PROPOSED METHOD OF INDOOR AIR POLLUTION CONTROL BY DILUTION WITH OUTSIDE AIR

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Abstract

Gaseous pollutants generated within the air conditioned space can be controlled by dilution. This is recognized in ASHRAE Standard 62-1981R, but no practical methodology is presented to compute the required percentage of outside air. This paper proposes such a method.

Simplified Dilution Model

The following assumptions are made:

1. The rate of generation of all contaminants in the building is constant and consists of two components:

   a. Contaminants produced by the occupants such as tobacco smoke.

   b. Contaminants produced by out-gassing from building materials.

2. For a given type of occupancy, it is possible to define an outside air ventilation rate which, if maintained constant during occupancy, will result in the statistical absence of discomfort due to contamination. It does this by diluting both contaminants produced by occupants and by the building materials.

   For this "ventilation rate" to be valid, it must be assumed that it produces a "steady state" mass balance - the rate of generation of contaminants balances the removal by dilution.

   The data published by ASHRAE is usually expressed in L/s per occupant. To make it useful for this methodology, it must be expressed as both L/s per occupant and L/s per m^2 of floor area. For instance, ASHRAE recommends 10 L/s per occupant for offices where smoking is allowed. Since the same reference also indicates that 7 occupants per 100 m2 should be used, it can be expressed as follows for the purpose of this method:

   $$R_1 = 10 \text{ L/s/occupant or}$$

   $$R_2 = 10 \times 7/100 = 0.7 \text{ L/s/m}^2$$

   V_u can then be computed as (R_1 x O) or (R_2 x A). In this form it is usable for our purposes.

3. The effect of contaminants in the outside air is neglected.

In a real HVAC system the amount of supply air, and with it the amount of outside air delivered to each room, is a function of the heat gain (heat loss in the case of a heating only system), and is therefore not proportional to either the number of occupants or the floor area. For

instance, a corner office occupied by one person will receive more outside air per occupant or per m^2 than an interior room with a lower heat gain but substantially more occupants, such as a typing pool.

Steady State Model

If, as must be assumed in the ASHRAE "ventilation rate", all occupants are in a single room, the ventilation rate is referred to in this paper as the "Uniform Ventilation Rate" (see preceding paragraph). It is assumed that if this rate is maintained, the contaminant concentration can be controlled below the threshold value x_{max}.

If the same number of occupants is distributed through a number of rooms with different characteristics, it is necessary to compute a new outside air ratio for the system such that x_{max} is not exceeded in any space (see Fig. 2).

If the the extent to which the system deviates from being homogeneous (all rooms have the same C and the same D) is expressed by the equations $F_1 = C_{min}/C_{av}$ or $F_2 = D_{min}/D_{av}$, the ratio of outside air to supply air in the system can be computed as

$$V = 1/(1/V_u - 1/F + 1) \text{ for } F_{min} = V_u \text{ and } F_{max} = 1 \qquad (1)$$

where $F = F_1$ or F_2, whichever is greater. If F is less than V_u, it indicates that there is at least one zone in the system which does not provide each occupant or each m^2 with the ASHRAE minimum ventilation rate R. For derivation of equation 1 see Appendix.

Fig. 1 shows equation 1, plotted for values of V_u ranging from 0.1 to 0.5.

It can be seen that for a reasonably homogeneous system with a V_u of 0.1, the difference between V_u and the actual required ventilation rate V is negligible; however, as the value of V_u increases, the effect of the factor F becomes more pronounced. In other words, systems with higher occupant density and serving rooms with widely differing load characteristics are more sensitive to the effect discussed in this paper.

Application to VAV Systems

As the VAV system modulates from its maximum (cooling design) air circulation rate to a minimum (heating design), the total air circulation rate reduces. As the outside air is admitted upstream of the supply fan (see Fig. 2) as a constant percentage of the total air circulated, its quantity will reduce proportionately. However, most systems are equipped with Economizer Cycle controls (Fig. 3). The amount of outside air added to the system at any given time therefore depends on two factors:

a. The percentage of outdoor air admitted by the economizer cycle control, and

b. The total amount of air circulated, to which the above percentage is applied.

Since we do not want the amount of outside air to fall below the number calculated by our method, some systems will require controls to set a lower limit to the amount of outside air introduced.

Since the turn-down due to load changes for exterior rooms tends to be larger than for interior rooms, the factors F_1 and F_2 will approach 1 as the cooling load reduces. This means that, as long as the amount of outside air added to the system is subject to the "low limit" control mentioned above, interior rooms will receive more fresh air during periods of heating or reduced cooling.

Transient and Intermittently Occupied Rooms

In every system there may be rooms, such as conference rooms or meeting rooms, which have high occupant density but are occupied only for limited periods of time, and then mostly by people who normally occupy other spaces in the system. In addition, there are spaces such as lobbies, waiting rooms, etc. which, while having fairly constant high density occupancies at all time, are occupied by transient people who do not stay there long. Both of these spaces will penalize the system with excessive amounts of fresh air, if they are included in the determination of the factor F. They should therefore be excluded.

Conclusion

Control of Indoor Air Pollution by dilution is effective and can be designed on a rational basis. Special care must be taken in the case of VAV systems to ensure continuation of the dilution effect when the system air supply is throttled back. It is suspected that much of the problems encountered with occupant complaints can be traced back to inadequate ventilation during these periods of reduced air circulation.

Appendix

Derivation of equation 1:

Contaminant level in space i: $x_i = x_e + k/C_i + m/D_i$ (2)

Since the emission rates k and m are not known, the equation can not be used in this form. In order to overcome this difficulty, the two pollution components (pollution generated by occupants and pollution generated by building materials) will be treated separately. Of the two resulting ventilation rates, the larger should then be used.

Pollution generated by Occupants

The maximum contaminant level will occur in the space with the smallest air circulation rate per occupant (C). If we limit this concentration to x_{max}, then

$$x_{max} = x_e + (k/C_{min}) \quad (3)$$

$$\text{Let } F_1 = C_{min}/C_{av} \quad \text{then} \quad (4)$$

$$x_e = x_{max} - [k/(C_{av}F_1)] \quad (5)$$

The contaminant level in the return air is:

$$x_r = x_e + (k/C_{av}) \quad (6)$$

Reference to the system diagram, Fig. 2, shows that:

$$Q_o x_o + Q_r x_r = Q_t x_e \quad (7)$$

By definition $x_o = 0$, thus $Q_r/Q_t = x_e/x_r$ (8)

Substituting equation (6) for x_r, we get:

$$Q_r/Q_t = x_e/[x_e + (k/C_{av})] \quad (9)$$

$$\text{Given } V_1 = Q_o/Q_t \text{ or } V_1 = 1 - (Q_r/Q_t), \text{ then:} \quad (10)$$

$$V_1 = (k/C_{av})/[x_e + (k/C_{av})] \quad (11)$$

Substituting equation (5) for x_e we get:

$$V_1 = (k/C_{av})/[x_{max} - (k/C_{av})(1/F_1 - 1)] \quad (12)$$

It is apparent from the above equation that if the entire air handling system consists of one space, $F_1 = 1$, and equation (12) reduces to:

$$V_1 = k/(C_{av}x_{max}) \quad (13)$$

The V_1 in equation (13) is by definition equal to V_u, which means that

$$x_{max} = k/(C_{av}V_u) \quad (14)$$

If equation (14) is substituted into equation (12), the term k/C_{av} cancels out and the result is:

$$V_1 = 1/(1/V_u - 1/F_1 + 1) \quad (15)$$

Pollution Generated By Building Materials

An analogous derivation results in the expression

$$V_2 = 1/(1/V_u - 1/F_2 + 1) \quad (16)$$

$$\text{where } F_2 = D_{min}/D_{av} \quad (17)$$

The actual ventilation rate to be used for the system is then the greater of V_1 or V_2.

Symbols Used

A = Room floor area, m^2
Q = Air circulation rate, L/s
k = Rate at which contaminants are generated, L/s of contaminant/occupant
m = Rate at which contaminants are generated, L/s of contaminant/m^2 of floor space
x = Contaminant level in air, L/s contaminant/L/s air
O = Number of occupants in space
C = Air circulation rate, L/s per occupant
D = Air circulation rate, L/s per m^2 of floor space
V = Outside air ratio, L/s outside air/L/s total air circulated
F = Dimensionless factor defined by equation (4) and (17)
R = ASHRAE ventilation rate, L/s/occupant or L/s/m^2

Subscripts

max = Maximum
min = Minimum
av = average
e = Entering
i = Space identification
o = Outside
r = Return
t = total
u = Uniform (refers to ASHRAE uniform ventilation rate)
1 = relating to occupants
2 = relating to building materials

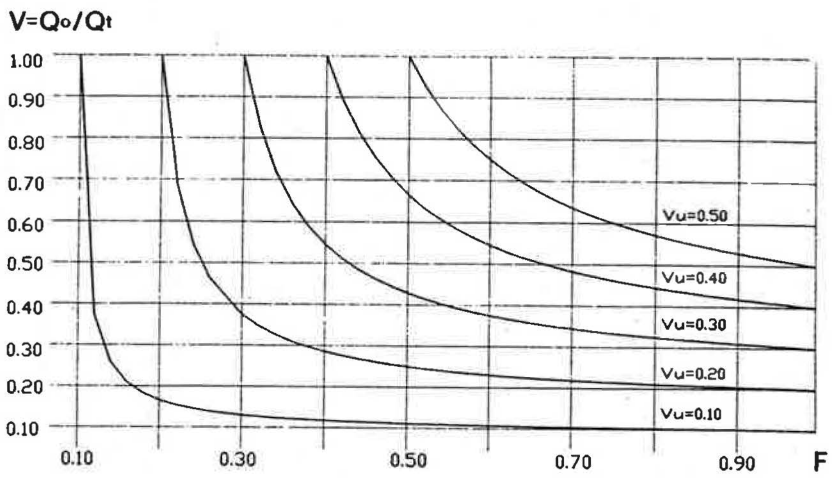

FIG. 1.

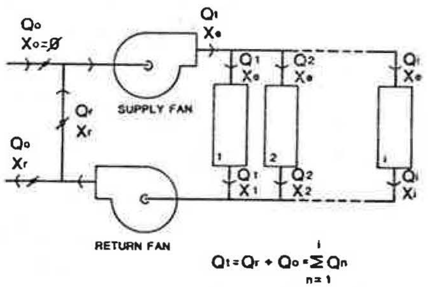

FIG. 2.

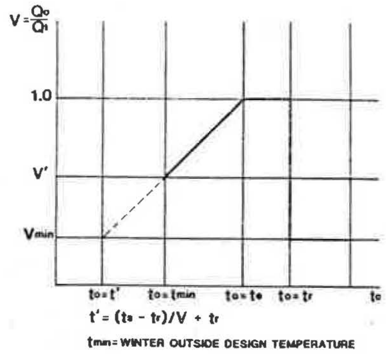

FIG. 3.

FAN STANDARDS IN CLEAN-ROOM TECHNOLOGY

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Abstract

Increasingly, clean-rooms are becoming the indispensable prerequisites for progress in the miniaturization of electronic components, for the more and more sophisticated technologies in surgery and intensive medicine as well as for the increasingly complex chemistry in the parmaceutical industry.

Clean-room technology with laminar air flow requiring appropriate air filtration at extreme air exchange rates has posed considerable problems not only to the constructors and operators of such systems but also to the fan manufacturers.

The fans employed in future clean room technology must mainly meet the following stringent requirements:

1. A maximum of operating safety
2. A low level of vibrations
3. A minimum of air turbulence
4. An optimized operating economy
5. Automatic monitoring

These demands could in the past not always be met by the fans usually employed. The fans therefore had a destructive influence on product quality and thus on the productivity of the manufacturing operation.

The variable pitch axial flow fan (Fig. 1) has had a firm place in industrial applications for more than three decades because of its excellent controllability and its supreme economy. This fan system proves to be particularly successful wherever stringent demands have to be met with regard to volume flow and pressure control, economy, operating safety and automatic monitoring.