

Taking the Heat out of Desiccants

Enhancing the dehumidification capability of traditional vapor-compression equipment widely used in commercial applications

The rising cost of natural gas, combined with the improving efficiency of conventional cooling equipment, favors the use of vapor-compression equipment for the dehumidification of air in commercial applications. In applications (e.g., industrial process) in which conventional vapor-compression equipment cannot achieve the required supply-air dew-point temperature (Figure 1), heat-activated desiccants often are used.

With interest in maintaining lower humidity levels in commercial applications growing, the use of desiccants to enhance the dehumidification capability of traditional vapor-compression equipment is on the rise.

A major barrier to the use of desiccants in commercial applications has been the high-temperature

heat required for regeneration. This heat can result not only in operating costs higher than those of a traditional cool-and-reheat system, but the unwanted transfer of heat to supply-air streams. This article will examine ways to minimize or avoid the use of “new-energy” regeneration heat when desiccants are used in commercial applications.

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SUPPLY-AIR DEW POINT

Conventional direct-expansion (DX) equipment often is limited to an approximately 45°F supply-air dew point. Specialty DX equipment may be able to achieve a 38°F dew point, but requires steady conditions and complex controls. The risk of freezing water prevents chilled-water systems from being used to dehumidify air much below a 42°F dew point. The addition of glycol to water allows fluid temperatures below 32°F, but because of coil frosting, dehumidification below a 38°F dew point is uncommon. If an application requires a supply-air dew point below these limits, a desiccant-dehumidification system likely will be used.

DEHUMIDIFICATION USING A COLD COIL

Dehumidification can be accomplished using a cold coil, with air passing through the coil cooled to below its dew point and water condensed

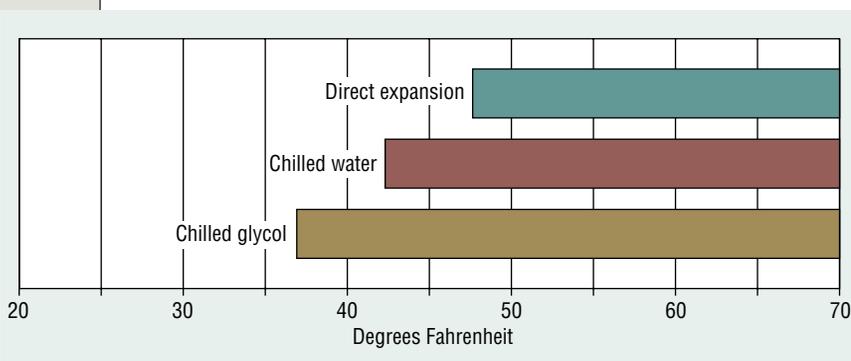


FIGURE 1. Supply-air dew-point capability of conventional cooling equipment.

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out of the air onto the surfaces of the coil (Figure 2). If dehumidification to a required supply-air dew point results in too much sensible cooling (too low of a dry-bulb temperature), air must be reheated to the desired supply-air dry-bulb temperature.

DEHUMIDIFICATION USING A HEAT-ACTIVATED DESICCANT

As a heat-activated desiccant adsorbs (removes) water vapor from air, it warms the air, typically requiring that the air be re-cooled. For the dehumidification process to continue, the desiccant must be regenerated with high-temperature heat. Adding heat changes the isotherm of the desiccant, decreasing the desiccant's ability to hold water vapor and lowering the desiccant's water-vapor content (Figure 3).

When the desiccant wheel again is exposed to the process-air stream, it

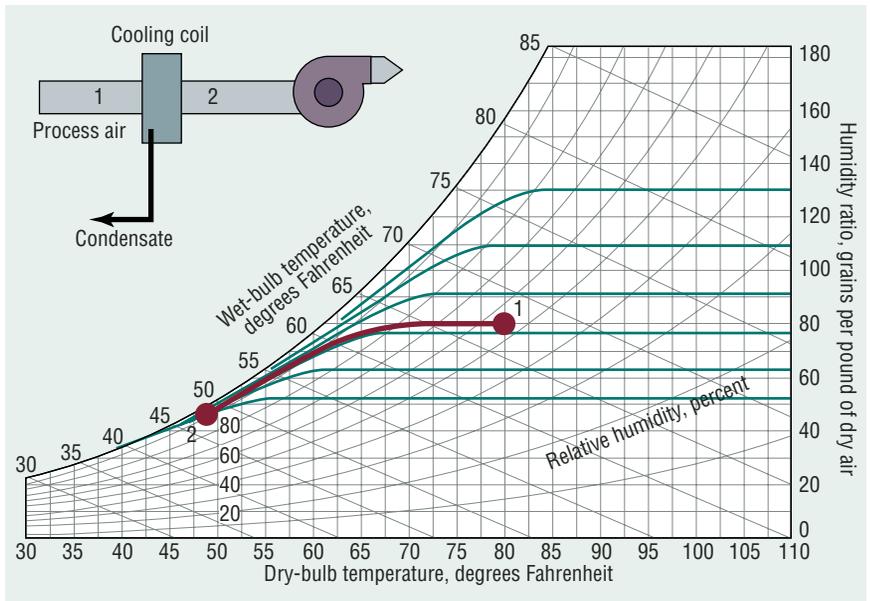


FIGURE 2. Dehumidification using a cold coil.

adsorbs water vapor, but adds both sensible heat carried over from

regeneration and the heat of adsorption (Figure 4). Heat of adsorption is the

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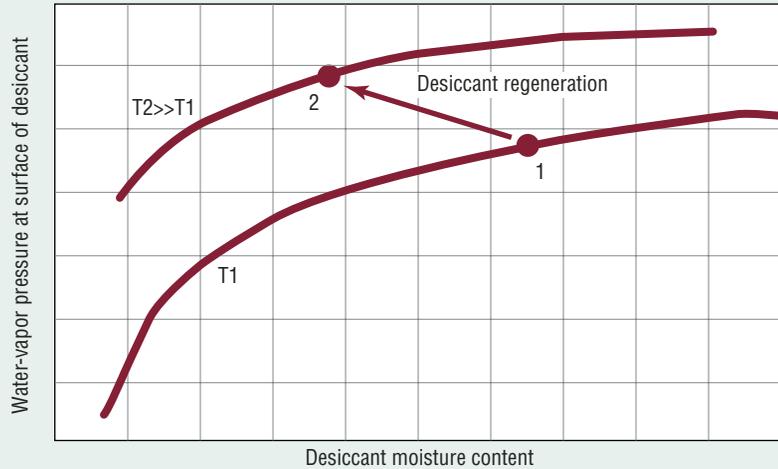


FIGURE 3. Effect of regeneration heat on desiccants.¹

energy exchanged during the process of adsorbing water vapor.

HEAT-ACTIVATED DESICCANT VS. VAPOR-COMPRESSION REFRIGERATION

With a heat-activated-desiccant wheel, the amount of energy required to dehumidify air is equal to the heat energy required for regeneration. The amount of heat energy required for regeneration depends on the isotherm properties of the desiccant. Most heat-

activated-desiccant wheels require 2,000 Btu to 2,400 Btu of heating energy for every pound of water transferred, while the temperature of the regeneration-air stream typically ranges from 180°F to 280°F.² Heat energy of this temperature can be recovered from some industrial processes or from on-site power-generation equipment, but is not readily available for recovery in most commercial applications. Therefore, this regeneration heat typically comes from

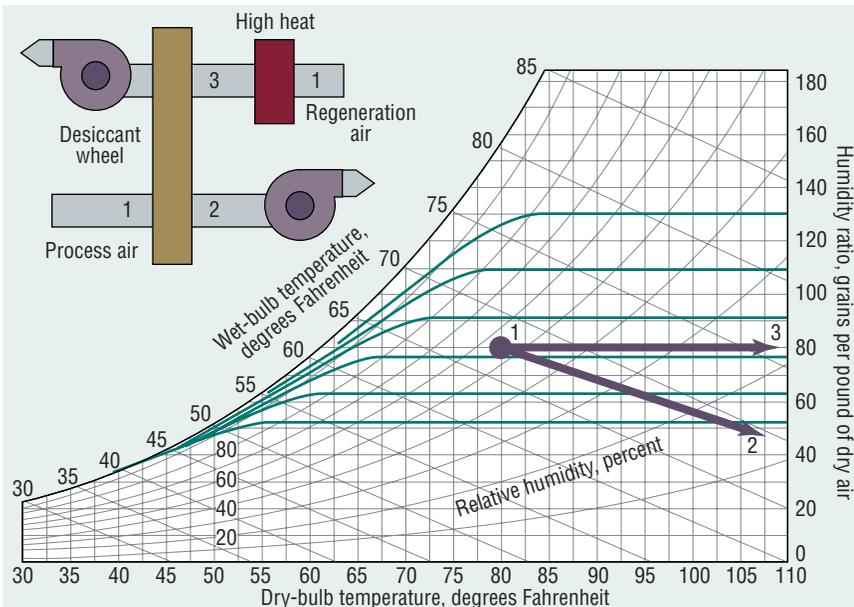


FIGURE 4. Dehumidification using a heat-activated desiccant.

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natural gas by way of either a direct-fired burner or a high-pressure steam boiler. Considering that the coefficient of performance (COP) of these regeneration-heat sources ranges from 0.80 to 0.89, a typical heat-activated-desiccant wheel requires 2,250 Btu to 3,000 Btu of input energy for every pound of water removed.

The sensible-heat ratio of a cooling coil typically ranges from 0.50 to 0.75. With the heat of vaporization of water equal to 1,060 Btu per pound of water, this equates to 2,120 Btu to 4,240 Btu of cooling energy for every pound of water removed. Considering that the COP of conventional vapor-compression equipment ranges from 4.0 to 6.0, a typical cooling coil needs 520 Btu to 800 Btu of input energy for every pound of water removed.

Considering moisture removal only, dehumidifying with heat-activated-desiccant equipment could require three to six times more input energy than dehumidifying with vapor-compression equipment. While the cost (per British thermal unit) of natural gas is significantly lower than the cost of electricity, the difference is not great enough to overcome the difference in efficiency. From 2005 to 2006, the national average

commercial natural-gas price was \$1.12 per therm, while the average commercial electricity price was 8.5 cents per kilowatt-hour. Without a source of waste heat, the cost of removing a pound of water would be 25- to 95-percent greater with heat-activated-desiccant equipment than it would be with vapor-compression equipment.

This quick comparison, however, does not consider the sensible-cooling or heating needs of a space. Most commercial applications require sensible cooling at the same time dehumidification is needed. This is opposite what a heat-activated-desiccant wheel provides: Supply air is heated 1,400 Btu to 1,800 Btu for every pound of water removed. Figure 5 compares the performance of a cold coil and a heat-activated desiccant dehumidifying air from 80°F dry bulb and 67°F wet bulb, or 79 grains per pound, to a 48°F dew point, or 49 grains per pound.

The heat-activated-desiccant wheel warms the air to approximately 110°F dry bulb, while the cold coil cools it to approximately 48°F dry bulb. If the supply-air dry-bulb temperature required to satisfy the space sensible-cooling load were 60°F, the heat-activated-desiccant system would use approximately the

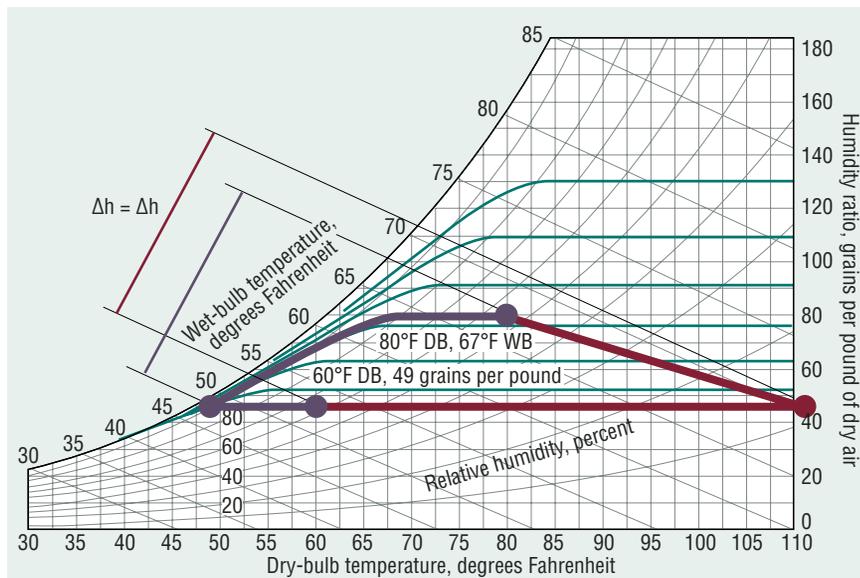


FIGURE 5. Cold-coil vs. heat-activated-desiccant performance.

same amount of total cooling energy as the cold-coil system ($\Delta h = \Delta h$), but three times the heating energy. At typical part-load conditions—and even at design conditions—a constant-volume cool-and-reheat system may require heat, but not nearly as much as that required by a heat-activated-desiccant system. In this example, if a warmer supply-air temperature were required, the desiccant system would use slightly less cooling energy, but still not enough to overcome the amount of high-temperature heat energy required for regeneration. When space sensible cooling is considered in the comparison, the vapor-compression system looks more efficient.

MINIMIZING REGENERATION HEAT IN DESICCANT SYSTEMS

To minimize the energy impact of desiccant-based dehumidification, pre-cool supply air with a conventional cooling coil before the air is dehumidified by the desiccant wheel (Figure 6). Some of the air will pass through the desiccant wheel and mix with air that bypassed the wheel. Some dehumidification responsibility will shift from the desiccant wheel to the more-efficient cooling coil. With the higher delta-T resulting from pre-cooling, however, more heat will be transferred from regeneration to the process. Because some process air will leak to the lower-pressure regeneration side, some pre-cooling will be wasted. The dehumidified supply air still will be heated by the desiccant, so re-cooling may be needed downstream of the wheel. Because this configuration reduces the work performed by the heat-activated-desiccant wheel, overall system efficiency can be improved.

The use of desiccants in commercial applications has evolved based on this premise: Push as much dehumidification work as possible to the cooling coil, and use the desiccant to lower the supply-air dew point below what the coil is able to achieve by itself. For example, if a conventional cooling coil can achieve a 48°F dew point, let the coil dehumidify

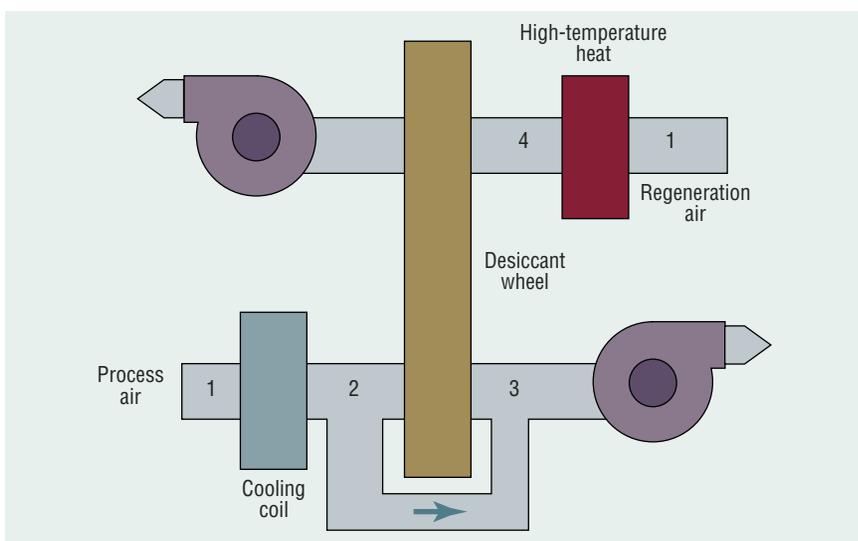


FIGURE 6. Pre-cooling to minimize the work performed by a heat-activated desiccant.

the air to that point; then use the desiccant wheel to lower the dew point further.

USING DESICCANTS WITHOUT ADDED REGENERATION HEAT

The next step to a more efficient system is eliminating the need for high-temperature regeneration heat. This can be accomplished using a desiccant with a different isotherm shape. The three most common isotherm shapes are shown in Figure 7. With Type I or Type II desiccants, isotherm

shape can be changed with the addition of heat (Figure 3). Type II desiccants commonly are used in industrial dehumidification processes because they can adsorb and hold water vapor, even when process air already is very dry. This makes the achievement of dew points below 0°F possible.

Switching to a Type III desiccant and configuring a desiccant wheel in series with a cooling coil can eliminate the need for regeneration heat. In this arrangement, the wheel removes water vapor by “riding” the desiccant isotherm

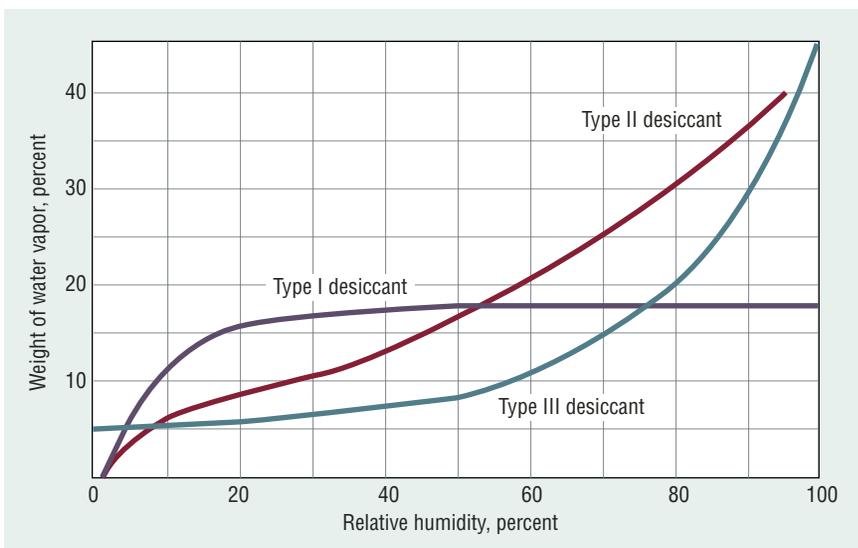


FIGURE 7. Typical desiccant isotherm shapes.³

(Figure 8). Because air leaves cooling coils at very high relative humidities, the desiccant is going to adsorb water vapor. On the other side of the wheel, the relative humidity (RH) of the regeneration air needs to be less than 70 percent for water vapor to be removed from the desiccant. In many applications, this is not difficult to achieve and may not require regeneration heat. Many environments requiring low supply-air dew points, such as surgery rooms and laboratories, require a high air-change rate, resulting in a large percentage of recirculated return air. The space sensible load heats the return air and lowers the RH of the air entering the upstream side of the series desiccant wheel, often to a level low enough to avoid the need to add heat for regeneration. With no heat added, the Type-III-isotherm wheel can be used in series with the cooling coil. In this arrangement, all of the dehumidification work is moved to the cooling coil, and the desiccant wheel is used to allow the system to achieve a supply-air dew point lower than the one the conventional cooling coil could achieve by itself. In addition, the wheel allows the coil to dehumidify with a warmer coil temperature, improving system efficiency.

DEHUMIDIFICATION USING A COLD COIL AND TYPE III SERIES DESICCANT WHEEL

Figure 9 shows a cold coil and Type III series desiccant wheel dehumidifying air from 80°F dry bulb and 67°F wet bulb, or 79 grains per pound, to a 48°F dew point, or 49 grains per pound. The air leaves the cooling coil at 54°F dry bulb and 97-percent RH. The desiccant wheel adsorbs water vapor to lower the dew point of the supply air to 48°F. Because of the heat of adsorption, sensible heat is added to the supply air, raising the dry-bulb temperature of the air leaving the wheel to 61°F. The opposite happens on the regeneration side of the wheel. Because the RH of the air passing through the upstream side of the wheel is 50 percent, the desiccant

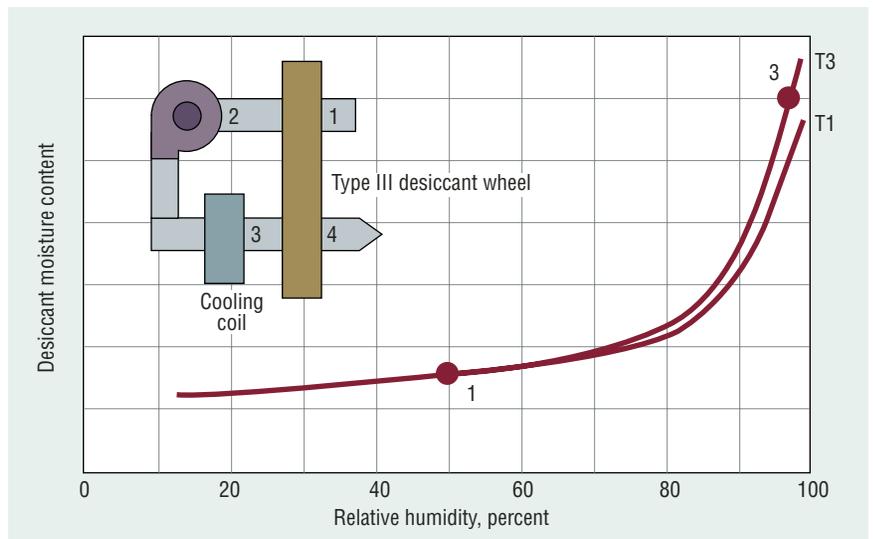


FIGURE 8. Type-III-isotherm desiccant wheel in series with a cooling coil.

is less able to hold water vapor, which is desorbed into the air. This water vapor is added upstream of the cooling coil, where it can be removed from the air stream by condensation. This process occurs without the addition of any heat to regenerate the desiccant.

A series desiccant wheel allows conventional vapor-compression equipment to achieve lower supply-air dew points and improves system efficiency. For the example in Figure 9, 45°F chilled water can be used, rather than the 40°F

to 42°F chilled water a cool-and-reheat system would require. This higher water temperature allows for a 10- to 15-percent increase in chiller efficiency. Additionally, the load on the cooling coil is reduced by 20 percent, and reheat is minimized.

The biggest improvement in efficiency occurs in systems requiring a supply-air dew point below 45°F because the Type III series desiccant wheel can eliminate the need to use heat-activated desiccants. The wheel can remove up to 24 grains

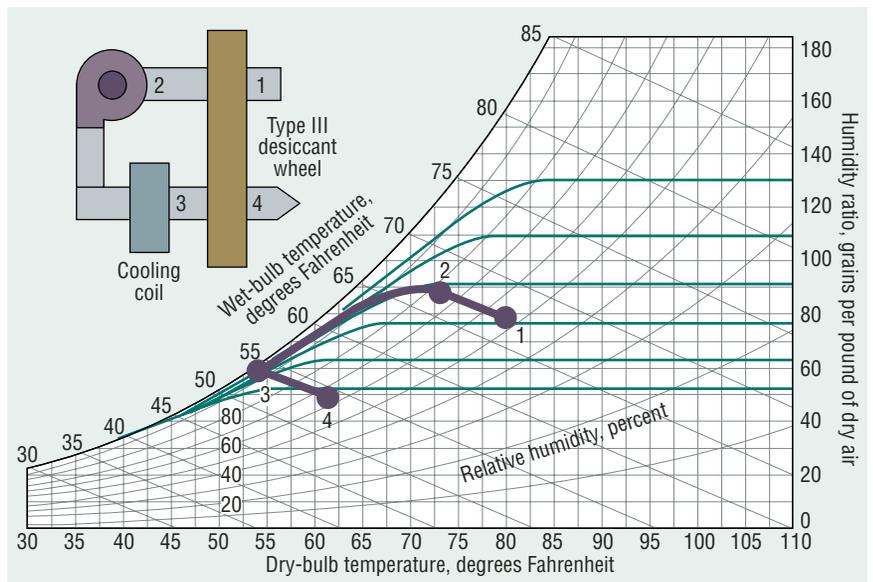


FIGURE 9. Psychrometrics of Type III series desiccant wheel.

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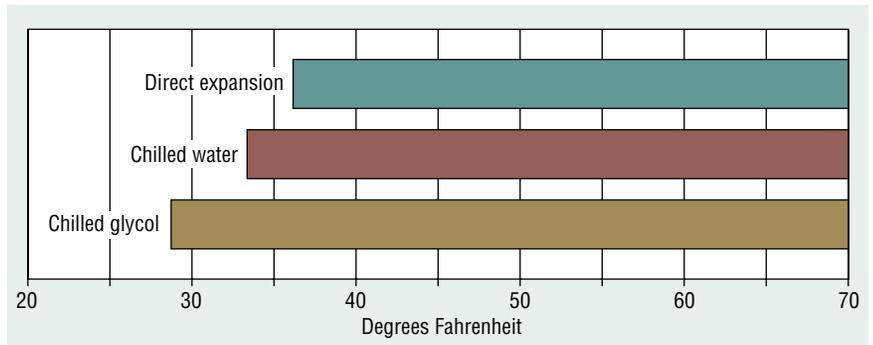


FIGURE 10. Extended supply-air-dew-point capability resulting from the addition of a series desiccant wheel.

per pound of water vapor from supply air, making possible the use of conventional vapor-compression equipment in a system that can dehumidify air down to a 25°F dew point (Figure 10). Below a supply-air dew point of 25°F, heat-activated desiccants or specialty refrigeration equipment with defrost cycles may be required.

COOL VS. HOT SUPPLY AIR

The temperature difference between the process and regeneration sides of a Type III series desiccant wheel is smaller than that of a traditional heat-activated-desiccant wheel. Additionally, the thermal mass of a Type III series desiccant wheel is less than that of a heat-activated-desiccant wheel. There-

fore, heat carryover is minimized. Also, the heat associated with adsorption can be less.⁴ The worst-case scenario is that the total heat added does not vary much from the heat of condensation: 1,100 Btu per pound-mass of water vapor removed. This is a 30- to 40-percent reduction compared with heat-activated desiccants, which transfer water vapor at 1,400 Btu per pound-mass to 1,800 Btu per pound-mass. Typically, the need for post-cooling is avoided.

Figure 11 is an infrared image of a Type-III-isotherm desiccant wheel.

EXAMPLE OF SERIES-DESICCANT-WHEEL INSTALLATION

The design criterion for each operating room in a hospital served by an air

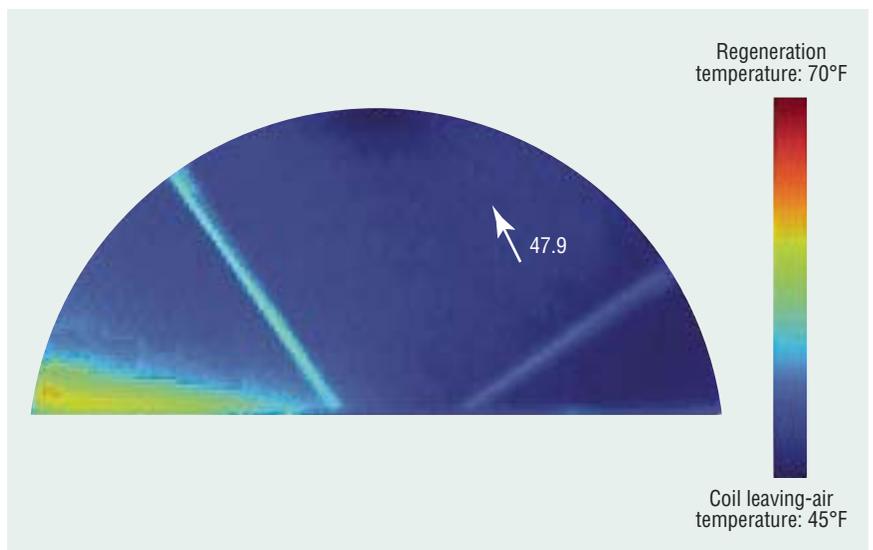


FIGURE 11. Infrared image of the cool adsorption process on a Type III desiccant wheel.

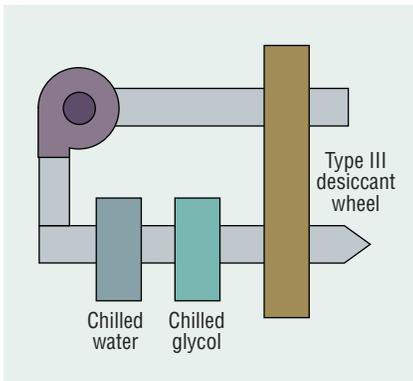


FIGURE 12. Series desiccant wheel with two cooling coils.

handler equipped with a Type III series desiccant wheel (Figure 12) is 62°F dry bulb and 50-percent RH. The hospital wishes to achieve even drier conditions, if possible.⁵ This will require a supply-air dew point of 42°F or less.

The existing water-cooled chiller supplies 45°F water to a cooling coil, lowering the supply-air dew point to 55°F. Next, a stand-alone air-cooled chiller supplies a 38°F glycol solution to a second cooling coil, lowering the dew point to 42°F. Finally, the series desiccant wheel lowers the supply-air dew point to 35°F. The wheel is regenerated by a mixture of outdoor air (35 percent) and recirculated return air (65 percent). Because the return air

is so dry, no heat is added to regenerate the desiccant.

Figure 13 shows supply-air conditions for an air handler in a hospital surgical wing. For more than 50 percent of the hours, the series-desiccant-wheel system delivered a supply-air dew point lower than what would have been achieved with chillers alone. During periods of full occupancy, the supply-air dew point was 36°F to 39°F, maintaining space conditions of 62°F dry bulb and 50-percent RH. No regeneration heat was added. This is important because a supply-air dry-bulb temperature of 50°F to 55°F is required to maintain the desired space temperature.

HEAT REMOVAL FOR DEDICATED OUTDOOR-AIR SYSTEMS

For applications with an insufficient amount of recirculating air, additional heat may be necessary to regenerate a desiccant. The most challenging scenario is a dedicated outdoor-air system in a climate with high outdoor humidity. Regeneration preheat may be needed to lower the RH of air entering the upstream side of a series desiccant wheel. However, a Type III series desiccant wheel will require only 5°F to 20°F of preheat to regenerate, compared with a 100°F temperature rise with a heat-

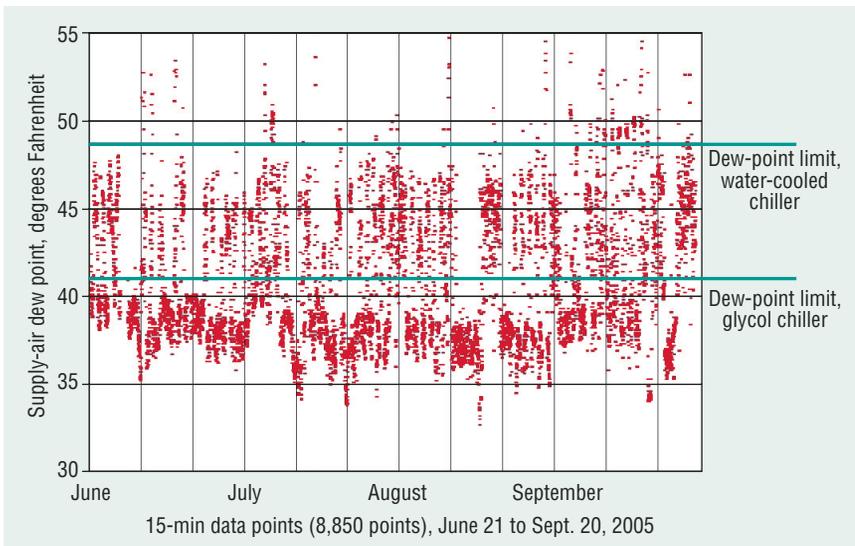
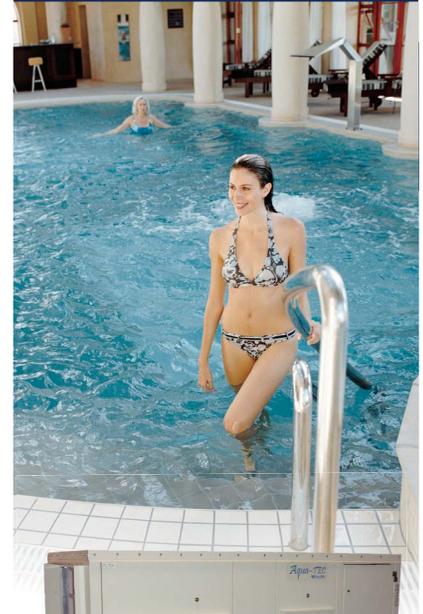


FIGURE 13. Monitored supply-air dew point, no regeneration heat added.

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DOAS air-handler configuration	DOAS-component arrangement	DOAS performance			Chiller data ¹		
		Water removed by coil, lb per hr	Water removed by desiccant, lb per hr	Supply temperature, °F	Chilled-water temperature, °F	Entering condenser temperature, °F	Chiller efficiency, kw per ton
Series-path Type-III-isotherm desiccant	Figure 14	635	0	59	45	85	0.557
Cold coil with recovered reheat	Figure 2	635	–	59	39	85	0.710
Dual-path Type-III-isotherm desiccant	Figure 6	555	80	60	45	85	0.557
Dual-path Type-III-isotherm desiccant, recovered preheat	Figure 6	555	80	60	45	110	0.814
Cold coil with new reheat	Figure 2	635	–	59	39	85	0.710
Dual-path heat-activated desiccant with pre-cooling coil	Figure 6, 65-percent bypass	513	122	80	45	85	0.557
Dual-path heat-activated desiccant with post-cooling coil	Figure 4	0	635	60	45	85	0.557

¹Centrifugal HCFC-123 chiller. Power does not include terminal-equipment fans, pumps, or cooling-tower energy.

²Heating-equipment efficiencies at 85 percent.

³For dual-path wheel units, 5-percent cross leakage to the regeneration side is assumed.

⁴Operating cost at design conditions based on 2005-2006 national commercial-retail-rate average reported by the U.S. Department of Energy: \$0.085 per kilowatt-hour and \$1.12 per therm.

⁵Without a post-cooling coil, space-cooling equipment will need to be 33-percent larger than the other systems to account for the added heat.

TABLE 1. Comparison of dedicated-outdoor-air-system (DOAS) configurations.

activated-desiccant wheel.

One method of avoiding the use of new energy for regeneration preheat is to pre-condition outdoor air with an energy-recovery wheel (or enthalpy wheel). When outdoor RH is high, the energy-recovery wheel will transfer water vapor to the exhaust-air stream, lowering the RH of air entering the upstream side of the Type III desiccant wheel.

The dual-path arrangement (Figure 4) also could be used to prevent the heating of outdoor air that is going to be cooled. The series wheel, however, still could be more efficient (Table 1).

In many systems, a dedicated outdoor-air unit may require colder chilled water (for dehumidifying outdoor air) than local HVAC units require for space sensible cooling. Using a Type III series desiccant wheel avoids the need to lower chilled-water temperature to meet the dehumidification load on a dedicated outdoor-air unit. Using a series arrangement and pre-cooling (pre-dehumidifying) outdoor air allows

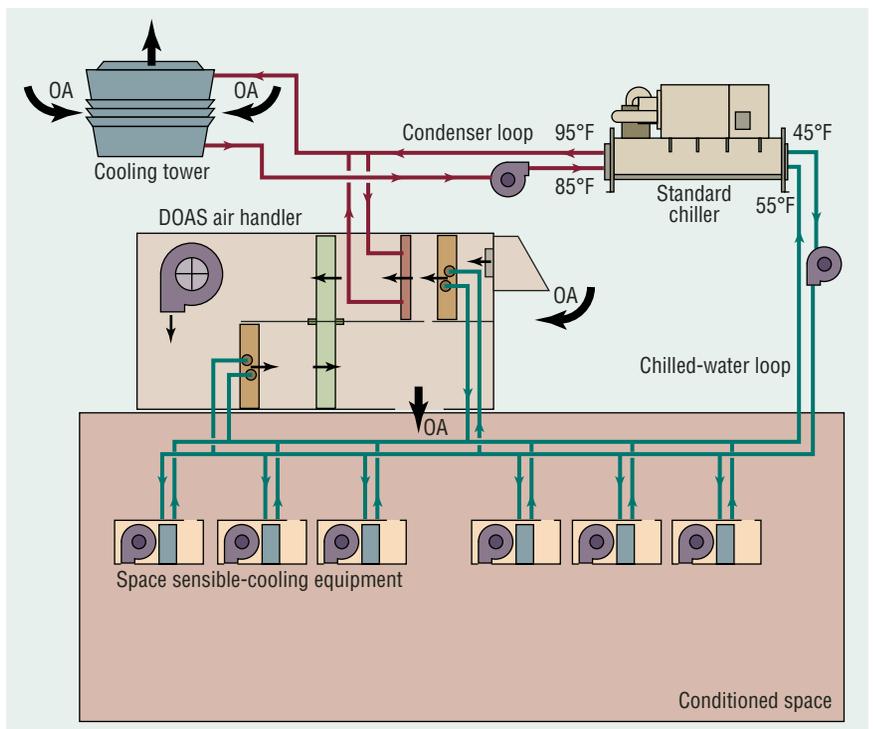


FIGURE 14. A dedicated outdoor-air system with a series desiccant wheel using condenser water for regeneration.

a system to use even lower-temperature heat for regeneration (Figure 14). In

Heating data ²			Cooling data				Energy at design hour			
Heat source	Regeneration or reheat temperature, °F	New regeneration heat or reheat input, Btuh	DOAS coil temperature, °F	DOAS tons cooling ³	Space sensible-cooling tons	Total cooling tons	Chiller kwh	DOAS fan kwh	Natural gas, therms	Operating cost at design hour ⁴
Condenser water	74		54/53	100	60	160	89	6.3	–	\$8.11
Condenser water	59		46	90	60	150	107	4.7	–	\$9.45
180°F hot water	105	238,700	53	82	62	144	80	9.2	2.39	\$10.27
Condenser water	105		53	82	62	144	117	9.2	–	\$10.75
180°F hot water	59	146,000	46	90	60	150	107	4.7	1.46	\$11.09
Gas heat	284	357,650	55	74	80 ⁽⁵⁾	154	86	6.2	3.58	\$11.82
Gas heat	284	1,587,500	60	95	62	157	87	8.3	15.88	\$25.92

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water-cooled systems, this may avoid the need to raise condenser-water temperature to obtain sufficient heat for regeneration. In this configuration, condenser-water temperatures as low as 85°F typically are adequate for regeneration.

An alternative approach is to use a dual-path arrangement (Figure 4 or Figure 6) with a Type III desiccant wheel. Such a configuration, however, requires a separate regeneration-air stream to be heated to 100°F to 110°F. Unless a chiller already is in a heat-recovery application, elevating condenser-water temperature to provide such temperatures will have a negative impact on chiller efficiency (Figure 15). Therefore, a dual-path arrangement may require new energy for regeneration heat. Still, this represents an efficiency improvement, compared with a traditional heat-activated-desiccant wheel.

Table 1 compares various dedicated-

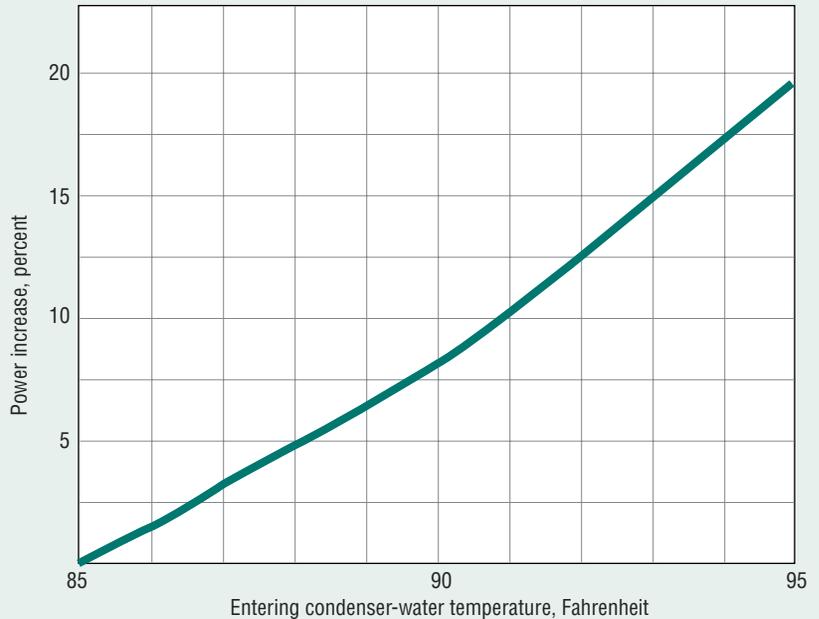


FIGURE 15. Impact on chiller efficiency of raising condenser-water temperature.⁶

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outdoor-air-system units using the configurations discussed in this article and the following design conditions:

- Ventilation airflow: 10,000 cfm.
 - Conditioned outdoor air to be delivered: 60°F dry bulb and 45 grains per pound-mass, 45.4°F dew point.
 - Outdoor conditions: 83°F dry bulb and 77°F dew point, 144 grains per pound.
 - Space sensible-cooling load: 75 tons.
- The series-path-Type-III-isotherm-desiccant configuration requires the most cooling capacity, but costs the least to

water, Type III desiccant, and heat-regenerated desiccant—for commercial buildings. Each system has its advantages based on the dew-point temperature required and the cooling and heat sources available. When comparing technologies, the overall effect they have on a system must be considered.

Type III desiccant wheels can be used in series with a cooling coil to improve the dehumidification capability of conventional cooling equipment. This often can be accomplished without the need to add heat for regeneration, which avoids

Engineers should evaluate all dehumidification-system types for commercial buildings.

operate. The dual-path-heat-activated-desiccant-with-pre-cooling-coil configuration, on the other hand, requires the least cooling capacity, but is one of the costlier alternatives to operate. Using an active desiccant to perform all dehumidification is inefficient, compared with using a coil to remove all or most water vapor. As Table 1 shows, the more dehumidification performed by a desiccant, the more heat that will be needed and added and, thus, the more energy that will be used.

Although, for a complete comparison, an annual analysis should be considered, several important rules can be learned by looking at this one point. When designing a dehumidification system, consider:

- The complete system effect. Looking at only required cooling capacity often leads to false conclusions.
- Cooling efficiency, along with the net cost of both utilities.
- That when building exhaust is available, exhaust-air energy recovery can be used with any of the configurations in Table 1 to pre-treat outside air.

CONCLUSION

Engineers should evaluate all dehumidification-system types—DX, chilled

the energy-related penalties associated with traditional heat-activated-desiccant wheels.

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