Standard 62.1-2004

Designing Dual-Path, Multiple-Zone Systems

By Dennis Stanke, Member ASHRAE

ANSI/ASHRAE Standard 62.1-2004, *Ventilation for Acceptable Indoor Air Quality*,[1] prescribes new minimum breathing zone ventilation rates and a new calculation procedure to find the minimum intake airflow for different ventilation systems. Previous articles discussed the new ventilation design requirements for single-zone, 100% outdoor air,[2] changeover-bypass VAV,[3] and single-path VAV[4] systems.

The detailed, step-by-step examples in these earlier articles proved to be a two-edged sword. Many readers were thankful for the calculation details. Others were daunted by the number of steps and apparently concluded that the calculations are too complex for ventilation system designers. However, in this author's opinion, Standard 62.1-2004 spells out more clearly what must be calculated and neither lengthens nor complicates the procedure substantially, compared to previous versions (since 1989).

Although some readers may find it off-putting, this article again includes design details to aid learning—this time, for an important set of ventilation systems wherein one or more zones receive ventilation air via two separate paths. Readers who persevere will find that Standard 62.1 offers designers the opportunity to account for secondary ventilation and thereby design energy-efficient systems that are less costly to install and operate.

Dual-Path, Multiple-Zone Systems

Many HVAC systems are configured as dual-path, multiple-zone, recirculating ventilation systems, which Appendix A in Standard 62.1-2004 describes as "systems that provide all or part of their ventilation by recirculating air from other zones without directly mixing it with outdoor air."

Dual-path systems (*Figure 1*) include a *primary ventilation path* (which supplies a mixture of first-pass outdoor air and centrally recirculated air) and a *secondary ventilation path* (which supplies only recirculated air). Secondary ventilation may be provided by either central recir-

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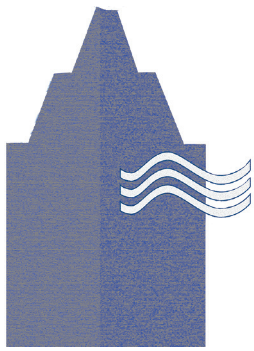
culation of return air from all zones, or local recirculation of return air from one, several, or all zones.

Dual-fan, dual-duct systems are dual-path ventilation systems with central secondary recirculation (*Figure 1*). One central fan supplies primary ventilation, and another central fan supplies secondary ventilation.

Series fan-powered (SFP) VAV systems, on the other hand, are dual-path ventilation systems with local secondary recirculation (*Figure 1*). A central fan supplies primary ventilation and local fans in the VAV boxes supply secondary ventilation.

Although any of these systems may be applied in various building types, the example in this article focuses on the design details for a series fan-powered VAV system applied in an office building. (Systems with central secondary recirculation are often easier to design, but systems with local secondary recirculation are more common and allow us to more completely illustrate the required calculations.)

Dual-path ventilation systems offer a unique benefit. Ventilation air is delivered not only in the primary airstream from the central system but also in the local secondary recirculation air from local air sources, such as the return plenum. As the example will show, secondary recirculation greatly improves system ventilation efficiency, which reduces outdoor air intake flow compared to single-path ventilation systems.



Proper accounting for secondary recirculation in compliance with Standard 62.1-2004 requires that the designer use the dual-path system equation in Appendix A. Of course, the use of this equation (and the higher system ventilation efficiency that results) as an alternative compliance approach is entirely voluntary.

Dual-path systems designed using simpler approaches, such as the Table 6-3 defaults or the single-path system equation in Appendix A (similar to the multiple-space system equation in Standard 62 since 1989), also comply. While these approaches entail simpler calculations, they do not result in the higher system efficiency and lower intake airflow that result from proper accounting of secondary recirculation.

The series fan-powered VAV system (*Figure 2*) in our example includes a central air-handling unit (with a modulating outdoor air damper), a variable-volume primary air fan, series fan-powered VAV boxes (with hot water reheat in perimeter zones), and a central relief fan to control building pressure. Zone temperature setpoint is maintained by adjusting the fraction of primary airflow to the VAV box. Plenum airflow increases as primary airflow decreases (and vice versa), so each series fan provides constant-volume, variable-temperature discharge airflow to its zone.

Zone Calculations

Our example office building (*Figure 3*) includes eight HVAC zones, each with a thermostat controlling one or more VAV boxes. Each VAV zone in this case is also a separate ventilation zone, which Section 3 defines as "one occupied space or several occupied spaces with similar occupancy category, occupant density, zone air-distribution effectiveness, and zone primary airflow per unit area."

As shown in earlier articles, design calculations usually begin with zone ventilation requirements and proceed to system intake airflow requirements. Following the prescribed steps for zone ventilation calculations in Section 6.2.2, we first find *each zone's outdoor airflow* (V_{oz}) (*Table 1*):

1. Use Equation 6-1 to find the minimum required breathing zone outdoor airflow ($V_{bz} = R_p \times P_z + R_a \times A_z$).
2. Look up zone air-distribution effectiveness (E_z) in Table 6-2 based on the zone air-distribution configuration.
3. Use Equation 6-2 to find the minimum required zone outdoor airflow ($V_{oz} = V_{bz} / E_z$) for each zone.

These calculations (detailed in a previous article[4]) are straightforward, once the zones are identified and a design population level (see inset) is established for each zone.

Series Fan-Powered System Calculations

Similar to previous versions, Standard 62.1-2004 acknowledges that multiple-zone recirculating ventilation systems deliver excess outdoor airflow to many zones, but that recirculation recovers some of that excess outdoor air. When the primary airstream contains sufficient outdoor air to properly ventilate the critical zone,* the same primary air overventilates all other zones to some degree. When recirculated, the unused outdoor air from overventilated zones reduces the required intake airflow. However, unused outdoor air that leaves the system (in relief air, for instance) without diluting contaminants reduces system ventilation efficiency.**

The standard defines two design approaches to find and correct for *system ventilation efficiency* (E_v). The default approach uses prescribed values for E_v (listed in Table 6-3 but not shown here), which depend upon the required fraction of outdoor air in the primary air supplied to the critical zone. The calculated approach determines E_v for the system using equations (in Appendix A). A previous article[4] applied each approach to the design of a multiple-zone, single-path ventilation system.

Either approach may be used to establish E_v for dual-path systems. However, the default efficiency values in Table 6-3 are based only on primary recirculation airflow to each zone.

* The critical zone, in terms of system ventilation, requires the highest fraction of outdoor air in the primary airstream; that is, it results in the lowest zone ventilation efficiency (E_{vz}).

** To avoid multiple-zone system inefficiency, some designers advocate dedicated/100% outdoor air systems, which typically deliver constant volume outdoor airflow directly to each ventilation zone. While this approach simplifies ventilation calculations, the design population must be assumed to be present in every zone. With no correction for system occupant diversity, many zones receive excess outdoor air during normal operation. Total excess outdoor air may be less than in recirculating systems, but outdoor air intake flow actually may be higher in some cases because no opportunity exists to recover and recirculate unused outdoor air.

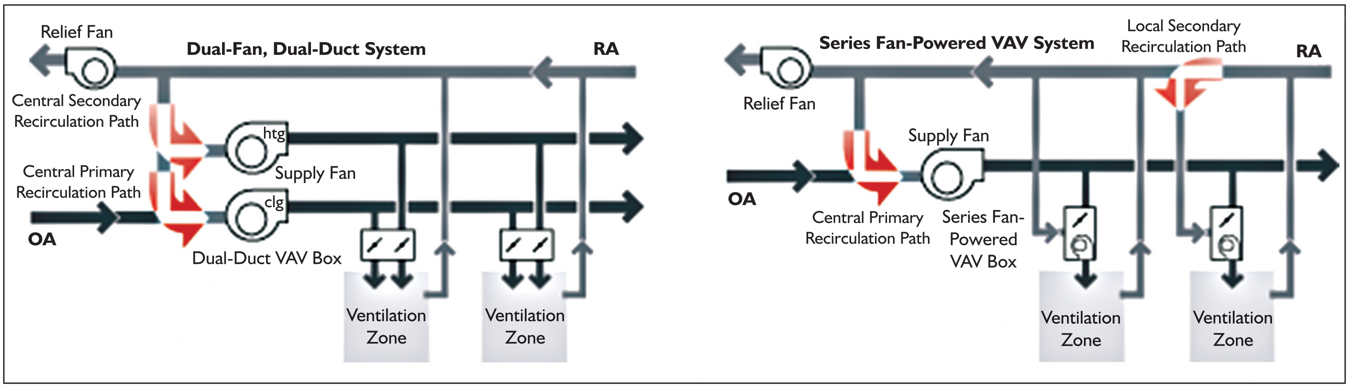


Figure 1: Dual-path, multiple-zone, recirculating ventilation systems.

Since the default approach takes no credit for ventilation contributed by secondary recirculation, it significantly overventilates dual-path systems.

With that in mind, this article covers only the calculated approach presented in Appendix A. It results in proper minimum ventilation for the critical zone and accounts for unused outdoor air leaving the system ($E_v < 1.0$). It also takes credit for unused outdoor air recirculating from all other overventilated zones, both at the central air handler and at each local fan-powered VAV box.

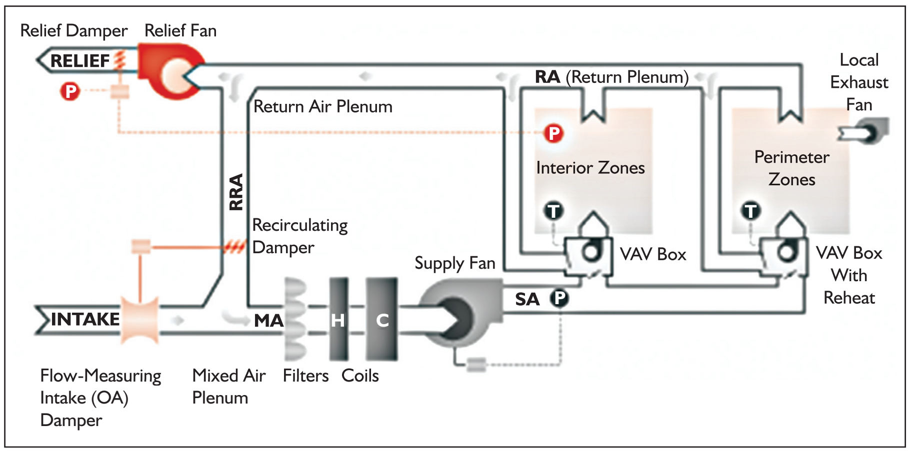


Figure 2: Series fan-powered variable-air-volume system.

<u>*Cooling Design*</u>

From our zone calculations, we now know how much outdoor airflow each zone must receive. The next step is to figure out the minimum system level *outdoor air intake flow* (V_{ot}) that will deliver the required zone outdoor airflow. Initially we will use the cooling design condition (when system heat gain—and, therefore, system primary airflow—is greatest), even though the worst-case ventilation condition (when the required outdoor air intake flow is greatest) may occur at the heating design condition for some systems. In most dual-path ventilation systems, the highest outdoor air intake flow during mechanical cooling is likely to occur when most non-critical zones receive design primary airflow; the critical zone is at (or near) its minimum primary airflow setting; and the primary fan delivers design airflow.

The procedure described here is similar to the one in Section 6.2.5 for multiple-zone recirculating systems (*Table 2*), but we used the definitions and equations from Appendix A to calculate system ventilation efficiency (E_v) rather than look up default values in Table 6.3. For this example, we used typical values for *zone primary airflow* (V_{pz}) at cooling design conditions and selected series fans that deliver the required primary airflow when the damper in each VAV box is open-to-primary/closed-to-plenum airflow.

(Note: Step 4 and Steps 9 through 12 may be applied to each zone as we did in our example, but experienced designers may choose to apply these steps only to zones deemed to be potentially critical, as described in Appendix A.)

4. For each zone (or selected zones), find the *zone discharge outdoor air fraction* (Z_d), according to the definition in Appendix A ($Z_d = V_{oz}/V_{dz}$), where V_{dz} is the *minimum* expected *zone discharge airflow*.† In most series fan-powered systems, diffuser airflow is the same at all operating conditions. If we only consider diffuser airflow (no transfer airflow), then minimum V_{dz} equals design V_{dz}. Note: For dual-path ventilation systems, the required fraction of outdoor air in the primary supply (Z_p) air delivered to the VAV box is not the same as the fraction needed in air delivered to the zone (Z_d) due to the influence of secondary recirculation. In our example, the south offices require 210 cfm of outdoor airflow ($V_{oz\text{-}clg}$), of which about 11% must be outdoor air ($Z_d = 210/1{,}900 = 0.111$).

5. For the system, solve Equation 6-7 ($D = P_s/\Sigma P_z$) to find *occupant diversity* using the expected peak *system population* (P_s) and design *zone population* for all zones. This step is optional, but it reduces overventilation by accounting for variations in occupancy among all zones. Estimating population (both zone and system) is key to the design process because of

†Although it is commonly considered as only supply diffuser airflow, discharge airflow may include any controlled airstream that discharges into the ventilation zone, such as transfer air from the return plenum or from adjacent zones.

its impact on system ventilation requirements. We estimated a maximum system population of 164 people in our example, so *D = 164/224 = 0.73*.

6. For the system, find the *uncorrected outdoor air intake* flow using Equation 6-6 $\left(V_{ou} = D \times \Sigma(R_p \times P_z) + \Sigma(R_a \times A_z)\right)$. This value represents the rate at which outdoor air (found in both first-pass intake air and unused recirculated air) is used up in the process of diluting indoor contaminants generated within the system. (In fact, a more appropriate name for it may be used outdoor air rate.) Any outdoor air introduced in excess of this value is unused outdoor air that helps provide dilution ventilation if recirculated and reduces system ventilation efficiency if exhausted. Our example system needs at least V_{ou} = 2,800 cfm of outdoor air in the breathing zones for proper dilution.

7. For the system, establish the *system primary airflow* ($V_{ps} = \Sigma V_{pz}$) at the cooling design condition. This equation uses the sum of the zone primary airflows at the condition analyzed, not the sum-of-peak zone primary airflows. At the cooling design condition, many zones need peak primary airflow while others need less than peak. That's because the sun moves and not all zone sensible loads peak simultaneously.

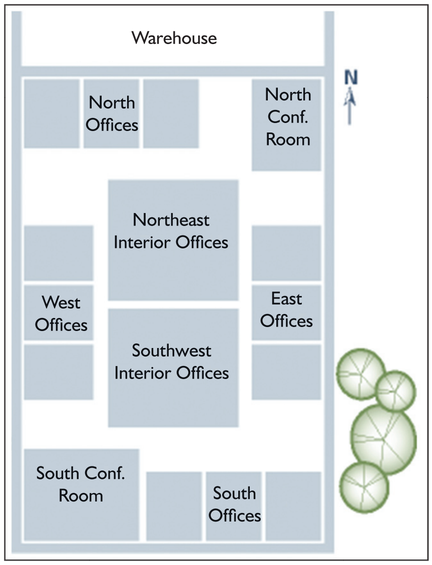


Figure 3: Multiple-zone office building.

It's reasonable to simply use the maximum primary fan airflow for V_{ps} at cooling design. Usually, the central primary VAV fan is selected to deliver block airflow, rather than sum-of-peak airflow, based on a system load diversity factor (LDF = system block load divided by sum-of-peak zone load). In our example office, we used a system load diversity factor of 0.70 based on load calculations, so the central fan delivers V_{ps} = 0.70 × 26,600 = 18,600 cfm at cooling design.

(During cooling operation, most dual-path systems exhibit their lowest system ventilation efficiency when system primary airflow is high and critical zone primary airflow is low. Since all non-critical zones are overventilated to a greater extent at this condition, excess outdoor air recirculates, but a large portion of it is lost in the relief air leaving the system.)

8. For the system, find the *average outdoor air fraction* X_s (or perhaps more accurately, the used outdoor air fraction) according to the definition in Appendix A ($X_s = V_{ou}/V_{ps}$).†† In our example, X_s = 2,800/18,600 = 0.15 at the cooling de-

†† The definition in Appendix A states that X_s represents the fraction of outdoor air intake flow in the system primary airflow. That wording does not strictly match the equation for X_s, which shows the outdoor air usage rate as a fraction of system primary airflow. However, the equation is correct according to the derivation of these equations. Perhaps the definition in the standard can be clarified quickly via the continuous maintenance process so that it more closely matches the defining equation.

Supplemental Recirculating Fan

Conference rooms are notorious for fouling up multiple-zone system ventilation calculations, especially in VAV systems. Why? Conference rooms have high design population density, so the fraction of outdoor air needed in the air supplied to the zone is also high, especially when cooling load is low.

Typically, in single-path systems, the minimum primary airflow must be set to a very high value to limit Z_p $(= V_{oz}/V_{pz})$ and ensure proper ventilation without requiring 100% outdoor air at the intake. In dual-path systems, a high minimum primary setting (which increases $E_p = V_{pz}/V_{dz}$), may not be necessary, since the secondary recirculation path can keep zone ventilation efficiency (E_{vz}) high, even with low primary recirculation to a zone. In either system, designers may want to include intermittent (or continuous) supplemental recirculating fans, which draw air from the return plenum and discharge it into the conference room.

How does this improve system ventilation efficiency? Fan operation adds another local recirculation path to the conference room. The air-conditioning diffusers provide some outdoor air (via primary and perhaps secondary recirculation) while the supplemental fan diffusers add a local recirculation path from the return plenum.

To solve the equations in Appendix A, discharge airflow into the zone (V_{dz}) includes diffuser airflow and any supplemental airflow from the return plenum. Increasing V_{dz} changes several of the parameters, but most importantly, it decreases Z_d (i.e., V_{oz}/V_{dz}) significantly. Zone ventilation efficiency E_{vz} rises. If it's the critical zone, system ventilation efficiency E_v rises too, decreasing outdoor air intake flow, V_{ot}. Supplemental recirculating fans work well and the equations in Appendix A can handle them, but such fans may or may not be allowed in all jurisdictions. Designers should check with code authorities before pursuing this design approach.

(The equations probably don't apply for zones with supplemental return fans, which result in increased transfer air from adjacent zones. This actually introduces a tertiary ventilation path so it's likely to improve ventilation, but it's too difficult to predict the fraction of unused outdoor air transferred from adjacent zones.)

						Cooling			Heating	
Procedural Step					1	2	3	2	3
Variable	R_p	P_z	R_a	A_z	V_{bz}	E_z	V_{oz}	E_z*	V_{oz}
Ventilation Zone	cfm/p	P	cfm/ft^2	ft^2	cfm		cfm		cfm
South Offices	5	18	0.06	2,000	210	1.0	210	0.8	260
West Offices	5	20	0.06	2,000	220	1.0	220	0.8	275
South Conference Room	5	30	0.06	3,000	330	1.0	330	0.8	410
East Offices	5	20	0.06	2,000	220	1.0	220	0.8	275
Southwest Interior Offices	5	50	0.06	10,000	850	1.0	850	1.0	850
Northeast Interior Offices	5	50	0.06	10,000	850	1.0	850	1.0	850
North Offices	5	16	0.06	2,000	200	1.0	200	1.0	200
North Conference Room	5	20	0.06	2,000	220	1.0	220	1.0	220

* For zones with reheat at the VAV box, discharge air temperature changes from cool to warm, so E_z changes from 1.0 to 0.8 at design heating conditions.

Table 1: Zone ventilation calculations.

sign conditions, that is, 15% of the outdoor air in the primary airstream is used to dilute contaminants.

9. For each zone (or selected zones), find the lowest fraction (E_p) of primary air in all air delivered to the zone according to the definition in Appendix A ($E_p = V_{pz}/V_{dz}$). Since proper ventilation airflow must be delivered to all zones at all load conditions, the lowest E_p in a zone is likely to result in the lowest *zone ventilation efficiency* (E_{vz}) in that zone. The lowest *possible* E_p for a zone occurs when zone primary airflow (V_{pz}) is at the minimum primary airflow setting. For our example, we arbitrarily assumed a 25% minimum primary airflow, so $E_p = V_{pz}/V_{dz} = 0.25 \times V_{dz}/V_{dz} = 0.25$ for each zone. (This does not mean that all zones are at 25% primary airflow at the cooling design condition. Rather, it means that the lowest possible E_p value for a zone is 0.25.)

Note: Although dual-path systems often can be ventilated properly with much lower minimum primary airflow settings, we arbitrarily used 25% minimums for our example calculations to allow easy comparison with the ventilation systems considered in earlier articles.

10. For each zone (or selected zones), establish the ventilation quality of locally recirculated return air (secondary air) using the *return air mixing efficiency* (E_r), defined in Appendix A. This efficiency varies from 0.0 to 1.0, depending on the location of the zone secondary air source with respect to the central system return air. *When zone secondary air has the same ventilation quality*—that is, the same fraction of unused outdoor air—*as the central return air*, then E_r = 1.0. (In dual-fan, dual-duct VAV systems, the return air supplied by the heating fan is the same as that recirculated by the main supply fan, so E_r = 1.0.) *When zone secondary air has the same ventilation quality as the zone served*, then E_r = 0.0. (In systems where return air from the zone is ducted to the inlet of the fan-powered VAV box serving the zone, then E_r = 0.0.)

In actual systems with fan-powered boxes, E_r is closer to 1.0 if the VAV box is located near the central return air inlet, and closer to 0.0 if located directly over the return grille of the zone served. In our example, the fan-powered boxes are located centrally, so we assumed E_r = 0.8 for all zones. (The standard does not require a calculation method for E_r, so the values used must be based on designer judgment. However, a current ASHRAE research project [RP-1276] will measure E_r values in a real building with fan-powered VAV boxes in various locations. This data should provide the information needed to help designers establish E_r values in the future.)

11. For each zone (or selected zones), according to the definitions in Appendix A, find:

Design Population

Standard 62.1-2004 allows the designer to use either the highest expected zone population or the average zone population when calculating breathing zone ventilation airflow using Equation 6-1. This design population option introduces design flexibility, but it also introduces a potential source of inconsistency among designers. Peak population is usually available, but if used, it increases zone ventilation requirements. Average population, on the other hand, decreases zone ventilation requirements, but it's not always easy to find. Or, more accurately, it's not always easy to predict the zone population profile.

For some occupancy categories—classrooms or perhaps churches and theaters, for example—zone population profile may be readily predictable based on a time-of-day schedule. However, for other categories—offices, conference rooms, retail areas and perhaps theaters—occupant level fluctuations over the averaging period T (found using Equation 6-9) are not easily established. In these areas, one designer's profile estimate may be considerably different than that of another designer. So, the definition for design population P_z is flexible but can lead to inconsistency among designers. Time will tell if a stricter, less flexible definition is needed.

					From Above										
Procedural Step							**4**	**5–8**	**9**	**10**	**11a**	**11b**	**11c**	**12**	**13–14**
Ventilation Zone	**Box Type**	**V_{pz} (clg design)**	**V_{fan}**	**V_{dz}**	**V_{pz-min}***	**V_{oz-clg}**	**Z_d**		**E_p**	**E_r**	**F_a**	**F_b**	**F_c**	**E_{vz}**	
		cfm	**cfm**	**cfm**	**cfm**										
South Offices	SFP Reheat	1,900	1,900	1,900	475	210	0.111	—	0.25	0.8	0.85	0.25	1	0.914	—
West Offices	SFP Reheat	2,000	2,000	2,000	500	220	0.110	—	0.25	0.8	0.85	0.25	1	0.915	—
South Conference Room	SFP Reheat	3,300	3,300	3,300	825	330	0.100	—	0.25	0.8	0.85	0.25	1	0.927	—
East Offices	SFP Reheat	2,000	2,000	2,000	500	220	0.110	—	0.25	0.8	0.85	0.25	1	0.915	—
Southwest Interior Offices	SFP VAV	7,000	7,000	7,000	1,750	850	0.121	—	0.25	0.8	0.85	0.25	1	0.901	—
Northeast Interior Offices	SFP VAV	7,000	7,000	7,000	1,750	850	0.121	—	0.25	0.8	0.85	0.25	1	0.901	—
North Offices	SFP VAV	1,600	1,600	1,600	400	200	0.125	—	0.25	0.8	0.85	0.25	1	0.897†	—
North Conference Room	SFP VAV	1,800	1,800	1,800	450	220	0.122	—	0.25	0.8	0.85	0.25	1	0.900	—
System…															
ΣV_{pz}			26,600												
(Step 5) $D = P_s/\Sigma P_z$††								0.73							
(Step 6) V_{ou}								2,800							
(Step 7) V_{ps}								18,600							
(Step 8) X_s								0.15							
(Step 13) E_v															0.897
(Step 14) V_{ot}															3,120

* Set at 25% of design primary airflow.
† Ventilation critical zone.
†† P_s (system population) = 164 people, and ΣP_z (sum of zone peak population) = 224 people.

Table 2: System ventilation calculations at cooling design.

a. The fraction of supply air to the zone from locations outside the zone $\left(F_a = E_p + (1 - E_p) \times E_r\right)$. For our example, $F_a = 0.25 + (1 - 0.25) \times 0.8 = 0.85$ for all zones.

b. The fraction of supply air to the zone from the primary airstream ($F_b = E_p$). For our example, $F_b = 0.25$ for all zones.

c. The fraction of outdoor air to the zone from outside the zone $\left(F_c = 1 - (1 - E_z) \times (1 - E_r) \times (1 - E_p)\right)$. For our example, $F_c = 1.0$ for all zones when cooling, because $E_z = 1.0$ for all zones.

12. For each zone (or selected zones), find *zone ventilation effectiveness* using Equation A-2 $\left[E_{vz} = F_a + X_s \times F_b - Z_d \times F_c)/F_a\right]$ for dual-path systems. In our example, $E_{vz} = (0.85 + 0.15 \times 0.25 - 0.111 \times 1.0)/0.85 = 0.914$ for the south offices.

13. For the system, find *system ventilation efficiency* using Equation A-3 (E_v = minimum E_{vz}), the lowest *zone ventilation efficiency* among the zones served by the system. In our example, we identify the north offices as the ventilation critical zone at the cooling design condition. It has the lowest zone ventilation efficiency, so $E_v = 0.897$. (We used three decimal places in our example because our numbers result in only slight differences in E_{vz} values.)

14. Finally, find *outdoor air intake flow* for the system by solving Equation 6-8 ($V_{ot} = V_{ou}/E_v$ in the body of the standard). In our example, $V_{ot} = 2{,}800/0.897 = 3{,}120$ cfm at the design cooling condition. In other words, about 17% of the 18,600 cfm primary airflow needed at the cooling design condition must be first-pass outdoor air.

When compared to the 4,310 cfm of first-pass outdoor air needed for a single-path system applied to the same building,[4] it's clear that dual-path systems can significantly reduce the required outdoor air intake flow.

Heating Design

We repeated the previous calculation steps (*Table 3*) for the heating design conditions, when system heat loss to outdoors is greatest, to see if worst-case ventilation occurs when primary airflow is very low. (We also could use the minimum outdoor air intake flow at the heating design condition to save energy by establishing separate minimum intake flow settings for summer and winter operation.)

Step 4 results in a *minimum discharge outdoor air fraction* (Z_{d-htg}) that's slightly higher than $Z_{d\text{-}clg}$ in the perimeter zones with reheat—0.138 vs. 0.111 in the south offices, for example. (Zone air-distribution effectiveness is somewhat lower when delivering heat from overhead diffusers, which makes $V_{oz\text{-}htg}$ somewhat higher than $V_{oz\text{-}clg}$.)

Since the zone and system populations don't change, Steps 5 and 6 yield the same values for both D and V_{ou}.

Step 7 is a little tricky: What is *system primary airflow* (V_{ps}) at the heating design condition? That depends on the system and the

Equations and Variables from Addendum 62*n*

[6-1] $V_{bz} = R_p P_z + R_a A_z$

[6-2] $V_{oz} = V_{bz}/E_z$

[6-3] $V_{ot} = V_{oz}$ single-zone systems

[6-4] $V_{ot} = \Sigma V_{oz}$ 100% outdoor-air systems

[6-5] $Z_p = V_{oz}/V_{pz}$

[6-6] $V_{ou} = D\Sigma_{allzones} R_p P_z + \Sigma_{allzones} R_a A_z = D\Sigma_{allzones} V_{bzp} + \Sigma_{allzones} V_{bza}$

[6-7] $D = P_s / \Sigma_{allzones} P_z$

[6-8] $V_{ot} = V_{ou}/E_v$ multiple-zone recirculating systems

[6-9a] $T = 3v/V_{bz}$ IP version

[6-9b] $T = 50v/V_{bz}$ SI version

where

$\boldsymbol{A_z}$ is zone floor area, the net occupiable floor area of the zone, ft² (m²)

$\boldsymbol{D}$ is occupant diversity, the ratio of system population to the sum of zone populations

$\boldsymbol{E_v}$ is ventilation efficiency of the system

$\boldsymbol{E_z}$ is air-distribution effectiveness within the zone

$\boldsymbol{P_s}$ is system population, the maximum simultaneous number of occupants in the area served by the ventilation system

$\boldsymbol{P_z}$ is zone population, the largest expected number of people to occupy the ventilation zone during typical usage (See caveats in Addendum 62n–Section 6.2.1.1)

$\boldsymbol{R_a}$ is area outdoor air rate, the required airflow per unit area of the ventilation zone determined from Addendum 62n–Table 6.1, cfm/ft² (L/s·m²)

$\boldsymbol{R_p}$ is people outdoor air rate, the required airflow per person determined from Addendum 62n–Table 6.1, in cfm/person (L/s·person)

$\boldsymbol{T}$ is averaging time period, minutes

$\boldsymbol{v}$ is ventilation-zone volume, ft³ (m³)

$\boldsymbol{V_{bz}}$ is breathing-zone outdoor airflow, the outdoor airflow required in the breathing zone of the occupiable space(s) of the ventilation zone, cfm (L/s)

$\boldsymbol{V_{ot}}$ is outdoor air intake flow, adjusted for occupant diversity and corrected for ventilation efficiency, cfm (L/s)

$\boldsymbol{V_{ou}}$ is the uncorrected outdoor air intake flow, cfm (L/s)

$\boldsymbol{V_{oz}}$ is zone outdoor airflow, the outdoor airflow that must be provided to the zone by the supply-air-distribution system at design conditions, cfm (L/s)

$\boldsymbol{V_{pz}}$ is zone primary airflow, the primary airflow that the air handler delivers to the ventilation zone; includes both outdoor air and recirculated return air

$\boldsymbol{Z_p}$ is zone primary outdoor air fraction, the fraction of outdoor air in the primary airflow delivered to the ventilation zone … for VAV systems, Z_p for design purposes is based on the minimum expected primary airflow, V_{pzm}.

weather. For proper system ventilation, we want to find the highest volume of outdoor air needed. Does this occur when all zones receive minimum primary airflow (an easy condition to check)? Or when perimeter zones receive minimum primary airflow and interior zones receive more than minimum (a more difficult condition to check)? For our example, we first assumed that all zones receive minimum primary airflow (V_{ps} = 6,650 cfm).

Step 8 results in *average outdoor air fraction* (X_s) of 0.42, compared to 0.15 at cooling design. In other words, the used outdoor air fraction increases when primary air decreases.

Steps 9, 10 and 11 result in E_p, E_r, F_a, and F_b values that are identical to those at the cooling design condition. F_c, on the other hand, is lower in the reheat zones because zone air-distribution effectiveness (E_z) is lower when delivering warm air.

Step 12 reveals that the south offices are now the critical zone, rather than the north offices, while Step 13 indicates that system ventilation efficiency (E_v) rises to 0.966. Higher efficiency means that outdoor air intake flow (V_{ot}; Step 14) drops to 2,900 cfm from the 3,120 cfm needed at the cooling design condition making cooling design the worst-case ventilation condition. This result is expected to be typical of dual-path recirculating systems because the system contains less excess outdoor air at reduced primary airflow.

Multiple-Zone Systems

Does the designer always need to calculate ventilation requirements at both heating and cooling design? Not necessarily. Some systems—series-fan powered and dual-fan, dual-duct VAV systems with constant volume discharge, for instance—almost surely need less intake air as primary airflow drops (as it does at part-cooling load) than when primary airflow peaks (as it does at cooling design condition). The same is true for parallel fan-powered and dual-fan, dual-duct VAV systems with VAV discharge airflow, since any zone in the heating mode has two ventilation paths. On the other hand, single-path systems, such as single-duct VAV and constant volume reheat systems, with low zone air-distribution effectiveness during heating, may need higher outdoor air intake flow heating design conditions.

Until we have more experience with these systems, or until more analytical research results tell us the answer, we can't draw absolute conclusions. Our advice to designers is to use an ASHRAE spreadsheet (you will find one at www.ashrae.org) or develop your own spreadsheet. Once you have entered the equations, it's easy to experiment and draw your own conclusions for your ventilation system design.

From Above															
Procedural Step							**4**	**5–8**	**9**	**10**	**11a**	**11b**	**11c**	**12**	**13–14**
Ventilation Zone	**Box Type**	**V_{pz} (htg design)**	**V_{fan}**	**V_{dz}**	**V_{pz-min}***	**V_{oz-htg}**	**Z_d**		**E_p**	**E_r**	**F_a**	**F_b**	**F_c**	**E_{vz}**	
		cfm	**cfm**	**cfm**	**cfm**	**cfm**									
South Offices	SFP Reheat	475	1,900	1,900	475	260	0.138	—	0.25	0.80	0.85	0.25	0.97	0.966†	—
West Offices	SFP Reheat	500	2,000	2,000	500	275	0.138	—	0.25	0.80	0.85	0.25	0.97	0.967	—
South Conference Room	SFP Reheat	825	3,300	3,300	825	410	0.125	—	0.25	0.80	0.85	0.25	0.97	0.981	—
East Offices	SFP Reheat	500	2,000	2,000	500	275	0.138	—	0.25	0.80	0.85	0.25	0.97	0.967	—
Southwest Interior Offices	SFP VAV	1,750	7,000	7,000	1,750	850	0.121	—	0.25	0.80	0.85	0.25	1.00	0.981	—
Northeast Interior Offices	SFP VAV	1,750	7,000	7,000	1,750	850	0.121	—	0.25	0.80	0.85	0.25	1.00	0.981	—
North Offices	SFP VAV	400	1,600	1,600	400	200	0.125	—	0.25	0.80	0.85	0.25	1.00	0.977	—
North Conference Room	SFP VAV	450	1,800	1,800	450	220	0.122	—	0.25	0.80	0.85	0.25	1.00	0.980	—
System…															
ΣV_{pz}			6,650												
(Step 5) D								0.73							
(Step 6) V_{ou}								2,800							
(Step 7) V_{ps}								6,650							
(Step 8) X_s								0.42							
(Step 13) E_v															0.966
(Step 14) V_{ot}															2,900

* Set at 25% of design primary airflow.
† Ventilation critical zone.

Table 3: System ventilation calculations at heating design.

What happens if our simple assumption about primary airflow (all zones at the minimum primary setting) at the heating design condition is too low? To find out, we recalculated system ventilation efficiency for our example with the perimeter zones at minimum primary airflow and the interior zones arbitrarily at 50% of design primary airflow. This is based on the assumption that these zones need more than minimum primary airflow (probably the norm in most systems) even when it's really cold outdoors.

Without going into detail, we found that *system primary airflow* (V_{ps}) rises to 11,000 cfm, *system ventilation efficiency* (E_v) drops to 0.917, and *minimum outdoor air intake flow* (V_{ot}) rises to 3,050 cfm. In this example system, the cooling design condition still requires the highest outdoor airflow. For an actual ventilation system, however, the designer should take care to base the design on worst-case intake airflow. A load analysis (for instance) can help determine the lowest primary airflow needed in each zone at all design conditions.

Other Dual-Path Ventilation Systems

We looked at ventilation system design for a series fan-powered VAV system in detail. What about other dual-path ventilation systems?

Parallel Fan-Powered VAV

The local (secondary) ventilation path in a parallel fan-powered VAV system only functions when the fans in the VAV boxes operate, which is during heating. The local ventilation path and the benefits of secondary recirculation disappear during cooling, when the local parallel fans are off.

If we applied such a system to our example building, we'd find that *zone* ventilation requirements don't change and that worst-case ventilation (highest required outdoor air intake flow) again occurs at the cooling design condition. At the heating design condition, system ventilation efficiency rises, and outdoor air intake flow is much less than that required at cooling design.

Dual-Fan, Dual-Duct VAV

Two central air handlers—one that delivers a cool mixture of first-pass outdoor air and centrally recirculated return air (primary ventilation), and another that delivers warm, centrally recirculated return air (secondary ventilation)—supply air to the dual-duct VAV boxes. The boxes can be controlled to deliver either constant-volume–variable-temperature *or* variable-volume–constant-temperature air to each zone. When configured with constant-volume–variable-temperature VAV boxes, dual-fan, dual-duct VAV systems usually have the highest system ventilation efficiency among all multiple-zone recirculating systems.

Applying this dual-fan, dual duct system to our example, again *zone* ventilation requirements do not change. Worst-case ventilation may occur at either the cooling design or heating design condition, depending on the zone minimum primary airflow settings.

The key difference between dual-fan, dual duct systems with central secondary recirculation and series fan-powered systems with local secondary recirculation is that E_r (the fraction of average system return air in the secondary air) always equals 1.0, because return air from all zones mixes at the central secondary recirculation fan. This means that system efficiency can be slightly higher and outdoor air intake flow can be slightly lower than would be the case when using a series fan-powered system. Of course, actual results may differ, depending on the control configuration (constant volume vs. variable volume) and minimum settings of the VAV boxes, but detailed design analysis is left for future articles.

Summary

Standard 62.1-2004 not only clarifies the calculation procedures that were always required by Standard 62, but it also adds an important new tool for designers. Earlier versions required that the designer use the multiple space equation to account for the inherent inefficiency of multiple-zone systems. However, that equation addressed only single-path ventilation systems and, therefore, did not "reward" dual-path systems for ventilating more efficiently. Standard 62.1-2004, through the dual-path equations presented in Appendix A, incorporates a more generalized multiple-space equation, giving ventilation credit where it is due.

This article demonstrated, in detail, how to calculate the worst-case ventilation for one type of dual-path ventilation system. We also noted that some types of dual-path systems ventilate more efficiently than others, and identified the tradeoffs that designers can consider when selecting and designing these systems to comply with Standard 62.1-2004.

References

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