Designing Dedicated Outdoor Air Systems

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Integration of dedicated outdoor air (OA) systems with parallel terminal systems is not new.[1] However, many engineers have dismissed the concept of a separate dedicated OA system on the basis of cost and space limitations. Recently, some industry leaders[2,3] have employed designs using the dedicated OA approach. The author's paper in *ASHRAE Transactions*[4] illustrates the complexity of the problem and raises serious questions about the ability of all-air VAV systems to actually meet the ventilation rate procedure of ANSI/ASHRAE Standard 62-1999, *Ventilation for Acceptable Indoor Air Quality*.

A separate dedicated OA ventilation system may be the only reliable method of meeting Standard 62-1999. It is also the simplest method. The separate dedicated OA concept can be expanded, if the air is properly conditioned, to completely meet the space latent loads, thus decoupling the space latent and sensible loads. Therefore, a new design/integration paradigm was developed with the following two basic tenets:

1. Employ a separate dedicated outdoor air system to deliver the proper ventilation air to each individual space and to remove all of the outdoor air and space latent cooling loads.
2. Remove the space sensible cooling loads, primarily independent of the dedicated outdoor air system, with a *parallel* mechanical system.

OA Supply Temperature and Dew-Point Temperature

Decoupling the space sensible and latent loads requires that the ventilation air be supplied with a sufficiently low humidity ratio to remove the moisture generated within the space, as well as that carried in by infiltration. If infiltration is low as required by ANSI/ASHRAE/IESNA Standard 90.1-1999, *Energy Standard for Buildings Except Low-Rise Residential Buildings,* the primary source of moisture often is the occupant.

Because the dedicated OA system is designed to deliver the required ventilation air to each occupant (15 to 20 scfm/person [7 to 9 L/s per person]), the change in humidity ratio of the ventilation air is easily computed for any given activity level. Typically, this requires a supply air dew-point temperature of about 45°F (7°C) to maintain a space dew-point temperature around 52°F (11°C) (support for this lower than customary space dew-point temperature follows).

Dehumidification and Cooling

Dehumidification can be accomplished using either active desiccants (solid or liquid) or cooling coils. As a general rule, cooling coils are a better choice when the required dew-point temperature is above 40°F (4°C). Active desiccants are a better choice when the dew-point temperature is below 40°F (4°C). Therefore, only cooling-based dehumidification is explored here.

Properly selected and maintained deep chilled-water cooling coils, capable of producing the approximately 45°F (7°C) dew-point temperature, appear to be the best choice. The coil capacity can be modulated to match the transient load conditions. Also, off peak air conditioning (ice or water thermal storage) can be used.

Reheat

In the event that the required supply air dew-point temperature is lower than the desired supply air dry-bulb temperature, reheat is necessary.

Winter Humidification

Winter humidification often is overlooked in current design practice. However, since it is recommended that the winter space relative humidity (RH) be maintained above 40% for optimal thermal comfort, and for minimizing microbial-related health risks,[5] humidification is important.

The significant OA flow rates required by Standard 62-1999 can lead to a very low wintertime indoor air relative humidity in many locations when humidification is not used.

Equipment Configurations

The three configurations presented in *Figure 1* are discussed in this article. In Configuration 1, free reheat is achieved with the runaround heat recovery coils surrounding the cooling coil. A heating coil is available to temper the supply air when there is insufficient heat in the incoming outdoor air to complete the job, and for winter conditions. A humidifier is

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Configuration	Design Cooling Coil Load (Tons)	Peak Humidification Load (kBtu/h)	Peak Heating Load (kBtu/h)	Annual Cooling Coil Energy Use (Ton-hours)	Annual RH Energy Use (kBtu)	Annual Heating Energy Use (kBtu)	Annual Humidification Energy (kBtu)
1	85	238	465	103,000	14,800	110,000	155,000
2	43	0	108	83,000	14,800	7,000	0
3	43	0	0	89,000	0	0	0
Conv. VAV	85	238	0	92,000	0	0	155,000

*Table 1: Load and energy analysis summary.**

used to provide winter humidification.

In Configuration 2, an enthalpy wheel is used to precondition the OA ahead of the cooling coil. The enthalpy wheel cools and dehumidifies the outdoor air in the summer, lowering the load on the cooling coil. In the winter, the enthalpy wheel can be used to heat and humidify the outdoor air, eliminating the need for the humidifier. Preheating is required to prevent frost formation on the wheel. The runaround heat recovery coils and heating coil perform essentially the same duty as in Configuration 1.

Configuration 3 is similar to Configuration 2, but with the runaround coils replaced with a sensible heat recovery wheel for reheat. The sensible wheel completely eliminates the need for the heating coil found in Configuration 2.[6]

Simulated Performance for the Three Configurations

Mathematical models and control algorithms were written for each of the configurations.[6] Atlanta's typical meteorological year (TMY) data was used to determine the peak design component loads and the annual component energy consumption. The simulation used 3,744 hours per year occupancy and 10,000 scfm (4719 L/s) of OA. The results are summarized in *Table 1*.

Based upon the results of the simulations presented in *Table 1,* the best configuration for dedicated outdoor air systems intended to decouple the space latent and sensible loads is Configuration 3. From this point on, Configuration 3 shall be referred to as dedicated outdoor air system (DOAS).

Operation of the DOAS

The operation of the DOAS is best understood with the help of a psychrometric chart. The psychrometric chart in *Figure 2* (with Atlanta weather data plotted) shows the four regions: A, B, C, and D where the OA may fall. The white lines are based upon the following conditions:

- State 3, 45°F (7°C) and saturated;
- State 4, 55°F (13°C) dry-bulb temperature and 45°F (7°C) dew-point temperature;
- State 5, 80°F (27°C) dry-bulb temperature and 55°F (13°C) dew-point temperature; and
- State 6, 70°F (21°C) dry-bulb temperature and 55°F (13°C) dew-point temperature.

A horizontal line represents 45°F (7°C) dew-point temperature, the supply air dew-point temperature. If the outside air conditions fall above that line, the air must be cooled and dehumidified to State 3 and then reheated to State 4 with the sensible wheel. The sensible cooling of the relief air from State 5 to State 6 is a result of energy extraction from the return air. An identical rate of heat is added to the supply air leaving the deep cooling coil (CC) at State 3, reheating it to State 4.

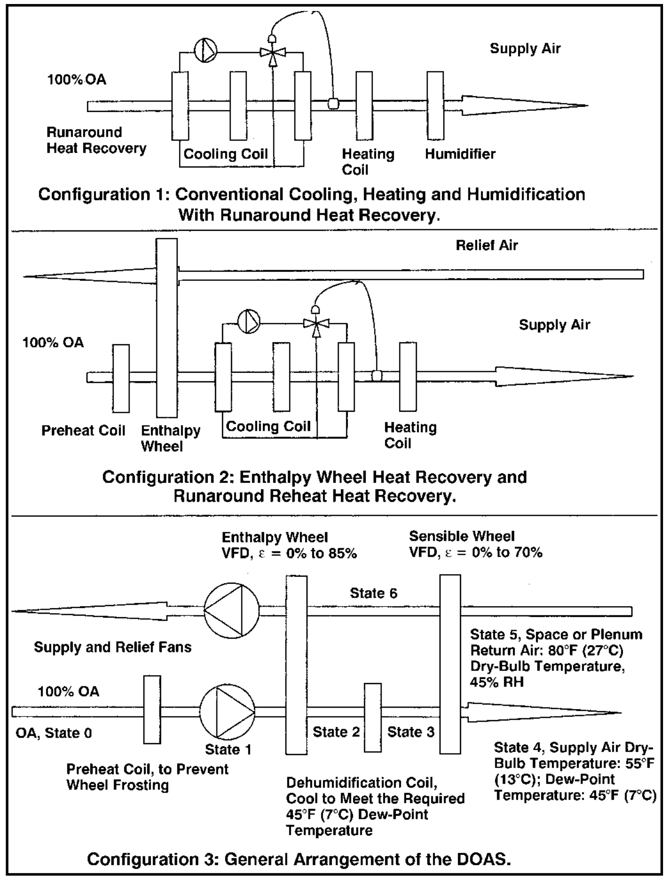

Figure 1: Dedicated outdoor air configurations analyzed.

A line of constant enthalpy passing through State 6 separates the area above the 45°F (7°C) dew-point temperature line into Regions A and B. The boundary between Regions A and B (h_6) separates OA conditions, where dehumidification is required, into the two regions. In Region A, full use of the enthalpy wheel dramatically reduces the cooling coil load. In Region B, any use of the enthalpy wheel increases the cooling coil load. Therefore, the enthalpy wheel must be off.

Another boundary is formed by the extension of a line through the return condition State 5 and the supply State 4. The line, which first appears at State 4 and proceeds to a humidity ratio of 0 gr/lb (0 g), divides the area below the 45°F (7°C) dew-point temperature line into Regions C and D. The boundary between Regions C and D separates the OA conditions, where humidification is required, into two regions. In region C, sensible cooling is required. In Region D, no sensible cooling is required. The equipment operating status is presented in *Table 2.*

* (tons × 3.517 = kW); (Btu/h × 0.2931 = W)

Region	Enthalpy Wheel Control	Cooling Coil Control	Sensible Wheel Control
A	100% speed for Max. Effectiveness	Modulate to Hold 45°F (7°C) LAT	Modulate to Hold 55°F (13°C) SAT
B	Off! Must Not Modulate	Modulate to Hold 45°F (7°C) LAT	Modulate to Hold55°F (13°C) SAT
C	Modulate to Required Dew-Point Temperature	Modulate to Hold 55°F (13°C) LAT	Will Modulate Off
D	Modulate to Required Dew-Point Temperature	Will Modulate Off	Modulate to Hold 55°F (13°C) SAT

Table 2: Control status of the DOAS equipment.

Selecting Supply Air Conditions for the DOAS

Given that most decisions are based on first-cost considerations, and then on energy costs, the following hypotheses must be tested when selecting the supply air conditions:

• Hypothesis 1: low supply air temperatures will reduce the parallel equipment load, and hence their first costs and energy transport costs. The supply air temperature could be as low as the required supply air dew-point temperature and as high as neutral.

• Hypothesis 2: lower supply air dew-point temperature will result in lower space dew-point temperature, allowing the parallel terminal equipment to operate at lower temperatures, further reducing the first costs and energy transport costs.

• Hypothesis 3: supply air temperatures of 55°F (13°C) will result in no more terminal reheat, and frequently less than is currently required with conventional VAV systems using box minimums as dictated by Standard 62-1999.

These hypotheses have been tested[7] and found to be true. The conclusion from that work is "select the supply air dew-point temperature low enough to maintain a summer space RH no greater than 40%, or a supply air dew-point temperature around 44°F (7°C). Likewise, the supply air dry-bulb temperature should be at or below 55°F (13°C)."

Meeting the Space Cooling Loads

Many options are available to meet the sensible cooling load not met by the 55°F (13°C) or colder ventilation air from the DOAS equipment. One option is ceiling radiant cooling panels.[8]

Some of the advantages of ceiling radiant cooling panels are:

• First cost (with experienced contractors) is about 15% less than installing a conventional air system;

• Long-term savings are dramatic, (i.e., approximately 20% to 30%, as a result of reduced fan power);

• The panels provide reduced operation and maintenance costs (minimal moving parts and no filters);

• Testing and balancing during commissioning before occupancy is simpler and less expensive to perform;

• Comfort levels can be better than those of other conditioning systems because radiant loads are treated directly and air motion in the space is at normal ventilation levels;

• Supply air quantities usually do not exceed those required for *ventilation and dehumidification*;

• A 100% outdoor air system may be installed with smaller penalties, in terms of refrigeration load, because of reduced outdoor air quantities (multiple spaces equation 6.1 of Standard 62-1999 does not apply to this situation);

• Wet surface cooling coils are eliminated from the occupied space, reducing the potential for septic contamination;

• The panel system can use the automatic sprinkler system piping (see NFPA *Standard* 13, Chapter 3, Section 3.6);

• A compact design;

• Vertical shaft space area/volume savings;

• A quick response to load dynamics; and

• Zoning is easily achieved, and can accommodate changing office scapes.

Ceiling cooling panels remove heat from the space by radiation and convection. For normal room and panel operating temperatures, the panel can remove 30 Btu/h·ft^2 (95 W/m^2) (about 15 Btu/h·ft^2 [47 W/m^2] by radiation and about 15Btu/h·ft^2 [47 W/m^2] by convection). This rate of heat removal can be placed in perspective with a few approximations. If one assumes that a typical VAV system delivering 55°F (13°C) air will supply 1 cfm/ft^2, such a system has the capability of removing 21.6 Btu/h·ft^2 (68 W/m^2) of sensible heat. If it also is assumed that the ventilation air supply is about 0.2 cfm/ft^2 and supplied at 55°F (13°C), the sensible load remaining is 80% of the 21.6 Btu/hr·ft^2 (68 W/m^2), or slightly more than 17 Btu/hr·ft^2 (54 W/m^2). Under these circumstances, the panel area to ceiling area ratio is 17/30, or less than 60% of the ceiling requires cooling panels. The percentage of ceiling used for cooling panels drops to about 50% when the ventilation supply air temperature drops to 45°F (7°C). A potential schematic of the thermal transport system-fire suppression transport system integration with the DOAS cooling coil and the chiller is shown and discussed in detail in 2001 *ASHRAE Transactions*.[9]

Conclusions and Recommendations

The paradigm shift in the design of building comfort control systems is in its early stages. The requirements of the new paradigm include:

• Separating the OA system from the space-conditioning systems to ensure proper ventilation in all occupied spaces,

• Conditioning the OA to handle all of the space latent load and as much of the space sensible load as is economically feasible without excessive reheat,

• Maximizing the cost-effective use of energy recovery equipment,

• Integrating the fire suppression and energy transport systems, and

• Using ceiling radiant sensible cooling panels for occupant thermal control where appropriate.

To facilitate transition the to the new paradigm, the following near-term goals must be achieved.

1. First, the paradigm must be implemented into buildings located in each of the four major quadrants of the United States.

2. Second, a professional development program must be created and deployed around the country.

3. Third, the U.S. ceiling radiant cooling panel industry must be developed/gear up to meet the potential $5 billion/year to $50 billion/year business opportunity.

For more information about dedicated outdoor air systems, see www.doas.psu.edu.

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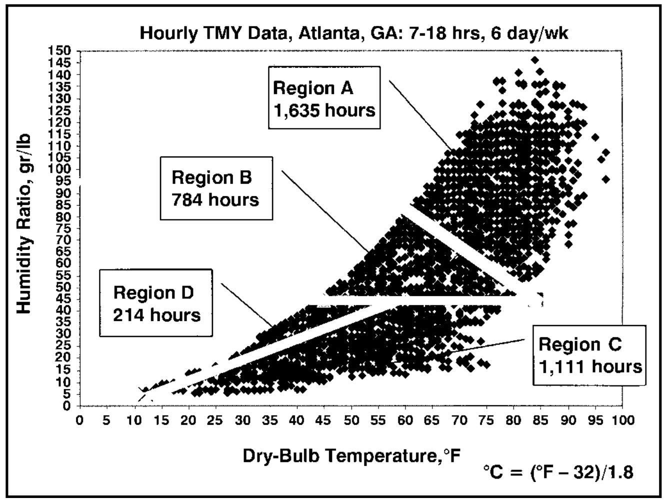

***Figure 2:** Atlanta weather data on the psychrometric chart, 3744 hours.*

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