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Use of Air Cleaners to Reduce Outdoor Air Requirements

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ABSTRACT

The proposed revision to ASHRAE Standard 62-1981, "Ventilation for Acceptable Indoor Air Quality," recommends a minimum of 15 cfm of outdoor air per person. This amount is needed to control occupant odors and guarantee that the concentration of carbon dioxide will not exceed 1000 ppm. Additionally, other recognized contaminants, including formaldehyde, office products, building materials, and tobacco smoke, will be maintained at acceptable levels. Most applications (i.e., offices) where the above contaminants can be expected to be found generally require more outdoor air. Air-cleaning systems that effectively remove the major contaminants can reduce the amount of outdoor air required. However, this generally requires an increase in the amount of recirculated air. A model is developed and equations are presented for calculating the amount of outdoor air required, space concentration of filtered contaminants, or the amount of recirculation needed. These parameters are dependent on the type of air distribution system (VAV or constant volume), supply temperature (constant or variable), and the use of outdoor air (constant or proportional). Also required are the air cleaner efficiency, ventilation efficiency, recirculation factor, and the flow reduction factor (with VAV systems). Sufficient design of air cleaning systems can reduce the amount of outdoor air required.

INTRODUCTION

Although submarines and spacecrafts operate as sealed environments, it is uneconomical and impractical to operate buildings in this manner. Ventilation with outdoor air is needed to replace the oxygen consumed and to dilute contaminants to an acceptable level. At the very least, carbon dioxide (CO_2) exhaled by occupants must be diluted to an acceptable level. According to ASHRAE Standard 62-1981 (ASHRAE 1981), the acceptable CO_2 level is 0.25% (2500 ppm). Research conducted since this standard was published has shown that 0.25% CO_2 in the indoor environment is too high. Occupant-generated

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odors are likely to develop (Berg-Munch et al. 1984) and the occupants will find the indoor atmosphere "stuffy" (Janssen and Wolff 1986). The World Health Organization, most foreign countries, and ASHRAE now agree that the CO_2 limit should be 0.10%. This increases the minimum outdoor air ventilation rate from 5 cfm per person, as specified in Standard 62-1981, to 15 cfm per person (Janssen 1987). The presence of other contaminants, such as tobacco smoke, increases the dilution air needed to as much as 60 cfm per person in smoking lounges. Where air-cleaning systems can be used to remove the contaminants from the return air, this air can be substituted for some of the outdoor air, thus reducing the energy needed to condition the ventilating air. However, there are interactions among the system variables that must be considered in designing an air-cleaning system.

DESIGN CONSIDERATIONS

A number of design considerations affect the performance of an air-cleaning system. The type of air cleaner used, its location in the system, and the amount of air passed through the air cleaner all have a direct bearing on the amount of reduction in contaminants.

Air-Cleaning System Selection

Particles are the easiest contaminants to remove from the air. General types of particle air cleaners are (1) media filters, (2) electrostatic air cleaners, (3) adsorbers/absorbers, (4) centrifugal separators, and (5) air washers. Centrifugal separators (cyclone) are dependent on the difference between the density of the air and the density of the particles and on the particle size. Very small particles have a relatively large surface area for their mass with increased aerodynamic drag in a moving airstream. Cyclone separators, therefore, are ineffective in removing very small particles and are used mainly for dust removal in industrial applications. Air washers are also used in industrial applications and in desiccant cooling systems. Chemical reagents can be used as the washing fluid to remove gases such as sulfur dioxide (SO_2) from power plant boiler gases. Air washer systems are used in atomic submarines to remove carbon dioxide (CO_2); however, the high cost and need for regular maintenance make air washer systems unattractive for use in commercial buildings.

Media filters and electrostatic air cleaners are used in building HVAC systems. Media filters consist of a porous material, frequently fibrous glass, through which the air is forced. The small air passages cause particles to be "strained out" by impact on the filter media. These filters range in performance from the dust-stop filters used in most warm-air furnaces to remove lint and large dust particles to high efficiency particulate air (HEPA) filters that can remove submicron particles. The collected particulate matter on media filters alters their performance. Filtration effectiveness tends to increase with filter loading, but the increased pressure loss across the filter reduces the airflow. Thus, while filtration efficiency may increase with use, the amount of air filtered tends to decrease. Media filters require regular maintenance with cleaning or replacement.

Activated carbon and activated alumina are adsorbers or absorbers. Activated carbon has a very large surface area due to the fine voids in its structure. Filtration of gases occurs by adhesion of gas molecules to the surface of the carbon structure; thus activated carbon can remove certain gases and vapors. In general, the effectiveness is related to the size of the molecules. Activated carbon is quite effective in removing odor molecules, which are large, but is ineffective in removing small molecules such as carbon monoxide (CO).

Activated alumina is a porous material similar to activated carbon. Since alumina is considerably more expensive, it is not usually used in the pure form. Activated alumina can be saturated with potassium permanganate, however, to be used as a chemically active air cleaner. Potassium is a strong oxidizing agent that is very effective in oxidizing odor molecules. Activated alumina treated with potassium permanganate is very effective in removing formaldehyde, which is converted to carbon dioxide and water.

Activated carbon and alumina require regular replacement. Activated carbon can be regenerated with steam; however, this is not considered to be a viable process for building HVAC system filters. The potassium permanganate in activated alumina filters is a consumable reagent, and the filter must be replaced when the reagent is exhausted.

Electrostatic air cleaners offer many desirable features. Particles passing through the charging section of the air cleaner receive an electric charge, which causes them to be attracted to oppositely charged plates in the collector section. Electrostatic air cleaners (EAC) can have a very high efficiency for collecting particles in the range of 0.01 micron to 5 microns. EACs are quite open and have a very small pressure loss. Regular cleaning is required to remove the collected material from the collector plates, which is accomplished by washing the collector section of the filter. The whole filter is removed and washed in residential and small commercial filters. Wash-in-place designs are available in larger electrostatic filters.

Although electrostatic air cleaners are mainly effective in removing particulate matter, there is evidence that they are somewhat effective against certain gases and vapors. Radon-decay products tend to attach to particles; thus, when the particles are removed, some radon prodgeny are also removed. Change of particle concentration in the room air changes the balance between the attached and unattached fraction of radon prodgeny. Thus, the effect on health risk is unclear.

The charging voltage in the charging section is high enough to ionize some gases. It is possible that gases such as CO_2 can be ionized and then removed. The removal efficiency is low, but there is evidence that a small amount of certain gases can be removed in this way.

This brief discussion of the different types of air-cleaning systems is presented to give the reader some idea of the options available.

System Factors

The objectives in using an air cleaner to increase the fraction of return air used can be satisfied by putting the air cleaner in one of two different locations; the air cleaner can be placed in the recirculated airstream or the mixed airstream. If the contaminants to be removed are all coming from the conditioned space (i.e., microorganisms, dust, tobacco smoke, and other respirable particles), filtration of the recirculated air is most efficient. However, if the outdoor air is also a source of filterable contaminants, it will be more desirable to filter the supply air. These locations are identified as A and B in Figure 1.

The volumetric flow through the air cleaner and its effectiveness determine the amount of filterable material that can be removed. When the volumetric flow is reduced in a variable air volume (VAV) system, the contaminant removal capacity also will be reduced. This fact is recognized by the flow reduction factor, F_r. The filter effectiveness is given by E_f and must be obtained from the manufacturer. This will be affected by the flow through the filters; thus, the increased effectiveness with reduced flow may compensate to some extent for a reduction in flow. In general, effectiveness increases as velocity through the air cleaner is reduced.

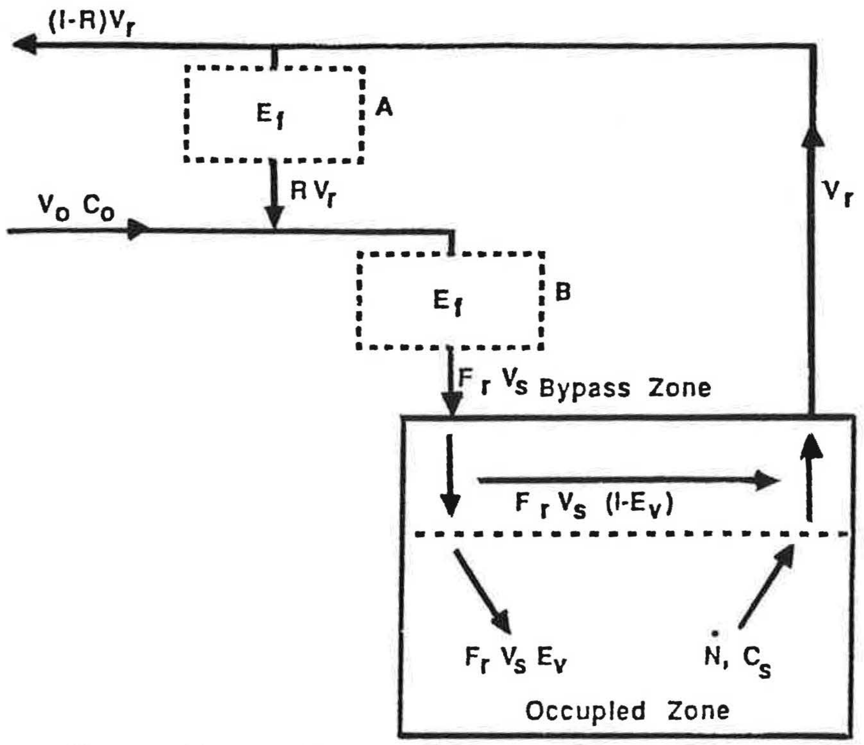


Figure 1 Proposed alternate air cleaner locations (source: proposed draft of ASHRAE Standard 62-1981R)

The recirculation flow factor, R, is the fraction of the return air that is recirculated. As the ventilation load increases, the amount of outdoor air needed to dilute occupant-generated carbon dioxide increases. The recirculation factor will then decrease.

The degree of mixing in the space determines the amount of space-generated contaminants that are contained in the return air. Many building systems are designed with the supply outlets and return inlets both in the ceiling. If the discharge velocity is not high enough and flow is not directed away from the ceiling, a portion of the supply may flow along the ceiling and above the "occupied zone" directly to the return inlet. This will dilute the contaminants in the return flow to a concentration lower than that in the occupied space. When some of this return flow is exhausted, ventilating air will be wasted. This factor is accounted for by the ventilation effectiveness, E_v, which is described in greater detail below.

The contaminant generation rate, $\dot{N}$, must be specified in order to determine the load on the system. If tobacco smoke or other identifiable contaminants are of primary concern, the particle generation rate may be found in the literature (i.e., Leaderer and Cain 1983; Thayer 1982).

The concentration of contaminants, C_o, in the outdoor air must be known or estimated. The concentration of contaminants in the conditioned space, C_s, can be either an independent variable specified by some standard or can be calculated on the basis of the flow variables, V_o and V_r.

A supply airflow can be either constant or variable volume. The flow reduction factor indicates a variable air volume (VAV) system. The supply air temperature can be either constant or variable. If the supply air temperature is constant, the supply air volume must be varied to follow the thermal load. A constant supply air temperature can be used in a VAV system. The outdoor airflow rate may be constant or can be adjusted to be inversely proportional to the supply air volume in a VAV system. The outdoor airflow rate must be maintained sufficiently to keep contaminants within acceptable limits.

Seven different permutations of these design variables are possible. The effects on the mass balance equations are shown in Table 1. These equations can be solved for the required outdoor air, the resulting contaminant concentration in the space, or the recirculation rate required. It will be found that it is generally necessary to increase the recirculation rate in order to minimize the outdoor air required.

Mixing Efficiency

If a fraction, S, of the supply air flows directly from the supply outlet to the return inlet without mixing at the occupied level, and if a fraction, F_r, of the return air is recirculated, it has been shown (Janssen 1987) that the mixing efficiency, E_v, is:

$$E_v = (1 - S)/(1 - F_r S) \quad (1)$$

This has been measured with tracer gas and found to be as low as 50% in a system where the supply air is discharged from a supply diffuser that is mounted on a wall at ceiling height with its flow directed to a return inlet also located at ceiling height on the opposite side of the room (Janssen 1982). The "Coanda" effect keeps the supply attached to the ceiling even under a cooling operation. In another room, with a 17-ft high ceiling and the return inlet located eight feet above the floor, the mixing efficiency for the occupied space was found to be equally poor.

Best mixing is accomplished when the supply outlet is high (in the ceiling) and the return inlet is low (near the floor) or vice versa. The supply and return locations should be properly distributed to avoid "dead spaces" in the room. Rooms with high ceilings are likely to have lower mixing efficiencies when the supply and return locations are both in the ceiling.

Table 1 is used in the revised ASHRAE Standard 62-1981R as an integral part of the air quality procedure. Examination of the relationships for each class in Table 1 shows that the HVAC designer needs to specify the following parameters, namely, the indoor contaminant level, C_s; outdoor contaminant level, C_o; filter effectiveness, E_f; ventilation effectiveness, E_v; flow reduction factor, F_r; return rate, V_r; the recirculation fraction, R; and the contaminant generation rate, $\dot{N}$

Table 2 reflects the results of a recent office study comparing the above referenced outdoor air requirements using the ventilation rate procedure and air quality procedure for various classes and systems for a nominal 22,000 ft² office building located in Burbank, California.

Following is an example of how the air quality procedure values tabulated in Table 2 were computed for office occupancies employing a Class II air distribution system.

Parameters assumed are:

$C_s = 75\ \mu g/m^3$

$C_o = 5\ \mu g/m^3$

$E_v = 0.8$

$E_f = 0.5$

$V_s = 329\ m^3/min$

TABLE 1

Effect of Filter Location and System Parameters on Required Outdoor Air, Space Contaminant Concentration, or Recirculation Rate

Class	Required Recirculation Rate				Required Outdoor Air	Space Contaminant Concentration	Required Recirculation Rate
	Filter Location	Flow	Temperature	Outdoor Air			
I	None	VAV	Constant	100%	$V_o = \frac{\dot{N}}{E_v F_r (C_s - C_o)}$	$C_s = C_o + \frac{\dot{N}}{E_v F_r V_o}$	Not applicable
II	A	Constant	Variable	Constant	$V_o = \frac{\dot{N} - E_v RV_r E_f C_s}{E_v (C_s - C_o)}$	$C_s = \frac{\dot{N} + E_v V_o C_o}{E_v (V_o + RV_r E_f)}$	$V_r = \frac{\dot{N} + E_v V_o (C_o - C_s)}{E_v RE_f C_s}$
III	A	VAV	Constant	Constant	$V_o = \frac{\dot{N} - E_v F_r RV_r E_f C_s}{E_v (C_s - C_o)}$	$C_s = \frac{\dot{N} + E_v V_o C_o}{E_v (V_o + F_r RV_r E_f)}$	$V_r = \frac{\dot{N} + E_v V_o (C_o - C_s)}{E_v F_r RE_f C_s}$
IV	A	VAV	Constant	Proportional	$V_o = \frac{\dot{N} - E_v F_r RV_r E_f C_s}{E_v F_r (C_s - C_o)}$	$C_s = \frac{\dot{N} + E_v F_r V_o C_o}{F_r E_v (V_o + RV_r E_f)}$	$V_r = \frac{\dot{N} + E_v F_r V_o (C_o - C_s)}{E_v F_r RE_f C_s}$
V	B	Constant	Variable	Constant	$V_o = \frac{\dot{N} - E_v RV_r E_f C_s}{E_v [C_s - (1 - E_f) C_o]}$	$C_s = \frac{\dot{N} + E_v V_o (1 - E_f) C_o}{E_v (V_o + RV_r E_f)}$	$V_r = \frac{\dot{N} + E_v V_o [(1 - E_f) C_o - C_s]}{E_v RE_f C_s}$
VI	B	VAV	Constant	Constant	$V_o = \frac{\dot{N} - E_v F_r RV_r E_f C_s}{E_v [C_s - (1 - E_f) C_o]}$	$C_s = \frac{\dot{N} + E_v V_o (1 - E_f) C_o}{E_v (V_o + F_r RV_r E_f)}$	$V_r = \frac{\dot{N} + E_v V_o [(1 - E_f) C_o - C_s]}{E_v F_r RE_f C_s}$
VII	B	VAV	Constant	Proportional	$V_o = \frac{\dot{N} - E_v F_r RV_r E_f C_s}{E_v F_r [C_s - (1 - E_f) C_o]}$	$C_s = \frac{\dot{N} + E_v F_r V_o (1 - E_f) C_o}{E_v F_r (V_o + RV_r E_f)}$	$V_r = \frac{\dot{N} + E_v F_r V_o [(1 - E_f) C_o - C_s]}{E_v F_r RE_f C_s}$

Source: Table 3, proposed draft, ASHRAE Standard 62-1981R

TABLE 2

Office Building
City of Burbank, California

22,000 FT^2 TOTAL FLOOR AREA

AIR DISTRIBUTION SYSTEM				
CLASS	FILTER LOC.	FLOW	TEMP.	OUTDOOR AIR
II	A	CONSTANT	VARIABLE	CONSTANT
III	A	VAV	CONSTANT	CONSTANT
IV	A	VAV	CONSTANT	PROPORTIONAL
V	B	CONSTANT	VARIABLE	CONSTANT
VI	B	VAV	CONSTANT	CONSTANT
VII	B	VAV	CONSTANT	PROPORTIONAL

DESIGN CONDITIONS

C_o = 5 $\mu g/m^3$
E_v = 0.8
F_r = 0.7
E_f = 0.5

CONTAMINATION GENERATION RATE $\dot{N}$ μg/-MIN.-PERSON	
TELECOMMUNICATIONS	$\dot{N}$ = 12
CONFERENCE ROOMS	$\dot{N}$ = 11
OFFICES	$\dot{N}$ = 7
RESTAURANT	$\dot{N}$ = 12
RECEPTION AREA	$\dot{N}$ = 10
CORRIDOORS & LOBBY	$\dot{N}$ = 7

ZONE NAME	ZONE AREA FT^2	NO. OF PEOPLE	VENTILATION RATE PROCEDURE OSA-CFM/PSN	AIR QUALITY PROCEDURE					
				CLASS II (Cs = 75) OSA CFM/PSN	CLASS III (Cs = 125) OSA CFM/PSN	CLASS IV (Cs = 50) OSA CFM/PSN	CLASS V (Cs = 15) OSA CFM/PSN	CLASS VI (Cs = 15) OSA CFM/PSN	CLASS VII (Cs = 35) OSA CFM/PSN
RESTAURANT	5,000	500	20	9.2	5.7	4.0	3.3	4.4	5.6
TELECOMMUNICATIONS	4,000	240	20	7.4	7.7	6.1	5.0	6.8	8.4
RECEPTION AREA	2,000	120	15	12.8	7.9	4.9	4.6	5.7	6.4
CONFERENCE ROOMS	2,200	110	20	23.6	14.6	7.4	6.1	8.1	9.5
OFFICES	7,700	54	20	59.0	36.0	31.8	27.5	34.0	37.0
CORRIDORS & LOBBY	1,100	10	50	40.7	25.2	25.0	21.2	26.0	28.7

$$V_r/V_s = 0.9$$

$$R = 0.8$$

$$\dot{N} = 7\ \mu g/min \cdot m^3$$

$$V_o = \frac{\dot{N} - E_v R V_r E_f C_s}{E_v (C_s - C_o)} \quad (2)$$

$$= \frac{(7 \times 1744) - (0.8)(0.8)(0.9 \times 329)(0.5)(75)}{0.8(75 - 5)}$$

$$= 92\ m^3/min \text{ or } 3255\ cfm$$

for 54 people, V_o is 59 cfm per person.

Table 2 compares the outdoor air requirement using Classes II through VII for various occupied spaces with the values obtained employing the ventilation rate procedure. At this point, we must emphasize the importance of using a consistent unit system. For example, if the outside air rate is given in cubic meters per person, then the contaminant generation rate should be in microgram per cubic meter per minute per person. Also, the assumed value of V_r/V_s should be rechecked now that V_o is known.

It is also evident from Table 1 that V_o would be positive as long as:

$$V_r < \frac{\dot{N}}{C_s \Phi} \quad (3)$$

where

$\Phi = R E_v E_f$ for constant volume systems, and

$\Phi = F_r E_v E_f$ for VAV systems

Estimation of the contamination generation rate, $\dot{N}$, needs careful attention compared to the other system parameters. The internal generation rate ($\mu g/m^3 \cdot min$) varies from 0.4 to 8.5 for particle size ranging from 0.3 micron to 10.0 microns (refer to Table 3). These values were obtained experimentally (Meckler 1985). In the absence of data, the ASHRAE Standard 62-1981 ventilation rate procedure can be used to estimate $\dot{N}$ for the case where no unusual indoor air contaminants (particulates) are known to be present.

For example, for the case of the restaurant occupancy referred to in Table 2, based on the ventilation rate procedure, the required outdoor air for ventilation is 20 cfm per person ($0.57\ m^3/min \cdot person$). Using the relationship for required outdoor air rate (found in Table 1 under Class I), namely,

$$V_o = \frac{\dot{N}}{E_v F_r (C_s - C_o)} \quad (4)$$

with V_o = 20 cfm/person ($0.57\ m^3/min \cdot person$), one can solve directly for $\dot{N}$. Assuming a C_s of 75 $\mu g/m^3$, C_o of 5 $\mu g/m^3$, E_v of 0.8, and F_r of 0.7, the contaminant generation rate, $\dot{N}$, can be calculated to be (0.57)(0.8)(0.7) (75 − 5) or 22.3 $\mu g/min$.

Analysis of the equations in Table 1 suggests that the outdoor air rate required using the filter in position A may be greater than if the same filter is located in position B, for same system operating parameters (i.e., temperature and flow conditions). Generally, an increase in outdoor air rate results if the filter is at position A instead of B and is equal to $(C_o E_f)/(C - C_o)$. For example, if C_s is 75 $\mu g/m^3$ and E_f is 0.8, then for C_o equals 5 $\mu g/m^3$, the increase in outdoor air is $(5 \times 0.8)/(75 - 5)$ or 0.06 (6%). If,

TABLE 3

Representative Values of C_s
for Various Particulate Sizes and Efficiencies (%)

PARTICLE SIZE (MICRONS)	INTERNAL GENERATION - N (MICROGRAMS/M^3 - MIN.)	FILTER EFFICIENCY (DIMENSIONLESS)	SUSPENDED PARTICLE CONCENTRATION C_s (MICROGRAMS/M^3)
0.3 - 0.5	0.4	0.56	26.53
0.5 - 0.7	0.55	0.60	13.515
0.7 - 1.0	0.85	0.67	6.683
1.0 - 3.0	2.0	0.825	2.374
3.0 - 5.0	4.0	0.98	0.224
5.0 - 7.0	6.0	0.995	0.0548
7.0 - 10.0	8.5	0.9975	0.0273

Note: 5 cfm/person = 0.0059 air change/minute of outside air

however, C_o increases to 20 $\mu g/m^3$, the increase in outdoor air becomes 0.18 (18%). When difficult outdoor air conditions are expected (i.e., high C_o) or when low indoor contaminant levels (i.e., C_s) are required to be maintained, lower outdoor air rates will be required if the same filter is positioned at location B instead of A. It should be noted, however, that the unit's fan energy consumption may be lower if the filter is at location A instead of B, particularly if a return or exhaust air fan is positioned between the occupied zone and location A, as illustrated in Figure 1.

It is also evident from Table 1 that the outdoor air rate required for VAV systems using constant outdoor air is less than proportional outdoor air by a factor of F_r. Also, constant flow systems generally require less outdoor air than VAV systems with filters in similar locations, as can be seen from Table 2.

INDOOR AIR QUALITY STANDARD

Present regulations do not require homes, offices, and other commercial or institutional buildings to be monitored for indoor contaminants. The "sick building syndrome" in a building involves people not being able to live or work comfortably without suffering some discomfort. Current indoor air quality standards prescribe minimum air quantities but do not require the air to be effectively distributed within occupied, conditioned spaces. Generally, in office buildings, the supply air outlet and return air inlet are both located at the ceiling (Meckler 1985), which may cause the circulation to be short-circuited and thus leave half of the room volume without proper ventilation through mixing and poor overall air circulation (Meckler 1972c-f). Lately, improved air filters and cleaning systems that remove particulate and gaseous contaminants from outside air and return air have received a great deal of interest from architects and engineers (Meckler 1972a-f).

ASHRAE Standard 62-1981 attempts to address indoor air quality issues, while other ASHRAE standards address indoor air quality from the thermal point of view. The current ASHRAE Standard 62-1981 was originally intended to achieve a balance between energy conservation and the health needs of individuals, but this dual approach has been controversial and not easily accepted. ASHRAE is in the process of revising Standard 62-1981 to increase the minimum outside air to be mixed with the supply air.

Having illustrated the use of the latest draft of the ASHRAE Standard 62-1981R air quality procedure for some representative office occupancies, it may be instructive to examine the following considerations:

1. Under what conditions can placement of one or more air filters at the air-handling (terminal) unit or in adjacent ductwork affect overall air filtration performance?
2. How can the use of air recirculation at part-load enhance the overall performance of VAV air distribution systems in maintaining acceptable indoor air quality within occupied spaces at reduced supply air rates?

One of the methods proposed, for VAV systems, involves the additional filtration and recirculation of ventilating air to provide enhanced overall filtration (NIBS 1985) to compensate for reduced supply airflow. VAV systems should be controlled so that the outside air drawn into the system is in the same proportion to the total amount of air reduced. However, this may be impractical due to increased cost.

Referring to Table 1, it can be seen that the term E_f (which corresponds to the overall net air filtration efficiency of one or more air filters located in positions A and/or B) can have a significant impact on the outdoor air rate, V_o.

It can be seen in Table 1 that the indoor air quality performance of tested VAV systems is particularly sensitive to selection of air cleaners, especially where increased air recirculation is contemplated as a design strategy for lowering outdoor air requirements (i.e., ventilation). Since it has been demonstrated (Meckler 1984, 1985) that VAV air distribution systems may be subject to indoor air quality problems, it may prove instructive to examine how air recirculation and judicious placement of air cleaners can be used to improve indoor air quality conditions in spaces served year-round.

EXAMPLE

Let us assume that we wish to serve the representative temperature control zones illustrated in Figure 3 by a Class III VAV air distribution system (refer to Table 1). Let us further assume that the effectiveness (or efficiency) of the primary air filter proposed by the HVAC designer is designated by E_1. Additionally assume that the designer is also considering some further contaminant control enhancement by adding an additional (i.e., secondary) air filter of effectiveness (or efficiency) E_2, either ahead of the primary air filter in position B, as shown in Figure 1, or at another alternate location. Accordingly, two alternate air filtration scenarios are proposed.

Scenario 1 consists of a ceiling-mounted unitary water-cooled air conditioner (or heat pump) with a constant-speed supply fan equipped with a bypass damper control. This control feature is capable of sensing static pressure buildup in the main supply air duct, as shown in Figure 2. Individual zone control dampers in their respective spaces modulate closed-in response to net space demands for heating and cooling and are commercially available.

To achieve the overall desired VAV airflow rate to all zones served, supply air is automatically bypassed to the inlet, thereby providing a natural path for air recirculation where the volume control damper must add more or less air frictional resistance. Therefore, locating the secondary air filter, E_2, in the bypass duct (provided that the primary air filter, E_1, is located at position B) would improve overall air filtration at no additional energy cost. Figure 3 shows locations of the primary air filter E_1 and the secondary air filter, E_2, arranged in a series/parallel bypass airflow or air recirculation configuration. Figure 4 illustrates an HVAC/space configuration similar to that shown in Figure 3 yet differing in several important details. Note that the terminal VAV unit's air conditioner supply fan is of the variable-speed type responding directly to net space demands for VAV flow, also sensed by static pressure build-up in the main supply air duct, and identified as Scenario 2.

In Scenario 2, the HVAC designer proposes to place a secondary air filter, E_2, in the position shown in Figure 5. Returning now to Scenario 1, notice that whenever less than 100% (i.e., or the design day cooling) supply air is needed, one can directly compute the fraction of the total supply air delivered by the terminal VAV unit fan that will automatically bypass through the secondary air filter, E_2. The quantity (1-X) represents the fraction of the total supply air delivered by the terminal VAV unit supply fan that is actually supplied to the conditioned zones.

Let us now compare Scenario 1, utilizing air recirculation, with Scenario 2, which employs a more conventional approach (i.e., pre- and post-filtration), the Class III, VAV air distribution system discussed earlier.

With respect to the Scenario 1 air filtration approach, as illustrated in Figure 3, one must first evaluate a seasonal average value for X. If we define $E_{p/s}$ as the overall filtration effectiveness for Scenario 1, we arrive at the following useful relationship:

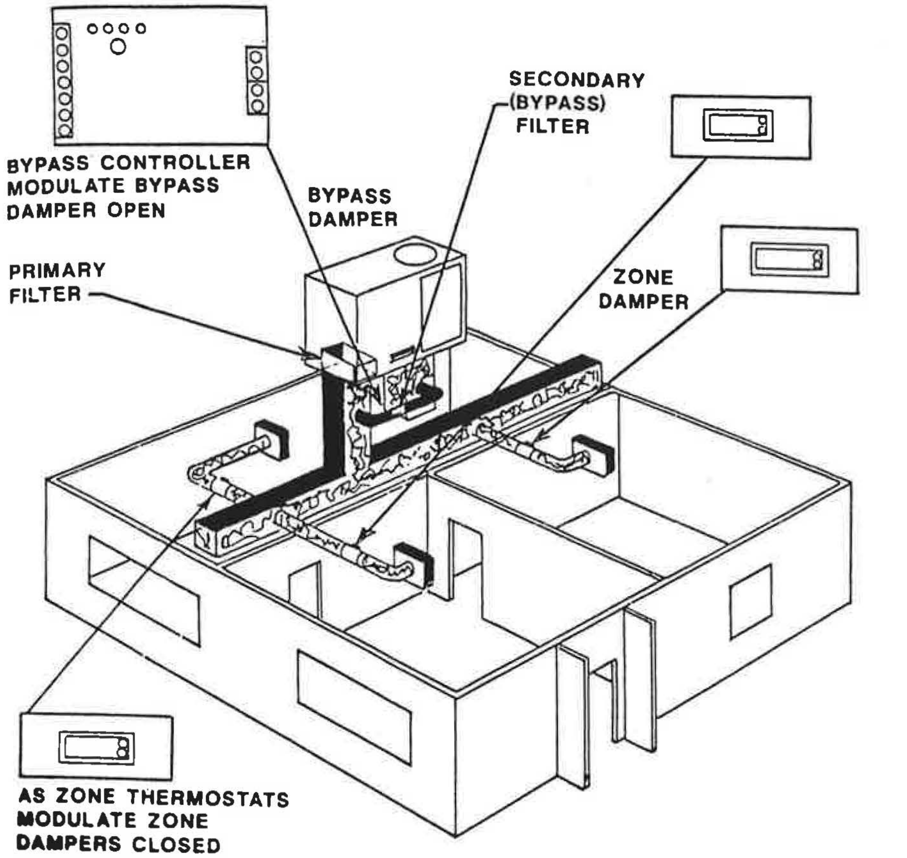


Figure 2 Air-conditioning system with a VAV unit (includes filters in both return and bypass positions); scenario 1: class III VAV air distribution alternate

$$E_{p/s} = 1 - \frac{(1 - E_l)}{1 + X[1 + (1 - E_1)(1 - E_2)]} \tag{5}$$

Similarly, if we define E_s as the overall air filtration effectiveness for Scenario 2, we arrive at a comparable relationship to that illustrated above (i.e., for the VAV series/parallel air recirculation concept) but for Scenario 2 VAV flow only, as follows:

$$E_s = 1 - (1 - E_1)(1 - E_2) \tag{6}$$

Therefore, by solving for either E_s or $E_{p/s}$ as described above, one can directly substitute either value for the term E_f in Table 1. Then, the percentage increase in air filtration effectiveness of Scenario 1 in comparison to Scenario 2 can be determined. For example, one can plot $\{[(E_{p/s} - E_s) \div E_s] \times 100\}$ versus X as shown in Figure 6. Referring to Figure 6, notice that whenever X is greater than or equal to 0.1 some improvement can be expected. For example, at an estimated annual average value of $X = 0.5$ for the spaces served, the corresponding contamination control enhancement due to the pro-

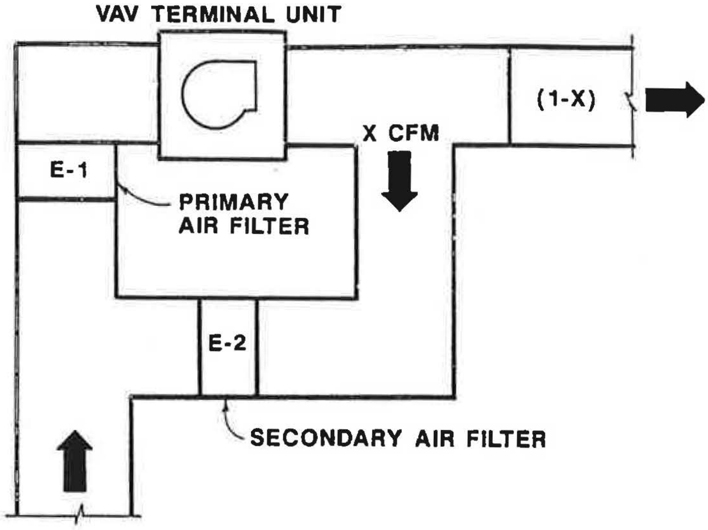


Figure 3 Series/parallel bypass flow air filtration; scenario 1: recycle series/parallel filtration system

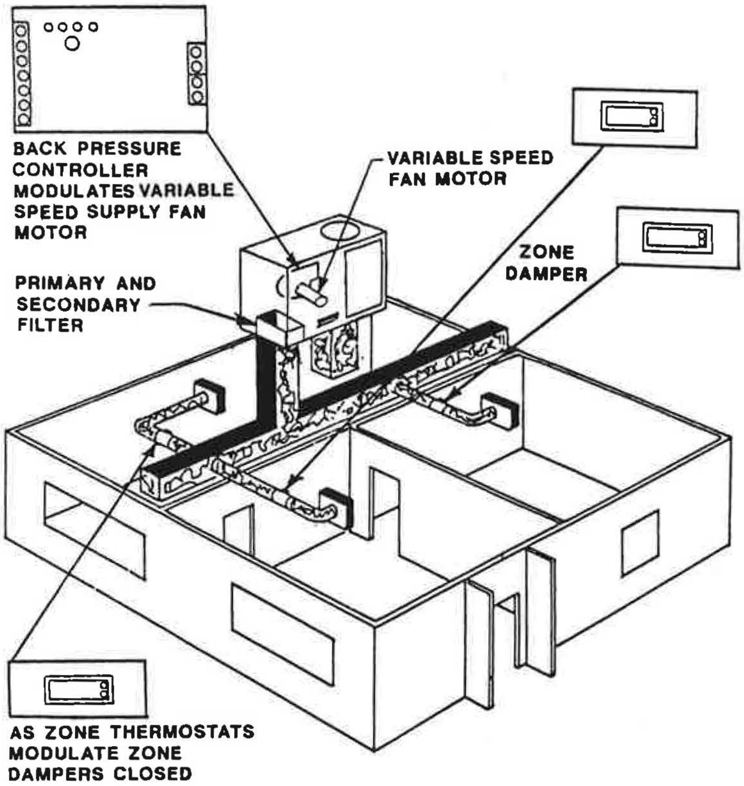


Figure 4 Air-conditioning system with a VAV unit (includes filter in only return air position); scenario 2: class III VAV air distribution alternate

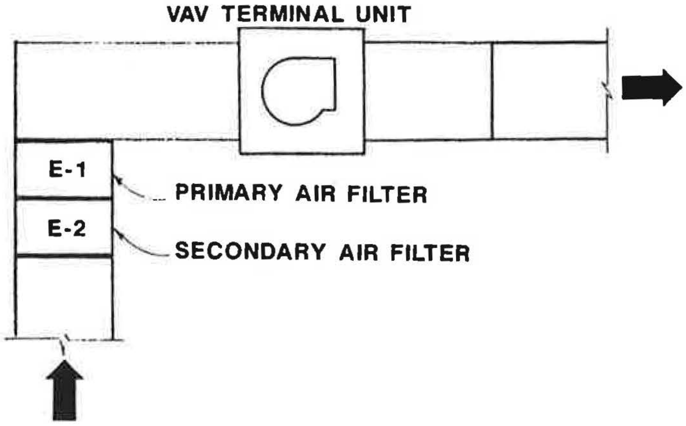
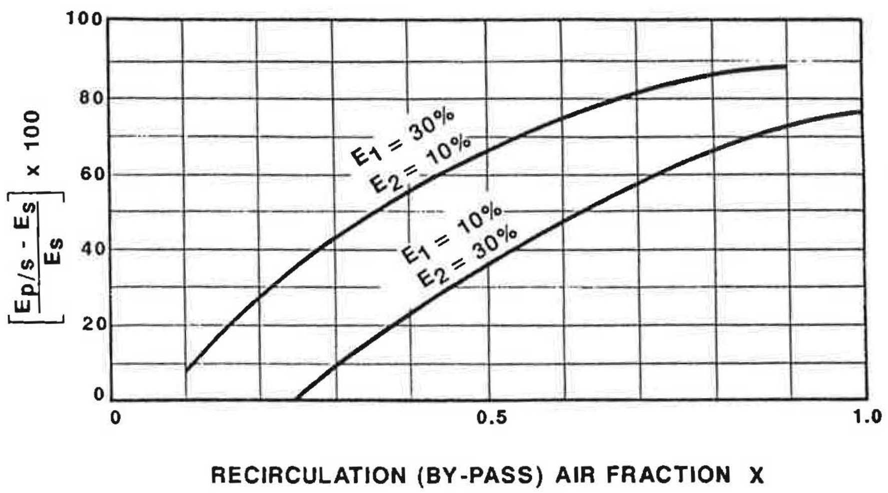


Figure 5 Series flow air filtration system; scenario 2: series flow air filtration



E_1 = FILTER EFFICIENCY OF SECONDARY FILTER LOCATED IN SCENARIO 2 BY-PASS DUCT POSITION

E_2 = FILTER EFFICIENCY OF PRIMARY AIR FILTER LOCATED IN MIXED OSA/RA POSITION

NOTE : FILTER EFFICIENCY DEFINED BY % CORRESPONDING TO ASHRAE STANDARD 52-76

Figure 6 Increase in overall filtration effectiveness vs. recirculation air fraction posed air recirculation approach ranges from 35% to 65% and is significant. Also note that the proposed "bypass" position, the pressure loss through the secondary air filter as shown in Figure 3, results in no net energy penalty.

The proposed air quality procedure of the new ASHRAE Standard 62-1981R can be used to reduce the amount of outdoor air required for given amounts of indoor contaminants over that required employing the prescriptive (i.e., alternate Ventilation Rate) procedure, thus reducing the associated energy cost for heating, cooling, humidifying, and dehumidifying outdoor air as well. Design and maintenance of HVAC systems should provide for comfortable and healthy indoor air consistent with energy optimization in buildings. Also, as a result of building occupancy during off-normal hours, selection of more unitary equipment (i.e., equipped with pre-integrated controls) capable of replacing (more expensive) built-up EMCS field-installed components and enhanced indoor air quality now appear to suggest a new design trend. Clearly the use of air recirculation in combination with adequate filtration (as a viable cost-effective alternative to increasing outdoor air rates in accordance with the proposed air quality procedure described earlier) now provides the HVAC designer with the means to substantiate his decisions when dealing with client concerns about increased energy usage.

New and remodeled buildings require HVAC systems that will ventilate, filter, and dilute pollutants when an unsafe level of contamination is reached. It is also necessary to eliminate areas that breed bacteria and mold (i.e., standing water in ducts). Contaminants (particulates) can be removed effectively by the use of available filtration and removal technology (Meckler 1968) such as electrostatic filters, ionization devices, and high efficiency particulate air (HEPA) filters (Meckler 1972a,b) that are commercially available in the micron range, where appropriate. In addition, some lithium chloride liquid desiccant cooling (i.e., air washer) systems are known to have bacteriostatic effects. Actually, energy conservation and indoor air quality goals are starting to come together through a greater awareness of their interdependence. Whatever design is finally selected, it is important that its intended purpose be carefully explained to operating personnel in order to ensure effective operation and to obtain the necessary commitment toward maintaining a healthy environment.

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DISCUSSION

H. Levin, University of California, Berkeley: Based on sensitivity analysis of your model's parameters, what are the key variables for which we have the most/least and best/worst data? Also, for design purposes, what does the cost benefit analysis show with respect to equipment specification/selection and operation costs?

M. Meckler: Although we have not performed a detailed sensitivity analysis of our spreadsheet model for those Table 1 equations which resulted in our Table 2 tabulations, by inspection, one can draw some insights. Clearly, the most variables for which we have the best data concern design/operating choices; namely: C_s, V_r, E_f, R and F_r. Those remaining variables for which we have the least/worst data are C_o, E_v, and $\dot{N}$. When computing V_o, for example, uncertainty in values of $\dot{N}$, E_v, and F_r (when applicable) may result in some under or overestimation. It is hoped, however, with improved space air distribution practices and published test results, present uncertainties in estimating E_v and F_r will be overcome. Also, better reporting of occupancy and ambient data may improve reliability of values for $\dot{N}$, C_o, and C_s.

Although we have not performed cost-to-benefit studies for the scenarios illustrated in Figures 3 and 5, clearly, system design factors must be taken into account when employing air recirculation in lieu of employing increased outdoor airflow for maximum benefit. The trade-off between energy and IAQ concerns, however, should act as a natural driver to suggest various means of achieving improved air cleaning without excessive energy penalties similar to those pointed out by operating comparisons illustrated in Figure 6.

G. Yan, Rhodon Consulting Group, Brockville, Ontario: Would carbon dioxide concentration not increase with recirculated air?

Meckler: Carbon dioxide concentration may rise above the 1000 ppm guidelines at some level of recirculation (i.e., corresponding to the point at which the combined outside air supply and infiltration amount may not be sufficient to maintain space air at or below this limit) in the absence of any provision for some CO_2 removal or introduction of additional outdoor air for a temporary period, say, in response to a CO_2 space sensor.

Yan: How do you tackle the 1000 ppm carbon dioxide guideline in your proposed design protocols?

Meckler: Where the 1000 ppm guideline for CO_2 is of concern, the same equations listed in Table 1 can be employed based on a known (or assumed) internal generation rate for CO_2 (as $\dot{N}$) and by setting C_s equal to or slightly less than 1000 ppm, C_o equal to that of the worst case ambient air condition anticipated and then solving for V_o for the air distribution system type approximating that identified in Table 1.

K.M. Hovey, Jr., Landis & Gyr Powers, Northbrook, IL: Other speakers have indicated the 62-1981R standard requires 15 cfm/person as an absolute minimum, you seem to be contradicting that. What's the story?

Meckler: It is my understanding that 15 cfm of outside air per person was selected in drafting the proposed ASHRAE Standard 621981R to ensure maintenance of a maximum allowable 1000 ppm CO_2 concentration when employing the prescriptive Ventilation Rate Procedure. When employing the alternate Indoor Air Quality Procedure, CO_2 must be considered along with all other known contaminants in establishing appropriate outdoor air quantities. CO_2 also has been recognized primarily as a surrogate for other known contaminants.

Hovey: Given that carbon dioxide cannot be removed with the current state-of-the-art and that a 1000 ppm CO_2 target implies 15 cfm/person as the minimum, is this alternate procedure only useful to determine ventilation requirements in excess of 15 cfm/person?

Meckler: Based on your premise that CO_2 cannot be removed, I would recommend that the equations given in Table 1 be used to solve directly for CO_2 using a value of C_s equal to 1000 ppm or less, if desired. This may result in a value of C_o equal to or greater than the 15 cfm/person value depending upon the criteria used in selecting the remaining variables (i.e., ventilation efficiency, internal and external CO_2 generation rate, etc). Employing 15 cfm outside air/person cannot be automatically assumed as the proper minimum under the Indoor Air Quality Procedure, particularly when other known contaminants may actually result in requiring higher outdoor air qualities.

C.N. Lawson, Carl Lawson and Associates, New Port Richey, FL: With the use of the two-filter (air cleaner) system on VAV, what is the carbon dioxide content that is removed?

Meckler: It is planned that both primary and secondary filters will have both particulate and gaseous removal capabilities. Although CO_2 serves primarily as a surrogate for other contaminants, it is still necessary to maintain no more than 1000 ppm CO_2 in occupied spaces. Accordingly, use of a CO_2 sensor is planned to act as an override to automatically increase outdoor airflow to the conditioned space in the event this recommended CO_2 limit is exceeded.

D. Stone, R. Douglas Stone and Associates, Orlando, FL: Has the parallel/series filter arrangement been installed? And how does one account for the variation in static? Does variable flow occur in the unitary system?

Meckler: The proposed parallel/series filter arrangement illustrated in Figure 3 is not yet operational. It is, however, under active development as an integral component of a proprietary line of unitary heat pumps and air conditioners also incorporating thermal energy storage. The bypass duct interconnecting the unitary unit supply and return air sections is already equipped with an automatic damper continuously operating to maintain a constant supply air static pressure at the unit discharge as individual downstream VAV dampers respond to zone demands for more or less supply air by their respective thermostats.

Referring to Figure 3, Notice that the airflow through the VAV terminal unit illustrated remains constant as the bypass damper (see Figure 2) is modulated. Therefore, the pressure drop across the VAV terminal unit remains the same except for gradual primary air filter loading effects. Although the airflow resistance through the secondary air filter changes with varying flows, the downstream VAV terminal unit static pressure sensor allows for immediate compensation through repositioning of the bypass control damper.

As illustrated in Figures 2 and 3, airflow through the primary filter remains constant at all times, while improved filtration of the VAV supply air delivered to downstream zones results from some serial flow through both the primary and secondary air filters at all part load conditions.

M.J. Hodgson, University of Pittsburgh, PA: The 1000 ppm figure to which you have referred assumes that humans are the primary source of indoor air pollution.

Meckler: Not necessarily, as it can result from other known sources within conditioned spaces. The value, is however, believed to be a safe limit for occupants for purposes of design based on currently available consensus information.

J.E. Janssen, Minnetonka, MN (co-author of this paper): Several questions addressed the issue of maintaining a 1000 ppm limit on CO_2 while reducing ventilation by using filters. The Air Quality Procedure is independent of the Ventilation Rate Procedure. Under the Air Quality Procedure, the outdoor airflow rate is unspecified, but the CO_2 limit of 1000 ppm is specified. An air cleaner can be used for an application such as a bar or cocktail lounge where tobacco smoke particles are a major contaminant. The outdoor airflow rate then can be reduced from 30 cfm to perhaps 20 or even 15 cfm under the air quality procedure. If a CO_2 sensor were used as a control, the outdoor airflow rate could be even lower under many conditions. However, when using an air cleaner that removes only particles, the designer must consider those other contaminants which will limit the reduction in outdoor dilution air that is possible. This is not a simple design task, but skillful designers may employ the method to achieve energy savings. Another factor to be considered is the increase in recirculation needed to achieve full air-cleaner effectiveness. The last column in Table 1 provides this information. Table 1 is being included in the Appendix of Standard 62-1981R to provide designers the means for calculating the performance of an air-cleaning system.