Dehumidification Enhancements

for 100-Percent-Outside-Air AHUs

Simplifying the decision-making process

Author's note: While this series of articles is written from a 100-percent-outside-air standpoint, most of the dehumidification technologies are applicable to systems that handle both outside air and return air.

By DONALD P. GATLEY, PE

Over the last 10 years, there has been a trend to providing outside air to buildings using dedicated outside-air units. This approach decouples the ventilation-air load from the air-conditioning systems that handle the space-cooling and heating loads. In many cases, this results in less-complicated units and simpler controls for the units that handle the space-cooling and heating loads. Dedicated outside-air units result in better control of building pressurization, a more consistent introduction of required outside-air quantities, and better management of water-vapor removal (often called "latent cooling") from the outside air, which is especially important at part-load operating conditions, which occur during 97-plus percent of all operating hours.

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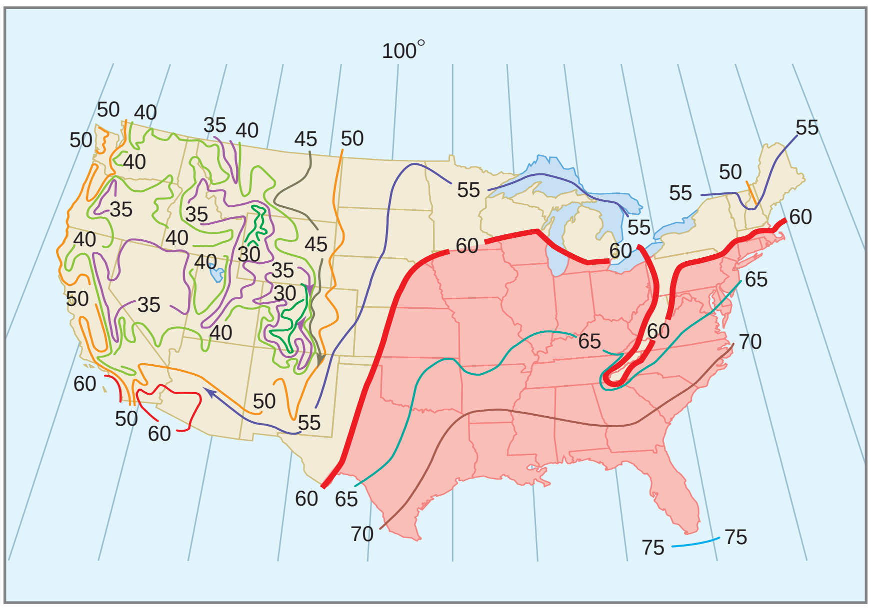

FIGURE 1. Mean dew-point temperature isolines for August (1946 to 1965).
Source: Climatic Atlas of the United States.

Some designers extend this concept and decouple both the outside-air load and the space latent load from space air-conditioning systems. In dedicated outside-air units, this is achieved by lowering the supply-air dew-point temperature far enough below the desired space dew-point temperature to offset space latent gains. This eliminates or minimizes condensed moisture at most if not all space air-conditioning units and in the case of a chilled-water system, may allow the sequential use of chilled water with the lowest-temperature chilled water supplied to the dedicated outside-air unit and then in series to the space air-conditioning units. Eliminating or minimizing condensed moisture at a space-conditioning unit's cooling coils provides indoor-air-quality benefits, reduces maintenance costs, and may eliminate the requirement for double-wall casings in the unit.

This is not a new concept. Willis H. Carrier introduced the under-window air-water induction air-conditioning sys-

	Dew point (F)	Vapor pressure (in. Hg)	Vapor pressure (mm Hg)	Humidity ratio (Gr/lb_{da})	Humidity ratio (lbs_{wv}/lb_{da})	Absolute humidity ($Grains/ft^3$)	Absolute humidity (lb/ft^3)
Run 1	212	29.9	760.0	NA	NA	NA	NA
Run 2	110	2.6	66.0	416	0.05945	26.45	0.003778
Run 3	104	2.2	55.4	344	0.04912	22.42	0.003203
Run 4	100	1.9	49.1	302	0.04320	20.04	0.002863
Run 5	90	1.4	36.1	218	0.03120	15.01	0.002144
Run 6	86	1.3	31.8	191	0.02733	13.32	0.001903
Run 7	80	1.0	26.2	156	0.02234	11.10	0.001585
Run 8	70	0.7	18.8	111	0.01584	8.10	0.001156
Run 9	68	0.7	17.5	103	0.01476	7.59	0.001084
Run 10	60	0.5	13.3	78	0.01109	5.82	0.000832
Run 11	50	0.4	9.2	54	0.00766	4.12	0.000589
Run 12	40	0.2	6.3	37	0.00522	2.88	0.000411
Run 13	32	0.2	4.6	27	0.00379	2.13	0.000304
Run 14	30	0.2	4.2	24	0.00346	1.95	0.000278
Run 15	20	0.1	2.6	15	0.00215	1.24	0.000177
Run 16	10	0.1	1.6	9	0.00132	0.78	0.000111
Run 17	0	0.0	1.0	6	0.00079	0.48	0.000068
Run 18	-10	0.0	0.6	3	0.00046	0.28	0.000041
Run 19	-20	0.0	0.3	2	0.00026	0.17	0.000024
Run 20	-30	0.0	0.2	1	0.00015	0.09	0.000014
Run 21	-40	0.0	0.1	1	0.00008	0.05	0.000007

TABLE 1. Equivalent ways of expressing the quantity of water vapor in air, based on sea-level altitude and a barometric pressure of 29.92-in. HG.

tem in 1937. This system supplied dehumidified air from a central air-handling unit (AHU) to venturi-induction nozzles in induction units under each window. The slightly lower pressure in the induction-nozzle area of the unit caused room air to flow over a cooling-heating coil in the unit. The coil was supplied with chilled water for cooling and hot water for heating. In the most popular version of this system, the central AHU supplied 100-percent outdoor air at a dew-point temperature several degrees below the design room temperature. Thus, the central AHU handled all of the outside-air cooling and dehumidification load and also the room-dehumidification (latent) load. Another popular system used the same central dehumidification and outside-air concept, but used under-window fan-coil units.

In the current era of fast-track design, downsizing, and "commodity-priced" fee structures, it should come as no surprise that designers sometimes do not have the time or inclination to evaluate technology options for dedicated outside-air units. That many manufacturers have their own non-generic marketing name for each of their equipment configurations or enhancements does not help the designer—nor do case studies in industry publications that compare an optimized system using the manufacturer's technology with non-optimized "conventional" equipment (e.g., an enhanced and optimized favored technology with 1950s conventional air conditioning and electric reheat).

This article simplifies the decision-making process and presents most of the technology choices.

FIRST THINGS FIRST

A simple analysis using space and outside-air dew-point temperatures is used to determine when the dehumidification of outside air is required. Dew-point temperature is a good indicator of the amount of water vapor in the air.

What is the desired space dew-point temperature? Most air conditioning is designed for space conditions of 75-F dry-bulb temperature and 50-percent relative humidity (RH). This combination of temperature and RH results in a dew-point temperature of 55 F.

That said, in what part of the United States is the dehumidification of outside air required? Basically, most of the country east of 100-degree west longitude. Figure 1 is a map from the *Climatic Atlas of the United States*[1] showing mean dew-point temperature isolines for August. Note that for most of the eastern half of the U.S., the average August dew-point temperature is above 60 F.

The first rule in maintaining a comfortable, non-mold-producing environment in buildings is always introduce outside air into the space at a dew-point temperature lower than the room design dew-point temperature (about 55 F). A corollary is never allow outside air having dew-point temperatures above the room dew-point temperature to infiltrate the building. This is achieved by maintaining each space in the building at a slight positive pressure relative to the outside whenever the outside dew-point temperature is above 55 F. This is not rocket science; however, these principles are violated in many designs and subsequent construction and operations-and-maintenance practices.

Contributing to designers' difficulty understanding systems and enhancements is the number of ways that meteorologists and dehumidification and air-conditioning practitioners quantify water vapor associated with air. Table 1 lists equivalent ways of expressing the quantity of water vapor in air.

DESIGN SCOPE AND TECHNOLOGY OPTIONS

Given the objective of introducing outside air into a building at a dew-point temperature of 55 F or lower, what are the options? The sidebar on Page 32 lists applicable technologies.

Note that many optimum-choice dehumidification systems are a combination of two or more of the technologies listed in the sidebar. Ordinary reheat with a cooling and dehumidifying coil, followed by hot-water, steam, or electric reheat coils, rarely is the optimum choice. The optimum usually incorporates an exhaust-air enthalpy-recovery wheel and/or a recuperative enhancement. Similarly, if a life-cycle evaluation points to a heat-powered active desiccant dehumidifier, the optimum choice often is this technology with an exhaust-air enthalpy-recovery wheel and/or recuperative heat-transfer devices.

Each building application and evaluation involves the outside climate, the desired supply-air temperature (which may be reset), operating hours, energy and maintenance costs, and installed first cost. The first step is determining what the author calls the "target technology," which is likely to be the final choice after detailed life-cycle costing. The second step is optimizing and performing true value engineering on the selected target technology with vendor and contractor input. Follow this with a package of preliminary design sketches and specifications sufficient to obtain rough price estimates. Send the package along with climate, operating-hour, energy-cost, financial, and space-constraint data to responsible vendors representing other technologies and challenge them to beat the target technology.

The choice of a target technology in this article is subject to the following assumptions:

■ The outside-air design dew-point temperature is 73 F or higher.

■ The supply-air dew-point design temperature is in the 40-to-55-F range. Many 100-percent outside-air systems decouple not only the ventilation load from the space-conditioning units, but the space latent (water-vapor removal) load as well. Thus, they supply air at a dew-point-temperature differential below the 55-F dew-point design temperature to offset the water vapor generated in spaces by occupants, activities, and processes.

■ The outside-air unit utilizes a draw-through fan with a rise in total fan and motor temperature of approximately 3 F.

Caveat Emptor: Let the Buyer Beware

With a few exceptions, the equipment covered in this series of articles is produced by relatively small manufacturers. Although most of these manufacturers have good intentions, because of price competition and pressure to get new or redesigned products to the marketplace quickly, the units that they supply to the field may not be as installation-friendly or as maintenance-free as their advertising suggests.

Owners should insist that equipment for their projects have a two-year history of successful field operation. They are advised to visit installations and talk to the people who have operated and maintained the equipment. If some components have a problematic track record, the owner should consider specifications that require the manufacturer to provide five- or 10-year full-field-maintenance and parts-replacement coverage for at least the problematic components.

For all equipment and systems, evaluate the level of commissioning, instrumentation, BAS monitoring and trending, check-test-start-up services, training, spare parts, and maintenance required. Decide if some of these services should be provided by the equipment manufacturer, the manufacturer's local representative or local service organization, an independent expert, or the design engineer.

Be especially wary of equipment substitutions and pseudo value engineering (cost cutting). Consider substitutions, but evaluate them with the same degree of thoroughness as the equipment originally evaluated and specified. In some cases, the substitute equipment may not have the same proven track record as the base product. In cases such as these, it may be prudent to require long-term maintenance contracts, extended warranties, or bonds that provide for the complete replacement of equipment that is not suitable for the intended application.

IS ENTHALPY RECOVERY FROM EXHAUST AIR ECONOMICAL?

The collection of exhaust air equal to 100 percent of the outside air is rarely possible because some of the outside air is used for building pressurization and some exhausts are too remote or contaminated for collection. From a life-cycle standpoint, it usually is cost-effective to collect as much building exhaust and relief air as possible—even if this is as low as 20 percent of the outside-air quantity. The reason for this is clearly demonstrated in Figure 2, which is based on exhaust air at 90 percent of outside air. The enthalpy exchange device reduces the size of the cooling units, significantly lowers operating cost, and, in many cases, allows industry-standard packaged roof-top air conditioners designed for 15- to 20-percent outside air to work effectively with greater quantities of outside air. The enthalpy exchange device not only saves cooling energy, but also saves heating and humidifying energy during the winter.

Most applications, including schools, hotels, high-rise dwellings, and office buildings, can life-cycle justify the collection of 90 percent of outside air. The exceptions are spaces such as kitchens and laundries, where the exhaust air is loaded with particulates and is too warm or humid for beneficial use during the summer.

SYSTEMS USING EXHAUST-AIR ENTHALPY RECOVERY

The choice of additional options depends on the desired supply-air dry-bulb temperature at full- and part-load operating conditions. If the supply-air dry-bulb temperature at full load and part load needs to be as cold as possible, then reheat devices are not required.

If the desired supply-air dry-bulb temperature is more than 3 F higher than the leaving-air dew-point temperature, then some form of reheat is required to raise the supply-air temperature from the coil and fan leaving-air temperature to the desired full- or part-load supply-air temperature. This reheat can be supplied by a sensible heat exchanger with one component located on the entering exhaust-air side of the enthalpy-recov-

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ery device and the other component located downstream from the primary cooling and dehumidifying coil. This heat exchanger also will increase the overall wintertime heat-recovery effectiveness of the unit.

If the exhaust-air sensible heat-recovery device cannot transfer sufficient heat to raise the supply-air dry-bulb temperature to the desired temperature, then one of the five recuperative run-around heat-exchange devices (to be discussed in Part 2) should be considered in combination with or independent of exhaust-air sensible recovery.

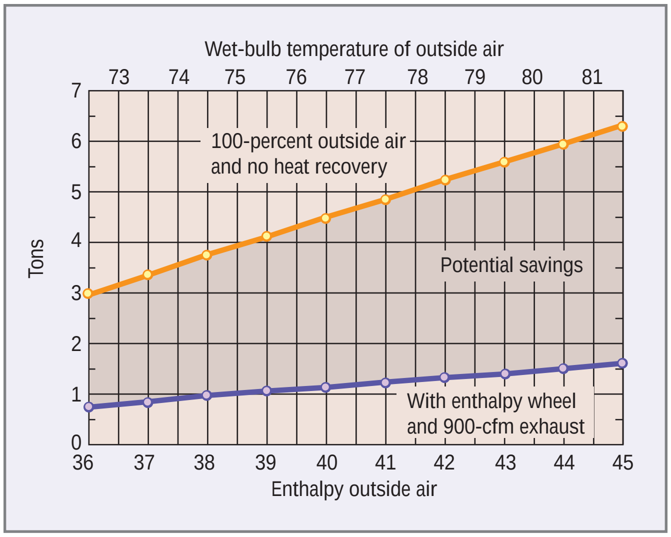

FIGURE 2. Outside-air cooling load with and without enthalpy-wheel recovery based on exhaust air at 90 percent of outside air.

If the desired supply-air dry-bulb temperature at full load and part load is greater than 90 F, then consider both a heat-powered active desiccant dehumidifier and a cooling and dehumidifying coil with one of the five recuperative enhancements possibly supplemented with hot-gas reheat or condenser-water reheat.

If the required dew-point temperature is in the 35-to-40-F range, which is below the lowest assumed value of 40 F, and the supply-air dry-bulb temperature is above 45 F, then consider replacing the sensible heat exchanger mentioned above with a passive desiccant-dehumidification wheel. This may permit the cooling and dehumidifying coil to maintain a frost-free condition with a nearly saturated leaving-air temperature of approximately 45 F. The passive desiccant wheel then adiabatically takes on additional water vapor from the saturated air, lowering the dew-point temperature to approximately 35 F. The latent cooling is offset by sensible heating, resulting in a dry-bulb temperature increase.

SYSTEMS INVOLVING ONLY THE OUTSIDE-AIR STREAM

If the desired supply-air dry-bulb temperature at full load and part load needs to be as cold as possible without any reheat, then reheat devices are not required.

If the supply-air dry-bulb temperature is more than 3 F higher than the leaving-air dew-point temperature, then some form of reheat is required to raise the supply-air temperature to the desired full- or part-load supply-air temperature. This reheat normally can be supplied by one of the five recuperative run-around heat-exchange devices that will be discussed in Part 2 of this article. If this heat is insufficient, then hot-gas reheat or condenser-water reheat should be provided to supplement the recuperative run-around heat exchange.

Heat-powered active desiccant dehumidifiers are less popular for 100-percent-outside-air applications without exhaust-air enthalpy heat exchangers because few can depress the outside-air dew-point temperature from 78-80 F to 55 F without precooling or the use of two active desiccant dehumidifiers in series.

Dehumidification Technologies

Ordered from most to least likely to be considered, the following are dehumidification technologies for providing 100-percent outside air in humid climates:

■ Exhaust-air/outside-air enthalpy exchangers (passive desiccant with equal sensible- and latent-recovery effectiveness).

■ Cooling and dehumidifying coil with sensible recovery from exhaust air (with or without evaporative precooling of exhaust) or recuperative dehumidification enhancements:

—Coil-loop run-around precooling and reheating coils (water or glycol)

—Heat-pipe run-around precooling and reheating coils

—Air-to-air flat-plate heat exchangers for precooling and reheat

—Rotary-wheel heat exchangers for precooling and reheat

—Refrigerant liquid subcooling/air-reheating coil (recuperative enhancement only)

■ Exhaust-air/supply-air rotary-wheel latent exchangers (passive desiccant with latent-recovery effectiveness greater than sensible-recovery effectiveness). This is a new technology that can be considered for use with cooling and dehumidifying coils at dew-point temperatures less than 45 F.

■ Heat-powered active desiccant dehumidifiers with a recuperative heat-transfer device between the first segment of the process air and the regeneration air upstream of the primary heat source.

■ Cooling and dehumidifying coil with refrigerant hot-gas or condenser-water reheat.

■ Cooling and dehumidifying coil with “new-energy reheat” using hot water, steam, or electric reheat coils.

CONCLUSION

Never let outside air pass through an air conditioner or enter a building at dew-point temperatures higher than 55 F.

REFERENCE

1) *Climatic Atlas of the United States*, National Oceanic and Atmospheric Administration, 1968 (reprinted 1977). ■

Part 2 of this three-part series will discuss technology options, including enthalpy exchangers, sensible recuperative enhancements, and hot-gas and condenser-water reheat.