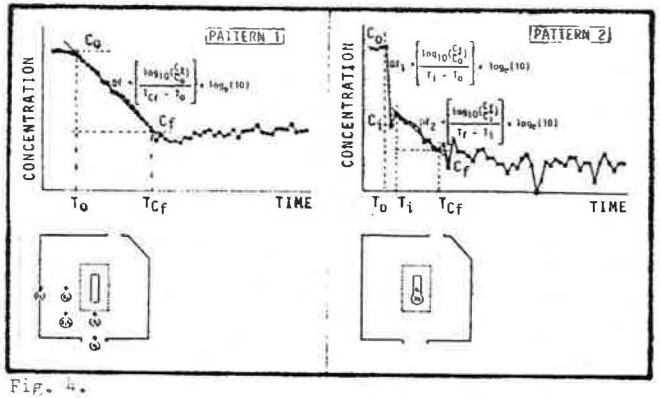

Fig. 4.

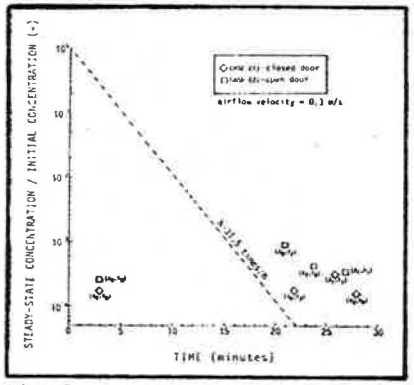

Fig. 5.

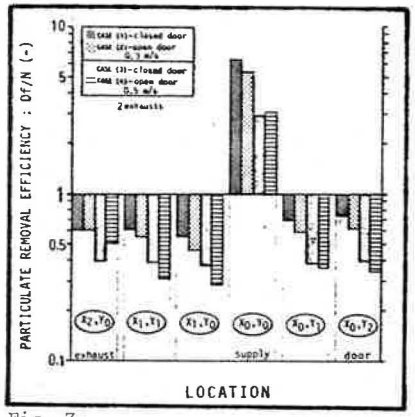

Fig. 7.

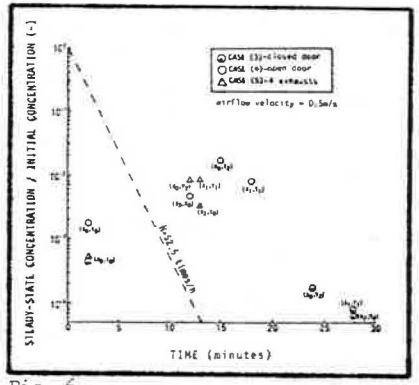

Fig. 6.

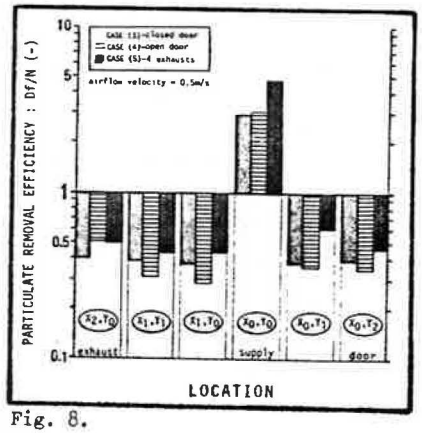

Fig. 8.

THE USE OF TRACER GAS AND ANALYTICAL MODELS TO STUDY DILUTION AND RECIRCULATION FROM LABORATORIES

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Abstract

A tracer technique for estimation of dilution and recirculation of the exhaust gases was developed and applied in tests conducted at two major laboratory buildings. The results of the tests demonstrated the great dependence of dilution and recirculation upon factors such as wind speed and direction, vent-to-intake distance, and stack height. The tracer data were also used to validate several analytical models predicting the minimum dilution. Although none of the models was outstanding relative to the others, the one accounting for the extra dilution caused by the stack height gave particularly good predictions. Finally, a model developed in the present study proved that the Gaussian diffusion equation can provide reasonable estimates of dilution around buildings. provided that the appropriate enhanced dispersion parameters are selected.

Introduction

The interaction of wind with a building results in complex local air motions that may cause reentry of the exhaust gases into the fresh air intakes contaminating the indoor air, and pose potential life threatening situations in the case of accidental releases. Unfortunately, the problem of contamination of the intakes from nearby effluents cannot be entirely avoided without exacting an unacceptable penalty in capital and operating cost. However, the degree of contamination can be controlled so that the concentrations of the emitted substances at the intakes remain within the acceptable levels. Establishment of such a protection requires a fundamental understanding of the dispersion of exhaust gases, as well as the development of methods for the prediction of the concentration fields around buildings. In addition, there is a great need of guidelines for choosing among the mathematical models, predicting the exhaust-to-intake dilution, in the case of a specific configuration.

The present study was conducted in an effort to provide a better understanding of the dispersion of pollutants around buildings. Specifically its objectives were as follows: 1) design of a tracer gas technique for estimation of the dilution and recirculation of exhaust gases back into the buildings; 2) application of the above technique in field tests conducted at two buildings of the University of Toronto; and 3) validation of several mathematical models estimating the stack-to-intake dilution of the exhaust gases, and subsequent development of guidelines for choosing the appropriate model for a specific configuration.

Tracer gas technique

The field tests were carried out at two multistorey, "airtight" complexes, namely the Wallberg and Erindale buildings, which house, among others, the University of Toronto departments of Chemical Engineering and Applied Chemistry, Biology, and Chemistry. Both these buildings are equipped with central heating, ventilating, and air-conditioning systems, and had experienced recirculation problems in the past due to the close proximity of the fume hood exhaust vents to the air intakes on the roofs.

The testing procedure involved the controlled release of an inert, nontoxic, and odorless gas (sulfur hexafluoride, SF_6) from specific vents of the buildings over a predetermined period of time. Starting with the time of release, sequential 15-min averaged samples of the return air were collected inside the fresh air intakes, and were analyzed for SF_6 by means of gas-solid chromatography with electron capture detection. Also, during each test the wind speed and direction were recorded by a 5-m tower installed on the rooftop, and instrumented with a windvane-cup anemometer.

The percentage of the exhaust gases recirculation, R, into the fresh air intakes was estimated from the equation:

$$R = 100 \frac{Vr}{Vt} \qquad (1)$$

where Vr and Vt are the recirculated and released volumes of the tracer, respectively. In addition, the stack-to-intake dilution of the exhaust gases, D, was estimated by dividing the exhaust concentration of SF_6 at the stack exit, Ce, with the maximum concentration recorded inside the intake, Cin:

$$D = \frac{Ce}{Cin} \qquad (2)$$

Results

During the present study, a total of 72 tracer release tests was conducted, under a variety of weather conditions, from June 1985 through February 1986, and altogether 90 hours of tracer and wind data and more than 600 samples were collected and analyzed.

The results demonstrated the dependence of dilution and recirculation upon factors such as wind speed and direction, vent-to-intake distance, and stack height, while the atmospheric stability was found to have no correlation. In addition, the tests provided data upon which recommendations for possible solutions to the recirculation problems, experienced in the two buildings, were made. Specifically, reentry rates as high as 9.5%, observed at the Wallberg building, indicated that there exists a potential for contamination of the indoor air, and an increase of the stack height was recommended to aleviate the problem. The tests at the Erindale building suggested that although some recirculation was present, the dilution between the intakes and the building's interior was large enough to prevent serious contamination of the indoor air. Nevertheless, an increase of the stack height was also recommended in some cases to prevent any serious problems.

Model validation

All the proposed models have been developed from wind-tunnel studies and are designed to account for the "worst case" (with the wind carrying the exhaust plume directly from the source toward the intake), so that they can be used to provide a conservative safeguard for design or modification purposes. The validation of each of the examined models is accomplished using graphs of the dilution levels observed during the tests vs. the dilution factors calculated by a particular model (see Figure 1). The degree of underprediction or overprediction provided by each model is expressed by the diagonal lines representing ratios of the observed to the calculated dilution factors.

Wilson (3), based on the results of over 5000 wind-tunnel measurements, proposed the following equation:

$$Dmin = 0.11 \frac{U\ S^2}{Ae\ Ve} \qquad (3)$$

where Dmin is the minimum dilution, U the wind speed, S the stretched-string exhaust-to-intake distance, Ae the area of the exhaust opening, and Ve the stack exit velocity. Figure 1 indicates that although equation 3 provided a lower bound of dilution to 98% of all measurements, it resulted in very conservative predictions in the case of stacks located close to the intakes ($S < 4m$), underestimating the observed values by 2 to 3 orders of magnitude. This was probably caused by the unrealistic behavior of the model in the case of short stack-to-intake distances, since for $S = 0$ it yields the physically impossible $Dmin = 0$ instead of the real $Dmin = 1$.

Halitsky (2), based on his measurements of concentration coefficients around building models, suggested the following equation:

$$Dmin = \left(1 + 0.132 \left(\frac{S^2}{Ae} \right)^{0.5} \right)^2 \qquad (4)$$

for the case where both the exhaust and intake are located on the same wing of the building. This model resulted in very conservative estimates of dilution underestimating the observed values by 100 to 1000 times in 74% of all measurements (Figure 2). One of the major reasons for this underestimation is that Halitsky did not attempt to simulate the random fluctuations of the atmospheric turbulence in the wind approaching the building models. In addition, the absence of the wind speed and stack height as parameters from the model, and the fact that it was developed from tests with flush vents and low emission velocities have also contributed to the low dilutions predicted.

Because none of the proposed models accounts for the dilution of the exhaust gases caused by the stack height, h, and the plume rise, Δh, Wilson (3) proposed an equation for the estimation of this additional dilution:

$$Dmin = \left(1 + 0.33 \left(\frac{U\ S^2}{Ae\ Ve} \right)^{0.5} \right)^2 \exp\left(\frac{(h + \Delta h)^2}{2\ \sigma_z^2} \right) \qquad (5)$$

where σz is the vertical plume deviation. As it is shown in Figure 3, the degree of protection provided by equation 5 is more reasonable than those provided by the previously mentioned models. This indicates that the very conservative predictions of equations 3 and 4 are indeed partly due to their inability to account for the extra dilution caused by the stack

height and plume rise. In the cases of stacks located very close to the intakes the predictions of equation 5 were completely unsuccesful since it overestimated the observed values by factors ranging from 100 to 10000.

Because neither the wind direction nor the turbulence intensity are elements of the existing models, an attempt was made to include these two parameters in a model, so that their effect could be examined. As a result, a new model was derived from the Gaussian plume formulation, accounting for the building effects by specifying enhanced dispersion parameters (1). This model includes the turbulence intensity, i, and the direction of the wind expressed in terms of the angle, f, formed between the wind and the stack-to-intake directions:

$$D = 0.8 \; \frac{i^2 \; S^2 \; U \cos^2(f)}{Ae \;\; Ve} \exp\left(\frac{(h + \Delta h)^2}{1.28 \; i^2 \; S^2 \cos^2(f)} \right) \exp\left(\frac{\tan^2(f)}{2 \; i^2} \right) \quad (6)$$

Equation 6 provided good estimates of dilution in the cases where $0^\circ < f < 45^\circ$ since the predicted values were within a factor of 10 from the observed ones (Figure 4). However, in the cases where $f > 45^\circ$ the model resulted in erroneous predictions of the observed dilutions. This behaviour was not unexpected since tan(f) increases very fast with f, with the tan(f) being infinite when $f = 90^\circ$. As a result, equation 6 yields high values as f approaches 90°, giving the theoretically correct but realistically erroneous value of $D = \infty$ when $f = 90^\circ$. In real situations the reverse-flow zones around a building and the random variations of the wind direction cause some of the exhaust gases to reenter the intakes even if the wind blows exactly to the opposite direction.

Conclusions

The conducted research demonstrates that the use of tracer gas technology offers a unique approach and a valuable tool for solving problems associated with air flow and gas dispersion around buildings. In addition, the model validation indicates that the degree of protection provided by each of the models varies largely with the stack-to-intake distance and the stack height. In particular, the use of such models in cases where the stacks are located very close to the intakes should be generally avoided.

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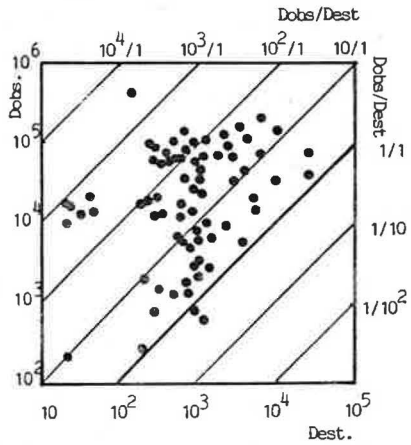

Figure 1 - Observed versus dilution factors estimated from equation 3.

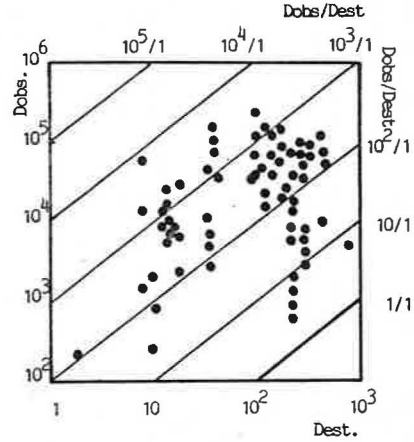

Figure 2 - Observed versus dilution factors estimated from equation 4.

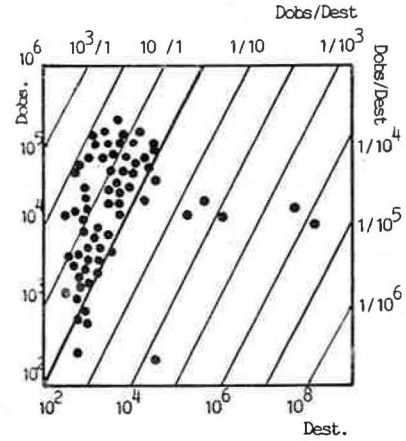

Figure 3 - Observed versus dilution factors estimated from equation 5.

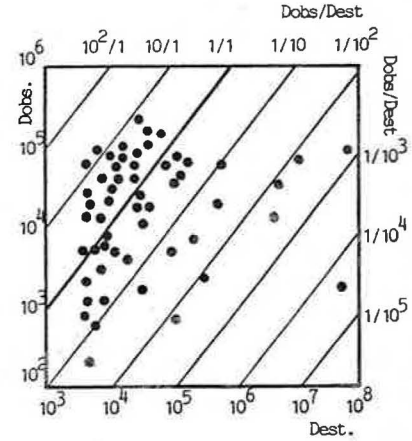

Figure 4 - Observed versus dilution factors estimated from equation 6.

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 contr 's 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

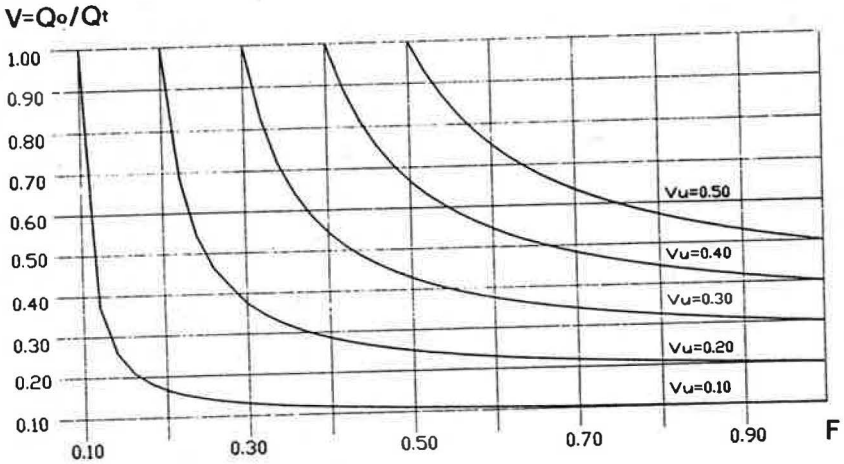

FIG. 1.

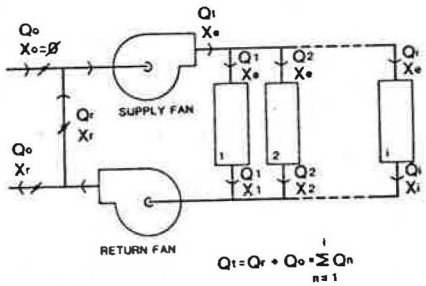

FIG. 2.

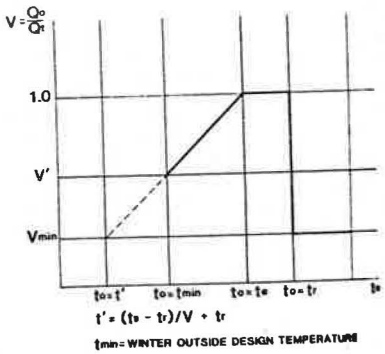

FIG. 3.

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.