# HOW *VENTILATION* INFLUENCES ENERGY CONSUMPTION AND *INDOOR AIR QUALITY*

**Until recently, natural ventilation was adequate for the indoor air environment of buildings. Today, with added insulation and tighter building envelope standards due to the energy situation, indoor air quality is more dependent on mechanical means than ever before. But mechanical means are expensive! The authors of this paper contend indoor air quality is attainable with cost-effective means if the proper constraints are maintained.**

**JAMES E. WOODS**
*Member ASHRAE*
**EDUARDO A. B. MALDONADO**
**GARY L. REYNOLDS**

INDOOR air quality has been recognized for centuries as an important factor to be controlled for the health and comfort of occupants. Until recently, the conventional method of control was dilution with outdoor air by natural or mechanical ventilation. However, in light of the current energy situation, ventilation systems often have been deactivated or the ventilation rates have been substantially reduced. In some of these cases, the concentrations of indoor contaminants have increased to levels that might have been harmful to the occupants.[1-4]

Recent evaluation of these trends have shown that energy efficiency and acceptable indoor air quality do not have to be incompatible. To the contrary, some control strategies may result in increased acceptability at reduced energy consumption rates and operating cost.

## HISTORIC PERSPECTIVE

Natural ventilation has long been used to dilute indoor contaminants. As major scientific advances were made and the nature of buildings changed, however, natural ventilation was no longer sufficient. By 1895, ventilation codes and standards were being adopted which specified minimum ventilation rates at 30 cfm per person.[5] To transport and diffuse these large quantities of ventilated air, the use of mechanical systems, although expensive to install and operate, was usually required. Thus, the problem of providing adequate indoor air quality at a reasonable cost was faced almost 100 years ago. This dilemma may also have contributed to a conventional interpretation of a *minimum* ventilation rate as a *maximum* for design purposes.

With further technological development, the minimum ventilation rates were gradually decreased to 5 cfm per person, as specified in current standards.[6] However, when ASHRAE Standard 90-75 was published, it stated that the *minimum* values "*shall* be used for design" purposes.[7] This statement effectively deleted the *recommended* ventilation rates that were given in ASHRAE/ANSI Standard 62-73.[8]

The recommended values were specified to provide odor-free environments whereas the minimum values were specified to accommodate fuel economy. The revised ANSI/ASHRAE/IES Standard 90A-1980 also requires the use of the minimum values.[9] To help resolve these conflicts, the revised ANSI/ASHRAE Standard 62-1981 now specifies *required* ventilation rates for "smoking" and "non-smoking" areas.[10] It also is more specific regarding the requirements of recirculation air flow rates and air cleaner efficiencies.

So, for the second time in recent history, we are faced with the problem of providing adequate indoor air quality at reasonable cost (i.e., energy consumption). This time, failure to find a solution may have more serious consequences. Since the interpretation of "*minimum* is *maximum*" is now centered around 5 cfm per person rather than 30 cfm per person as it was 100 years ago, a safety factor of 6 has been removed. Thus, any miscalculation may result in greater risks to the occupants.

## CONTROL STRATEGIES

Control of indoor environments is currently based on three sets of criteria: environmental, economic, and energy consumption. Thus, the situation we face calls for solutions based on optimization theory. This rigorous method has seldom, if ever, been used in the design of ventilation systems now in operation. Rather, ventilation systems have been conventionally designed to provide acceptable rates of outdoor air for dilution of indoor contaminants at "design loads" and for thermal control when appropriate, but at rates sufficiently low to meet specified energy and economic criteria.[11]

The three common methods of controlling the mass quality of indoor air are source control, dilution control, and removal control. These three functions are not always independent. A simple steady-state mass balance for a system which does not recirculate air indicates that a relationship among these control methods can be expressed as

$$C_s = C_o + \frac{\dot{N} - \dot{E}}{\dot{V}} \qquad (1)$$

where $C_s$ is the indoor concentration, $C_o$ is the outdoor concentration, $\dot{N}$ is the net generation rate of contaminant indoors, $\dot{E}$ is the removal rate of contaminant, and $\dot{V}$ is the air flow rate (outdoor air and infiltration).

*Source control* involves methods which minimize the net generation rates of contaminants indoors, $\dot{N}$. These methods currently include

- Isolation of the source from the indoor environment, such as product substitution or prohibition of smoking in certain areas.[12]
- Containment of the source by treatment with paints or other barriers is also currently attempted.[13]
- Local exhaust such as biological cabinets in laboratories or kitchen range hoods.[14]

Source control is probably the most cost effective and energy efficient method of indoor air quality control, but also provides the least assurance of control to the occupants, if other methods are not present.

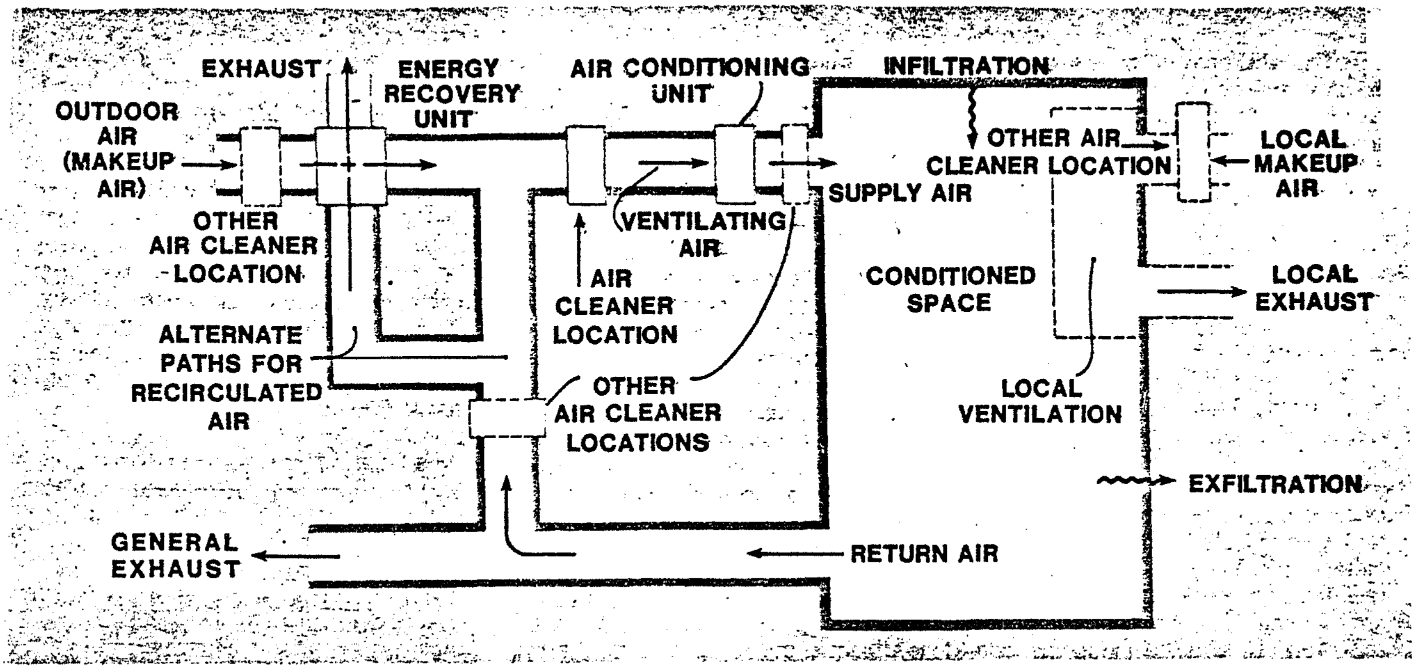
*Dilution control*, pertaining to methods that affect $\dot{V}$, is the most common method of indoor air quality control in non-industrial facilities. This method of control may be achieved by

- Infiltration through cracks around windows, doors, and construction joints.[15,16]
- Natural ventilation through open windows and doors, and through other openings or vents designed for that purpose.[17,18]
- Forced or mechanical ventilation systems which include supply or exhaust fans, and which also may include dampers and filters.[19]

Dilution control methods may be applied independently or in combination. Their relationships are shown schematically in Fig. 1. Note that when dilution control is employed, the presence of a contamination source, or generation rate $\dot{N}$, is implied.

For the simple case of no air recirculation for the occupied space (*i.e.*, 100% ventilation with outdoor air) and no contaminant removal control (*i.e.*, $\dot{E} = 0$), Eq. (1) indicates that the concentration indoors, $C_s$ varies inversely with the air flow rate $\dot{V}$. This relationship is the basis for the ventilation rates commonly specified in current standards.[20]

*Mr. Woods is professor of mechanical engineering at the Architecture and Research Institute of Iowa State University in Ames, Iowa. Mr. Maldonado is a graduate research assistant at the department of mechanical engineering at Iowa State, as well. Mr. Reynolds, also with the same faculty, is an instructor and research associate.*




Typical residential forced air systems have been designed to recirculate 100% of the supply air and infiltration has been depended upon for dilution control. In this case, a filter, if present, is located in the recirculation air stream. The steady-state indoor concentration can be expressed by Eq. (1) if the air flow rate, $\dot{V}$, is taken as the infiltration rate, $\dot{V}_I$, only. As energy conservation efforts have been adopted, the amount of infiltration air rates have often been reduced to less than 0.5 air changes per hour in new construction. [21]

Although reductions in existing residences have been reported as low as new construction, reductions in infiltration are still substantial. In these energy efficient residences, indoor air quality may be significantly degraded if care is not taken to provide alternative control strategies. [22] Conversely, with the alternative strategies, drafts and cold wall effects caused by infiltration can be reduced for improved thermal comfort as well as energy savings.

Forced air systems in commercial facilities vary considerably. For these systems, the steady-state indoor concentration can also be expressed by Eq. (1) with the air flow rate, $\dot{V}$, assumed to be the sum of the variable infiltration and outdoor air rates, ($\dot{V}_o$ + $\dot{V}_I$). As in residential facilities, energy conservation efforts in commercial buildings have resulted in reduced infiltration rates. In addition, the ventilation systems are often deactivated during reduced occupancy periods. If the major contamination sources within a facility are the occupants, deactivation may not cause degradation of the indoor air quality for the remaining occupants. Conversely, if the sources of contamination are processes or materials which are independent of the occupancy density, deleterious effects on the remaining occupants could result.

Thermostatically-controlled, mixed-air systems are of particular concern today because of their energy implications. Advice or demands have recently caused resetting of the setpoints of the mixed-air controllers to higher values, deactivation of the thermostatic function of these systems, manual adjustment to a minimum amount of ventilation, or complete deactivation of the systems. [23-25] Unfortunately, efforts have usually been counterproductive. These systems, with either temperature or enthalpy control (*i.e.*, "economizer" systems), were probably designed to provide supply air conditions to meet cooling requirements imposed by thermal loads. Thus, the amount of outdoor air introduced for cooling may exceed the minimum required for mass air quality control. Moreover, the proper use of mixed-air control allows refrigeration equipment to remain deactivated when thermal conditions of the outdoor air satisfy the cooling loads. Conversely, if these mixed-air control systems are improperly operated (*i.e.*, set to higher setpoints), thermal comfort may be degraded or additional refrigeration loads may be required which would increase energy consumption.

*Removal control,* which pertains to methods that affect $\dot{E}$, is commonly used as an alternative to, or in combination with, dilution control to reduce indoor concentrations by means of air cleaning devices. These can be located either in the forced air systems, as shown in Fig. 1, or directly in the occupied space.

Air cleaning devices for residential or commercial systems may be classified as

- Particle removal devices which include mechanical filters and electronic air cleaners. [26]
- Gas and vapor removal devices which contain sorbents such as activated charcoal or activated alumina. [27]

The removal rate of the indoor contaminant, $\dot{E}$, in Eq. (1), can be expressed as

$$\dot{E} = \epsilon \dot{V}_u C_u \qquad (2)$$

where $\epsilon$ is the efficiency of contaminant removal device, $\dot{V}_u$ is the air flow rate through the contaminant removal device, and $C_u$ is the concentration of contaminant in the airstream entering the contaminant removal device.

The upstream air flow rate, $\dot{V}_u$, is usually known explicitly. For fan-filter modules, $\dot{V}_u$ is given as part of the rating of the device. In central systems with 100% recirculated air or 100% outdoor air, $\dot{V}_u$ is the same as the air flow rate through the system fan. In mixed-air systems, $\dot{V}_u$ is the sum of the outdoor and recirculated air flow rates. It should be noted that $\dot{V}_u$ in larger systems may vary depending on changes in system resistances or variable volumetric flow rates. [28]

The contaminant concentration in the air entering the contaminant removal device, $C_u$, should be equal to the concentration in the occupied space for fan-filter modules and 100% recirculated air central systems. However, if stagnation or stratification exists within the occupied space, the concentrations to which the occupants may be exposed could be significantly different than those entering the air cleaning equipment. [29,30] For 100% outdoor air systems, $C_u$ should be the same as the ambient air. But, if the outdoor air intake is not carefully located, these concentrations can be influenced by local effects, such as automobile emissions, particle entrainment, or short-circuiting from system exhaust ducts. The concentrations $C_u$, in mixed-air control systems can vary between those in the occupied space and those in the outdoor air, depending upon the mode of mixed air control.

The efficiency of the contaminant removal device, $\epsilon$, must be expressed as a dimensionless fraction and determined as the complement of the penetration, P, of the contaminant through the device

$$\epsilon = 1 - P \quad (3)$$

where $P = C_d/C_u$ and $C_d$ is the concentration of the contaminant in the airstream leaving the contaminant removal device.

As seen from Eq. (1), an increase in the removal rate, E, has the same effect as reducing the generation rate, $\dot N$ (*i.e.*, $\dot N - \dot E \rightarrow 0$). The result is to decrease the difference between the indoor and outdoor concentrations. In fact, with sufficiently high values of air cleaner efficiency, $\epsilon$, and of air flow rate through the air cleaners, $\dot V_u$ the removal rate, $\dot E$, will exceed the generation rate, $\dot N$, and the indoor concentration, $\dot C_s$, can be less than the outdoor concentration, $C_o$, independent of the value of the air flow rate, $\dot V$.

## NEW STRATEGIES

Many current standards allow the specified ventilation rates to be reduced if certain precautions are taken, such as the provision of mechanical ventilation, installation of air cleaning devices, and thermal treatment of the air.

These standards, however, do not provide criteria for evaluating the acceptability of the indoor air quality after the ventilation rates have been reduced.

A rational criterion for reducing the amount of outdoor air can be expressed as follows

*"Outdoor air required for dilution control may be reduced, if alternative source control and removal control strategies are sufficient to provide the same quality of indoor air as would be achieved by dilution control."*

This rationale may be implemented in the design stage by minimizing life-cycle-costs through optimization of N, $\dot V$, and $\dot E$, while maintaining the indoor concentration, $C_s$, at the same value.

The techniques required to apply source control to reduce N are reasonably well-known and have been described as part of current strategies. Application of removal control, $\dot E$, as an alternative to dilution control, $\dot V$, while maintaining the indoor air quality, $C_s$, is not, however, a commonly used control strategy.

**Steady-State Recirculation.** A control strategy can be derived from steady-state mass balances for 100% outdoor air and for recirculated air systems. These systems are shown schematically in Fig. 2. The steady-state mass balances for these systems can be expressed as

100% outdoor air [Fig. 2a]

$$\dot V_o(C_s - C_o) = \dot N \quad (5)$$

Recirculating air [Fig. 2b]

$$(\dot V_m C_o + \dot V_r C_s)(1 - \epsilon) + \dot N = (\dot V_m + \dot V_r)C_s \quad (6)$$

Recirculating air [Fig. 2c]

$$\dot V_m C_o + (1 - \epsilon)\dot V_r C_s + \dot N = (\dot V_m + \dot V_r)C_s \quad (7)$$

where $\dot V_r$ is the recirculated air flow rate, $\dot V_m$ is the reduced outdoor air flow rate, and the other terms are as previously defined.

This recirculation control strategy imposes the same indoor air quality (*i.e.*, $C_s$) for the same space with the same generation rate $\dot N$. Thus, Eqs. (5) and (6) can be combined to obtain a mixed-air ratio, $\dot V_r/\dot V_o$, for the system with the air cleaner in the mixed air:

$$\frac{\dot V_r}{\dot V_o} = \frac{K - 1}{K_\epsilon}\left[1 - \frac{K - (1 - \epsilon)}{K - 1}\right]\frac{\dot V_m}{\dot V_o} \quad (8)$$

And, Eqs. (5) and (7) can be combined to obtain a mixed-air ratio for the system with the air cleaner in the recirculated air:

$$\frac{\dot V_r}{\dot V_o} = \frac{K - 1}{K_\epsilon}\left[1 - \frac{\dot V_m}{\dot V_o}\right] \quad (9)$$

In Eqs. (8) and (9), the parameter K is defined as the ratio of the contaminant concentrations of the indoor air to the outdoor air. Note that the parameter K in these equations must be equal to, or greater than, unity as the basis for this rationale was a system without air cleaning devices, which supplied only outdoor air either by natural or mechanical ventilation (*i.e.*, Eq. (5)).

ANSI/ASHRAE Standard 62-1981 contains a recirculation criterion based on a simplification of Eqs. (8) and (9). In this Standard, it is assumed that $K > 1$ for most contaminants. Thus, Eqs. (8) and (9) have been reduced to:

$$\dot V_r = \frac{\dot V_o - \dot V_m}{\epsilon} \quad (10)$$

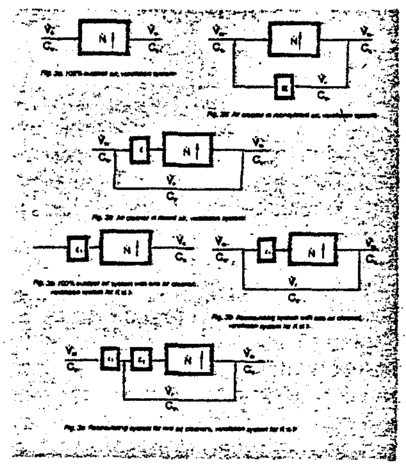
Eq. (10) conservatively estimates $\dot V_r$, since $\dot V_r$, as calculated by Eqs. (8) and (9), is less than $\dot V_r$ calculated from Eq. (10) for the same circumstances. This Standard also specifies that the total air flow of a recirculation system (*i.e.*, $\dot V_r + \dot V_m$) can never be less than the air flow required for a 100% outdoor air system ($\dot V_o$). Furthermore, $\dot V_m$ must also be at least 5 cfm per person at all times.

A control strategy can also be developed for recirculating systems where it is required that the concentrations of contaminants inside the space be less than the respective outdoor concentrations (*i.e.*, $K < 1$). In this case, the 100% outdoor air system, which serves as the reference, must then have an air cleaner (efficiency $\epsilon_1$) to remove the contaminants from the air supply, as shown in Fig. 3. A steady-state mass balance on the 100% filtered outdoor air system may be expressed as:

$$K = \frac{\dot N}{\dot V_o C_o} + (1 - \epsilon_1) \quad (11)$$

Eq. (11) shows that the required air cleaner efficiency is a function of the indoor generation rate which must be known prior to designing the system. The recirculation criterion for cases where $K < 1$ can be obtained from steady-state mass balances for the systems. Thus, for the system with a single air cleaner in the mixed-air stream, Fig. 3b, the mixed-air ratio may be expressed as:

$$\frac{\dot V_r}{\dot V_o} = \frac{\frac{\dot N}{\dot V_o C_o} + \frac{\dot V_m}{\dot V_o}[(1 - \epsilon_1) - K]}{K\epsilon_1} \quad (12)$$



And, for a system with two air cleaners, Fig. 3c, the mixed-air ratio may be expressed as:

$$\frac{\dot V_r}{\dot V_o} = \frac{\frac{\dot N}{\dot V_o C_o} + \frac{\dot V_m}{\dot V_o}[(1 - \epsilon_1)(1 - \epsilon_2) - K]}{K\epsilon_2} \quad (13)$$

Note that this technique also can be used to express mixed-air ratios for other variations of air cleaner locations.

**Variable Generation Rates.** The steady-state recirculation strategies assume that the generation rate, $\dot N$, remains constant during operation, even though it may have been minimized. In many cases, however, $\dot N$ varies with changes in occupancy load, processes, etc. When significant changes in $\dot N$ are expected, strategies should be applied which maintain control at minimum life-cycle costs.

Two control strategies can be identified that are sensitive to changes in $\dot N$: programmed systems and continuous feedback systems. In both strategies, the basic objective is to take advantage of time and space to maintain indoor concentrations at acceptable levels for the occupants at reduced life-cycle costs.

Control strategies for programmed systems are not new, but their interactions with other energy conservation strategies (*e.g.*, reduced infiltration) have presented some new problems. Therefore, more care in defining the rate of change in indoor concentrations is now required when the system is operated at reduced capacity or is deactivated.

Methods for determining lead or lag times for system activation for periods of occupancy are presented in ANSI/ASHRAE 62-1981. When contaminants are generated primarily by the occupants, activation of the supply air systems may lag (*i.e.*, delay) occupancy depending on the air supply rate and the volume of the space. During system operation, the amount of ventilation air is determined by the design (*i.e.*, full-load) occupancy.

Unlike the programmed systems, control strategies that utilize feedback are not limited by fixed amounts of ventilation air during system operation. Although proportional or two-position controllers for contamination control in industrial environments are used, adoption of these techniques to residential and commercial systems are new. The advantage to be gained with such systems as $CO_2$ controlled variable ventilation is to more closely match system performance (*i.e.*, cost of operation) to *demand* (*i.e.*, maintenance of acceptable air quality).[31]

## CONCLUSIONS

Today's technology could provide significant improvements in conventional methods of ventilation control. There is a potential to reduce energy consumption and life-cycle costs of the systems while increasing the quality of the indoor environment. To implement improved control strategies, the following barriers will have to be removed.

1. Interactions among air quality, thermal, lighting, spatial, and acoustic indoor environmental factors are seldom considered. Interactive control algorithms should be developed. Indoor environmental standards including these factors should also be developed. A joint effort between government and technical societies may be required.

2. Standard methods for evaluating components (*e.g.*, air cleaners) and systems (*e.g.*, variable ventilation control systems) are required. These standards should be developed by voluntary standards organizations such as ANSI, ASTM, and ASHRAE.

3. Commercialization of appropriate sensors, controllers, and controlled devices should be accelerated. Passive elements which would be sensitive to the common contaminants found indoors should be marketed. The sensors that are now available have been developed for industrial applications and are too expensive for widespread application.

4. Low first costs continue to dominate the construction industry. Therefore, incentives should be made available to building owners and designers to incorporate the improved control strategies.

5. Little information is available on indoor air quality control. A concentrated effort to provide educational material to all levels of society should be undertaken. Formal education for design professionals is needed. Informal and formal education should be developed for building owners and operators, for occupants and the general public, and for governmental inspection and enforcement officials.

## REFERENCES


1. Repace, J. L. and A. H. Lowrey. "Indoor Air Pollution, Tobacco Smoke and Public Health." *Science* 208: 464-472 (1980).
2. Hollowel, C. D. and G. W. Traynor. *Combustion-Generated Indoor Air Pollution*. Lawrence Berkeley Laboratory Report LBL-7832, University of California, (April 1978).
3. Andersen, I., G. R. Lindqvuist and L. Molhave. "Indoor Air Pollution Due to Chipboard Used as a Construction Material." *Atmos. Environ.* 9: 1121-1127 (1975).
4. Colli, R. and P. E. McNall, Jr. *Radon in Buildings*. Proceedings of a Roundtable Discussion of Radon in Buildings held at the National Bureau of Standards, Gaithersburg, MD, June 15, 1980. National Bureau of Standards Special Publication 581. U.S. Government Printing Office, Washington, DC, (1980).
5. Klauss, A. K., R. H. Tull, L. M. Roots, and J. R. Pfafflin. "History of the Changing Concepts in Ventilation Requirements." *ASHRAE Journal* 12 (6): 51-55 (1970).
6. Woods, J. E. "Ventilation, Health & Energy Consumption: A Status Report." *ASHRAE Journal* (7): 23-27 (1979).
7. *ASHRAE Standard 90-75. Energy Conservation in New Building Design*. American Society of Heating, Refrigerating and Air Conditioning Engineers, Circulation Sales Department, New York (1975).
8. *ASHRAE Standard 62-73. Standards for Natural and Mechanical Ventilation*. American Society of Heating, Refrigerating and Air Conditioning Engineers, Circulation Sales Department, New York (1973).
9. *ASHRAE Standard 90A-1980. Energy Conservation in New Building Design*. American Society of Heating, Refrigerating and Air Conditioning Engineers, Circulation Sales Department, New York (1980).
10. *ASHRAE Standard 62-1981. Ventilation for Acceptable Indoor Air Quality*. American Society of Heating, Refrigerating and Air Conditioning Engineers, Circulation Sales Department, New York (In Press).
11. *Op. cit.*, Woods.
12. State of Iowa. The Code of 1979, Vol. 1, Chapter 98A, (*Smoking Prohibitions*) Section 98A.2, p. 514. Published by the State of Iowa, Des Moines (1979).
13. Auxier, J. A., W. H. Shinpaugh, G. D. Derr, and D. J. Christian. "Preliminary Studies of the Effects of Sealants on Radon Emanation from Concrete." *Health Physics* 27: 390-392 (1974).
14. Department of Health, Education and Welfare; National Institute of Health. "Recombinant DNA Research — Guidelines." *Federal Register* 41 (131): 27909-27943 (1976).
15. *Ibid. Federal Register* 41.
16. Grimsrud, D. T., M. H. Sherman, A. K. Blomsterberg, and A. H. Rosenfeld. "Infiltration and Air Leakage Comparisons: Conventional and Energy Efficient Housing Designs." In: *Changing Energy Use Futures*, Vol. III, R. A. Fazzolare and C. B. Smith, Editors, pp. 1351-1358. Published by Pergamon Press, Oxford (1979).
17. Brundrett, G. W. "Window Ventilation and human behaviour." In: *Indoor Climates*, P. O. Fanger and O. Valbjorn, Editors, pp. 317-330. Published by the Danish Building Research Institute, Copenhagen (1979).
18. Evans, B. H. "Energy Conservation with Natural Air Flow Through Windows." *ASHRAE Trans.* 85 (Part 2): 641-650 (1979).
19. Woods, J. E. and R. G. Nevins. "Ventilation Requirements and Energy Conservation." *Proceedings of the Sixth International Congress of Climatistics*, pp. IV-11.1 to 11.25, 1975.
20. *Op. cit.*, ASHRAE Standard 62-1981.
21. *Op. cit.*, Grimsrud.
22. *Op. cit.*, Hallowel et. al. and Colli et. al.
23. Public Law 95-619 (November 9, 1978), *National Energy Conservation Act*. 92 Stat. 3204. U.S. Government Printing Office, Washington, DC, (1978).
24. Public Law 94-385 (August 14, 1976), *Energy Conservation and Production Act*. 90 Stat. 1125. U.S. Government Printing Office, Washington, DC, (1976).
25. Department of Energy. "Emergency Building Temperature Restrictions. Final Rule." 10 CFR Part 490, *Federal Register* 44 (130): 39354-39368 (1979).
26. *ASHRAE Handbook and Product Directory: 1979 Equipment Volume*. Chapter 10, "Air Cleaners." American Society of Heating, Refrigerating and Air Conditioning Engineers, New York (1979).
27. Turk, A. "Adsorption." In: *Air Pollution*, Vol. IV: Engineering Control of Air Pollution. Edited by A. C. Stern, third edition, Academic Press, New York (1977).
28. *ASHRAE Handbook and Product Directory: 1976 Systems Volume*. Chapter 3, "All-Air Systems." American Society of Heating, Refrigerating and Air Conditioning Engineers, New York (1976).
29. Nevins, R. G. *Air Diffusion Dynamics*. Business News Publishing Company, Birmingham, NJ (1976).
30. Kusuda, T. "Control of Ventilation to Conserve Energy While Maintaining Acceptable Indoor Air Quality." *ASHRAE Trans.* 82 (Part I): 1169-1181 (1976).
31. Ogasaware, S., H. Taniguchi, and C. Sukehira. "Effect of Energy Conservation by Controlled Ventilation: Case Study in a Department Store." *Energy and Buildings* 2 (1): 3-8 (1979). □□