air is cooled and partially dehumidified by the conventional rooftop air conditioner, A–C–D. After being precooled, a portion of the cool, saturated air leaving the coil is passed through the desiccant wheel, D–E, while 50–67% of it is bypassed around the desiccant and recombined with the desiccated air to provide cool, dry, fresh air ventilation to the building being served. What cannot be illustrated in Fig. 2 are the relative equipment size, blower power, and various energy saving advantages afforded by this ‘‘post-coil’’ desiccant approach. These are enumerated and explained in Section 2.

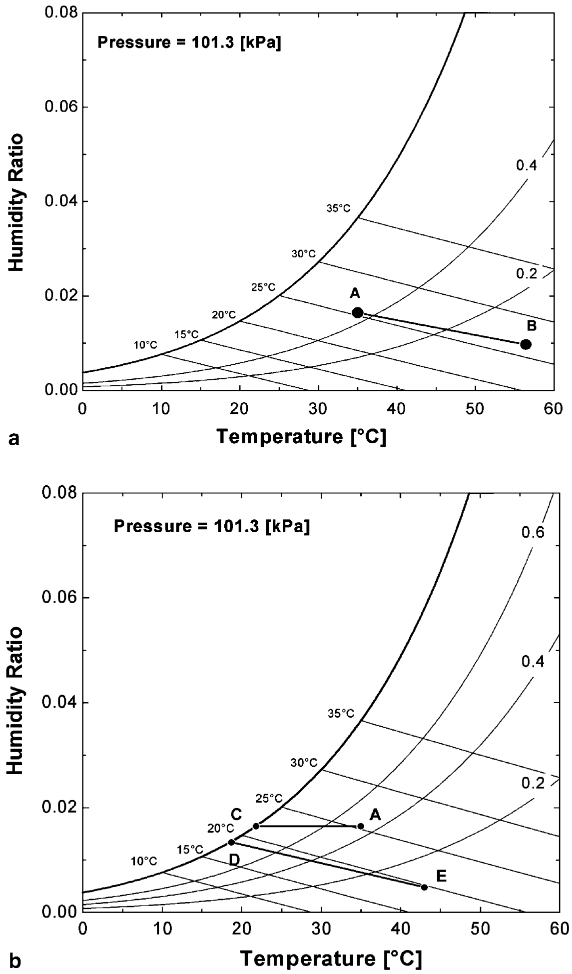

Fig. 2. Psychrometric effect of (a) "pre-coil" hot and humid air and (b) "post-coil" pre-cooled outside air passed through an active desiccant wheel.

2. Discussion

2.1. The active desiccant module (ADM) approach

The ADM module includes an active desiccant wheel sized to process approximately 33% of the air that passes across the cooling coil of the packaged rooftop unit. 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 dew point and 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 dew point and at a room neutral temperature (20–25 °C).

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. Since all packaged units utilize forward curve fans, the external static capability is limited. Also, the desiccant wheel in this application should be optimized for best performance at moderate regeneration temperatures and with saturated inlet conditions.

The ADM also integrates a direct-fired burner and fan to heat/move 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 a combined cooling, heating and power (CHP) system where waste heat is available for regeneration of the active desiccant wheel.

2.2. Advantages of the ADM approach

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 simulated comparison made between the ADM/rooftop packaged system combination and a customized 35 kW packaged unit designed to handle 100% outdoor air. The comparison assumes that each system will process 0.71 m^3/s of outdoor air from a cooling season design conditions of 29.4 °C with a 23.6 °C dew point to a 13.3 °C dew point at room neutral temperatures. It also assumes that in order to avoid over-cooling the space, the outdoor air will be reheated to 20 °C prior to its introduction to the space. The energy analyses assume continuous operation and use utility costs of $.07/kwh for electricity and gas at $4.50/million BTU ($4.27/MJ) for natural gas.

As shown in Table 1, the first obvious advantage is that the refrigeration capacity of mechanical cooling 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

Table 1
Comparison of active desiccant module (ADM) approach with a custom packed DX rooftop unit approach

	ADM rooftop combination	Custom DX rooftop (over-cool and reheat)
Cooling capacity required (kW)	17.6 kW	35.2 kW
Reheat energy required (kJ/h)	0	34,200
Regeneration energy rqd. (kJ/h)	35,300	N/A
Supply dew point used for analysis	13.3 °C (56 °F)	13.3 °C (56 °F)
Annual cooling energy cost	$1360	$2480
Unit approximated size ($H \times W \times L$) (cm)	$79 \times 117 \times 117$	$85 \times 118 \times 211$

requirements, this reduces the amount of compressor cycling since the smaller rooftop is fully loaded far more frequently. This also minimizes the problem of condensate re-evaporation from the cooing coil mentioned previously.

Table 1 also compares the estimated energy consumption for both system approaches. 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 during peak utilization seasons.

Table 2 compares the ADM/packaged rooftop approach with two previously marketed active desiccant system configurations. The first places the active desiccant wheel prior to the packaged rooftop system. The second is the traditional "desiccant based cooling system" (DBC) which has also been installed upstream of the packaged rooftop system, but in addition to the active desiccant wheel includes a sensible only recovery wheel and evaporative cooling section. These components were added to the DBC to remove much of the heat of adsorption from the incoming fresh air stream prior to its delivery to the cooling system and to preheat the regeneration air stream. This DBC approach is discussed extensively in the ORNL Active Desiccant-Based Preconditioning Phase 2 Market report [12].

Data presented in Table 2 highlights many of the benefits offered by the ADM approach. The ADM provides the desired dehumidification capacity using only 17.6 kW of mechanical cooling capacity compared to 29.5 kW of conventional cooling required by the active desiccant preconditioning approach (installed upstream of the cooling coil). It also utilizes only 40% of the regeneration energy 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 of outdoor ventilation air. The corresponding reduction in the active desiccant wheel diameter results in a much smaller final product. Maintaining a module size compatible with that of the packaged cooling equipment is a significant advantage. It also results in far less fan power being utilized by the combined ADM/rooftop system approach to move air.

When 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 makes the ADM the obvious choice.

Table 2
Comparison of the ADM-rooftop combination with other active desiccant system approaches previously marketed for preconditioning outdoor air supplied to conventional cooling systems

	ADM/rooftop combination	Active desiccant preconditioning	Traditional DBC preconditioning
Cooling capacity required (kW)	17.6 kW	29.5 kW	8.8 kW
Air process by active wheel (m^3/s)	0.24	0.71	0.71
Regeneration energy rqd. (kJ/h)	35,300	87,200	64,900
Supply dew point used for analysis	13.3 °C (56 °F)	13.3 °C (56 °F)	13.3 °C (56 °F)
Annual cooling energy cost	$1360	$2620	$1560
Approx. unit size ($H \times W \times L$) (cm)	$79 \times 117 \times 117$	$132 \times 168 \times 168$	$132 \times 168 \times 270$
Relative cost of manufacturing	1	2.2	3

2.3. Control options and advantages offered by the ADM/rooftop approach

One of the key advantages offered by the ADM/rooftop approach is the number of control options available. 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 dew point. 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 much the same way a standard rooftop unit cycles both the cooling coil and heating source.

Another control option is modulation of air bypassed around the active desiccant wheel or, conversely, processed through the active desiccant wheel. By moving more air through the desiccant wheel, dryer, warmer air will be delivered by the ADM system. By bypassing more air, cooler, less dry air will be provided. The ability 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 internal static pressure and provide more outdoor air to the space.

By modulating the rotational speed of the desiccant wheel, the amount of reheat provided can be increased or decreased to help match the space sensible load conditions without significantly impacting the supply air dew point 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 .25 rpm to .5 rpm, this is accomplished. During conditions where the outdoor air is hot, sunny 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 .25 rpm). This control scheme allows the regeneration energy to serve a dual purpose, both dehumidification and reheat as needed.

3. Results

The 17.6 kW rooftop unit by itself and the unit with an added desiccant module were subjected to performance testing in an engineering laboratory. The test facility provides two air streams of carefully conditioned air to simulate indoor and outdoor conditions. Table 3 indicates the range of conditions over which the two units were tested. Data presented in this table shows the latent load performance of both systems and how the added ADM allows the conventional rooftop to provide dry, room neutral outdoor ventilation air in a simple, energy efficient manner. In the combined unit, the very dry warm air leaving the desiccant wheel mixes with cooler evaporator air bypassed around the wheel to reach the supply air temperature and humidity range desired in the ventilation air stream.

To simplify this presentation of the data, both systems were operated as 100% outdoor air preconditioning units. Post-evaporator air temperatures from the rooftop unit are within 0.5 °C of the dew point under these operating conditions. Room neutral supply temperature from the

ADM/rooftop combination system was defined to be no greater than 25 °C and no less than 20 °C. To highlight the dehumidification capability of the ADM approach and to simplify the data a constant regeneration temperature of 93.3 °C was utilized to obtain the data in Table 3.

By decreasing the bypass air fraction, drier air could be delivered from the system, but the delivered air temperature would increase. Decreasing the rotational speed of the active desiccant wheel could provide cooler air at the cost of dehumidification capacity. Decreasing the $m^3/s/kW$ processed by the unit would result in cooler, drier air at a reduced capacity. Many operational parameters can be adjusted for this ADM/rooftop approach, all of these can be considered as control options. The performance data presented in Table 3 was obtained by operating the ADM/rooftop equipment in a manner that made concise presentation of the results possible.

Table 3 shows that two tests were conducted at the 18.3 °C, 17.0 °C dew point, outdoor air condition. This is to highlight the ability of the ADM to handle outdoor air conditions that are cool and humid, without the use of the vapor-compression air conditioning section. This avoids potential coil frosting, compressor failure, and costly control mechanisms and it reduces energy consumption. The first 18.3 °C dry bulb/17.0 °C dew point test shows the latent performance that would be delivered by the ADM/rooftop combination if both were in operation. As indicated, a 6.7° dew point supply air stream can be delivered at this condition. The second point shows how the targeted supply air conditions are met by only operating the ADM section. Table 3 also highlights the increased latent capacity made possible by the ADM module. As shown, the ADM increased the latent capacity of the conventional 17.6 kW rooftop by 50–130% without increasing the airflow delivered or the amount of conventional cooling capacity utilized.

Table 3
Performance summary for the ADM/rooftop system tested in laboratory

Outdoor air inlet test conditions		Latent load processed by 17.6 kW rooftop alone		Latent load processed byADM/rooftop combination	
Temp. (°C)	Dew point (°C)	Latent (kW)	Dew point (°C)	Latent (kW)	Dew point (°C)
35.0	21.8	7.0	18.7	13.4	13.9
29.4	23.7	12.7	17.7	19.0	13.3
29.4	21.1	7.7	17.2	13.4	12.8
35.0	19.6	4.6	17.4	10.5	12.8
23.9	23.7	14.1	16.9	20.0	12.5
29.4	17.9	6.0	14.2	11.3	10.0
23.9	19.6	9.1	14.3	14.1	10.0
21.1	17.9	8.8	12.1	13.4	7.8
18.3	17.0	8.8	10.9	12.7	6.7
18.3	17.0	Coil off	Coil off	6.0	12.8
32.2	14.0	0.1	13.9	5.6	9.4

Notes: ADM coupled with a standard, single-stage, 17.6 kW rooftop packaged unit.
Packed unit operated at 0.040 $m^3/s/kW$ (0.708 m^3/s) air flow rate.
Bypass fraction and regeneration temperature selected to achieve delivery of conditioned air at space-neutral temperature (20–25 °C) and at or below 13.9 °C dew point.
Rooftop condenser temperature maintained at 26.7 °C.
Regeneration inlet humidity conditions maintained at 17.9 °C dew point.

4. Conclusions

Laboratory testing of an active desiccant module specifically designed to be integrated with a conventional rooftop air conditioner confirmed this combination is an effective ventilation air pre-conditioning system when compared to alternate approaches currently used to precondition outdoor air in 100% outdoor air applications.

A simplified energy simulation analysis showed the cost of operating the ADM/rooftop hybrid system would be 45% less than the over-cooling/reheat packaged equipment routinely applied today. Based on actual test data, the ADM approach required only one half the installed condensing capacity required by the conventional approach, and was able to deliver outdoor air at a much lower dew point without over-cooling the occupied space.

Compared to previous active desiccant system approaches, the ADM was determined to be far more energy efficient, compact, and cost effective. The ADM equipment described here was specifically designed to integrate 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 importantly, it allows for a cost-effective way to apply conventional, "off-the-shelf" rooftop units to process 100% outdoor air streams.

Utilizing a design in which the desiccant processes only a portion of the cold, saturated air from a pre-cooling process, the ADM/rooftop system makes the most effective and efficient use of the desiccant component, dramatically reduces the dimensions and cost of the desiccant module, and allows desiccant regeneration with a 93.3 °C air stream. All of these features make this ventilation air preconditioning approach much more suitable for combined cooling, heating, and power (CHP) for buildings or integrated energy systems (IES) applications where the desiccant system can use waste heat from a thermally driven source of power for desiccant regeneration.

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