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STATUS -- VENTILATION MODELS FOR INDOOR AIR QUALITY

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ABSTRACT

As measurements are essential for performance assessment, models are essential for design. Both empirical and rational models are being developed for predicting the effectiveness of ventilation for acceptable indoor air quality. In this status report, models for contaminant generation rates, and dilution and removal control are introduced through a simple, one-compartment model. Models for predicting air quality within and between occupied spaces are then reviewed based upon assumptions of uniform and non-uniform mixing, including respiration requirements, dynamics of window and doors, pressurization control, and thermal interactions. Systems models are then reviewed including psychrometrics, dilution and removal control, and optimal control strategies. From this status review, areas for further development are identified.

INTRODUCTION

Air quality may be defined as the nature of air that affects an individual's health and well-being. More technically, air quality is an indicator of how well air satisfies three requirements for human occupancy: 1) Thermal acceptability; 2) Normal concentrations of respiratory gases (i.e., oxygen and carbon dioxide); and 3) Suppression of other contaminants below levels that are deleterious or cause odor discomfort.

Prescriptive methods to ventilate occupied spaces for acceptable air quality can be traced back to about 1600 AD, when King Charles I declared that ceilings had to be at least 10 feet high and windows had to be higher than they were wide. Current building codes have not progressed much since then. For example, model codes in the US state that occupied spaces shall be provided with:

> "Natural light by means of exterior glazed openings with an area not less than one tenth of the total floor area, and natural ventilation by means of openable exterior openings with an area of not less than one twentieth of the total floor area or shall be provided with artificial light and a mechanically operated ventilating system...supplying a minimum of 5 cubic feet per minute of

> outside air with a total circulated of not less than 15 cubic feet per minute per occupant in all portions of the building during which time as the building is occupied" (ref. 1).

Ventilation codes and standards are intrinsically prescriptive. For example, the "Ventilation Rate Procedure" in the ASHRAE Standard 62-1981 (ref. 2) specifies the amount of outdoor air to be supplied to various occupied spaces. Such prescriptive codes and standards provide design criteria which can be inspected and evalualted during design and construction, but they provide no assurances that occupant exposure will be acceptable. Prescriptive codes and standards are conventionally established by consensus processes which are highly dependent upon historically accepted values and depend little upon scientific or mathematical techniques such as modeling.

Conversely, performance criteria and standards have been established by various governmental agencies for outdoor air and for occupational workspaces to minimize human exposure to air pollutants. ASHRAE 62-1981 also introduced an "Indoor Air Quality Procedure" which recommends both objective and subjective criteria with which to evaluate the performance of occupied, non-industrial spaces. These criteria and standards are more frequently based on scientific and medical data than are prescriptive codes and standards and they can be enforced when the place is occupied, but they offer little guidance on how to design or construct systems that will provide the required control.

The dichotomy of these codes and standards provides an excellent example of the need for an holistic basis for ventilation and indoor air quality control. The development and application of validated mathematical and physical models can provide the designer, the building operator, the occupant, and the public official with tools that can relate the design of the physical system to the expected exposure of the occupants, the related dose, and the resultant human response.

FUNDAMENTAL MODELS

A simple, one-compartment model of an occupied space, Fig. 1, demonstrates the various factors that affect indoor exposure: indoor and outdoor sources, and methods of source, removal and dilution control.

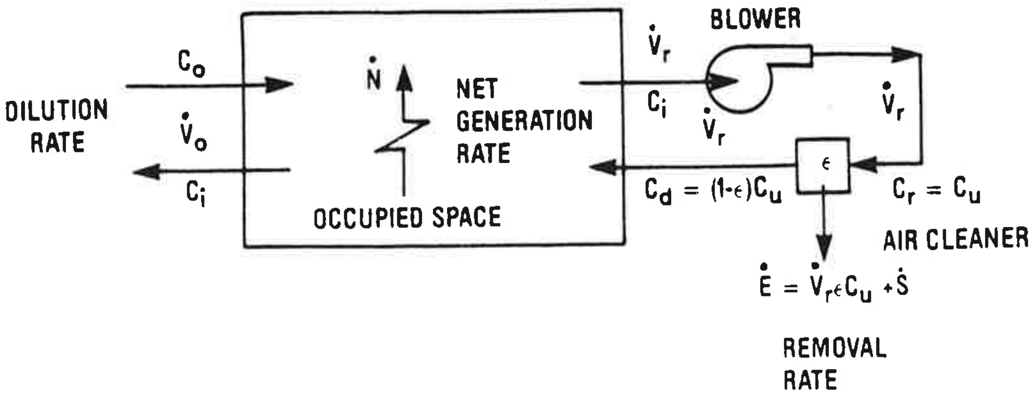

Fig. 1. One-compartment, uniformily-mixed, steady-state model for indoor air quality control.

A mass-balance of this uniformly mixed space in steady-state may be expressed as:

$$\Delta C = C_i - C_o = \frac{\dot{N} - \dot{E}}{\dot{V}_o} \tag{1}$$

This model identifies the three most common methods of control (i.e., source control, removal control, and dilution control). In this model, source control, $\dot{N}$, may be represented by isolation, product substitution, or local exhaust; removal control, $\dot{E}$, may be represented by passive mechanisms such as settling or sorption, and active mechanisms such as fan-filter modules, clean benches, or central forced air systems with recirculated air; and dilution control, $\dot{V}_o$, may be represented by infiltration, natural ventilation, or mechanical ventilation.

This model also indicates that the indoor air concentration of a contaminant, C_i, should be expected to exceed the outdoor concentration, C_o, unless the removal rate exceeds the net generation rate or the dilution rate is infinitely large. Although the latter control method is impractical, the former method is commonly used for applications such as cleanrooms. This relationship also identifies a reasonable control strategy: to achieve an acceptable ΔC, first attempt to reduce the net generation rate, then apply techniques that will increase the removal rate, then use techniques that will increase the dilution rate as necessary.

Models for Generation Rates

Maybe the most significant factor to be quantified in air quality models, the generation rates of contaminants establish an essential link between the concentration to be maintained indoors (i.e., performance criteria) and the conventional methods of removal and dilution control (i.e, prescriptive criteria). In this paper, the net generation rate, $\dot{N}$, is considered to be the difference between the emission rate from the indoor source (sometimes called the "Source Strength") and the remission rate by local source control (e.g., local exhaust). Thus, the net generation rate may be considered to be the factor that causes exposure: If the net generation rate can be eliminated, exposure will not occur; otherwise, the contaminant must enter the occupied space for removal or dilution, and some exposure must exist if occupants are present.

Literally thousands of gases, vapors, particulates, and radionuclides are emitted indoors from three primary sources: 1) the human occupants; 2) materials and furnishings within the occupied space; and 3) processes conducted within the occupied spaces (ref. 3). The indoor sources of these emissions vary from "single point" (e.g., occupants, tobacco smoking) to large "surface areas" (e.g., building material outgassing). Contaminant emissions may also be independent of other indoor environmental conditions (e.g., tobacco smoking), or they may be interactive with the environment (e.g., bacterial growth, formaldehyde outgassing).

Independent Emissions. The simplest assumption for modeling emissions is that the rates are independent of other environmental factors. With this assumption, contaminant emission rates may be considered to be constant for specific time intervals which are dependent on the mobility of the source (e.g, human occupants) or by the frequency of the occurance (e.g., cigarette smoking). Emission rates from these types of sources are usually normalized to a specific factor such as mass rate per occupant for human emissions, or total mass per cigarette. Therefore, the calculation of emission rate profiles from these sources also requires predictions of the activities which occur in the space being evaluated. Some examples of independent emission rates are shown in Table 1.

TABLE 1

Examples of independent emission (i.e., net generation) rates from indoor sources of contaminants (Ref. 3)

Source	Net Generation Rate	Comments
Human Activity		Adult,
Respiration	O_2 = -38 g/hr per	sedentary
	CO_2 = 32 g/hr per	activity;
	H_2O = 12 g/hr per	values may
Skin Transport	H_2O = 17 g/hr per	increase 10x
	Particulates = 7 x 10^6	due to
	Skin scales/min per	activity.
	Bacteria = 4/skin scale	
Building Materials		
Masonry	Radon = 0.2 - 2.0 pCi/kg hr	
Wood products	HCHO = 0.035 - 0.41 mg/hr m^2	
Processes		
Tobacco Smoking	Particulates = 31.0 mg/cig	
	CO_2 = 443 mg/cig	
	CO = 51.6 mg/cig	
	Pyridines = 1.3 mg/cig	
	Aldehydes = 114 ug/cig	
	NO_x = 79 ug/cig	
	Hydrogen cyanide = 65 ug/cig	
	Acrolein = 11 ug/cig	

Interactive Emissions. The assumption that emissions are independent of environmental factors simplifies calculations, but significant errors in predicting exposures can result. Two types of interactions can be identified: 1) interaction with thermal factors and 2) interactions with other contaminants.

Formaldehyde is an example of a contaminant affected by thermal interactions. Net generation rates have been reported to double as the indoor dry-bulb temperature increases 6 K or the relative humidity increases 30% (ref. 4). Moreover, HCHO emissions have been related to the cyclic differences in temperature and moisture content between the occupied space and the material containing the HCHO.

Radon is an example of a contaminant affected by interactions with other contaminants. When radon is in equilibrium with its progeny, due primarily to high concentrations of airborne particulates, the radiation dose due to the progeny may be 500 times that due to the radon itself (ref. 5). Moreover, the net generation rate of radon into the occupied space tends to increase when the moisture content of the soil decreases, the moisture content of building materials increases, or the indoor relative humidity decreases.

The development of quantitative relationships for describing emission rates as functions of environmental variables is just beginning. However, it now seems reasonable to assume that interactive models will require simultaneous solutions of thermal and mass balances.

Models for Removal Control

In its simplest form, the removal rate of particulate and gaseous contaminants can be estimated from the term E in Eq. 1. These removal rates may be considered to be comprised of two components: passive removal mechanisms, $\dot{S}$, and active removal mechanisms, $\dot{F}$.

Passive Removal. Within indoor spaces, the concentrations of particulate and gaseous contaminants may be reduced by settling, condensation, ion diffusion charging, thermophoresis, photophoresis, and sorption (ref. 6). For the case of particulate settling, the passive removal rate, $\dot{S}$, (sometimes called the "Sink Strength") may be expressed as (ref. 7):

$$\dot{S} = G_r C_i \tag{2}$$

where G_r = Settling factor ($G_r = KA_sV_t$)

K = Settling rate coefficient, defined as a control factor which can be minimized or maximized depending on the control objective.

A_s = Settling area (i.e., surface area of settling).

V_t = Terminal velocity of the particulates in "still air", which can be estimated by the Stokes relationship and Cunningham correction factor.

Evidence is beginning to accumulate which indicates that reactive gases, such as ozone, sulfur dioxide, nitrogen dioxide, and carbon monoxide may be adsorbed or react with building surfaces, furniture and furnishings within buildings (ref. 8). As these contaminants are removed from the air, their reactions with these materials are likely to lead to corrosion and deterioration. Procedures to calculate passive removal rates of gases and vapors from occupied spaces are not well known at this time.

Active Removal. The only practical method available today by which indoor concentrations can be controlled to values below those found outdoors is active removal. This control method has been employed for several years in critical applications such as cleanrooms and critical care facilities in hospitals. To date, these applications have been primarily limited to removal control of particulates; gas and vapor removal control is generally not practiced in non-industrial applications.

The active removal rate may be expressed as:

$$F = \varepsilon V_r C_u \qquad (3)$$

where ε = Air cleaner efficiency = (1 - P).

P = Penetration of contaminant through air cleaner = C_d/C_u.

V_r = Volumetric air flow rate through air cleaner.

C_u = Concentration of contaminant, upstream of air cleaner ($C_u = C_i$).

C_d = Concentration of contaminant, downstream of air cleaner.

The air cleaner efficiency, ε, must be expressed in terms of the contaminant to be removed. Concentrations of gases, vapors, and non-viable particulates are usually expressed as a volume or mass of contaminant per unit volume of air (e.g., parts per million, ppm, or ug/m^3), or mass of contaminant per unit mass of air (mg/kg air). Other examples are number of particles of a specific size range per unit volume of air (e.g., No./m^3 air) or number of colony forming units of viable particles per unit volume of air (CFU/m^3 air). Thus, contaminant penetration and air cleaner efficiencies must be carefully expressed in consistent terms.

In applications where particles, gases, and vapors must be removed, it is often necessary to install the air cleaners sequentially, first to remove particulates then to remove the gases and vapors. The overall efficiency of a sequential array of air cleaners may be expressed for a specific contaminant as:

$$\varepsilon = 1 - \prod_{i=1}^{n} (1 - \varepsilon_i) \tag{4}$$

Estimates of ε or ε_i can be made from rational models such as the Darcy and Chen Equations for media-type filters (ref. 9); the Deutsch Equation for electronic air cleaners (ref 10); and from the Freundlich and Langmuir Equations, the Brunauer, Emmett and Teller (BET) Equations, or the Turk Equation (refs. 11, 12) for gas and vapor removal devices. However, the efficiencies for particulate, gas, and vapor removal are more frequently estimated from empirical data for the various types of devices.

As shown in Eq. 3, the active removal rate is a function of three variables. Therefore, a high air cleaner efficiency may not be sufficient for removal of the contaminant load. Rather, for a specified contaminant concentration challenging the air clearer, C_u, the product of the air flow rate through the air cleaner and its efficiency, εV_r, is the critical factor. In other words, for a given room concentration, $C_u = C_i$, the same removal effectiveness can be expected from two devices, one of which has half the efficiency of the other if it delivers twice as much air. It should also be noted that for the same removal effectiveness, εV_r, the removal rate will decrease as the room becomes cleaner (i.e., C_u decreases).

Models for Dilution Control

As mentioned in the introduction, the historical and most commonly used method of air quality control is the introduction of outdoor air for dilution of the indoor concentrations. Dilution control, V_o Fig. 1 and Eq. 1, may be the most energy intensive and costly of the control methods available today. Conversely, dilution control can provide energy savings (e.g., use of an economizer cycle) while improving indoor air quality. Dilution control can be provided by passive methods (i.e., infiltration and natural ventilation) and by active methods (i.e., mechanical ventilation).

Infiltration and Natural Ventilation. Infiltration is usually considered to be unwanted air leakage through cracks, joints, and connections in the building envelope, whereas natural ventilation is usually considered to be desired air exchange through intentional openings in the envelope such as windows and doors.

Because of the large number of variables associated with infiltration and natural ventilation, completely rational models do not exist yet. However, significant work has been accomplished at the National Research Council of Canada, Division of Building Research, where semi-empirical models have been developed for predicting infiltration through low and tall buildings as functions of wind and stack (i.e., thermal) effects (ref. 13). The simpler and more common method of estimating infiltration rates for small buildings is the "Crack Method" which has been used for many years for estimating thermal loads and energy consumption (ref. 14).

The fundamental limitation of these methods is that they assume that the air is uniformly mixed within the occupied spaces. Evidence now exists which indicates that the turbulance intensity on windward sides of buildings causes sufficient variations in the pressure differences to affect the mixing within and between the occupied spaces (ref. 13). Moreover, the placement and utilization of windows, and exterior and interior doors significantly affects the natural ventilation rates of the occupied spaces. Needless to say, much additional work is needed in this area of modeling. It is obvious at this time, however, that the simple methods of estimating infiltration for energy calculations are insufficient for air quality modeling.

Mechanical Ventilation. For the simple case of no recirculated air within the occupied space (i.e., 100% outdoor air) and no removal control (i.e, $\dot{E} = 0$), Eq. 1 indicates that the concentration indoors varies inversely with the dilution rate, $\dot{V}_o$. This relationship is the basis for the prescriptive ventilation rates commonly specified in ventilation codes and standards. In the simple case of mechanical supply of outdoor air directly to a room or zone of a building, the dilution rate is usually expressed explicitly. However even in this case, the assumption of uniform mixing within the occupied space may lead to significant errors in correctly estimating the amount of dilution air supplied to a given location within the occupied space.

ROOM MODELING

In ASHRAE Standard 62-1981, ventilation is defined as: "Th process of supplying and removing air by natural or mechanica means to and from any space. Such air may or may not b conditioned". It also defines ventilation air as: "That portio of supply air which is outdoor air plus any recirculated ai that has been treated for the purpose of maintaining acceptabl indoor air quality". The combined strategies of removal an dilution control, shown in Fig. 1 and Eq. 1, are compatible wit these definitions. Yet, this model and the related engineerin definitions of ventilation are not necessarily synonymous wit the definition of ventilation commonly assumed by physiologist and life-scientists: "The inspiration of fresh air followed b the expiration of some alveolar gas". The fundamental differenc is that the engineer considers ventilation air as that enterin the room or zone with a building (i.e., room ventilation), whil the physiologist considers ventilation air as that bein respired (i.e, respiratory ventilation).

Ventilation Ratio

To couple the concepts of room and respiratory ventilation a two-compartment model, as shown in Fig. 2, may be considere to consist of a room compartment, and a respiratory compartment Assuming uniform mixing within each compartment, a steady-stat ratio between the room and respiratory ventilation rates (i.e. $\dot{V}_o/\dot{V}_a$ = "Ventilation Ratio") can be expressed as follows:

$$\frac{\dot{V}_o}{\dot{V}_a} = \frac{C_e - C_i}{C_i - C_o} \qquad (5$$

where $\dot{V}_o$ = Room ventilation rate.

$\dot{V}_a$ = Respiratory ventilation rate.

C_e = Contaminant concentration in expired air.

C_i = Contaminant concentration in inspired air = Contaminant concentration in uniformily mixed room air.

C_o = Contaminant concentration in outdoor air.

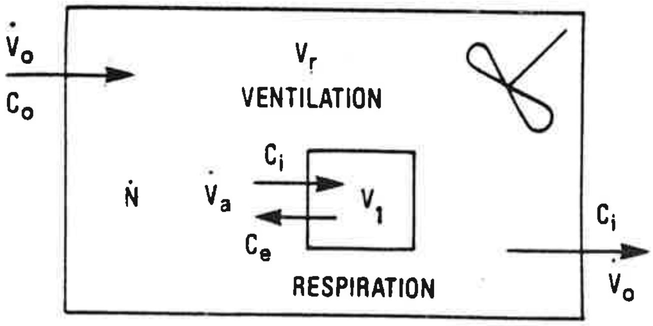

Fig. 2. Two-compartment model of the coupling between room and respiratory ventilation.

By comparing concentrations of indoor contaminants with the resultant ventilation ratios, a rationalization between performance and prescriptive criteria is possible. For example, ventilation ratios for various CO_2 standards are shown in Fig. 3, assuming that the concentrations of the expired air and outdoor air are 3.8 and 0.04%, respectively:

- NASA has set a maximum long-term exposure level of 1.0% (ref. 15).
- The US Navy has set maximum long-term exposure levels of 0.5 - 0.7% (ref. 16).
- The US OSHA has set a time-weighted average of 0.5% (ref. 17).
- ASHRAE has recommended that continuous exposures to concentrations of 0.25% not be exceeded (ref. 2).
- The Japanese indoor standard in now set at 0.1% (ref. 18).
- The World Health Organization has recommended that concentrations of 0.1% not be exceeded (ref.19).
- The Scandinavian Countries are currently recommending that concentrations of 0.07 - 0.08% not be exceeded (ref. 20).

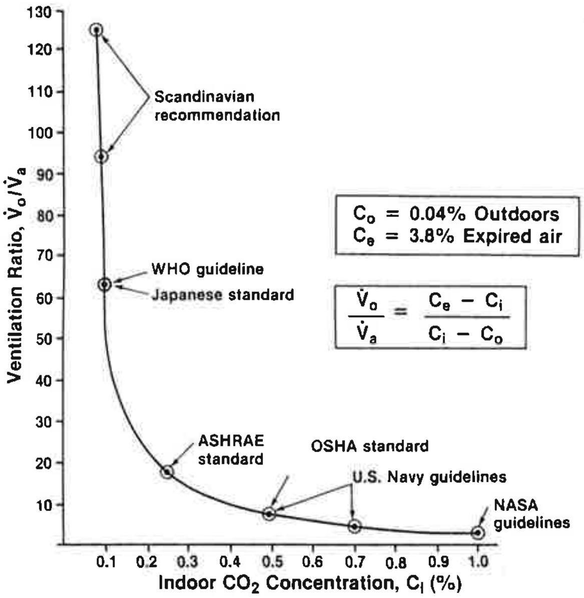

Fig. 3. Relationship between the ventilation ratio, $\dot{V}_o/\dot{V}_a$, and the concentration of CO_2 within a uniformly-mixed occupied space, C_i.

At sedentary activity, the normal respiration rate of an adult is about 0.5 m^3/hr (0.3 cfm). Thus, a comparison between the CO_2 concentrations and the ventilation ratios in Fig. 3 indicates that the room ventilation rates for sedentary adults vary from 1.5 m^3/hr (0.9 cfm) to meet the NASA criterion, to 62 m^3/hr (37 cfm) to meet the recommendations of the Scandinavian Countries. Moreover, respiration rates can increase by factors 10 greater than sedentary, thus, required room ventilation rates must also compensate for the increased activity levels if the CO_2 concentrations are to be maintained.

While the NASA, Navy, OSHA, and ASHRAE standards apparently have been set for direct exposure to CO_2, the other recommendations have probably been set to correlate with concentrations of other contaminants that also exist with the specified CO_2 levels and that are the sources of objectionable indoor conditions (e.g., odorous, stale air). Moreover, evidence also exists that interactive effects may be detected between thermal conditions and CO_2 concentrations (ref. 21). If thermal control and removal control of odorous contaminants are employed, would it be acceptable to reduce the room ventilation rate as this model implies? To answer this question in terms of design alternatives, models of human responses to the concentrations and the thermal interactions will be needed.

Ventilation Efficiency

The models presented so far have assumed that the air within the occupied space is uniformly mixed. However, as described by Skaret in the previous paper and by others at this symposium, thermal and contaminant stratification can occur within the occupied space, resulting in occupant exposures much higher than predicted by models that assume uniform mixing.

When heating and cooling loads are not controlled by forced air systems, natural ventilation may be the primary means of dilution control. However, in most cases today, especially in non-residential facilities, the primary method of control is mechanical ventilation which may incorporate a combination of dilution and removal control strategies into the forced air heating and cooling systems. Whether natural or mechanical ventilation or a combination is employed, the effectiveness of the system for air quality control is dependent upon two system characteristics: the room air exchange rate, and the air flow patterns within the room.

If the room air distribution is not sufficient to dilute or remove contaminants from the location of most likely (i.e., critical) exposure, the effectiveness of the system will be impaired as excessive air exchange rates will probably be used as compensation, with the expected results of increased energy consumption, non-uniform mixing, and drafty or uncomfortable subjective responses. Thus, air distribution patterns within the room may be as important to the effectiveness of the ventilation system as the room air exchange rate.

For purposes of this discussion, ventilation efficiency is defined as the fraction of the ventilation air supplied to the room that actually ventilates the occupied space. Note that this definition does not specify the quality of the ventilation air.

Thermal Control. The location of the terminal air devices (e.g., supply diffusers, return and exhaust grilles and registers, etc.) is critical if high ventilation efficiencies are to be achieved. Empirical models for use in selecting the location of supply diffusers for acceptable thermal control have been available for more than 10 years (ref. 22). Systems designed by these methods are usually based on the assumption that the locations of the return and exhaust air devices are not important if the supply diffusers are properly located tc provide acceptable thermal conditions. As a result, many systems exist today which provide thermal uniformity within the occupied space, but which also have significant contaminant stratification. In these systems, the contaminants can reach thermal equilibrium within the occupied space and therefore they are not detected by thermal analysis.

Air Quality Control. For acceptable air quality control, care must be taken in the location of the supply and return or exhaust devices. For example, the common practice of locating both supply and return air devices in the ceiling, or on opposing high sidewalls has been shown to result in less than 50% of the supply air reaching the occupants (i.e., ventilation efficiency <50%) (ref. 21). In this rational model shown in Fig. 4, the ventilation efficiency, n_v, has been derived as an infinite series expression in terms of a room stratification factor, S, and a system recirculation factor, R:

$$\begin{aligned}\dot{V}_v &= \dot{V}_o - \dot{V}_e \\ &= \dot{V}_o(1 - S(1 - R) - S^2R(1 - R) \\ &\quad - S^3R^2(1 - R) - \ldots)\end{aligned} \tag{6}$$

Therefore:

$$n_v = \frac{\dot{V}_v}{\dot{V}_o} = \frac{(1 - S)}{(1 - SR)} \tag{7}$$

where $\dot{V}_v$ = Flow rate of ventilation air to occupied space.

$\dot{V}_o$ = Flow rate of outdoor air into the system.

$\dot{V}_e$ = Flow rate of outdoor air that is exhausted without mixing in the occupied space.

n_v = Ventilation efficiency.

R = Fraction of total supply air that is recirculated.

$S = (I_o - I_\infty)/I_o$ (8)
= Fraction of the total supply air that is stratified and bypasses the occupied space.

I_o = Initial apparent tracer gas decay rate.

I_∞ = Steady-state tracer gas decay rate.

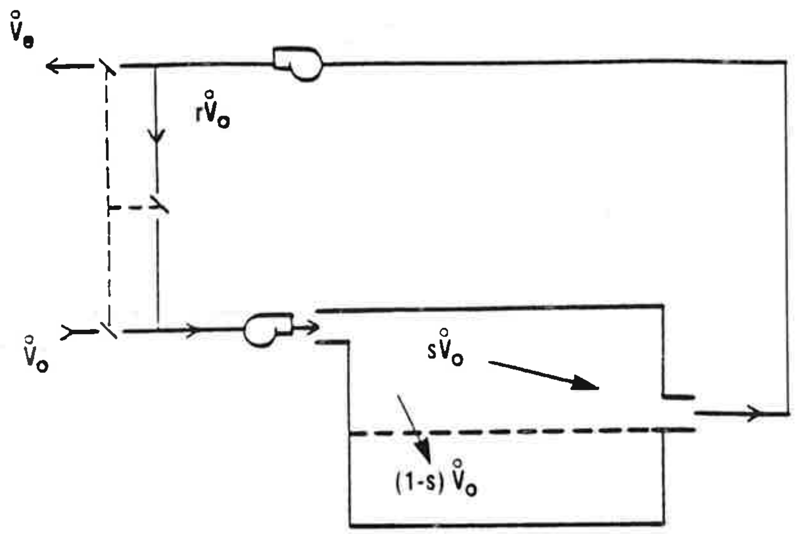

Fig. 4. Two-compartment model of recirculation system with stratification from which to predict ventilation efficiency.

In this model, the stratification factor, S, can be associated with the location of the supply and return air terminal devices, and the recirculation factor, R, can be associated with the most common types of ventilation systems. It should be noted that this model lends itself more to providing information for prescriptive criteria (i.e., ventilation rates) than for performance criteria (i.e., acceptable concentrations or exposures). Work is needed in the development of models which will link the prescriptive and performance criteria, yet be sensitive to physical parameters such as building and system characteristics.

SYSTEM MODELING

With the exception of room type air cleaning devices, most procedures to maintain acceptable air quality within occupied spaces will involve the application of the HVAC systems. Even the use of local exhaust fans requires the replacement of the exhaust air with outdoor air which must be heated or cooled to provide thermal acceptability. Thus, the interaction of air quality and thermal control for occupied spaces is closely coupled.

The basic components of HVAC systems include: 1) An energy supply (e.g., gas or electricity); 2) Energy conversion systems (e.g., hot water generator or boiler, refrigerating system, humidifier, blowers, pumps); 3) Thermal and ventilation transport mechanism (e.g., ductwork and piping); and 4) Control systems (e.g., thermostats, humidistats, valves, dampers, air cleaners).

Central, forced-air systems may be constant air volume (i.e., CAV) or variable air volume (i.e, VAV) systems. The basic difference is that the CAV system provides the same amount of air flow rate into an occupied space, independently of thermal load, whereas the VAV system reduces the air flow rate into the occupied space as a function of thermal load. The VAV system is inherently more energy efficient for applications where variable thermal loads exist, as the reduction in fan-power can be significant. Conversely, because VAV systems can reduce their air flow rates to occupied spaces in response to thermal loads, they may at times operate at less than required ventilation capacities.

For either a CAV or a VAV system, shown in Fig. 5, the thermal and air quality interactions between the HVAC system and the room has been expressed as (ref. 23):

$$K = \frac{(1 - \varepsilon)H + M + Q}{(1 - \varepsilon)H + M + \varepsilon} \tag{9}$$

where K = Room acceptability factor = x_1/x_o

H = Outdoor air ratio = $\dot{m}_o/\dot{m}_m = (h_r - h_m)/(h_r - h_o)$

M = Passive to active air exchange ratio = $\dot{m}_i/z\dot{m}_m$

Q = Room Contamination factor = $\dot{N}/z\dot{m}_m x_o$

ε = Air cleaner efficiency

x = Contaminant concentration per unit mass of air

h = Enthalpy per unit mass of air

$\dot{m}$ = Mass air flow rate

$\dot{N}$ = Contaminant generation rate within room

z = fraction of room supply air to system supply air flow rate = $\dot{m}_s/\dot{m}_m$)

and subscripts

i = Infiltration and natural ventilation rates into the room (i.e., psychrometric condition 5 in Fig. 5).

o = Outdoor air (i.e., psychrometric condition 5 in Fig. 5).

m = Mixed air in HVAC system (i.e., psychrometric condition 6 in Fig. 5).

r = Recirculated air into HVAC system (i.e., psychrometric condition 4 in Fig. 5).

s = Supply air to room (i.e., psychrometric condition 8 in Fig. 5).

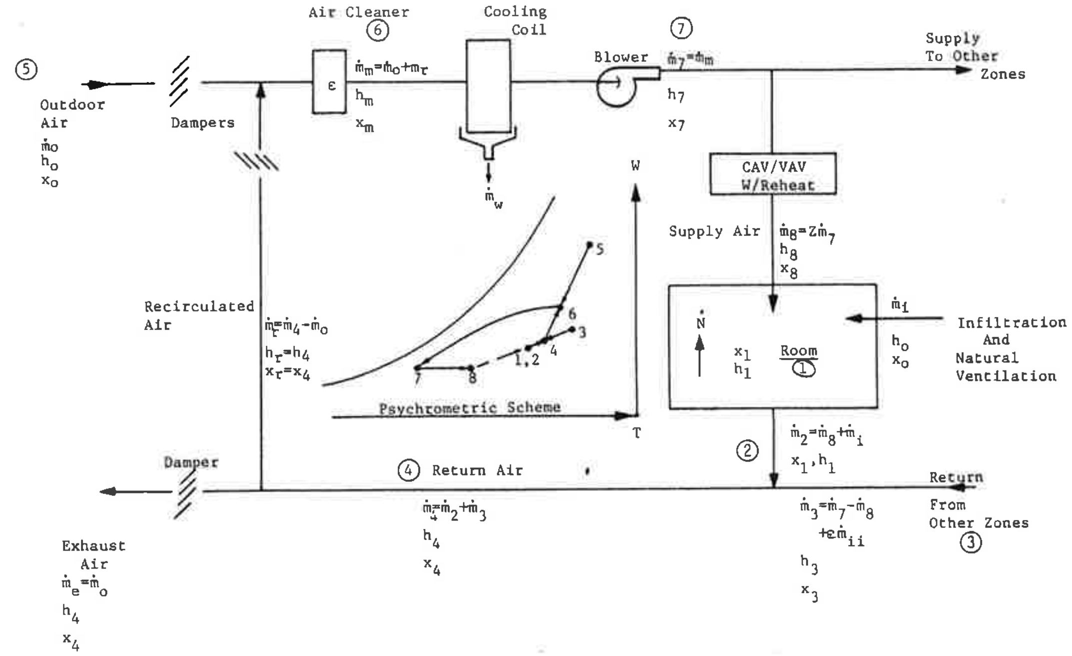

Fig. 5. Schematic and psychrometric representations of an HVAC system which provides thermal and air quality control to occupied spaces.

Note that it is necessary to express the variables in Eq. 9 in terms of air mass rather than air volume, as isothermal conditions cannot be assumed throughout the systems and changes in air density must be considered.

Because of the number of variables involved in this rational model of thermal and air quality interactions, simple relationships are hard to describe. However, computer simulations make optimization of these factors practical. It may be noted here, however, that Eq. 9 contains several of the factors needed for a complete system analysis:

- The acceptability factor, K, may be selected to be greater or lesser than one as required, dependent on contaminant, health risk, and comfort.

- The outdoor air ratio, H, can be determined as a function of outdoor air psychrometrics, h_o, as a function of dilution requirements for thermal or air quality control, h_m, and as a function of thermal comfort, h_r.

- The passive to active air exchange ratio, M, can be determined as a function of wind and stack effects on infiltration and natural ventilation, $\dot{m}_i$, and as a function of the variable or constant air flow rate to the room, $z\dot{m}_m$.

- The room contamination factor, Q, can be determined as a function of the contaminant generated within the room, $\dot{N}$, and as a function of the the amount of outdoor contaminant transported into the room by the HVAC system, $z\dot{m}_m x_o$.

- The air cleaner efficiency, ε, can be determined as a function system pressure drop.

This model is an example of a simple, steady-state approach to system simulation. Significant additional work is needed in coupling the system models to models of the occupied space so that stratification factors can be incorporated.

CONCLUSION

In this status report, developments in predicting system performance to meet "performance criteria" have been discussed. It may be generally concluded that significant progress has been made in the last ten years in this regard. However, much work is still needed before we will be able to predict indoor air quality with the same sense of confidence that we have in predicting the thermal performance of a building. Specifically, additional work is needed in four areas:

1) Human Responses. Predictive models of health and comfort responses to exposure of indoor contaminants are needed. Models for predicting thermal sensations may serve as a reference.

2 Occupied Spaces. If predictions of exposures are to be made, at least two sets of models are needed:
   - Models of contaminant emission and net generation rates which include interactive effects of thermal and other contaminants.
   - Interactive models of thermal and contaminant stratification which include functional relationships between locations of terminal air devices and imposed loads.

3) Systems. If control of indoor air quality is to be simulated, dynamic models to predict the performance of HVAC systems in response to outdoor and indoor disturbances are required. These models should incorporate:
   - o Environmental conditions required for acceptable human response.
   - o Characteristics of the occupied spaces.
   - o Economic and Energy implications.

4) Implementation. If models of ventilation for indoor air quality are to be implemented on a wide basis, work is needed to:
   - o Validate proposed models and simulation techniques.
   - o Develop design and operation procedures which incorporate the validated models and simulation techniques.

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