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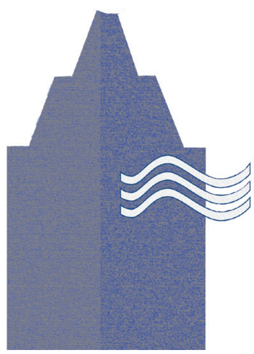
ous building types, we narrow our discussion to a single-duct VAV system, with throttling VAV boxes for interior zones and reheat VAV boxes in perimeter zones, applied in an example office building.

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.

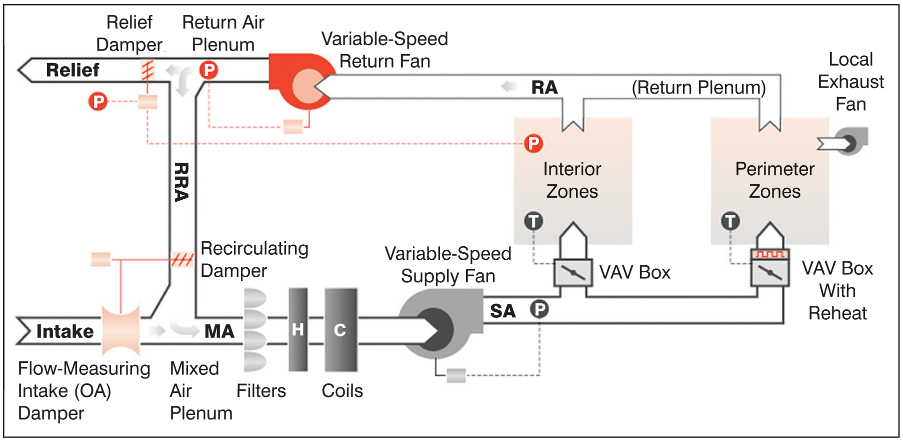


Figure 1: Variable air volume reheat system.

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.

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.

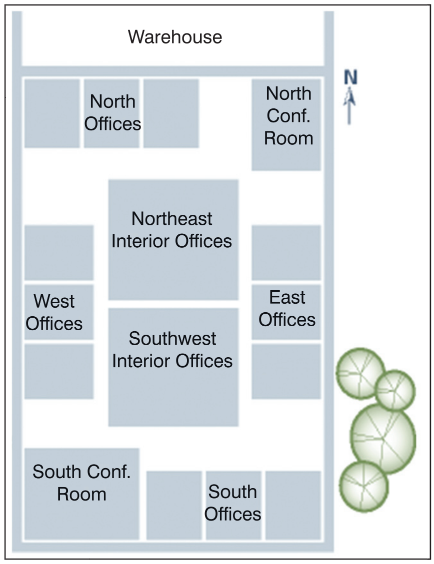


Figure 2: Multiple-zone office building.

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.

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.

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.

		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	$Z_{d\text{-}clg}$		$E_{vz\ clg}$	
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	$Z_{d\text{-}htg}$		$E_{vz\ htg}$	
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.

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 overventilation. 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 changeover-bypass VAV systems." ASHRAE Journal 46(11):22–32.