Standard 62-2001 Addendum 62*n*

Single-Path Multiple-Zone System Design

By Dennis Stanke, Member ASHRAE

ANSI/ASHRAE Standard 62-2001, *Ventilation for Acceptable Indoor Air Quality,*[1] as modified by Addendum 62*n*,[2] prescribes new minimum breathing-zone ventilation rates and new calculation procedures to find intake airflow for different ventilation systems. Previous articles[3,4] discussed the design of "simple" ventilation systems (single-zone, 100% outdoor-air, and changeover-bypass VAV) in compliance with Addendum 62*n* requirements. Here, we examine the design of a more complex set of ventilation systems, namely single-path, multiple-zone recirculating systems.

Although the Ventilation Rate Procedure in Standard 62 has required specific calculations (Equation 6-1) for multiple-zone systems since 1989, the calculation procedure was sketchy at best; consequently, it was widely misunderstood and largely ignored by designers. Addendum 62*n* includes a detailed calculation procedure for multiple-zone system design. Use of this procedure is expected to increase consistency among designers and reduce the tendency to design multiple-zone systems—especially VAV systems—that provide inadequate ventilation for some fully occupied zones. Addendum 62*n* also includes operational control options that can be used to modulate ventilation capacity as ventilation load and/or efficiency varies, but these options are left to a future article. The following discussion covers only design calculations.

Many HVAC systems are configured as "single-supply" or single-path, multiple-zone, recirculating ventilation systems. For instance, constant-volume systems with terminal reheat, traditional constant-volume multizone systems, single-duct VAV systems, and single-fan dual-duct VAV systems all provide ventilation from a single source or path. (A single-fan, dual-duct system supplies air to each space using two different ducts, but the air in each duct contains the same fraction of outdoor air, because one fan—a single source—delivers the same air mixture to each duct.) Other systems have multiple ventilation paths, including dual-fan, dual-duct VAV systems and VAV systems with fan-powered or induction terminal units. Single-duct VAV systems with series fan-powered boxes are always dual-path ventilation systems, but those with parallel fan-powered boxes are single-path with the local fan off and dual-path with it on. Although any of these HVAC systems may be used in vari-

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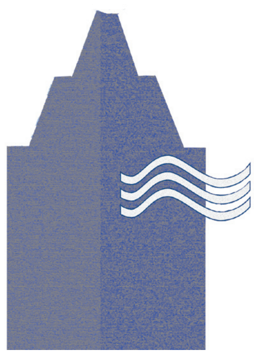
ous building types, we narrow our discussion to a single-duct VAV system, with throttling VAV boxes for interior zones and reheat VAV boxes in perimeter zones, applied in an example office building.

Demonstrating Compliance by Example

Our example system (*Figure 1*) includes a central air handler, with a modulating outdoor-air damper that may be controlled as an economizer; a variable-volume supply fan to deliver primary air; cooling-only, throttling VAV boxes in the interior zones; throttling VAV boxes with electric reheat in the perimeter zones; a central return fan; and a central relief damper for building pressure control. Although we won't discuss system control details here, it's important that we share the same mental "picture" of the VAV system we're designing:

• Intake airflow is sensed and maintained by adjusting the intake damper position. (Often, the return- and outdoor-air dampers are linked such that closing the outdoor-air damper opens the return-air damper proportionately. Alternately, these dampers can be controlled separately to reduce fan energy while maintaining proper intake airflow, but this has no impact on ventilation requirements at design conditions.)



• Primary air temperature is sensed and maintained by sequentially adjusting the heating-coil control valve, economizer dampers, and cooling-coil control valve.

• Duct pressure is sensed and maintained at setpoint by adjusting the primary fan capacity (via fan speed, for instance, or inlet guide vane position).

• Zone temperature is sensed and maintained at the cooling setpoint by adjusting the setpoint for VAV-box primary airflow.

• VAV-box airflow is sensed and maintained at setpoint by adjusting the position of the VAV-box damper.

• For zones that need reheat, zone temperature is sensed and maintained at the heating setpoint by adjusting reheat capacity (electric reheat or a hot water valve) and, thereby, discharge air temperature.

• Return air plenum pressure (at the central air handler) is sensed and maintained by adjusting return fan capacity.

• Building pressure is sensed and maintained between set limits by adjusting the relief (central exhaust) damper position.

Since multiple-zone systems provide the same primary air mixture to all zones, the fraction of outdoor air in the primary airstream must be sufficient to deliver the outdoor airflow needed by the "critical" zone—the zone needing the greatest fraction of outdoor air in its primary airstream. In the past, many designers simply added the zone outdoor airflow requirements and set the intake airflow to match this sum, which resulted in a very low outdoor-air fraction and many underventilated zones.

Some designers went to the other extreme, finding the highest fraction of outdoor air needed by any zone in the system and setting the intake airflow to provide this fraction at all times. This approach considers only first-pass outdoor air, giving no credit for unused recirculated outdoor air, and results in a very high outdoor-air fraction and overventilation in all zones.

Proper design in compliance with Addendum 62*n* calculation procedures strikes a balance between these extremes, appropriately accounting for both critical-zone needs and unused, recirculated outdoor air.

Let's look at an example office building (*Figure 2*). We assumed that thermal comfort can be achieved using only eight VAV thermostats, with each thermostat controlling one or more VAV boxes. We considered each of these "comfort zones" (or "HVAC zones" per ASHRAE Standard 90.1-2001) as a separate "ventilation zone."

According to Addendum 62*n*, a ventilation zone is "one occupied space or several occupied spaces with similar occupancy category, occupant density, zone air-distribution effectiveness, and zone primary airflow per unit area."

Most (but not all) HVAC zones qualify as ventilation zones. The area and population for each zone in this example were selected to help illustrate the calculations rather than to reflect typical zone sizes or population densities.

To comply with Addendum 62*n*, our design calculations begin by finding the ventilation needs at the zone level and conclude by determining the required intake airflow at the system level.

Zone Ventilation Calculations

Following the procedure under "zone calculations" in Section 6.2.1, we found zone outdoor airflow (V_{oz}) for each zone (*Figure 3*):

1. Referring to Addendum 62*n*, Table 6.1 (not shown), look up the prescribed minimum *people outdoor-air rate* (R_p) and the prescribed minimum building *area outdoor-air rate* (R_a). In our example office building, each zone needs 5 cfm/person and 0.06 cfm/ft². Using these values, along with the design *zone population* (P_z) and *zone floor area* (A_z), find the minimum *breathing-zone outdoor airflow* by solving Equation 6-1 ($V_{bz} = R_p \times P_z + R_a \times A_z$). Either peak or average expected occupancy may be

used to establish P_z; we used peak population in all zones. (An earlier article[3] covered population-averaging calculations in detail. See www.ashrae.org for the most current version.)

For our example, the west offices need $V_{bz} = 5 \times 20 + 0.06 \times 2{,}000 = 100 + 120 = 220$ cfm for proper ventilation in the breathing zone.

2. Look up *zone air-distribution effectiveness* (E_z), based on the air-distribution configuration and the default values presented in Addendum 62*n*, Table 6.2 (not shown). All of our example zones use overhead diffusers and ceiling returns, and they all receive 55°F primary air, so $E_z = 1.0$ when cooling. If the thermostat calls for heat in any of the perimeter zones, primary air is reheated and discharged at 95°F; so, $E_z = 0.8$ when heating.

3. Find the minimum *zone outdoor airflow* by solving Equation 6-2 ($V_{oz} = V_{bz} / E_z$) for both cooling and heating operation. For example, the west offices need $V_{oz} = 220/1.0 = 220$ cfm at the diffusers when cooling, and $V_{oz} = 220/0.8 = 275$ cfm when heating.

System Ventilation Calculations

As in Standard 62-1989, -1999, and -2001, Addendum 62*n* recognizes that multiple-zone recirculating systems must overventilate some zones to properly ventilate all zones. It also recognizes that "unused" outdoor air recirculated from overventilated zones reduces the required intake airflow, but that unused outdoor air that leaves the building (by exhaust or exfiltration) increases the required intake airflow. Proper accounting results in a ventilation credit for recirculated outdoor air and a ventilation debit for exhausted outdoor air.

Addendum 62*n* makes this accounting straightforward by requiring a specific calculation procedure to determine the minimum *outdoor-air intake flow* based on the *system ventilation efficiency* inherent in every multiple-zone recirculating system.

Earlier versions of the standard required use of the "multiple-space" equation, $Y = X / (1 + X - Z)$, to find the fraction of intake air needed. This approach resulted in about the same intake airflow as Addendum 62*n* for single-path systems; but without a clear procedural explanation, the equation was widely misunderstood and largely ignored by designers.

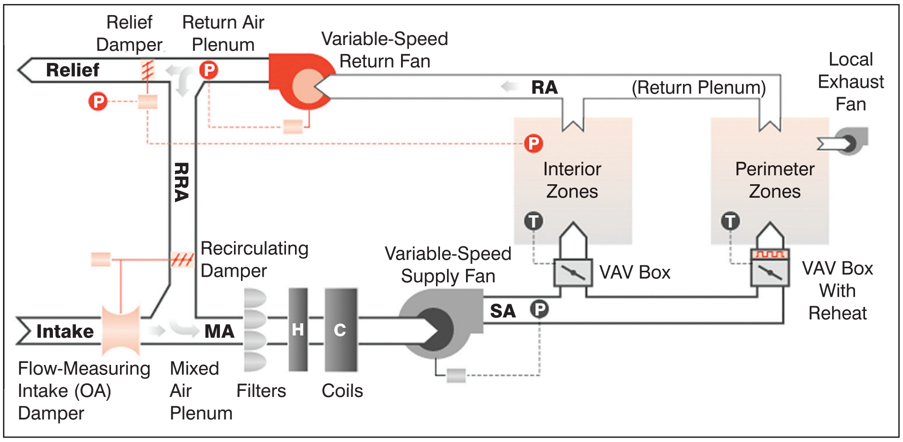


***Figure 1:** Variable air volume reheat system.*

Designs based on the 62*n* procedure result in proper ventilation for the critical zone at worst-case design conditions while allowing credit for "good" outdoor air that recirculates from all other overventilated zones.

From the zone calculations that we completed earlier, we know how much outdoor airflow must reach the diffusers in each zone. Now, let's figure out the minimum required intake airflow for the system at design conditions.

Before we start, we should recognize something that Addendum 62*n* implies but doesn't explain: *The "worst-case" or highest required intake airflow may or may not occur at the design cooling condition* (when system primary airflow is highest). In some cases, it may actually occur at the design heating condition (when zone primary airflow values are very low). With

Averaging Zone Population for Ventilation System Design

In earlier versions of the standard, only "intermittent occupancy" zones (at peak population for three hours or less) could be designed for ventilation at the average population (but not less than one-half of the peak population). Now, any zone may be designed for average population. According to the "short-term conditions" section of Addendum 62*n*, the system must be designed to deliver the required outdoor airflow to each occupied breathing zone.

However, if occupancy or intake airflow varies, the ventilation system design may be based on average conditions over a specific time period rather than on peak conditions. The averaging time T for a given zone is determined according to Equation 6-9 ($T = 3\,v/V_{oz}$) using zone volume and the breathing-zone outdoor airflow that would be needed at peak population. T equals three zone time constants, the time it takes for contaminant concentration to achieve a nearly steady-state value in response to a step change in contaminant source. When applied to population, this averaging approach replaces the population-averaging option for "intermittent occupancy" spaces, found in previous versions of the standard,

Averaging time may be applied to make design adjustments when changing conditions in the zone can be predicted. For instance, if zone population fluctuations are predictable, then the design breathing-zone outdoor airflow may be calculated based on the highest average population over any T-minute period.

this in mind, we'll need to check the required intake airflow at both design cooling *and* design heating because it's ultimately the worst-case outdoor-air intake flow that will establish the required capacities for the heating and cooling coils.

For our example, we tried to use "reasonable" values for *zone primary airflow* (V_{pz}) at design cooling load. We arbitrarily set all minimum primary airflow settings (V_{pz}-min) to 25% of design cooling airflow. We assumed that each reheat box enters reheat mode after its primary airflow decreases to the minimum setting and the zone temperature drops below the heating setpoint. Reheat operation continues until the zone temperature exceeds the heating setpoint.

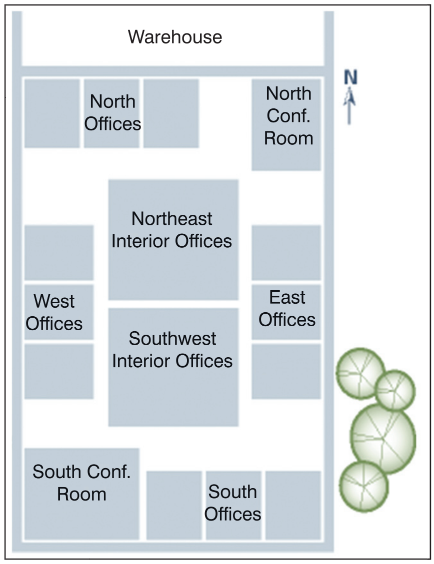


Figure 2: Multiple-zone office building.

Case 1: Ventilation Calculations for "Default" Cooling Design

Building on our earlier zone-level calculations (*Figure 3*), we followed the step-by-step, "multiple-zone recirculating systems" procedure to find the minimum, system-level, *outdoor-air intake flow* (V_{ot}) at the design cooling condition (*Figure 4*):

4. For each zone, find the *zone primary outdoor-air fraction* by solving Equation 6-5 ($Z_p = V_{oz}/V_{pz}$) using the *zone outdoor airflow* (V_{oz}) values for cooling from Step 3 and the minimum primary airflow setting. As an example, at minimum primary airflow, the south offices need $Z_p = 210/475 = 0.44$ when delivering cool air.

5. Addendum 62*n* allows the designer to use a either default value for *system ventilation efficiency* (E_v) using Table 6.3 (not shown) or a calculated value (found using equations in Appendix G). In this case, we used Table 6.3 and the highest *zone primary outdoor-air fraction* among the zones served ("max Z_p" = 0.50 for the north offices) to look up the corresponding default *system ventilation efficiency* (E_v). From that value, we can interpolate to find $E_v = 0.65$.

6. Find *occupant diversity* according to Equation 6-7 ($D = P_s/\Sigma P_z$) by using the expected peak *system population* (P_s) and the sum of design zone populations. For our example, we expect a maximum system population of 164 people, so $D = 164/224 = 0.73$.

7. Find the *uncorrected outdoor-air intake flow* for the system by solving Equation 6-6 $\left(V_{ou} = D \times \Sigma(R_p \times P_z) + \Sigma(R_a \times A_z)\right)$. Without correcting for zone ventilation effectiveness and system ventilation efficiency, we find that the system needs V_{ou} = 2,800 cfm of outdoor air at the breathing zones.

8. Finally, find *outdoor-air intake flow* for the system by solving Equation 6-8 ($V_{ot} = V_{ou}/E_v$). In our example, V_{ot} = 2,800/0.65 = 4,310 cfm at the design cooling condition.

But, is this really the worst-case (highest volume) intake airflow? What happens at design heating conditions?

Case 2: Ventilation Calculations for "Default" Heating Design

Let's find the minimum system-level *outdoor-air intake flow* (V_{ot}) for the design heating condition. The procedure is the same one that was just described for default cooling design in Case 1. It builds on the zone-level calculations that were completed earlier (*Figure 3*), but in this case, we assume that each space receives minimum primary airflow at the design outdoor heating condition (*Figure 5*).*

4. For each zone, find the *zone primary outdoor-air fraction* by solving Equation 6-5 ($Z_p = V_{oz}/V_{pz}$) with the *zone outdoor*

* Some readers might deem this to be a radical assumption because interior zones typically need more than minimum cooling airflow, even on the coldest day. But, it's an assumption that is likely to require a high intake airflow, which is useful for this demonstration.

Design Cooling Condition

For single-path VAV systems, the worst-case condition for ventilation (that is, the lowest system ventilation efficiency and the highest required intake airflow) in the cooling mode usually occurs when the VAV primary airflow for the system is at its highest value. Since almost all VAV systems exhibit load diversity (all zones don't require peak cooling airflow simultaneously), the critical zone can be assumed to be delivering minimum primary airflow with the central fan at cooling-design or "block" primary airflow. In some cases, worst-case ventilation in the cooling mode may actually occur at a central fan airflow that's slightly lower than block airflow. If a system doesn't have much load diversity (all interior zones, for example)—and if the critical zone requires a lot of primary airflow—then the central fan may or may not be at block airflow when the critical zone is at minimum primary airflow.

How can you find out the system primary airflow at the worst-case ventilation condition? Simply assume that primary airflow at the fan is the sum of all noncritical-zone peak airflow values plus the minimum primary airflow for the critical zone. At this condition, the difference between X_s and Z_p will be greatest, so system ventilation efficiency will be at its lowest value and outdoor-air intake flow will be at its highest values—the worst-case condition. (Operationally, this worst-case condition may not actually occur, since it assumes that the critical zone requires minimum primary airflow even when fully occupied; this might be the case for some perimeter zones, for example, during cold weather.)

Procedural Step						Cooling 1	Cooling 2	Cooling 3	Heating 2	Heating 3
Variable		R_p	P_z	R_a	A_z	V_{bz}	E_z*	V_{oz}	E_z**	V_{oz}
Ventilation Zone	Box Type	cfm/p	p	cfm/ft²	ft²	cfm		cfm		cfm
South Offices	Reheat	5	18	0.06	2,000	210	1.0	210	0.8	260
West Offices	Reheat	5	20	0.06	2,000	220	1.0	220	0.8	275
South Conference Room	Reheat	5	30	0.06	3,000	330	1.0	330	0.8	410
East Offices	Reheat	5	20	0.06	2,000	220	1.0	220	0.8	275
Southwest Interior Offices	VAV	5	50	0.06	10,000	850	1.0	850	1.0	850
Northeast Interior Offices	VAV	5	50	0.06	10,000	850	1.0	850	1.0	850
North Offices	VAV	5	16	0.06	2,000	200	1.0	200	1.0	200
North Conference Room	VAV	5	20	0.06	2,000	220	1.0	220	1.0	220

* For zones with a throttling VAV box, discharge air is usually cool whenever the zone is occupied (morning warmup usually occurs before occupancy).
** For zones with terminal reheat, discharge air temperature can be either cool or warm when the zone is occupied, so E_z drops from 1.0 when cooling to 0.8 when heating.

Figure 3: Zone ventilation calculations.

airflow (V_{oz}) values for heating from Step 3 and the minimum primary airflow setting. At minimum primary airflow, the south office needs $Z_p = 260/475 = 0.55$ when delivering warm air.

5. Using Table 6.3 (not shown) and the highest *zone primary outdoor-air fraction* among the zones served ("max Z_p" = 0.55 for the south, west, and east offices) to look up the corresponding default *system ventilation efficiency* (E_v), we find that $E_v = 0.60$.

6. Find *occupant diversity* according to Equation 6-7 ($D = P_s/\Sigma P_z$), as shown previously. In our example, $D = 164/224 = 0.73$.

7. Find the *uncorrected outdoor-air intake flow* for the system from Equation 6-6 $\left(V_{ou} = D \times \Sigma(R_p \times P_z) + \Sigma(R_a \times A_z)\right)$. Once again, without correcting for zone air-distribution effectiveness and system ventilation efficiency, our system needs $V_{ou} = 2{,}800$ cfm of outdoor air.

8. Finally, find *outdoor-air intake flow* for the system by solving Equation 6-8 ($V_{ot} = V_{ou}/E_v$). In our example, $V_{ot} = 2{,}800/0.60 = 4{,}670$ cfm at design heating conditions.

The system is less efficient at this heating condition than it was at the design cooling condition (*system ventilation efficiency* of 0.60 in heating vs. 0.65 in cooling). So, using the "default" approach (Table 6.3), worst-case/highest *outdoor-air intake flow* occurs at the design heating condition ($V_{ot} = 4{,}670$ cfm), assuming that all zones receive minimum primary airflow.

Multiple-Zone Systems

In multiple-zone recirculating systems, such as constant-volume reheat systems and all varieties of VAV systems, one air handler supplies a mixture of outdoor air and recirculated return air to two or more ventilation zones. The required outdoor-air intake flow only can be determined by properly accounting for system ventilation efficiency. Why?

These ventilation systems include an unavoidable "built-in" inefficiency. This inefficiency exists because the intake airflow must be sufficient to ventilate the *critical zone*—the zone that requires the highest fraction of outdoor air in its primary airstream. Since a multiple-zone system delivers the same primary air mixture to each ventilation zone, proper minimum ventilation in the critical zone overventilates all other zones. As a result, some outdoor air leaves the building via the relief, exhaust, and exfiltration airstreams without performing useful dilution.

This inefficiency isn't necessarily "bad;" it simply must be recognized and accounted for in system ventilation calculations.

Case 3: Ventilation Calculations for "Calculated" Cooling Design

As mentioned previously, Addendum 62*n* allows the designer to use either a default or calculated value for *system ventilation efficiency* (E_v). We used the default approach in Cases 1 and 2. Now, let's look at the calculated approach, which uses the equations found in Appendix G.

Again, we build on the zone-level calculations (*Figure 3*) to find the minimum system-level *outdoor-air intake flow* (V_{ot}) needed at the design cooling condition (*Figure 6*):

4. Find the *minimum discharge outdoor-air fraction* ($Z_d = V_{oz}/V_{dz}$) for each zone, using the *zone outdoor airflow* (V_{oz}) for cooling operation. Notice that this fraction differs from the *primary outdoor-air fraction* ($Z_p = V_{oz}/V_{pz}$) in the "default" approach. In this case, we're interested in the fraction of outdoor air in the airstream that discharges into the zone—not in the primary airstream from the air handler.†

5. Find *occupant diversity* according to Equation 6-7 ($D = P_s/\Sigma P_z$) using expected *peak system population* (P_s) and *design zone population*; as in the "default" approach (Case 1), $D = 164/224 = 0.73$.

6. Find the *uncorrected outdoor-air intake flow* for the system by solving Equation 6-6 $\left(V_{ou} = D \times \Sigma(R_p \times P_z) + \Sigma(R_a \times A_z)\right)$. Again, without correcting for zone air-distribution effectiveness and system ventilation efficiency, the system needs $V_{ou} = 2{,}800$ cfm of outdoor air.

7. Establish the *system primary airflow* ($V_{ps} = LDF \times \Sigma V_{pz}$-

† This nuance makes no difference for single-path systems ($V_{pz} = V_{dz}$), but becomes an important distinction for dual-path systems with local recirculation, as we'll see in future articles.

		From Figure 3		From Table 6.3		
Procedural Step				**4**	**5**	**6–8**
Ventilation Zone	V_{pz} **(Design) cfm**	$V_{pz\text{-}min}$ **cfm**	$V_{oz\text{-}clg}$ **cfm**	$Z_{p\text{-}clg}$	E_v	
South Offices	1,900	475	210	0.44	—	—
West Offices	2,000	500	220	0.44	—	—
South Conference Room	3,300	825	330	0.40	—	—
East Offices	2,000	500	220	0.44	—	—
Southwest Interior Offices	7,000	1,750	850	0.49	—	—
Northeast Interior Offices	7,000	1,750	850	0.49	—	—
North Offices	1,600	400	200	0.50*	0.65	—
North Conference Room	1,800	450	220	0.49	—	—
System…						
(Step 6) D						0.73
(Step 7) V_{ou}						2,800
(Step 8) V_{ot}						4,310

* For ventilation-critical zones with a throttling VAV box, discharge air is usually cool whenever the zone is occupied (morning warmup usually occurs before occupancy).

***Figure 4:** System ventilation calculations for default efficiency cooling design (Case 1).*

		From Figure 3		From Table 6.3		
Procedural Step				**4**	**5**	**6–8**
Ventilation Zone	V_{pz} **(Design) cfm**	$V_{pz\text{-}min}$ **cfm**	$V_{oz\text{-}htg}$ **cfm**	$Z_{p\text{-}htg}$	E_v	
South Offices	1,900	475	260	0.55*	0.60	—
West Offices	2,000	500	275	0.55*	—	—
South Conference Room	3,300	825	410	0.50	—	—
East Offices	2,000	500	275	0.55*	—	—
Southwest Interior Offices	7,000	1,750	850	0.49	—	—
Northeast Interior Offices	7,000	1,750	850	0.49	—	—
North Offices	1,600	400	200	0.50	—	—
North Conference Room	1,800	450	220	0.49	—	—
System…						
(Step 6) D						0.73
(Step 7) V_{ou}						2,800
(Step 8) V_{ot}						4,670

* For ventilation-critical zones with a throttling VAV box, discharge air is usually cool whenever the zone is occupied (morning warmup usually occurs before occupancy).

***Figure 5:** System ventilation calculations for default efficiency heating design (Case 2).*

peak). In VAV systems, primary airflow to each zone varies with load. Of course, system primary airflow also varies but it never can be more than the central fan can deliver. (The system is always least efficient when primary airflow is high and critical-zone[††] airflow is low because all noncritical zones are overventilated at this condition.) The central VAV fan usually is selected to deliver "block," not "sum-of-peak," airflow. In our example office, we assumed a system load diversity factor (*LDF*) of 0.70, so the central fan delivers $V_{ps} = 0.70 \times 26{,}600 = 18{,}600$ cfm at the design cooling load.

8. Find the *average outdoor-air fraction* ($X_s = V_{ou}/V_{ps}$) for the system. In our example, $X_s = 2{,}800/18{,}600 = 0.15$ at the design cooling condition.

9. For each zone, find *zone ventilation effectiveness* using Equation G-1 ($E_{vz} = 1 + X_s - Z_d$) for single-path systems.

10. Find *system ventilation efficiency* using Equation G-3 (E_v = minimum E_{vz}). In our example, $E_v = 0.65$ at the design cooling condition. As in the "default" approach (Case 1), the north offices are the ventilation-critical zone.

11. Finally, find *outdoor-air intake flow* for the system by solving Equation 6-8 ($V_{ot} = V_{ou}/E_v$). In our example, $V_{ot} = 2800/0.65 = 4310$ cfm at the design cooling condition.

This is identical to the intake requirement we found using the "default" approach. Why? The "default" approach is based on an assumed *average outdoor-air fraction* (X_s) of 0.15. By coincidence, that value matches this example's average outdoor-air fraction at design cooling. In most cases, however, these numbers will differ.

[††] We refer to the zone that requires the highest fraction of outdoor air in its discharge (primary plus recirculated) airstream as the "ventilation critical zone."

Now that we know the minimum intake at the design cooling condition, let's use the "calculated" approach to find the minimum intake for the design heating condition. The highest of these two intake values is the worst-case intake airflow.

Case 4: System Ventilation Calculations for Calculated Heating Design

As in the "default" approach for heating design (Case 2), assume that all spaces receive minimum primary airflow at the design heating condition. Building on the zone-level calculations (*Figure 3*), we'll follow the same steps that we used in Case 3 to calculate efficiency and intake airflow for cooling design (*Figure 7*).

4. For each zone, find the *minimum discharge outdoor-air fraction* ($Z_d = V_{oz}/V_{dz}$), using the appropriate V_{oz} value for heating operation. For example, the south offices need $Z_d = 260/475 = 0.55$ when heating.

5. Find *occupant diversity* according to Equation 6-7 ($D = P_s/\Sigma P_z$), $D = 164/224 = 0.73$.

6. Find the *uncorrected outdoor-air intake flow* for the system by solving Equation 6-6 $\left(V_{ou} = D \times \Sigma(R_p \times P_z) + \Sigma(R_a \times A_z)\right)$; as before, $V_{ou} = 2{,}800$ cfm.

7. Establish the *system primary airflow* (V_{ps}). For design heating calculations, we assume that all zones receive minimum primary airflow at worst case, so $V_{ps} = 6{,}650$ cfm in our example.

8. Find the *average outdoor-air fraction* ($X_s = V_{ou}/V_{ps}$) for the system. In our example, $X_s = 2{,}800/6{,}650 = 0.42$ at the design heating condition.

9. For each zone, find *zone ventilation effectiveness* using Equation G-1 ($E_{vz} = 1 + X_s - Z_d$).

Procedural Step	From Figure 3			4	5–8	9	10–11
Ventilation Zone	V_{pz} (Design) cfm	$V_{pz\text{-}min}$ cfm	$V_{oz\text{-}clg}$ cfm	$Z_{d\text{-}clg}$		$E_{vz\text{-}clg}$	
South Offices	1,900	475	210	0.44	—	0.71	—
West Offices	2,000	500	220	0.44	—	0.71	—
South Conference Room	3,300	825	330	0.40	—	0.75	—
East Offices	2,000	500	220	0.44	—	0.71	—
Southwest Interior Offices	7,000	1,750	850	0.49	—	0.66	—
Northeast Interior Offices	7,000	1,750	850	0.49	—	0.66	—
North Offices	1,600	400	200	0.50	—	0.65*	—
North Conference Room	1,800	450	220	0.49	—	0.66	—
System…							
(Step 5) D					0.73		
(Step 6) V_{ou} [2]					2,800		
(Step 7) V_{ps}					18,600		
(Step 8) X_s					0.15		
(Step 10) E_v							0.65
(Step 11) V_{ot}							4,310

* For ventilation-critical zones with a throttling VAV box, discharge air is usually cool whenever the zone is occupied (morning warmup usually occurs before occupancy).

Figure 6: System ventilation calculations for calculated efficiency cooling design (Case 3).

Procedural Step	From Figure 3			4	5–8	9	10–11
Ventilation Zone	V_{pz} (Design) cfm	$V_{pz\text{-}min}$ cfm	$V_{oz\text{-}htg}$ cfm	$Z_{d\text{-}htg}$		$E_{vz\text{-}htg}$	
South Offices	1,900	475	260	0.55	—	0.87*	—
West Offices	2,000	500	275	0.55	—	0.87*	—
South Conference Room	3,300	825	410	0.50	—	0.92	—
East Offices	2,000	500	275	0.55	—	0.87*	—
Southwest Interior Offices	7,000	1,750	850	0.49	—	0.94	—
Northeast Interior Offices	7,000	1,750	850	0.49	—	0.94	—
North Offices	1,600	400	200	0.50	—	0.92	—
North Conference Room	1,800	450	220	0.49	—	0.93	—
System…							
(Step 5) D					0.73		
(Step 6) V_{ou} [2]					2,800		
(Step 7) V_{ps}					6,650		
(Step 8) X_s					0.42		
(Step 10) E_v							0.87
(Step 11) V_{ot}							3,220

* For ventilation-critical zones with a throttling VAV box, discharge air is usually cool whenever the zone is occupied (morning warmup usually occurs before occupancy).

Figure 7: System ventilation for calculated efficiency heating design (Case 4).

10. Find *system ventilation efficiency* using Equation G-3 (E_v = minimum E_{vz}). In our example, $E_v = 0.87$ at the design heating condition. As before, the south, west, and east offices are equally "critical" for design heating calculations. Notice, too, that the ventilation system is much more efficient at this condition. When the *average outdoor-air fraction* (X_s) approaches the *critical zone's outdoor-air fraction* (Z_d), less unused air is exhausted; consequently, system ventilation efficiency rises.

11. Finally, find *outdoor-air intake flow* for the system by solving Equation 6-8 ($V_{ot} = V_{ou}/E_v$). In our example, V_{ot} = 2,800/0.87 = 3,230 cfm at the design heating condition.

The system is more efficient at the design heating condition than it was at the design cooling condition (*system ventilation efficiency* of 0.87 in heating vs. 0.65 in cooling). So, using the "calculated" approach (Appendix G), worst-case/highest *outdoor-air intake flow* occurs at the design cooling condition (V_{ot} = 4,310 cfm).

Reviewing our previous calculations, if we simply use the default table to find system ventilation efficiency (Cases 1 and 2), our example design needs *outdoor-air intake flow* of 4,670 cfm, which occurred at the design heating condition. If we use the more complicated but more accurate calculations in Appendix G (Cases 3 and 4), our example design needs *outdoor-air intake flow* of 4,360 cfm, which occurred at the design cooling condition. Since either approach is allowed, the designer can comply using either of these intake airflow values.

Assuming that our system controls can maintain the minimum required intake airflow, we can now size both the cooling coil and the heating coil for worst-case outdoor-air intake flow.

What About Part-Load Operation?

To comply with Addendum 62*n*, we need to find the highest minimum *outdoor-air intake flow* (V_{ot}), which we've called "worst-case" intake airflow. We could apply optional adjustments (averaging) for "short-term conditions" in our worst-case calculations, but we chose not to do so in the preceding discussion. In some cases, averaging adjustments can lower the worst-case intake value. In others, averaging can be used to assure proper ventilation when either supply-fan capacity or outdoor-air intake flow varies.

Adjustments for short-term conditions can help the designer find the appropriate worst-case minimum intake flow. Having found this value, the system can be designed to maintain this intake airflow during all occupied hours. In VAV systems, where both primary airflow and mixing-box pressure change in response to zone demands for cooling, this usually requires some means for sensing intake airflow and modulating the outdoor-air damper to maintain the minimum airflow setting.

But, do we really need to treat the worst-case outdoor airflow at all operating conditions, without regard to current ventilation needs? No.

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}.

In multiple-zone recirculating systems, system ventilation efficiency almost always increases as primary fan airflow decreases—provided, of course, that design efficiency is properly calculated at the worst-case condition (that is, with low primary airflow to the critical zone).

Although we must design the system with sufficient capacity for worst-case intake airflow, we could operate it at many conditions with less-than-worst-case intake and still comply with Addendum 62*n*. To do so, our design could incorporate one of the optional "dynamic reset" approaches presented in Addendum 62*n*, using a control approach that resets intake airflow to match current requirements at part-cooling load.

In a future article, we'll examine part-load operation and optional dynamic reset in detail. For now, we simply note we always must design for worst-case intake flow (as discussed earlier), regardless of any "dynamic reset" control options we may choose to implement. In other words, dynamic reset does not alter the worst-case outdoor-air intake flow needed to comply with the standard.

Summary

Historically, Standard 62 required both zone- and system-level calculations for the design of single-path, multiple-zone ventilation systems (like throttling VAV systems). Unfortunately, the calculation procedures were unclear and frequently misinterpreted or ignored by designers. As a result, many multiple-zone systems were improperly ventilated.

Addendum 62*n* clarifies the multiple-zone system calculations to reduce both underventilation and unnecessary over-ventilation. It allows a simple "default" approach, as well as a more accurate "calculated" approach for determining system ventilation efficiency.

As shown here, either calculation procedure can be readily applied to single-path VAV systems at the design conditions for both cooling and heating, to provide a compliant determination of worst-case minimum outdoor-air intake flow.

References

1. ANSI/ASHRAE Standard 62-2001, *Ventilation for Acceptable Indoor Air Quality.*

2. ANSI/ASHRAE Addendum *n* to ANSI/ASHRAE Standard 62-2001.

3. Stanke, D. 2004. "Addendum 62*n*: single-zone and dedicated-OA systems." ASHRAE Journal 46(10): 12–21.

4. Stanke, D. 2004. "Standard 62-2001 Addendum 62*n*: ventilation for change-over-bypass VAV systems." ASHRAE Journal 46(11):22–32.

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.

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* 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.

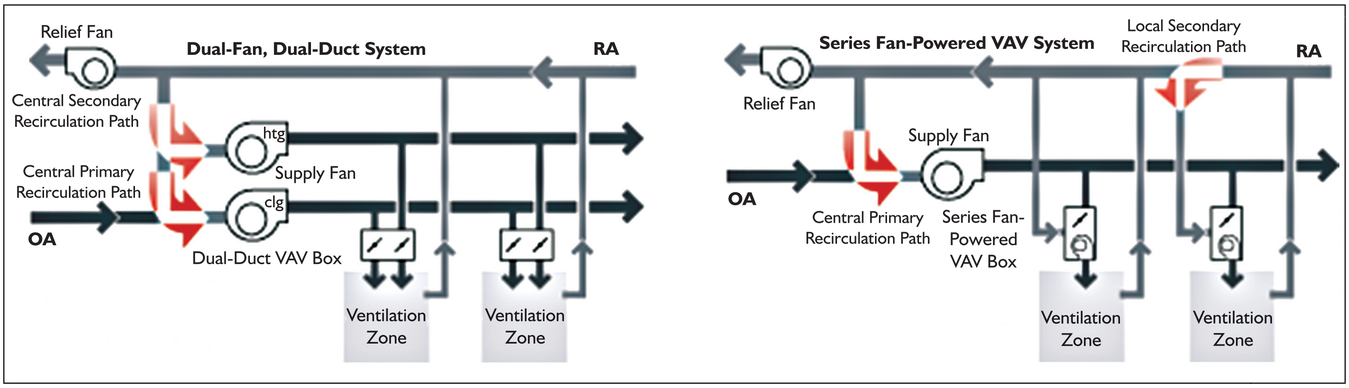

Figure 1: Dual-path, multiple-zone, recirculating ventilation systems.

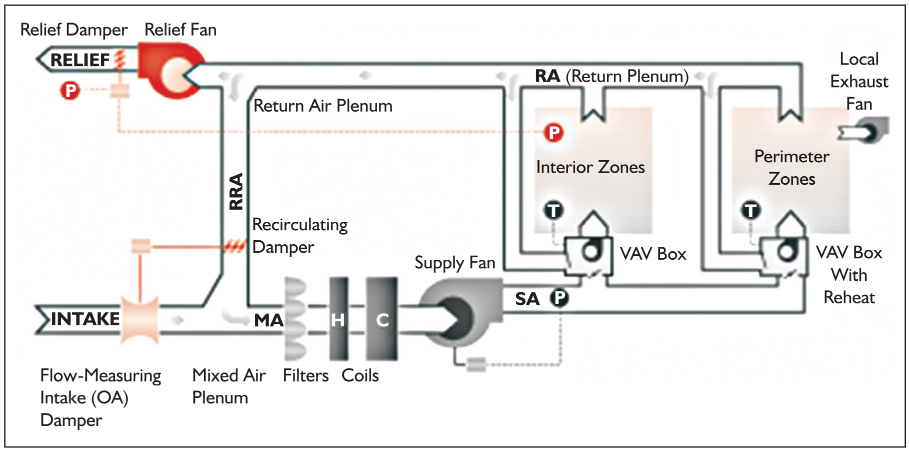

Figure 2: Series fan-powered variable-air-volume system.

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.

Cooling Design

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.

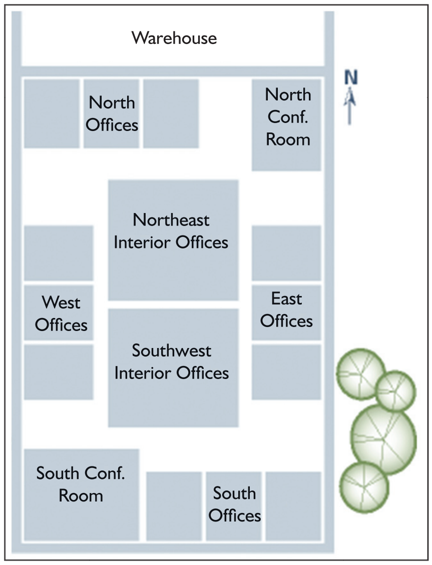

Figure 3: Multiple-zone office building.

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 $(F_a = E_p + (1 - E_p) \times E_r)$. 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 $(F_c = 1 - (1 - E_z) \times (1 - E_r) \times (1 - E_p))$. 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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3. Stanke, D. 2004. "Standard 62-2001 Addendum 62*n*: Ventilation for changeover-bypass VAV systems." *ASHRAE Journal* 46(11): 22–32.
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Standard 62-2001 Addendum 62*n*

Single-Path Multiple-Zone System Design

By Dennis Stanke, Member ASHRAE

ANSI/ASHRAE Standard 62-2001, *Ventilation for Acceptable Indoor Air Quality,*[1] as modified by Addendum 62*n*,[2] prescribes new minimum breathing-zone ventilation rates and new calculation procedures to find intake airflow for different ventilation systems. Previous articles[3,4] discussed the design of "simple" ventilation systems (single-zone, 100% outdoor-air, and changeover-bypass VAV) in compliance with Addendum 62*n* requirements. Here, we examine the design of a more complex set of ventilation systems, namely single-path, multiple-zone recirculating systems.

Although the Ventilation Rate Procedure in Standard 62 has required specific calculations (Equation 6-1) for multiple-zone systems since 1989, the calculation procedure was sketchy at best; consequently, it was widely misunderstood and largely ignored by designers. Addendum 62*n* includes a detailed calculation procedure for multiple-zone system design. Use of this procedure is expected to increase consistency among designers and reduce the tendency to design multiple-zone systems—especially VAV systems—that provide inadequate ventilation for some fully occupied zones. Addendum 62*n* also includes operational control options that can be used to modulate ventilation capacity as ventilation load and/or efficiency varies, but these options are left to a future article. The following discussion covers only design calculations.

Many HVAC systems are configured as "single-supply" or single-path, multiple-zone, recirculating ventilation systems. For instance, constant-volume systems with terminal reheat, traditional constant-volume multizone systems, single-duct VAV systems, and single-fan dual-duct VAV systems all provide ventilation from a single source or path. (A single-fan, dual-duct system supplies air to each space using two different ducts, but the air in each duct contains the same fraction of outdoor air, because one fan—a single source—delivers the same air mixture to each duct.) Other systems have multiple ventilation paths, including dual-fan, dual-duct VAV systems and VAV systems with fan-powered or induction terminal units. Single-duct VAV systems with series fan-powered boxes are always dual-path ventilation systems, but those with parallel fan-powered boxes are single-path with the local fan off and dual-path with it on. Although any of these HVAC systems may be used in vari-

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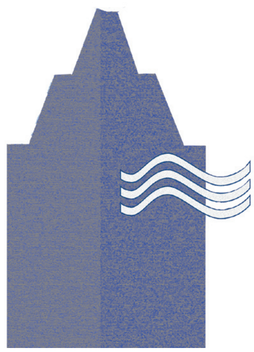
ous building types, we narrow our discussion to a single-duct VAV system, with throttling VAV boxes for interior zones and reheat VAV boxes in perimeter zones, applied in an example office building.

Demonstrating Compliance by Example

Our example system (*Figure 1*) includes a central air handler, with a modulating outdoor-air damper that may be controlled as an economizer; a variable-volume supply fan to deliver primary air; cooling-only, throttling VAV boxes in the interior zones; throttling VAV boxes with electric reheat in the perimeter zones; a central return fan; and a central relief damper for building pressure control. Although we won't discuss system control details here, it's important that we share the same mental "picture" of the VAV system we're designing:

• Intake airflow is sensed and maintained by adjusting the intake damper position. (Often, the return- and outdoor-air dampers are linked such that closing the outdoor-air damper opens the return-air damper proportionately. Alternately, these dampers can be controlled separately to reduce fan energy while maintaining proper intake airflow, but this has no impact on ventilation requirements at design conditions.)

• Primary air temperature is sensed and maintained by sequentially adjusting the heating-coil control valve, economizer dampers, and cooling-coil control valve.



• Duct pressure is sensed and maintained at setpoint by adjusting the primary fan capacity (via fan speed, for instance, or inlet guide vane position).

• Zone temperature is sensed and maintained at the cooling setpoint by adjusting the setpoint for VAV-box primary airflow.

• VAV-box airflow is sensed and maintained at setpoint by adjusting the position of the VAV-box damper.

• For zones that need reheat, zone temperature is sensed and maintained at the heating setpoint by adjusting reheat capacity (electric reheat or a hot water valve) and, thereby, discharge air temperature.

• Return air plenum pressure (at the central air handler) is sensed and maintained by adjusting return fan capacity.

• Building pressure is sensed and maintained between set limits by adjusting the relief (central exhaust) damper position.

Since multiple-zone systems provide the same primary air mixture to all zones, the fraction of outdoor air in the primary airstream must be sufficient to deliver the outdoor airflow needed by the "critical" zone—the zone needing the greatest fraction of outdoor air in its primary airstream. In the past, many designers simply added the zone outdoor airflow requirements and set the intake airflow to match this sum, which resulted in a very low outdoor-air fraction and many underventilated zones.

Some designers went to the other extreme, finding the highest fraction of outdoor air needed by any zone in the system and setting the intake airflow to provide this fraction at all times. This approach considers only first-pass outdoor air, giving no credit for unused recirculated outdoor air, and results in a very high outdoor-air fraction and overventilation in all zones.

Proper design in compliance with Addendum 62*n* calculation procedures strikes a balance between these extremes, appropriately accounting for both critical-zone needs and unused, recirculated outdoor air.

Let's look at an example office building (*Figure 2*). We assumed that thermal comfort can be achieved using only eight VAV thermostats, with each thermostat controlling one or more VAV boxes. We considered each of these "comfort zones" (or "HVAC zones" per ASHRAE Standard 90.1-2001) as a separate "ventilation zone."

According to Addendum 62*n*, a ventilation zone is "one occupied space or several occupied spaces with similar occupancy category, occupant density, zone air-distribution effectiveness, and zone primary airflow per unit area."

Most (but not all) HVAC zones qualify as ventilation zones. The area and population for each zone in this example were selected to help illustrate the calculations rather than to reflect typical zone sizes or population densities.

To comply with Addendum 62*n*, our design calculations begin by finding the ventilation needs at the zone level and conclude by determining the required intake airflow at the system level.

Zone Ventilation Calculations

Following the procedure under "zone calculations" in Section 6.2.1, we found zone outdoor airflow (V_{oz}) for each zone (*Figure 3*):

1. Referring to Addendum 62*n*, Table 6.1 (not shown), look up the prescribed minimum *people outdoor-air rate* (R_p) and the prescribed minimum building *area outdoor-air rate* (R_a). In our example office building, each zone needs 5 cfm/person and 0.06 cfm/ft². Using these values, along with the design *zone population* (P_z) and *zone floor area* (A_z), find the minimum *breathing-zone outdoor airflow* by solving Equation 6-1 ($V_{bz} = R_p \times P_z + R_a \times A_z$). Either peak or average expected occupancy may be

used to establish P_z; we used peak population in all zones. (An earlier article[3] covered population-averaging calculations in detail. See www.ashrae.org for the most current version.)

For our example, the west offices need $V_{bz} = 5 \times 20 + 0.06 \times 2{,}000 = 100 + 120 = 220$ cfm for proper ventilation in the breathing zone.

2. Look up *zone air-distribution effectiveness* (E_z), based on the air-distribution configuration and the default values presented in Addendum 62*n*, Table 6.2 (not shown). All of our example zones use overhead diffusers and ceiling returns, and they all receive 55°F primary air, so $E_z = 1.0$ when cooling. If the thermostat calls for heat in any of the perimeter zones, primary air is reheated and discharged at 95°F; so, $E_z = 0.8$ when heating.

3. Find the minimum *zone outdoor airflow* by solving Equation 6-2 ($V_{oz} = V_{bz} / E_z$) for both cooling and heating operation. For example, the west offices need $V_{oz} = 220/1.0 = 220$ cfm at the diffusers when cooling, and $V_{oz} = 220/0.8 = 275$ cfm when heating.

System Ventilation Calculations

As in Standard 62-1989, -1999, and -2001, Addendum 62*n* recognizes that multiple-zone recirculating systems must overventilate some zones to properly ventilate all zones. It also recognizes that "unused" outdoor air recirculated from overventilated zones reduces the required intake airflow, but that unused outdoor air that leaves the building (by exhaust or exfiltration) increases the required intake airflow. Proper accounting results in a ventilation credit for recirculated outdoor air and a ventilation debit for exhausted outdoor air.

Addendum 62*n* makes this accounting straightforward by requiring a specific calculation procedure to determine the minimum *outdoor-air intake flow* based on the *system ventilation efficiency* inherent in every multiple-zone recirculating system.

Earlier versions of the standard required use of the "multiple-space" equation, $Y = X/(1 + X - Z)$, to find the fraction of intake air needed. This approach resulted in about the same intake airflow as Addendum 62*n* for single-path systems; but without a clear procedural explanation, the equation was widely misunderstood and largely ignored by designers.

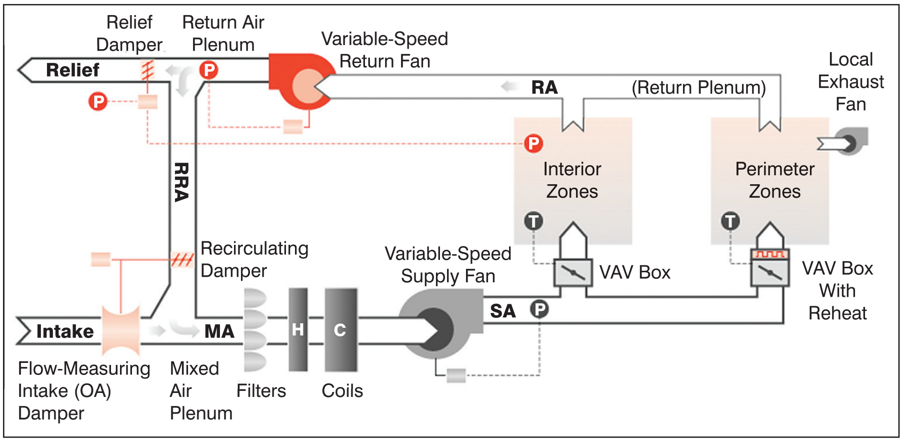


***Figure 1:** Variable air volume reheat system.*

Designs based on the 62*n* procedure result in proper ventilation for the critical zone at worst-case design conditions while allowing credit for "good" outdoor air that recirculates from all other overventilated zones.

From the zone calculations that we completed earlier, we know how much outdoor airflow must reach the diffusers in each zone. Now, let's figure out the minimum required intake airflow for the system at design conditions.

Before we start, we should recognize something that Addendum 62*n* implies but doesn't explain: *The "worst-case" or highest required intake airflow may or may not occur at the design cooling condition* (when system primary airflow is highest). In some cases, it may actually occur at the design heating condition (when zone primary airflow values are very low). With

Averaging Zone Population for Ventilation System Design

In earlier versions of the standard, only "intermittent occupancy" zones (at peak population for three hours or less) could be designed for ventilation at the average population (but not less than one-half of the peak population). Now, any zone may be designed for average population. According to the "short-term conditions" section of Addendum 62*n*, the system must be designed to deliver the required outdoor airflow to each occupied breathing zone.

However, if occupancy or intake airflow varies, the ventilation system design may be based on average conditions over a specific time period rather than on peak conditions. The averaging time T for a given zone is determined according to Equation 6-9 ($T = 3\,v/V_{oz}$) using zone volume and the breathing-zone outdoor airflow that would be needed at peak population. T equals three zone time constants, the time it takes for contaminant concentration to achieve a nearly steady-state value in response to a step change in contaminant source. When applied to population, this averaging approach replaces the population-averaging option for "intermittent occupancy" spaces, found in previous versions of the standard,

Averaging time may be applied to make design adjustments when changing conditions in the zone can be predicted. For instance, if zone population fluctuations are predictable, then the design breathing-zone outdoor airflow may be calculated based on the highest average population over any T-minute period.

this in mind, we'll need to check the required intake airflow at both design cooling *and* design heating because it's ultimately the worst-case outdoor-air intake flow that will establish the required capacities for the heating and cooling coils.

For our example, we tried to use "reasonable" values for *zone primary airflow* (V_{pz}) at design cooling load. We arbitrarily set all minimum primary airflow settings (V_{pz}-min) to 25% of design cooling airflow. We assumed that each reheat box enters reheat mode after its primary airflow decreases to the minimum setting and the zone temperature drops below the heating setpoint. Reheat operation continues until the zone temperature exceeds the heating setpoint.

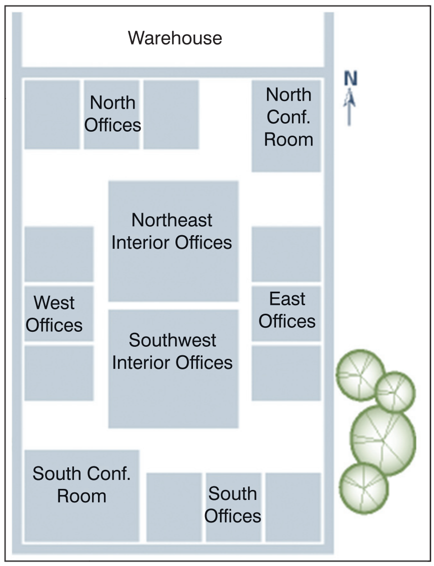


Figure 2: Multiple-zone office building.

Case 1: Ventilation Calculations for "Default" Cooling Design

Building on our earlier zone-level calculations (*Figure 3*), we followed the step-by-step, "multiple-zone recirculating systems" procedure to find the minimum, system-level, *outdoor-air intake flow* (V_{ot}) at the design cooling condition (*Figure 4*):

4. For each zone, find the *zone primary outdoor-air fraction* by solving Equation 6-5 ($Z_p = V_{oz}/V_{pz}$) using the *zone outdoor airflow* (V_{oz}) values for cooling from Step 3 and the minimum primary airflow setting. As an example, at minimum primary airflow, the south offices need $Z_p = 210/475 = 0.44$ when delivering cool air.

5. Addendum 62*n* allows the designer to use a either default value for *system ventilation efficiency* (E_v) using Table 6.3 (not shown) or a calculated value (found using equations in Appendix G). In this case, we used Table 6.3 and the highest *zone primary outdoor-air fraction* among the zones served ("max Z_p" = 0.50 for the north offices) to look up the corresponding default *system ventilation efficiency* (E_v). From that value, we can interpolate to find $E_v = 0.65$.

6. Find *occupant diversity* according to Equation 6-7 ($D = P_s/\Sigma P_z$) by using the expected peak *system population* (P_s) and the sum of design zone populations. For our example, we expect a maximum system population of 164 people, so $D = 164/224 = 0.73$.

7. Find the *uncorrected outdoor-air intake flow* for the system by solving Equation 6-6 $\left(V_{ou} = D \times \Sigma(R_p \times P_z) + \Sigma(R_a \times A_z)\right)$. Without correcting for zone ventilation effectiveness and system ventilation efficiency, we find that the system needs V_{ou} = 2,800 cfm of outdoor air at the breathing zones.

8. Finally, find *outdoor-air intake flow* for the system by solving Equation 6-8 ($V_{ot} = V_{ou}/E_v$). In our example, $V_{ot} = 2{,}800/0.65 = 4{,}310$ cfm at the design cooling condition.

But, is this really the worst-case (highest volume) intake airflow? What happens at design heating conditions?

Case 2: Ventilation Calculations for "Default" Heating Design

Let's find the minimum system-level *outdoor-air intake flow* (V_{ot}) for the design heating condition. The procedure is the same one that was just described for default cooling design in Case 1. It builds on the zone-level calculations that were completed earlier (*Figure 3*), but in this case, we assume that each space receives minimum primary airflow at the design outdoor heating condition (*Figure 5*).*

4. For each zone, find the *zone primary outdoor-air fraction* by solving Equation 6-5 ($Z_p = V_{oz}/V_{pz}$) with the *zone outdoor*

* Some readers might deem this to be a radical assumption because interior zones typically need more than minimum cooling airflow, even on the coldest day. But, it's an assumption that is likely to require a high intake airflow, which is useful for this demonstration.

Design Cooling Condition

For single-path VAV systems, the worst-case condition for ventilation (that is, the lowest system ventilation efficiency and the highest required intake airflow) in the cooling mode usually occurs when the VAV primary airflow for the system is at its highest value. Since almost all VAV systems exhibit load diversity (all zones don't require peak cooling airflow simultaneously), the critical zone can be assumed to be delivering minimum primary airflow with the central fan at cooling-design or "block" primary airflow. In some cases, worst-case ventilation in the cooling mode may actually occur at a central fan airflow that's slightly lower than block airflow. If a system doesn't have much load diversity (all interior zones, for example)—and if the critical zone requires a lot of primary airflow—then the central fan may or may not be at block airflow when the critical zone is at minimum primary airflow.

How can you find out the system primary airflow at the worst-case ventilation condition? Simply assume that primary airflow at the fan is the sum of all noncritical-zone peak airflow values plus the minimum primary airflow for the critical zone. At this condition, the difference between X_s and Z_p will be greatest, so system ventilation efficiency will be at its lowest value and outdoor-air intake flow will be at its highest values—the worst-case condition. (Operationally, this worst-case condition may not actually occur, since it assumes that the critical zone requires minimum primary airflow even when fully occupied; this might be the case for some perimeter zones, for example, during cold weather.)

							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	**Box Type**	cfm/p	p	cfm/ft²	ft²	cfm		cfm		cfm
South Offices	Reheat	5	18	0.06	2,000	210	1.0	210	0.8	260
West Offices	Reheat	5	20	0.06	2,000	220	1.0	220	0.8	275
South Conference Room	Reheat	5	30	0.06	3,000	330	1.0	330	0.8	410
East Offices	Reheat	5	20	0.06	2,000	220	1.0	220	0.8	275
Southwest Interior Offices	VAV	5	50	0.06	10,000	850	1.0	850	1.0	850
Northeast Interior Offices	VAV	5	50	0.06	10,000	850	1.0	850	1.0	850
North Offices	VAV	5	16	0.06	2,000	200	1.0	200	1.0	200
North Conference Room	VAV	5	20	0.06	2,000	220	1.0	220	1.0	220

* For zones with a throttling VAV box, discharge air is usually cool whenever the zone is occupied (morning warmup usually occurs before occupancy).
** For zones with terminal reheat, discharge air temperature can be either cool or warm when the zone is occupied, so E_z drops from 1.0 when cooling to 0.8 when heating.

Figure 3: Zone ventilation calculations.

airflow (V_{oz}) values for heating from Step 3 and the minimum primary airflow setting. At minimum primary airflow, the south office needs $Z_p = 260/475 = 0.55$ when delivering warm air.

5. Using Table 6.3 (not shown) and the highest *zone primary outdoor-air fraction* among the zones served ("max Z_p" = 0.55 for the south, west, and east offices) to look up the corresponding default *system ventilation efficiency* (E_v), we find that E_v = 0.60.

6. Find *occupant diversity* according to Equation 6-7 ($D = P_s/\Sigma P_z$), as shown previously. In our example, $D = 164/224 = 0.73$.

7. Find the *uncorrected outdoor-air intake flow* for the system from Equation 6-6 $\left(V_{ou} = D \times \Sigma(R_p \times P_z) + \Sigma(R_a \times A_z)\right)$. Once again, without correcting for zone air-distribution effectiveness and system ventilation efficiency, our system needs V_{ou} = 2,800 cfm of outdoor air.

8. Finally, find *outdoor-air intake flow* for the system by solving Equation 6-8 ($V_{ot} = V_{ou}/E_v$). In our example, $V_{ot} = 2{,}800/0.60$ = 4,670 cfm at design heating conditions.

The system is less efficient at this heating condition than it was at the design cooling condition (*system ventilation efficiency* of 0.60 in heating vs. 0.65 in cooling). So, using the "default" approach (Table 6.3), worst-case/highest *outdoor-air intake flow* occurs at the design heating condition (V_{ot} = 4,670 cfm), assuming that all zones receive minimum primary airflow.

Multiple-Zone Systems

In multiple-zone recirculating systems, such as constant-volume reheat systems and all varieties of VAV systems, one air handler supplies a mixture of outdoor air and recirculated return air to two or more ventilation zones. The required outdoor-air intake flow only can be determined by properly accounting for system ventilation efficiency. Why?

These ventilation systems include an unavoidable "built-in" inefficiency. This inefficiency exists because the intake airflow must be sufficient to ventilate the *critical zone*—the zone that requires the highest fraction of outdoor air in its primary airstream. Since a multiple-zone system delivers the same primary air mixture to each ventilation zone, proper minimum ventilation in the critical zone overventilates all other zones. As a result, some outdoor air leaves the building via the relief, exhaust, and exfiltration airstreams without performing useful dilution.

This inefficiency isn't necessarily "bad;" it simply must be recognized and accounted for in system ventilation calculations.

Case 3: Ventilation Calculations for "Calculated" Cooling Design

As mentioned previously, Addendum 62*n* allows the designer to use either a default or calculated value for *system ventilation efficiency* (E_v). We used the default approach in Cases 1 and 2. Now, let's look at the calculated approach, which uses the equations found in Appendix G.

Again, we build on the zone-level calculations (*Figure 3*) to find the minimum system-level *outdoor-air intake flow* (V_{ot}) needed at the design cooling condition (*Figure 6*):

4. Find the *minimum discharge outdoor-air fraction* ($Z_d = V_{oz}/V_{dz}$) for each zone, using the *zone outdoor airflow* (V_{oz}) for cooling operation. Notice that this fraction differs from the *primary outdoor-air fraction* ($Z_p = V_{oz}/V_{pz}$) in the "default" approach. In this case, we're interested in the fraction of outdoor air in the airstream that discharges into the zone—not in the primary airstream from the air handler.†

5. Find *occupant diversity* according to Equation 6-7 ($D = P_s/\Sigma P_z$) using expected *peak system population* (P_s) and *design zone population*; as in the "default" approach (Case 1), $D = 164/224 = 0.73$.

6. Find the *uncorrected outdoor-air intake flow* for the system by solving Equation 6-6 $\left(V_{ou} = D \times \Sigma(R_p \times P_z) + \Sigma(R_a \times A_z)\right)$. Again, without correcting for zone air-distribution effectiveness and system ventilation efficiency, the system needs V_{ou} = 2,800 cfm of outdoor air.

7. Establish the *system primary airflow* ($V_{ps} = LDF \times \Sigma V_{pz}$-

† This nuance makes no difference for single-path systems ($V_{pz} = V_{dz}$), but becomes an important distinction for dual-path systems with local recirculation, as we'll see in future articles.

		From Figure 3			From Table 6.3	
Procedural Step				4	5	6–8
Ventilation Zone	V_{pz} (Design) cfm	$V_{pz\text{-}min}$ cfm	$V_{oz\text{-}clg}$ cfm	$Z_{p\text{-}clg}$	E_v	
South Offices	1,900	475	210	0.44	—	—
West Offices	2,000	500	220	0.44	—	—
South Conference Room	3,300	825	330	0.40	—	—
East Offices	2,000	500	220	0.44	—	—
Southwest Interior Offices	7,000	1,750	850	0.49	—	—
Northeast Interior Offices	7,000	1,750	850	0.49	—	—
North Offices	1,600	400	200	0.50*	0.65	—
North Conference Room	1,800	450	220	0.49	—	—
System…						
(Step 6) D						0.73
(Step 7) V_{ou}						2,800
(Step 8) V_{ot}						4,310

* For ventilation-critical zones with a throttling VAV box, discharge air is usually cool whenever the zone is occupied (morning warmup usually occurs before occupancy).

***Figure 4:** System ventilation calculations for default efficiency cooling design (Case 1).*

		From Figure 3			From Table 6.3	
Procedural Step				4	5	6–8
Ventilation Zone	V_{pz} (Design) cfm	$V_{pz\text{-}min}$ cfm	$V_{oz\text{-}htg}$ cfm	$Z_{p\text{-}htg}$	E_v	
South Offices	1,900	475	260	0.55*	0.60	—
West Offices	2,000	500	275	0.55*	—	—
South Conference Room	3,300	825	410	0.50	—	—
East Offices	2,000	500	275	0.55*	—	—
Southwest Interior Offices	7,000	1,750	850	0.49	—	—
Northeast Interior Offices	7,000	1,750	850	0.49	—	—
North Offices	1,600	400	200	0.50	—	—
North Conference Room	1,800	450	220	0.49	—	—
System…						
(Step 6) D						0.73
(Step 7) V_{ou}						2,800
(Step 8) V_{ot}						4,670

* For ventilation-critical zones with a throttling VAV box, discharge air is usually cool whenever the zone is occupied (morning warmup usually occurs before occupancy).

***Figure 5:** System ventilation calculations for default efficiency heating design (Case 2).*

peak). In VAV systems, primary airflow to each zone varies with load. Of course, system primary airflow also varies but it never can be more than the central fan can deliver. (The system is always least efficient when primary airflow is high and critical-zone[††] airflow is low because all noncritical zones are overventilated at this condition.) The central VAV fan usually is selected to deliver "block," not "sum-of-peak," airflow. In our example office, we assumed a system load diversity factor (*LDF*) of 0.70, so the central fan delivers $V_{ps} = 0.70 \times 26{,}600 = 18{,}600$ cfm at the design cooling load.

8. Find the *average outdoor-air fraction* ($X_s = V_{ou}/V_{ps}$) for the system. In our example, $X_s = 2{,}800/18{,}600 = 0.15$ at the design cooling condition.

9. For each zone, find *zone ventilation effectiveness* using Equation G-1 ($E_{vz} = 1 + X_s - Z_d$) for single-path systems.

10. Find *system ventilation efficiency* using Equation G-3 (E_v = minimum E_{vz}). In our example, $E_v = 0.65$ at the design cooling condition. As in the "default" approach (Case 1), the north offices are the ventilation-critical zone.

11. Finally, find *outdoor-air intake flow* for the system by solving Equation 6-8 ($V_{ot} = V_{ou}/E_v$). In our example, $V_{ot} = 2800/0.65 = 4310$ cfm at the design cooling condition.

This is identical to the intake requirement we found using the "default" approach. Why? The "default" approach is based on an assumed *average outdoor-air fraction* (X_s) of 0.15. By coincidence, that value matches this example's average outdoor-air fraction at design cooling. In most cases, however, these numbers will differ.

[††] We refer to the zone that requires the highest fraction of outdoor air in its discharge (primary plus recirculated) airstream as the "ventilation critical zone."

Now that we know the minimum intake at the design cooling condition, let's use the "calculated" approach to find the minimum intake for the design heating condition. The highest of these two intake values is the worst-case intake airflow.

Case 4: System Ventilation Calculations for Calculated Heating Design

As in the "default" approach for heating design (Case 2), assume that all spaces receive minimum primary airflow at the design heating condition. Building on the zone-level calculations (*Figure 3*), we'll follow the same steps that we used in Case 3 to calculate efficiency and intake airflow for cooling design (*Figure 7*).

4. For each zone, find the *minimum discharge outdoor-air fraction* ($Z_d = V_{oz}/V_{dz}$), using the appropriate V_{oz} value for heating operation. For example, the south offices need $Z_d = 260/475 = 0.55$ when heating.

5. Find *occupant diversity* according to Equation 6-7 ($D = P_s/\Sigma P_z$), $D = 164/224 = 0.73$.

6. Find the *uncorrected outdoor-air intake flow* for the system by solving Equation 6-6 $\left(V_{ou} = D \times \Sigma(R_p \times P_z) + \Sigma(R_a \times A_z)\right)$; as before, $V_{ou} = 2{,}800$ cfm.

7. Establish the *system primary airflow* (V_{ps}). For design heating calculations, we assume that all zones receive minimum primary airflow at worst case, so $V_{ps} = 6{,}650$ cfm in our example.

8. Find the *average outdoor-air fraction* ($X_s = V_{ou}/V_{ps}$) for the system. In our example, $X_s = 2{,}800/6{,}650 = 0.42$ at the design heating condition.

9. For each zone, find *zone ventilation effectiveness* using Equation G-1 ($E_{vz} = 1 + X_s - Z_d$).

	From Figure 3						
Procedural Step				**4**	**5–8**	**9**	**10–11**
Ventilation Zone	V_{pz} (Design) cfm	$V_{pz\text{-}min}$ cfm	$V_{oz\text{-}clg}$ cfm	$Z_{d\text{-}clg}$		$E_{vz\text{-}clg}$	
South Offices	1,900	475	210	0.44	—	0.71	—
West Offices	2,000	500	220	0.44	—	0.71	—
South Conference Room	3,300	825	330	0.40	—	0.75	—
East Offices	2,000	500	220	0.44	—	0.71	—
Southwest Interior Offices	7,000	1,750	850	0.49	—	0.66	—
Northeast Interior Offices	7,000	1,750	850	0.49	—	0.66	—
North Offices	1,600	400	200	0.50	—	0.65*	—
North Conference Room	1,800	450	220	0.49	—	0.66	—
System…							
(Step 5) D					0.73		
(Step 6) V_{ou}[2]					2,800		
(Step 7) V_{ps}					18,600		
(Step 8) X_s					0.15		
(Step 10) E_v							0.65
(Step 11) V_{ot}							4,310

* For ventilation-critical zones with a throttling VAV box, discharge air is usually cool whenever the zone is occupied (morning warmup usually occurs before occupancy).

Figure 6: System ventilation calculations for calculated efficiency cooling design (Case 3).

	From Figure 3						
Procedural Step				**4**	**5–8**	**9**	**10–11**
Ventilation Zone	V_{pz} (Design) cfm	$V_{pz\text{-}min}$ cfm	$V_{oz\text{-}htg}$ cfm	$Z_{d\text{-}htg}$		$E_{vz\text{-}htg}$	
South Offices	1,900	475	260	0.55	—	0.87*	—
West Offices	2,000	500	275	0.55	—	0.87*	—
South Conference Room	3,300	825	410	0.50	—	0.92	—
East Offices	2,000	500	275	0.55	—	0.87*	—
Southwest Interior Offices	7,000	1,750	850	0.49	—	0.94	—
Northeast Interior Offices	7,000	1,750	850	0.49	—	0.94	—
North Offices	1,600	400	200	0.50	—	0.92	—
North Conference Room	1,800	450	220	0.49	—	0.93	—
System…							
(Step 5) D					0.73		
(Step 6) V_{ou}[2]					2,800		
(Step 7) V_{ps}					6,650		
(Step 8) X_s					0.42		
(Step 10) E_v							0.87
(Step 11) V_{ot}							3,220

* For ventilation-critical zones with a throttling VAV box, discharge air is usually cool whenever the zone is occupied (morning warmup usually occurs before occupancy).

Figure 7: System ventilation for calculated efficiency heating design (Case 4).

10. Find *system ventilation efficiency* using Equation G-3 (E_v = minimum E_{vz}). In our example, $E_v = 0.87$ at the design heating condition. As before, the south, west, and east offices are equally "critical" for design heating calculations. Notice, too, that the ventilation system is much more efficient at this condition. When the *average outdoor-air fraction* (X_s) approaches the *critical zone's outdoor-air fraction* (Z_d), less unused air is exhausted; consequently, system ventilation efficiency rises.

11. Finally, find *outdoor-air intake flow* for the system by solving Equation 6-8 ($V_{ot} = V_{ou}/E_v$). In our example, V_{ot} = 2,800/0.87 = 3,230 cfm at the design heating condition.

The system is more efficient at the design heating condition than it was at the design cooling condition (*system ventilation efficiency* of 0.87 in heating vs. 0.65 in cooling). So, using the "calculated" approach (Appendix G), worst-case/highest *outdoor-air intake flow* occurs at the design cooling condition (V_{ot} = 4,310 cfm).

Reviewing our previous calculations, if we simply use the default table to find system ventilation efficiency (Cases 1 and 2), our example design needs *outdoor-air intake flow* of 4,670 cfm, which occurred at the design heating condition. If we use the more complicated but more accurate calculations in Appendix G (Cases 3 and 4), our example design needs *outdoor-air intake flow* of 4,360 cfm, which occurred at the design cooling condition. Since either approach is allowed, the designer can comply using either of these intake airflow values.

Assuming that our system controls can maintain the minimum required intake airflow, we can now size both the cooling coil and the heating coil for worst-case outdoor-air intake flow.

What About Part-Load Operation?

To comply with Addendum 62*n*, we need to find the highest minimum *outdoor-air intake flow* (V_{ot}), which we've called "worst-case" intake airflow. We could apply optional adjustments (averaging) for "short-term conditions" in our worst-case calculations, but we chose not to do so in the preceding discussion. In some cases, averaging adjustments can lower the worst-case intake value. In others, averaging can be used to assure proper ventilation when either supply-fan capacity or outdoor-air intake flow varies.

Adjustments for short-term conditions can help the designer find the appropriate worst-case minimum intake flow. Having found this value, the system can be designed to maintain this intake airflow during all occupied hours. In VAV systems, where both primary airflow and mixing-box pressure change in response to zone demands for cooling, this usually requires some means for sensing intake airflow and modulating the outdoor-air damper to maintain the minimum airflow setting.

But, do we really need to treat the worst-case outdoor airflow at all operating conditions, without regard to current ventilation needs? No.

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

A_z is zone floor area, the net occupiable floor area of the zone, ft² (m²)

D is occupant diversity, the ratio of system population to the sum of zone populations

E_v is ventilation efficiency of the system

E_z is air-distribution effectiveness within the zone

P_s is system population, the maximum simultaneous number of occupants in the area served by the ventilation system

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)

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)

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)

T is averaging time period, minutes

v is ventilation-zone volume, ft³ (m³)

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)

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

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

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)

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

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}.

In multiple-zone recirculating systems, system ventilation efficiency almost always increases as primary fan airflow decreases—provided, of course, that design efficiency is properly calculated at the worst-case condition (that is, with low primary airflow to the critical zone).

Although we must design the system with sufficient capacity for worst-case intake airflow, we could operate it at many conditions with less-than-worst-case intake and still comply with Addendum 62*n*. To do so, our design could incorporate one of the optional "dynamic reset" approaches presented in Addendum 62*n*, using a control approach that resets intake airflow to match current requirements at part-cooling load.

In a future article, we'll examine part-load operation and optional dynamic reset in detail. For now, we simply note we always must design for worst-case intake flow (as discussed earlier), regardless of any "dynamic reset" control options we may choose to implement. In other words, dynamic reset does not alter the worst-case outdoor-air intake flow needed to comply with the standard.

Summary

Historically, Standard 62 required both zone- and system-level calculations for the design of single-path, multiple-zone ventilation systems (like throttling VAV systems). Unfortunately, the calculation procedures were unclear and frequently misinterpreted or ignored by designers. As a result, many multiple-zone systems were improperly ventilated.

Addendum 62*n* clarifies the multiple-zone system calculations to reduce both underventilation and unnecessary over-ventilation. It allows a simple "default" approach, as well as a more accurate "calculated" approach for determining system ventilation efficiency.

As shown here, either calculation procedure can be readily applied to single-path VAV systems at the design conditions for both cooling and heating, to provide a compliant determination of worst-case minimum outdoor-air intake flow.

References

1. ANSI/ASHRAE Standard 62-2001, *Ventilation for Acceptable Indoor Air Quality.*
2. ANSI/ASHRAE Addendum *n* to ANSI/ASHRAE Standard 62-2001.
3. Stanke, D. 2004. "Addendum 62*n*: single-zone and dedicated-OA systems." ASHRAE Journal 46(10): 12–21.
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Addendum 62*n*

Single-Zone & Dedicated-OA Systems

By **Dennis Stanke,** Member ASHRAE

ANSI/ASHRAE Standard 62, *Ventilation for Acceptable Indoor Air Quality*,[1] has been modified by Addendum 62*n*[2], whose ventilation requirements alter the very heart of Standard 62. Addendum 62*n* contains a long-awaited update to the minimum prescribed ventilation rates — last updated in ASHRAE Standard 62-1989 — and it incorporates ventilation airflow "additivity" for dilution of both people-source and building-source contaminants. The updated version of the once-familiar table of prescribed breathing-zone ventilation rates now contains both per-person and per-unit-area values for each occupancy category.

Addendum 62*n* updates the calculation procedure for zone ventilation airflow, incorporating an adjustment for air distribution effectiveness. It also updates the calculation procedure for system intake airflow for different ventilation systems and clarifies the prescribed approach for multiple-zone system design — required for years but also widely misunderstood and largely ignored by designers. (Incorrect intake calculations often result in multiple-zone recirculating systems that provide too little ventilation — especially for some fully occupied VAV zones.)

Finally, the addendum specifically identifies some operational control options that can reduce (or increase) intake airflow to match ventilation capacity with a changing ventilation "load," saving preconditioning energy while maintaining the required ventilation to the occupants.

What Are the New Ventilation Rates?

Table 1 shows minimum breathing-zone ventilation rates* for several important occupancy categories, comparing the prescribed rates from Addendum 62*n*, Table 6.1, with the previous rates from Standard 62-2001. The minimum cfm/person rate (R_p) dropped for many categories (except for some retail categories, where it increased from the previously prescribed "zero" per-person rate) because the ventilation basis for people-source contaminants changed from satisfying *unadapted visitors* to satisfying *adapted occupants*. However, the addition of a minimum cfm/ft² rate

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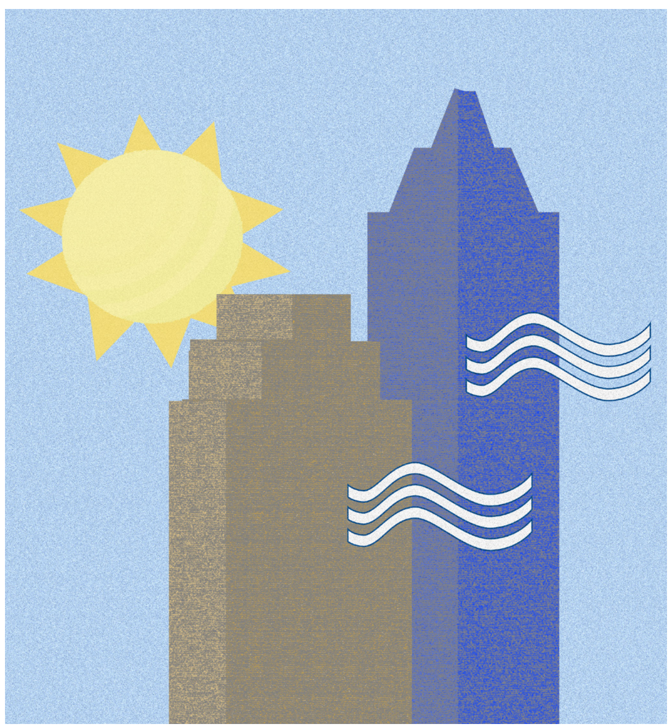
* Although Addendum 62*n* shows ventilation rates in both IP and SI units, this paper uses IP, except in selected specific calculations. This is because 62*n* uses rational conversions, not mathematical.

(R_a) for each category to dilute building-source contaminants moderates those drops in the "effective" per-person breathing-zone rates (i.e., the sum of the people- and area-related rates, divided by zone population). But be careful of simple, general comparisons; new default occupant densities and new options for population averaging can result in significant changes to "traditional" ventilation rates.

Underlying the prescribed rates, which only apply to no-smoking areas, is the premise that all other "general" requirements in the standard are met (i.e., that the drain pan drains, humidity is limited, filters are used, and so on). Of course, compliance has always meant meeting general requirements, but Standard 62-2001 and its addenda extend and clarify these requirements, moving many design decisions from "good practice" to "mandatory." As in the past, Table 6.1 includes default occupant densities for each category, but as before, these default values should only be used if reasonable for the application or if the actual design density is unknown.

What Is the New Procedure?

Addendum 62*n* prescribes a step-by-step procedure for ventilation system design to help designers *consistently* find the minimum ventilation airflow required in the breathing zone, in the ventilation zone, and at the outdoor air intake.



Zone Ventilation Calculations

Addendum 62*n* includes the following three-step procedure, which results in correct application of the prescribed rates to *each ventilation zone*.

1. Referring to *Table 1* (an excerpt from 62*n*, Table 6.1), look up both the people-source ventilation rate (R_p in cfm/person) *and* the building-source ventilation rate (R_a in cfm/ft^2). Establish the *zone floor area* and *zone population*. The latter is the largest (or average) number of occupants expected to occupy the zone during normal use. Using these values, solve Equation 6-1 ($V_{bz} = R_p \times P_z + R_a \times A_z$) to find the required outdoor airflow for the *breathing zone*.

2. Referring to *Table 2* (an excerpt from 62*n*, Table 6.2) look up the default value for zone air-distribution effectiveness (E_z), which is based on selection and placement of supply diffusers and return grilles. This effectiveness value, which is similar to "air-change effectiveness" (described in ASHRAE Standard 129) for well-mixed spaces, allows the designer to account for any ventilation air (delivered by the diffusers) that bypasses the breathing zone.

3. Solve Equation 6-2 ($V_{oz} = V_{bz} / E_z$) to find the outdoor airflow required in the air supplied to each ventilation zone. This volume of outdoor air must be supplied for ventilation, regardless of heating or cooling airflow requirements.

Let's use numbers to demonstrate how these three steps determine the needed outdoor airflow for a north-facing office area with overhead supply diffusers and return grilles. Referring to *Table 1,* the "office space" occupancy category requires a "people outdoor air rate" of R_p = 5 cfm/person** and an "area outdoor air rate" of R_a = 0.06 cfm/ft^2. If the office comprises 1,500 ft^2 of open-plan space and is occupied by 10 people (150 ft^2/person) solving Equation 6-1 ($V_{bz} = 5 \times 10 + 0.06 \times 1500 = 50 + 90 = 140$), tells us that at least 140 cfm (67 L/s) of outdoor air must be delivered to the breathing zone. The default zone air-distribution effectiveness for this configuration is 1.0 when cooling and 0.8 when heating. Now, we can solve Equation 6-2 and learn that our example office space requires at least ($V_{oz} = V_{bz} / E_z = 140/1.0 = 140$ cfm) 14 cfm/person of outdoor airflow when cooling and at least ($V_{oz} = 140/0.8 = 175$ cfm) 17.5 cfm/person when heating.

Since 1989, Standard 62 has required 20 cfm/person for office-space breathing zones, so this particular office space needs less outdoor air per person to comply with 62*n*. However, lower occupant density might require *more* airflow per person, and higher occupant density (common in private offices) would require *much less* airflow per person. If our example was an "executive" office, designed for five people (300 ft^2/person), it would require 115 cfm or 23 cfm/person (10.9 L/s per person), but if it was an "technical professional" office, designed for 20 people (75 ft^2/person), it would require 190 cfm or only 9.5 cfm/person (4.6 L/s per person).

System Ventilation Calculations

After determining outdoor airflow (V_{oz}) for each ventilation zone, outdoor air intake flow (V_{ot}) is calculated for the ventilation system as a whole. The procedure for finding the required outdoor air intake flow varies with the configuration of the

** cfm × 0.4719 = L/s; ft^2 × 0.0929 = m^2

	62-2001, Table 2		62*n*, Table 6.1			Per Person Rate	
Occupancy Category	**People Outdoor Air Rate**	**Area Outdoor Air Rate**	**People Outdoor Air Rate**	**Area Outdoor Air Rate**	**Default Occupant Density**	**62-2001**	**62*n***
			R_p	R_a			
	cfm/person	cfm/ft²	cfm/person	cfm/ft²	#/1000 ft²	cfm/person	cfm/person
Office Space	20	—	5	0.06	5	20	17
Conference/Meeting	20	—	5	0.06	50	20	6.2
Art Classroom	15	—	10	0.18	20	15	19
Classroom (Ages 5 – 8)	15	—	10	0.12	25	15	14.8
Classroom (Ages 9+)	15	—	10	0.12	35	15	13.4
Lecture Classroom	15	—	7.5	0.06	65	15	8.4
Multiuse Assembly	15	—	7.5	0.06	100	15	8.1
Retail Sales	—	0.30	7.5	0.12	15	20	15.5

Table 1: Comparison of breathing zone ventilation rates for several occupancy categories.

ventilation system. Addendum 62*n* defines three configurations: single-zone, 100% (or "dedicated") outdoor air, and multiple-zone recirculating systems.

Single-Zone Systems. In a single-zone ventilation system, one air handler supplies one ventilation zone with a mixture of outdoor air and recirculated return air. Single-zone rooftop units, packaged terminal air conditioners, classroom unit ventilators, and so on, are single-zone systems. For these systems, Addendum 62*n* defines required outdoor air intake flow as equal to the required zone outdoor airflow according to Equation 6-3 ($V_{ot} = V_{oz}$). Apparently for simplicity, this equation does not account for supply-duct leakage, which would increase V_{ot}, or recirculation of outdoor air that bypasses the breathing zone, which would reduce V_{ot}.

Dedicated Outdoor Air Systems. In a "dedicated" outdoor air system (DOAS), called a "100%-outdoor air system in Addendum 62*n,* one air handler serves the ventilation requirements of one or more ventilation zones, delivering the appropriate minimum outdoor airflow — without recirculated return air — to each zone. Terminal units (e.g., fan-coil units, water-source heat-pumps, or even chilled ceiling panels) handle the thermal loads within each zone. (Some designers call this type of ventilation system a "hybrid system," because it comprises a dedicated unit for ventilation air-handling and separate terminal units to handle thermal loads in occupied zones.) Like single-zone systems, all outdoor air entering the central air handler (assuming negligible duct leakage) reaches the ventilation zone diffusers. The required intake airflow is defined as the sum of the zone outdoor airflow values, according to Equation 6-4 ($V_{ot} = \Sigma V_{oz}$). Again for simplicity, the equation does not account for supply-duct leakage, which

Air Distribution Configuration	E_z
Ceiling Supply of Cool Air	1.0
Ceiling Supply of Warm Air and Floor Return	1.0
Ceiling Supply of Warm Air At Least 8°C (15°F) Above Space Temperature and Ceiling Return	0.8
Ceiling Supply of Warm Air Less Than 8°C (15°F) Above Space Temperature and Ceiling Return Provided That the 0.8 m/s (150 fpm) Supply Air Jet Reaches to Within 1.4 m (4.5 ft) of Floor Level.	1.0

Table 2: Zone air distribution effectiveness (E_z) for several configurations. Not all configurations from Table 6.2 are listed here.

Making Sense of Additivity

ASHRAE Standard 62 specifies minimum ventilation rates that are intended to result in indoor air that's free of harmful concentrations of known contaminants and that satisfies the senses of at least 80% of the occupants. (Though the rates prescribed in the standard seem to be based primarily on dilution of odors and irritants, they are presumed to be sufficient to adequately dilute potentially harmful contaminants as well.) Occupant satisfaction relates to the perceived intensity of odors and/or irritants from various indoor contaminant sources. These contaminants originate both from occupants (and their activities) and from the building (and its furnishings). While the relationships are complex, most experts agree that adding the outdoor airflow needed to dilute one odor or irritant to that needed to dilute another generally is the best simple model for dilution of odor and irritation effects. Accounting for the "additive" effect of contaminant sources really isn't new. Since 1989, the standard did so behind the scenes: Dilution rates for building-related contaminants were added to the per-person dilution rate for each occupancy category. For example, the standard previously required 20 cfm per person for offices: 15 cfm to dilute people-related odors and an additional 5 cfm to dilute building-related odors.

Using Equation 6-1 of Addendum 62*n*, engineers now can independently account for people-related and building-related contaminants using two ventilation rate requirements: one rate per occupant (cfm/person) and the other per unit of occupiable floor area (cfm/ft²). To determine the required ventilation, simply multiply the per-person rate by the number of people in the space and the per-unit rate by the floor area; then, add the resulting airflow values together.

					Cooling		Heating	
Ventilation Zone	People Outdoor Air Rate	Zone Population	Area Outdoor Air Rate	Zone Floor Area	Zone Ventilation Efficiency	Zone Outdoor Airflow	Zone Ventilation Efficiency	Zone Outdoor Airflow
	R_p	P_z	R_a	A_z	E_z	V_{oz}	E_z	V_{oz}
	cfm/person		cfm/ft^2	ft^2		cfm		cfm
South Offices	5	20	0.06	2,000	1.0	220	0.8	275
West Offices	5	20	0.06	2,000	1.0	220	0.8	275
North Offices	5	20	0.06	2,000	0.9	244	0.8	275
East Offices	5	20	0.06	2,000	1.0	220	0.8	275
Interior Offices	5	100	0.06	20,000	1.0	1,700	0.8	2,125
North Conference Room	5	14.4*	0.06	2,000	0.9	213	0.8	240
South Conference Room	5	23.1**	0.06	3,000	0.9	328	0.8	369
Total Zone-Level Outdoor Airflow					ΣV_{oz} =	3,150	ΣV_{oz} =	3,830
Single-Zone Systems: Total Intake Air							ΣV_{oz} =	3,830
100%-Outdoor-Air System							V_{ot} =	3,830

* Average population (72% of 20-person peak population)
** Average population (77% of 30-person peak population)

Table 3: A ventilation design example with calculations for a small office building.

would increase V_{ot}. And, even though Equation 6-4 increases V_{ot} to account for breathing-zone bypass (when $E_z < 1.0$), non-recirculating systems cannot reuse this bypassed air or any "unused" outdoor air from partially occupied zones; as a result, in some cases, dedicated outdoor air systems may actually require more intake airflow than multiple-zone systems.

Multiple-Zone Recirculating Systems. In multiple-zone recirculating systems, such as constant-volume reheat systems and nearly all varieties of VAV systems, one air handler supplies a mixture of outdoor air and recirculated return air to two or more ventilation zones. The required outdoor air intake flow can only be determined by properly accounting for *system ventilation efficiency*. The reason for this is because the intake airflow must be sufficient to ventilate the *critical zone*, which is the zone that requires the highest fraction of outdoor air in its primary airstream. Since a multiple-zone system delivers the same primary air mixture to each ventilation zone, proper minimum ventilation in the critical zone overventilates all other zones. As a result, some "unused" outdoor air recirculates (reducing required V_{ot}) while some leaves the building via the relief, exhaust, and exfiltration air streams (increasing required V_{ot}). Addendum 62*n* recognizes this behavior and accounts for it by incorporating system ventilation efficiency in the ventilation calculations.

System ventilation efficiency (E_v) can be determined using the default maximum values found in Addendum 62*n*, Table 6.3. These default values are based on the critical-zone ventilation fraction, found using Equation 6-5 ($Z_p = V_{oz}/V_{pz}$). Alternatively, system ventilation efficiency can be determined using the more accurate calculation procedure found in normative Appendix G of Addendum 62*n*. In either case, having found E_v, outdoor air-intake flow (V_{ot}) for the multiple-zone system must be determined using Equations 6-6, 6-7 and 6-8. Equation 6-6 $\left(V_{ou} = D \times \Sigma(R_p \times P_z) + \Sigma(R_a \times A_z)\right)$ establishes the required *uncorrected outdoor air-intake flow* while allowing the designer to account for system population diversity determined with Equation 6-7 ($D = P_s / \Sigma P_z$), provided breathing-zone outdoor airflow is based on peak (not average) zone population. Equation 6-8 ($V_{ot} = V_{ou}/E_v$) finds the *minimum outdoor air-intake flow* by dividing the uncorrected outdoor air intake flow by the system ventilation efficiency.

Ventilation Design Examples

To demonstrate the use of the design procedure prescribed in Addendum 62*n*, this article considers the design of two ventilation systems—a single-zone system and a 100%-outdoor air system. Each system is applied in two different building types: an office and a school. (Future articles will discuss the design procedure for multiple-zone ventilation systems.)

An Office Building

Let's review the design of a ventilation system for a small office building that contains the ventilation zones described in *Table 3*.

We began by looking up the outdoor air rates (R_p and R_a) in Table 6.1 and then used Equation 6-1 to find the breathing-zone outdoor airflow (V_{bz}) for each zone. In our example, we used the peak population as the expected occupancy for each office area. For the conference rooms, however, we elected to use the optional population-averaging approach described in Section 6.2.5. We used Table 6.1 default occupant densities for some zones and significantly higher densities (which may be more typical in individual offices) for others. According to a recent industry report,[3] average occupant density for some office workers exceeds 15 people per 1,000 ft^2 — significantly higher than the default density of five people per 1,000 ft^2 in Table 6.1.

Next, we established the zone air-distribution effectiveness (E_z)

for each zone based on the supply-air distribution configuration and the values shown in Table 6.2 (*Table 2*). Although the use of these values appears to be mandatory, it's reasonable to consider them as defaults, which are to be used whenever actual values are unknown. Higher-than-default values decrease required intake airflow and should be used cautiously (when justified by measurement or experience), since underventilation could result. Lower-than-default values, which should be used if more-than-typical bypass is expected, increase required intake airflow and may increase first cost and operating cost. For instance, if the ceiling-mounted supply diffusers and return grilles are tightly spaced, it may be reasonable to assume a lower E_z value during cooling, as we did for the north offices and the conference rooms in our example.

In our example office, the zone air-distribution effectiveness for heating is less than for cooling—i.e., $E_z = 0.80$ (or less) vs. 1.00. That's because our design uses ceiling supply and return, and delivers supply air that's warmer than 90°F (32°C) during heating operation. The lower E_z value for heating accounts for the tendency of warm supply air to float above the cooler, denser breathing-zone air.

We then used Equation 6-2 to find each zone's minimum outdoor airflow (V_{oz} in *Table 3*) during cooling and heating.

At this point, we knew the *zone*-level outdoor airflow requirements, but how much outdoor airflow must enter the building at the intake(s)? The answer varied with the type of ventilation system selected.

Single-Zone System Design. Initially, we assumed that each ventilation zone was served by a single-zone, constant-volume rooftop unit, making it necessary to use Equation 6-3 to find the outdoor air intake flow (V_{ot}) required at each rooftop unit. Because we also assumed that all intake air reaches the supply diffusers (no significant duct leaks), the minimum intake airflow for each rooftop unit equals the minimum zone outdoor airflow ($V_{ot} = V_{oz}$) for the ventilation zone it serves. As shown in *Table 3*, the highest intake

Averaging Zone Population for Ventilation Design

In earlier versions of the standard, only "intermittent occupancy" zones (at peak population for three hours or less) could be designed for ventilation at the average population (but not less than one-half of peak population). Now, any zone may be *designed* for average population.

According to Addendum 62*n*, Section 6.2.5, the system must be designed to deliver the required outdoor airflow to each occupied breathing zone, but the design may be based on averages, in some cases. For instance, if occupancy (or supply airflow or intake airflow) varies, ventilation system design may be based on average population over a specific time period rather than on peak population. The averaging time T for a given zone is determined according to Equation 6-9, using zone volume and the breathing-zone outdoor airflow that would be needed at peak population. This "averaging" concept replaces the traditional population-averaging approach for intermittent occupancy.

Why averaging time T? The indoor concentration of any contaminant can be modeled using a first order ordinary differential equation. As most engineers vaguely remember, the solution to such an equation has the form ($C_i = C_o - e^{-t/\tau}\ldots$). Space time-constant is space volume divided by the outdoor airflow rate ($\tau = v/V_o$). In response to a "step-change" in contaminant sources, the concentration in the space rises exponentially, reaching 95% of its steady state value after three time constants. So, Equation 6-9 finds a three-time-constant averaging period, giving the zone a reasonable chance to respond to various short-term conditions in the space.

Averaging time may be applied to make design adjustments when changing conditions in the zone can be predicted. For instance, if fluctuations in zone population can be predicted, design breathing zone outdoor airflow may be calculated based on the highest average population over any T hour period. *Table 4* shows estimated population profile, averaging time and average population for several of the example zones used in this article. A cautionary note: overly aggressive zone-population averaging applied with the new, lower prescribed breathing-zone rates can sometimes lead to very low intake rates and inadequate ventilation.

Ventilation Zone	Zone Floor Area	Zone Pop.	Breathing-Zone Outdoor Airflow	Ceiling Height	Averaging Time	Percentage of Peak Population by Time Interval										Avg. Zone Pop.
	A_z	P_z	V_{bz}		T	Operating Time "Blocks"									Avg.	
	ft²	People	cfm	ft	hr	8–9	9–10	10–11	11–12	12–1	1–2	2–3	3–4	4–5		People
Example Office	1,500	10	140	10	5.5*	100	100	100	100	60	80	80	100	100	91**	9.1
North Conference Room	2,000	20	220	10	4.5	50	50	50	100	100	50	50	50	50	72	14.4
South Conference Room	3,000	30	270	10	3.7	50	50	50	100	100	50	50	50	50	77	23.1
Art Classroom	2,000	40	760	12	1.6	75	75	50	100	0	0	100	0	0	81	32.4
Multiuse Assembly Area	3,000	300	2430	20	1.2	0	0	0	100	50	0	75	50	0	92	276

* For example, $T = 3\times v/V_{bz} = 3 \times 1500 \times 10/140 = 321$ minutes or about 5.5 hours.

** For example, by inspection, highest average population as a percentage of peak = $(4 \times 100 + 1 \times 60 + 0.5 \times 80)/5.5 = 91\%$, so $P_z\text{-}avg = 91 \times 10/100 = 9.1$ people.

Table 4: Estimated population profile, averaging time, and average population for several of the example zones used.

airflow needed for each zone occurs during heating operation. The resulting total outdoor airflow for the system is 3,830 cfm.

Heating operation requires more intake airflow than cooling because we assumed that the supply-air temperature during heating is quite high, and that the discharge velocity of the diffusers is quite low. If each rooftop unit has only one minimum setting for the outdoor air damper, we would set it for the higher minimum airflow and size the heating and cooling coils for the corresponding percentage of outdoor air. If the rooftop can accommodate two minimum settings, however, we could reduce intake airflow to the lower value during cooling operation, thereby reducing the percentage of outdoor air and the required cooling coil capacity.

An even better approach might be to simply lower the supply-air heating temperature to 90°F (32°C) or less while increasing the diffuser discharge velocity. According to Table 6.2, this would raise the maximum zone–air-distribution effectiveness to 1.0 and reduce the heating intake airflow for each zone to match the cooling intake airflow. The benefits include less intake airflow year-round, only one minimum outdoor air-damper setting, and increased comfort since reduced discharge temperature and longer throws result in less vertical temperature stratification.

Dedicated Outdoor Air-System Design. To determine the effect of a different ventilation system on outdoor air intake flow, we replaced the single-zone systems for our office with a dedicated (100%) outdoor air system. In this case, we assumed each ventilation zone is served by a single-zone, constant-volume heat pump (with no outdoor air intake). We also assumed all outdoor air is delivered directly to the ceiling mounted heat pumps from a central, constant-volume, dedicated outdoor air unit. This dedicated unit preconditions the outdoor air and delivers it through a ventilation duct system to each heat pump. Each heat pump, in turn, delivers both outdoor air and locally recirculated air from the plenum to the ventilation zone it serves.

For this system configuration, we found outdoor air intake flow (V_{ot}) at the dedicated outdoor air unit using Equation 6-4. Assuming negligible duct leakage, all intake air reaches all supply diffusers, so minimum intake airflow simply equals the sum of the minimum zone-outdoor airflow values.

In this example, we assumed that the preconditioned outdoor air mixes with local return air, and that the mixture is then cooled or heated by the heat pump before it enters the ventilation zone. In this configuration, the highest of the heating or cooling V_{oz} value for each zone must be used to find V_{ot}. With the values for zone air-distribution effectiveness shown in *Table 3*, and assuming that all zones are in heating, as might be the case in cold weather for the first few hours of operation, we found that the dedicated outdoor air unit must handle 3,830 cfm of outdoor air.

In a system with several ventilation zones, it might seem reasonable that accounting for system population diversity could lower the intake requirement. This is not the case, however. Fluctuations in zone population can be incorporated (via

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

A_z is zone floor area, the net occupiable floor area of the zone, ft² (m²)

D is occupant diversity, the ratio of system population to the sum of zone populations

E_v is ventilation efficiency of the system

E_z is air-distribution effectiveness within the zone

P_s is system population, the maximum simultaneous number of occupants in the area served by the ventilation system

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)

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)

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)

T is averaging time period, minutes

v is ventilation-zone volume, ft³ (m³)

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)

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

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

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)

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

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}.

Ventilation Zone	People Outdoor Air Rate	Zone Population	Area Outdoor Air Rate	Zone Floor Area	Cooling: Zone Ventilation Efficiency	Cooling: Zone Outdoor Airflow	Heating: Zone Ventilation Efficiency	Heating: Zone Outdoor Airflow
	R_p	P_z	R_a	A_z	E_z	V_{oz}	E_z	V_{oz}
	cfm/person		cfm/ft²	ft²		cfm		cfm
South Classrooms (Age 9+)	10	140	0.12	4,000	1.0	1,880	0.8	2,350
West Classrooms (Age 9+)	10	140	0.12	4,000	1.0	1,880	0.8	2,350
North Lecture Classrooms	7.5	260	0.06	4,000	1.0	2,190	1.0	2,190
East Lecture Classrooms	7.5	260	0.06	4,000	1.0	2,190	1.0	2,190
Interior Offices	5	5	0.06	1,000	1.0	85	0.8	106
North Art Classroom	10	32*	0.18	2,000	1.0	680	0.8	850
South Multiuse Assembly	7.5	276**	0.06	3,000	1.0	2,250	1.0	2,250
Total Zone-Level Outdoor Airflow					ΣV_{oz} =	11,200	ΣV_{oz} =	12,300
Single-Zone Systems: Total Intake Air							ΣV_{oz} =	12,300
100% Outdoor Air System					V_{ot} =	11,200		

* Average population (81% of 40-person peak population)
** Average population (92% of 300-person peak population)

Table 5: Ventilation calculations for example school building served by single-zone system and 100% outdoor air system.

averaging) into zone ventilation calculations, but population diversity for the entire system cannot. In other words, even though the system as a whole never contains the sum-of-zones design population, each zone must be ventilated as though it is occupied at design (peak or average) population. A dedicated outdoor air system does not receive credit for recirculated outdoor air from over-ventilated zones, and typically does not modulate ventilation airflow to the zones served. When determining intake airflow for the dedicated outdoor air unit, each zone must be considered to be at design occupancy.

A School Building

Next, let's review two ventilation system designs for a different type of building: a school that contains the ventilation zones described in *Table 5*.

As before, we followed the three-step procedure in 62*n* to find the zone-level ventilation requirements. To do so, we first used Equation 6-1 and Table 6.1 to find V_{bz} for each zone. For zone population, we used the expected peak occupancies for the classrooms and office areas, but we used average population for the art classroom and multiuse assembly area.

Second, we established the zone air-distribution effectiveness for each zone configuration, based on the values in Table 6.2 (see *Table 2*). We assumed that supply air during heating is less than 90°F (32°C), but we also assumed that the classroom and office diffusers do not provide sufficient velocity to deliver 150 fpm (0.75 m/s) air at the 4.5-ft (1.4 m) level. Therefore, we used E_z = 1.0 for the lecture rooms and assembly area, but we used an E_z value of 0.8 for the classrooms and offices during heating operation.

Third, we used Equation 6-2 to find the minimum outdoor airflow needed for each zone in both cooling and heating modes.

Because the minimum required outdoor air intake flow depends on the type of ventilation system, we again looked at two system types.

Zone Air-Distribution Effectiveness

All supply air leaving the ventilation zone diffusers may not actually arrive in the breathing zone. During heating operation, for instance, some portion of warm air from an overhead diffuser may simply float on cooler air in the space and never drop into the breathing zone (between 3 and 72 in. from the floor). Or, during cooling operation, some portion of cool air from an overhead diffuser may "short circuit" to a nearby (poorly placed) return grille, never reaching the breathing zone.

The fraction of the air supplied to a space that actually reaches the breathing zone can be characterized by zone air-distribution effectiveness (E_z). It is similar to "air-change effectiveness" (ACE) for well-mixed spaces, which can be determined in the lab using the methods in ANSI/ASHRAE Standard 129, *Measuring Air Change Effectiveness*. For laminar flow spaces, like some displacement and underfloor systems, ACE depends on ceiling height, so it is not necessarily a good indicator of the fraction of supply air reaching the occupants. (Both ACE and E_z values may be high with a nine-foot [2.75 m] ceiling, but with a 30-foot [9 m] ceiling, ACE would be higher than E_z, not equal to it; using an inappropriately high value [that is, E_z = ACE], would lead to underventilation when using Equation 6-2 to find zone outdoor airflow.)

Note that "air diffusion performance index" (ADPI), defined in the ASHRAE Handbook and ANSI/ASHRAE Standard 113, *Method of Testing for Room Air Diffusion,* is not directly related to air-distribution effectiveness; a poorly performing diffuser that dumps cold air onto an occupant's head has a low ADPI but a high E_z value.

Single-Zone System Design. First, we assumed that each ventilation zone is served by a single-zone, constant-volume unit ventilator or rooftop unit. Equation 6-3 ($V_{ot} = V_{oz}$) is used to find the outdoor air intake flow needed at each unit; it simply equals the minimum zone outdoor airflow. As shown in *Table 5*, some zones need more intake airflow in heating mode than in cooling because of lower zone air-distribution effectiveness. The total outdoor airflow for the system in this case is 12,300 cfm. As mentioned earlier, lowering supply air temperature when heating, and/or selecting better diffusers can reduce the required intake airflow (and unit capacity) as well as increase comfort by reducing vertical temperature stratification.

Dedicated Outdoor Air-System Design. Next, we assumed that each ventilation zone is served by a single-zone, constant-volume blower-coil unit (or a recirculating air handler) without an outdoor air intake. Further, we assumed that all outdoor air enters the system through a central, constant-volume, dedicated outdoor air unit. The entering outdoor air is preconditioned, so it is dry and cool (or neutral during heating operation), and is ducted to and discharged directly into each zone through "ventilation" diffusers. Given the system configuration, we used Equation 6-4 ($V_{ot} = \Sigma V_{oz}$) to find the minimum required outdoor air intake flow at the dedicated outdoor air unit; it's simply the sum of the minimum zone-outdoor airflow values

For this school example, preconditioned outdoor air enters each zone directly, and it is never "hot." We ignored the somewhat higher *heating* values and instead used the *cooling* V_{oz} values, which means that the dedicated outdoor air unit must handle 11,200 cfm of outdoor air, a little less total intake airflow than we needed using single-zone recirculating systems. As in the previous example, we cannot account for system population diversity in this constant-volume system, even though peak population does not occur in all zones simultaneously.

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What About Part-Load Operation?

In the preceding discussion, we found the minimum "design" outdoor air intake flow for two different ventilation systems in an example office and an example school. Operationally, however, ventilation load varies with changes in supply-air temperature and population. Addendum 62*n* explicitly permits dynamic reset of intake airflow so ventilation capacity can be matched to ventilation load. This can save outdoor air-conditioning energy during periods of low ventilation load and avoid underventilation problems during periods of high ventilation load. Because "demand-controlled ventilation" is a broad topic, it will be addressed separately in a future article. Suffice to say, single-zone systems may be cycled on and off, or the intake damper minimum setting may be modulated in response to population changes. But, in dedicated outdoor air systems, the 100% outdoor air unit must deliver design ventilation air to all zones whenever any zone is occupied, unless individual zones include modulating dampers to control outdoor airflow based on zone ventilation demand. Note that the ASHRAE energy standard[4] requires energy recovery on 100% outdoor air units delivering 5,000 cfm (2360 L/s) or more; a dedicated outdoor air system with energy recovery may use less energy than multiple single-zone systems without energy recovery.

Anything Else?

For zones with strong contaminant sources, Addendum 62*n* prescribes minimum exhaust airflow to remove contaminants from the building. Minimum exhaust rates for nineteen zones are prescribed in Table 6.4 (not included here). This may be regarded as a significant change from Standard 62-2001, which specified exhaust rates for only a few zones and did not clarify whether it was just minimum exhaust flow, or both minimum exhaust and outdoor air supply flow (e.g., public restrooms). While most zones require either one or the other, some zones — such as art classrooms — require both minimum supply and exhaust airflow. Makeup air from outdoors (to replace exhausted air) may be supplied to a zone by any combination of first-pass outdoor air or outdoor air transferred from other zones.

Summary

The Addendum 62*n* "heart transplant" certainly changes the ventilation standard. There are no guarantees, but these expectations seem reasonable: that the generally lower breathing-zone rates will reduce some designer complaints about overventilation, that people- plus building-ventilation "additivity" will reduce effective per-person ventilation in high-density spaces but increase it in low-density spaces, and that the prescribed calculation procedures will reduce underventilation in some systems while increasing calculation consistency among designers. By changing the prescribed rates and at the same time encouraging better design procedures, Addendum 62*n* may lower ventilation system costs for some systems and improve indoor air quality in others. Time will tell.

References

1. ANSI/ASHRAE Standard 62-2001, *Ventilation for Acceptable Indoor Air Quality*.
2. ANSI/SHRAE Addendum *n* to ANSI/ASHRAE Standard 62-2001.
3. International Facility Management Association. 2004. *Project Management Benchmarks Report*. In "IFMA Research Shows Office Space Shrinking." Cited Aug. 20, 2004. Available from: www.ifma.org/about/prdetail.cfm?id=331&actionbig=3&actionlil=153 .
4. ANSI/ASHRAE Standard 90.1-2001, *Energy Standard for Buildings Except Low-Rise Residential Buildings.*