MACROSCOPIC MODEL OF INDOOR AIR QUALITY AND AUTOMATIC CONTROL OF VENTILATION AIRFLOW

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ABSTRACT

Computer models are useful in understanding how pollutants are distributed in the indoor environment. Two types of models for indoor air quality that are currently utilized are microscopic models, which use a two- or three-dimensional fluid mechanics code to describe airflow and pollutant distribution in a ventilated room, and macroscopic models, which describe pollutant transport through a multiple-zone ventilation system. In this paper, a macroscopic pollutant transport model is employed to simulate several methods aimed at control of indoor pollutants.

An office zone and a meeting room zone, including the transient airflows and heating and cooling loads, together with the HVAC system, were modeled. Comparisons were then made between several strategies for outside airflow rate to determine their ability to control indoor pollutant levels and their impact on heating and cooling energy use. The results of this study indicate that an automatic outside airflow control strategy based on CO_2 concentration can control pollutants as well as the strategy of a fixed flow of outside air such as that recommended in the 1989 ASHRAE standard. The potential heating and cooling energy savings for the automatic control strategies, as compared to the fixed flow rate recommended by ASHRAE, ranged from 10% for CAV systems to as much as 50% for VAV systems.

INTRODUCTION

The current emphasis in building ventilation control is a quest for an indoor environment that is energy efficient and healthy for building occupants and enhances worker productivity. Before solutions to indoor air quality problems can be addressed, some discussion of the relevant concepts is appropriate.

The indoor air environment is quite different from what exists outdoors. First, there are fewer air changes; a ventilated building may have between 0.4 and 10 air changes per hour (ach), while outdoors, an 8 kmh (5 mph) breeze will result in 3,600 air changes in an hour (Meyer 1983). Second, indoor air is not part of the biologic and climatic air cycles, so there is no natural purification process. Lastly, the relatively constant temperatures that exist indoors reduce convection and turbulence and lead to poor mixing of the air.

The results of a survey of 466 buildings performed by the National Institute for Occupational Safety and Health (NIOSH) by Stolwijk (1987) are summarized in Table 1. According to this study, inadequate ventilation was found to be responsible for just more than half of the cases of "sick" buildings. Problems caused by contamination released inside the building and building material contamination are probably also an indication of inadequate ventilation, which would bring the total fraction of indoor air quality problems related to insufficient flow of outdoor air to 72%.

Adequacy of ventilation is not always indicated by a simple measure of outdoor airflow rate. A study by Dillon et al. (1987) of the airflow patterns in one office area showed that the observed air change rate at the level of occupancy (below the office partitions) was about one-half of the average obtained strictly from the airflow rate. This was largely due to the short-circuiting of the airflow from the supply directly to the exhaust. The amount of short-circuiting existing in a room is a function of the placement of inlet and outlet vents, the obstructions present, and the temperature profile in the room.

Many solutions have been proposed for the indoor air quality problems that have surfaced since the early 1970s. The most obvious solution, and the one applied most often prior to the last two decades when energy use became a concern, is to simply increase the amount of outdoor air ventilation. This will probably fix the problem—but at a high energy cost. There are alternatives that reduce the sources of pollution. Materials that give off objectionable gases could be coated to reduce the rate of gas evolution, or the materials could be replaced with less objectionable alternative materials. Localized sources of pollutants, such as copy machines, can be isolated from the rest of the circulation airstream by providing an area exhaust to the outside.

Improper maintenance or design of ventilation systems can also lead to indoor air pollution problems. Ventilation ducts have been found to contain dirt and biological growth that give off contaminants. In some buildings, outdoor air

TABLE 1
Sources of Indoor Air Quality Problems

SOURCE OF PROBLEM	% OF CASES
Contamination Released Inside the Building (copy machines, tobacco smoke, cleaning agents)	17
Contamination from the Outside (car exhaust, recycle from building ventilation exhaust)	11
Building Material and Fabric Contamination (formaldehyde, solvents, glues, fiberglass)	3
Microbal Contamination (bacteria, etc., from ducts, humidifiers, cooling towers)	5
Inadequate Ventilation (inadequate intake, poor maintenance, poor distribution)	52
Unknown	12

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inlet vents have been positioned either too close to the exhaust vent or near a source of outdoor pollution, such as a loading dock where vehicles are frequently idling. Room inlet diffusers and exhaust vents are sometimes located so that a significant fraction of the ventilation air bypasses the occupied zone. Design problems such as these are expensive to remedy after the fact. When other air quality solutions are not feasible or do not eliminate the problem completely, the only alternative is to adjust the outdoor air ventilation rate.

The use of CO_2 concentration as a measure of the quality of indoor air is an idea that dates back to Meyer (1983). Since the mid-1970s, several studies have determined that CO_2 concentration can be a reliable indicator of indoor air quality where the major sources of indoor pollution are related to occupancy (Berk et al. 1979; Liptak 1979). Also, since the metabolic production and the outdoor air concentration of CO_2 are known, measuring the indoor CO_2 concentration provides a means of determining the actual outdoor air exchange rate for the space. This method has been shown to compare favorably with tracer gas methods using SF_6 and airflow measurements (Turiel and Rudy 1985). Figure 1 shows the correlation between occupancy and CO_2 concentration during the course of a day in an office waiting room.

With the above limitations in mind, several systems have been proposed for controlling the amount of outdoor air delivered to a ventilated space based on the CO_2 concentration present (Liptak 1979; Kusuda 1976; Vaculik 1987). The control value could be either a limit on concentration or the rate of change of concentration. The system could respond by increasing or decreasing the outdoor airflow in proportion to the concentration. A minimum flow of outdoor air could also be provided to account for indoor pollution sources that are not related to occupancy. This would provide a flow of outside air that is adjusted automatically for varying occupancy and would make use of the storage capacity of the air space before an increase in outdoor airflow rate is implemented. A 1976 study by the National Bureau of Standards (NBS, now the National Institute of Standards and Technology, NIST) for a specific building showed that such a system could save up to 40% of the energy cost over a constant outdoor airflow rate system based on 10 L/s (20 cfm) of outside air per person (Kusuda 1976).

In a 1985 study, Sterling and Sterling (1985) showed that CO_2 responds differently to changes in ventilation than do hydrocarbons, CO, and particulates, which are all dependent on outdoor concentrations. It appears that CO_2 can be used as an indicator of the relationship between the outdoor air ventilation rate and occupancy but not as an overall indicator of air quality, since the CO_2 level will not respond to changes in indoor pollutant sources unrelated to occupancy or outdoor pollutant sources.

APPROACH

This paper describes models for the dispersal of indoor pollutants by a ventilation system through a multiple-zone building (Knoespel 1990). Various methods of controlling the amount of outdoor air introduced into a building's ventilation system are explored, and the corresponding annual energy use of these methods is compared. Methods for controlling outdoor airflow use the level of CO_2 as an indicator of the amount of occupant-related pollutants in an indoor environment. The objective is to determine if a strategy to control outdoor airflow based on the actual level of pollutants present can result in meaningful energy savings when compared to a constant-airflow system based on the 1989 ASHRAE ventilation standard (ASHRAE 1989).

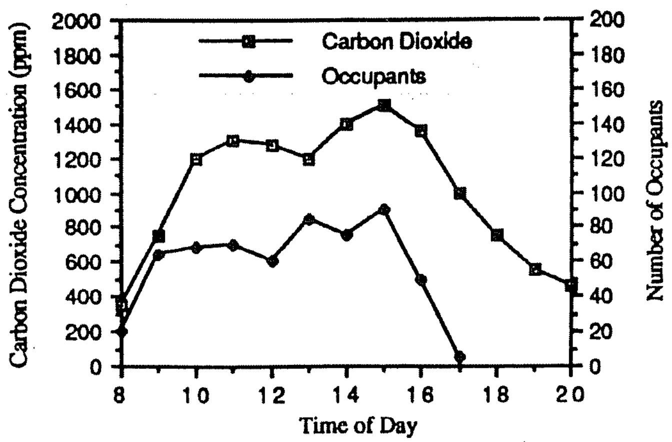

Figure 1 *Variation of CO_2 concentration and occupancy*

POLLUTANT TRANSPORT MODEL

Model Development

Although room airflow models have been in existence for nearly 20 years, the modeling of multiple-zone airflows is a more recent activity, with the earliest documentation only having been published about 10 years ago. The transport of indoor air pollutants through several zones within a building has been studied using computer models for only about the last five years.

Two basic types of multi-zone air quality models are presently in use. The first type of multiple-zone airflow model uses a network technique. The airflow in a building is described as a network of interconnected nodes representing each building zone, the ventilation system ductwork, and ambient conditions. Each node is assumed to be at a uniform pressure, temperature, and pollutant concentration, and nodes are connected by airflow paths such as windows, doors, ducts, and infiltration paths. The sum of the mass or energy flows at each node is zero, and each path includes a resistance that relates the mass flow rate to the pressure drop between nodes. Models of this type will generally include the driving forces of stack effect, wind pressure, and forced (circulation) airflow. The node and element equations are solved to yield the steady-state mass flows between all nodes and the pressures. For a unique solution, at least one nodal pressure (usually for the ambient node) must be specified. A model of this type is used in the NIST program AIRNET, described by Walton (1989a, b). In his survey of air infiltration and indoor air quality models, Haghighat (1989) describes several other programs of this type that are used for building energy use, air quality, and smoke migration studies.

The second type of multiple-zone air quality model is a simplification of the first type in that it assumes the interzonal airflows are already known. The model assumes that the nodes have a capacitance for pollutant and uses a pollutant mass balance for each node to arrive at pollutant concentrations. The equations for pollutant concentration as a function of time are in the form of an initial-value problem. This allows the determination of pollutant concentrations throughout a period of time during which the pollutant source and interzonal flows may be varying. The Environmental Protection Agency (EPA) program INDOOR uses this model (Sparks 1988).

The indoor pollutant transport module developed for a transient system simulation program (TRNSYS, UW 1988) is based on the nodal pollutant mass balance model, rather than the network model, for two reasons. First, since annual simulations were required, the nodal mass balance was the more practical model in terms of computational time. Second, the ventilation system's airflows and infiltration and interzonal airflows were input.

In addition to a variable ventilation system airflow, the pollutant transport module needed to be capable of simulating the variable pollutant source level in each zone corresponding to a changing occupation level in the zones throughout the day. It was also desirable to model the pollutant source and airflow characteristics that result in a non-uniform distribution of the pollutant. Lastly, as this was to be a transient simulation program for ventilation systems in large office buildings, the circulation time of the air through the ventilation system had to be included.

The pollutant transport terms in a ventilation system are shown in Figure 2. Air flows into and out of each zone, carrying with it the pollutants in various concentrations. For an individual zone, the possible airflow paths are the circulation flow in and out, infiltration flow in, and interzonal flow in and out. The zone may also contain a pollutant source.

The pollutant transport model is based on a mass balance of pollutant in each zone, which relates mass flow in, mass flow out, and the change in mass present in the zone as given by Equation 1. The subscript p in this and the following equations indicates pollutant quantities:

$$\frac{dm_p}{dt} = \dot{m}_{p,in} - \dot{m}_{p,out}. \tag{1}$$

The mass terms can be written in terms of volume concentrations, which are the partial volume of the pollutant divided by the air volume. The mass flow rate of pollutants can be represented by the mass flow rate of the airflow times the pollutant concentration. The density of air is essentially constant, and the mass flow rate of air can be replaced by the volume flow rate. The mass of pollutant in the room is volume concentration times the room volume. These assumptions allow the mass balance to be written as

$$\frac{d(CV_a)}{dt} = \dot{V}_{a,in} C_{in} - \dot{V}_{a,out} C_{out}. \tag{2}$$

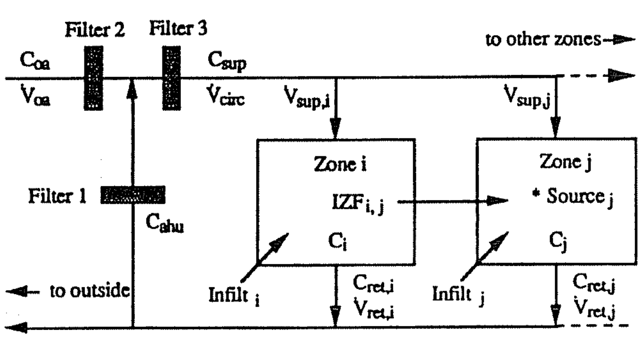

Figure 2 Ventilation system modeled

Both sides of Equation 2 are divided by the air volume in the zone (which is essentially equal to the nominal zone volume for dilute pollutant concentrations). The final form of the general pollutant balance equation for a zone as used in the pollutant transport model is shown in Equation 3:

$$\frac{dC}{dt} = \frac{\dot{V}_{a,in}}{V_a} C_{in} - \frac{\dot{V}_{a,out}}{V_a} C_{out}. \tag{3}$$

Equation 3 is now expanded to include all possible airflow paths into or out of zone i as shown in Figure 2. The result, Equation 4, has four inflow terms and two outflow terms. The volume flows are now all airflow rates, so the subscript a has been dropped:

$$\frac{dC_i}{dt} = \frac{S_i}{V_{a,i}} + \frac{\dot{V}_{inf,i}}{V_{a,i}} C_{oa} + \frac{\dot{V}_{sup,i}}{V_{a,i}} C_{sup} - \frac{\dot{V}_{ret,i}}{V_{a,i}} \varepsilon_{c,i} C_i + \frac{\sum_j (\dot{V}_{izf,j,i} C_j)}{V_{a,i}} - \frac{\left(\sum_j \dot{V}_{izf,i,j}\right) C_i}{V_{a,i}}. \tag{4}$$

The first term on the right-hand side of Equation 4 is the pollutant volume source located in zone i. The second term accounts for the air infiltration volume flow rate into the zone. The third term represents the air circulation flow through the zone with a variable-supply air concentration. The fourth term represents the return airflow. The average concentration in the zone is modified by the pollutant-removal effectiveness, ϵ_c for the zone. The removal effectiveness, as described by Seppänen (1986), is the return duct pollutant concentration in the zone divided by the average concentration, and it accounts for the fact that the pollutant may not be fully mixed in the zone air. Values of the effectiveness less than unity imply that, since the room concentration is larger than the exhaust concentration, there is "short-circuiting" of the supply air. Values of the effectiveness greater than unity imply that the source is located close to the return duct, and pollutants are effectively exhausted rather than mixing with room air.

The last two terms in Equation 4 are for the interzonal flows of pollutant. The next to the last term represents the volume airflow from zone j into zone i. This is multiplied by the average concentration in zone j and summed over all zones to arrive at the total interzonal pollutant flow into zone i. The last term represents the interzonal pollutant flow out of zone i.

The air volume flow rate returning to the ventilation system from zone i is calculated by a balance of all the other airflows for the zone:

$$\dot{V}_{ret,i} = \dot{V}_{sup,i} + \dot{V}_{inf,i} + \dot{V}_{izf,j,i} - \dot{V}_{izf,i,j}. \tag{5}$$

The supply air concentration is calculated by a two-step process. First, the concentration at the inlet of the air-handling unit C_{ahu} is calculated by summing the pollutant flows from all of the zones, as given by Equation 6. The term $C_{i,t-DT}$ represents the concentration that left zone i at time *t-DT*. *DT* is the return air time delay and is equal to the zone return air path volume divided by the return air volume flow rate for the zone:

$$C_{ahu} = \frac{\sum_i (\dot{V}_{ret,i}\,\varepsilon_{c,i}\,C_{i,t-DT})}{\sum_i \dot{V}_{ret,i}}. \tag{6}$$

The second step is to calculate C_{sup}, the supply air pollutant concentration given by Equation 7. This accounts for the fact that some of the circulation air may he exhausted to the outside and replaced with an equal volume of outdoor air:

$$C_{sup} = \frac{\left(\sum_i \dot{V}_{sup,i} - \sum_i \dot{V}_{oa,i}\right) C_{ahu} + \left(\sum_i \dot{V}_{oa,i}\right) C_{oa}}{\sum_i \dot{V}_{sup,i}}. \tag{7}$$

The pollutant transport model represented by Equations 4 through 7 was implemented as a subroutine for the transient system simulation program (UW 1988). The concentration differential equation was solved using an iterative modified Euler method, as used in other components of the simulation program. The time step was set independently of the simulation time step because the concentration transients, in response to changing circulation and outside airflows and changing source levels, require a much shorter time step than is needed by the other component models. The time step used for most of the solutions was 30 seconds, while the simulation time step was 15 minutes.

The circulation and outdoor airflows are calculated externally to the pollutant transport model and input at each simulation time step. The building information, such as zone volumes, zone removal effectiveness, infiltration, and interzonal flows, are read in from a data file at the beginning of the simulation. The interzonal flows used in the pollutant transport model are constants during occupancy. When the ventilation system is shut down, the interzonal flows are set equal to zero.

The zone pollutant-removal effectiveness, ϵ_c, was treated as a constant in the model. There are several variables that could change the value of the removal effectiveness, but there is not enough information available at present to correlate a change in a variable to a corresponding change in ϵ_c. However, when a pollutant source is no longer present in a zone, it is reasonable to assume that conditions will approach the well-mixed state. The value of ϵ_c for a zone is set equal to 1.0 when there is no pollutant source in the zone.

Model Validation

The pollutant transport model was compared to results from the EPA indoor air quality model INDOOR to ensure its accuracy and validity. The calculations of the INDOOR program have been compared with test measurements at the EPA test house and with the NIST indoor air quality model CONTAM with good results (Sparks et al. 1988, 1989). Five different runs with identical inputs were made to compare the pollutant transport model's predictions to those of INDOOR. The predicted concentration values from the two programs are essentially identical.

OUTDOOR AIRFLOW CONTROL

Model Development

In order to test various schemes for indoor air quality control, a component was developed to allow control of the flow of outdoor air based on the pollutant concentration calculated by the pollutant transport component. There are three varieties of flow controller: a proportional controller, a purge controller, and a temperature-based economizer controller. For the variable-air-volume (VAV) system, the circulation airflow based on the zone sensible load was also calculated.

The control action could be based on either the maximum zone return air duct concentration, the mixed return air concentration at the inlet of the air-handling unit, or the supply air concentration. For this study, only the maximum zone return air duct concentration was used because it was thought to provide the best control. The flow of outdoor air is increased when the controlling concentration value exceeds a high-limit setpoint and is decreased when the controlling concentration value falls below a low-limit setpoint.

The outdoor airflows determined by the flow controller are limited. The low limit enables setting a base or minimum value for an outdoor airflow controller and can be used to specify a constant value for outdoor airflow to model noncontrolled ventilation systems. The maximum value for outdoor airflow is also the constant circulation flow rate for a constant-air-volume (CAV) system. Priority is always given to providing the outdoor airflow rate required to control the pollutant level. Therefore, the outdoor airflow rate determined by the controller based on the pollutant concentration is also the minimum circulation airflow rate for the VAV system and the minimum outdoor airflow rate for the temperature-based economizer system. The ventilation system was started up and shut down once each day at a scheduled time.

Control Strategies

Three types of automatic outdoor airflow control based on CO_2 concentration in the zone were evaluated. The proportional flow controller increases the flow of outside air by 20% of the maximum circulation airflow rate at each simulation time step if the concentration is above the high-limit setpoint. The increases continue at each time step until the maximum flow is reached or the concentration falls below the high limit. When the concentration falls below the low-limit setpoint, the flow of outdoor air is decreased by 20% of the maximum circulation airflow rate at each time step. These decreases continue at each time step until the minimum outdoor airflow is reached.

The purge controller operates similarly to the proportional controller, except that when the high concentration limit setpoint is exceeded, the system switches to 100% outdoor air at the maximum circulation airflow rate. The flow drops to the minimum outdoor airflow rate when the concentration falls below the low-limit setpoint.

The proportional and temperature-based economizer controller operates first as a proportional controller based on CO_2 concentration. When the outdoor air temperature is less than the indoor air temperature but greater than the specified coil outlet temperature, the outdoor airflow rate is set equal to the circulation airflow rate (100% outside air). When the outdoor air temperature is less than the specified coil outlet temperature, outdoor air is mixed with recirculated room air in an attempt to achieve the specific coil outlet temperature without requiring any energy removal by the coil. For this case, the ratio of outdoor airflow to circulation airflow is calculated as shown in Equation 8:

$$\dot{V}_{oa} = \dot{V}_{circ} \frac{(T_{zone} - T_{coil})}{(T_{zone} - T_{oa})} . \tag{8}$$

In the above equation, $\dot{V}_{oa}$ is the outdoor air volume flow rate, $\dot{V}_{circ}$ is the circulation air volume flow rate, T_{zone} is the zone air temperature, T_{coil} is the coil air outlet temperature, and T_{oa} is the outdoor air temperature.

In a VAV system, the circulation air enters the zone at the temperature of the coil outlet. The flow rate is adjusted so that the flow of air reaching a zone will be just enough to meet the cooling or heating load of that zone. The flow is determined as

$$\dot{V}_{circ} = \frac{\dot{Q}_{sens}}{\rho_a C_p (T_{zone} - T_{coil})} \tag{9}$$

where $\dot{V}_{circ}$ is the volume flow rate of circulation air, $\dot{Q}_{sens}$ is the zone sensible heating or cooling load in units of energy per unit time, ρ_a is the air density, C_p is the air specific heat, T_{zone} is the zone air temperature, and T_{coil} is the coil air outlet temperature. The zone load, $\dot{Q}_{sens}$, is positive for a cooling load and negative for a heating load. The coil temperature is set less than the zone temperature for a cooling situation and greater than the zone temperature for a heating situation.

Since the circulation airflow rate is never less than the outdoor airflow rate required to control the level of pollutant, the circulation airflow rate may be greater than that required to meet the zone sensible load. When this occurs in cooling situations, the air is reheated at the zone inlet to prevent overcooling the zone. In heating situations, the temperature of the zone inlet air is lowered.

Building Model

The office area that was simulated is based on a model of the ninth floor of an office building located in Jacksonville, Florida. The single-zone model developed by Ruud (1990) for a study of building thermal storage was modified to add a small meeting room as a second zone. The building is typical of a modern office building with a glass curtain-wall exterior supported by structural steel. The office on the ninth floor is approximately 1,300 m^2 (14,000 ft^2) in area and has an air volume of 3,370 m^3 (119,000 ft^3). The meeting room zone has a floor area of 31 m^2 (340 ft^2) and an air volume of 81 m^3 (2,850 ft^3) and is surrounded by the office zone. The maximum occupancy of the office zone in the model is 100 people and of the meeting room is 10 people. These occupancy levels are varied through the business day to simulate actual conditions. A carbon dioxide generation rate of 5×10^{-6} m^3/s (1.77×10^{-4} ft^3/s) per occupant was used as the source strength (Appendix D of the ASHRAE ventilation standard for an activity level of 1.2 met).

The HVAC system was operated between the hours of 5:00 a.m. and 9:00 p.m., seven days a week. The CAV and maximum VAV circulation airflow rates used in the simulations were six volume ach. When the HVAC system was on, an interzonal flow from the meeting room to the office of 0.5 ach was included. An infiltration flow of 0.2 ach was included for the office zone. There was no infiltration for the meeting room zone, since it had no exterior walls. The ceiling plenum is used for the return airflow path, with a volume equal to the zone's floor area and 1 meter deep.

Control Strategies

A set of six year-long simulations for both CAV and VAV systems was performed using weather data from Madison, Wisconsin, and Miami, Florida, to produce a total of 24 base cases. The set of six simulations was made up of three constant outdoor airflow rate situations and one simulation of each of the three methods of automatic outdoor airflow control. For the automatic control simulations, the high-limit setpoint was 1,000 parts per million (ppm) by volume (0.001), and the low-limit setpoint was 800 ppm (0.0008).

The first constant outdoor airflow simulation used the flow rate recommended in *ASHRAE Standard 62-1989*. For a multiple-zone system, the ASHRAE standard provides Equation 10 to correct for anticipated uneven pollutant loads in the different zones in order to ensure that the zone with the greatest pollutant load (called the "critical zone" in ASHRAE) receives a sufficient supply of ventilation air:

$$Y = \frac{X}{(1 + X - Z)} . \tag{10}$$

In the above equation, Y is the corrected fraction of outdoor air for the supply airstream, X is the uncorrected fraction of outdoor air based on combined zone volumes and occupancy levels, and Z is the outdoor air fraction calculated for the critical zone.

The required outdoor air fractions are calculated by multiplying the occupancy level by the required outdoor airflow rate per person from the ASHRAE standard and dividing by the nominal circulation airflow rate. For an office area, the required flow rate is 10 L/s (20 cfm) of outdoor air per person. In these simulations, the critical zone was the meeting room, since the number of persons per unit floor area here was more than twice as high as for the office zone. The resulting outdoor air fraction for this situation was 0.38, which translates to an outdoor airflow rate of 2.3 ach using a circulation airflow rate of 6 ach. If the uncorrected fraction of outdoor air had been applied, the outdoor airflow rate would have been only 1.1 ach.

The second constant outdoor airflow rate scenario that was simulated used a "typical" value for outdoor airflow of 0.7 ach. This typical value is an average of more than 3,000 measured outdoor airflows from 14 different office buildings reported by Persily (1989). The last "constant" outdoor airflow scenario was really a temperature-based economizer simulation with a constant minimum outdoor airflow of 0.7 ach.

The three automatic outdoor airflow control scenarios did not include a minimum value for outdoor airflow. Control of pollutant concentration was to be accomplished by the controller alone. This was felt to be the most challenging test of the ability of a controller to keep the pollutant concentration at a reasonable level. In actual practice, a base minimum outdoor airflow would be required to dilute the indoor pollutants that are not related to human occupancy. Comparison simulations with a 1 ach minimum outdoor airflow rate were performed and are discussed later.

ENERGY USE AND POLLUTANT CONTROL

This section compares the various methods for achieving an acceptable indoor air quality using outdoor airflow. The performance of the fixed outdoor airflow rate as recommended in the ASHRAE standard is compared to those methods that provide automatic control of the outdoor airflow rate. The objective of this comparison is to determine if an automatic system for controlling the flow rate of outdoor air can provide a level of protection from occupant-generated pollutants equivalent to that provided by the ASHRAE standard at reduced energy use.

Tables 2 and 3 list the estimated annual energy use for the six scenarios described in the previous section for Madison for CAV and VAV systems, respectively. The tables list both the energy required by the heating/cooling coil and the total of coil and reheat energies. Both are shown in the tables because some HVAC systems are able to utilize "free" reheat energy from the condenser air conditioner. In all cases, the automatic flow control systems have a significant energy advantage over the constant airflow rate control. The relative energy uses are similar for the Miami location, although the magnitudes are higher. The Madison results will be discussed primarily; all results are given by Knoespel (1990).

When reheat energy is included in a CAV system, the automatic flow control systems have a smaller advantage, but in the VAV system, the advantage is about the same whether or not the reheat is included. As expected, the VAV system uses substantially less energy than the CAV system. For the VAV systems, the energy advantage of automatic flow control over the ASHRAE constant flow is substantially greater than it is for the CAV system.

The automatic flow control systems use about the same amount of energy as the constant typical outdoor airflow scenario. This implies that the automatic systems could be installed to provide control of pollutants with little or no energy penalty, unlike the typical office building with an outdoor airflow rate that does not provide adequate pollutant control.

Pollutant-Removal Ability Compared to ASHRAE Standard

The energy-saving potential of the automatic flow control systems is apparent. In this section, the ability of the various outdoor airflow schemes to remove pollutants will be compared. Table 4 lists the average outdoor airflow rates in air changes per hour and in liters per second per person for the Madison simulations.

The nominal flow rate of outdoor air per person recommended by the ASHRAE standard is 10 L/s. The typical and automatic flow scenarios require only one-half to one-third as much outdoor air on the average as prescribed by the ASHRAE standard. The two exceptions are the temperature-based flow control scenarios for the CAV system, where the outdoor airflow rates are only one-third less than the ASHRAE standard.

Figures 3 through 5 compare the concentration of carbon dioxide in the meeting room that results from the ASHRAE outdoor airflow rate to that resulting from the typical flow, proportional flow control, and 100% purge control, respectively. The simulation is for the 10th of July for the Miami VAV system. The relative inadequacy of the typical outdoor airflow rate to limit pollutant level is apparent from Figure 3. The two automatic systems control the pollutant concentration to levels comparable to those resulting from the ASHRAE standard during the hours that the building is occupied.

In Figures 4 and 5, the dip in the concentration levels for the two automatic control systems at 5:00 a.m. is from the dilution effect of the office air mixing with the meeting room air when the ventilation system is turned on. The spikes in concentration level for the proportional control (Figure 4) are due to the rapid increase in carbon dioxide concentration when people arrive in the morning and after lunch. The 15-minute simulation time step allowed an increase before it was controlled. An actual control system with a more rapid response time should be able to provide better control in this situation.

TABLE 2
Energy Use by CAV Systems in Madison

Control Scheme	Coil Energy (GJ)	Percent Savings Over ASHRAE	Coil + Reheat Energy (GJ)	Percent Savings Over ASHRAE
ASHRAE OA Flow	2262	-	2913	-
Typical OA Flow	1691	25	2496	14
Typical + Temperature	1780	21	2538	13
Proportional Control	1664	26	2504	14
Proportional + Temperature	1811	20	2648	9
100% Purge Control	1626	28	2466	15

TABLE 3
Energy Use by VAV Systems in Madison

Control Scheme	Coil Energy (GJ)	Percent Savings Over ASHRAE	Coil + Reheat Energy (GJ)	Percent Savings Over ASHRAE
ASHRAE OA Flow	1778	-	1944	-
Typical OA Flow	924	48	963	50
Typical + Temperature	954	46	997	49
Proportional Control	1012	43	1074	45
Proportional + Temperature	1004	44	1058	46
100% Purge Control	972	45	1056	46

TABLE 4
Comparison of Average Outside Airflow Rates for Madison

Control Scheme	CAV System		VAV System	
	Average Outdoor Air Flow Rate (ach)	Per Person Outdoor Air Flow Rate (L/s)	Average Outdoor Air Flow Rate (ach)	Per Person Outdoor Air Flow Rate (L/s)
ASHRAE OA Flow	2.30	19.9	2.30	19.9
Typical OA Flow	0.70	6.1	0.70	6.1
Typical + Temperature	1.75	15.2	0.94	8.2
Proportional Control	0.64	5.6	0.93	8.1
Proportional + Temperature	1.92	16.6	1.07	9.3
100% Purge Control	0.54	4.7	0.83	7.2

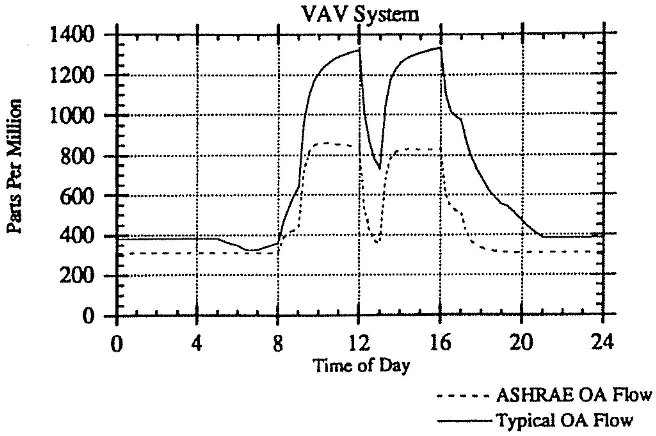

Figure 3 *Comparison of carbon dioxide concentration levels vs. time of day for ASHRAE and typical outside airflow rates*

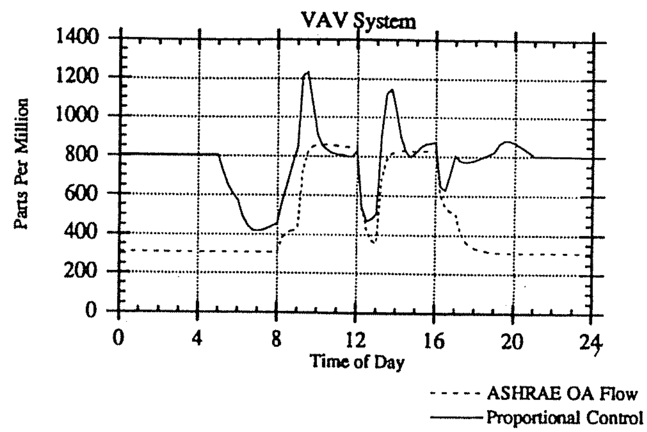

Figure 4 *Comparison of carbon dioxide concentration levels vs. time of day for ASHRAE outside airflow rate and proportional control*

Relative Pollutant-Removal Ability of CAV and VAV Systems

Figures 6 through 9 give the pollutant concentration histogram summaries for the ASHRAE, proportional control and purge control simulations using VAV and CAV systems, respectively, for Madison. The histograms show the fraction of occupied hours during the year that the CO_2 concentration fell into one of the 50 ppm concentration intervals.

As expected, due to lower occupant density in the offices, the pollutant concentrations in the office zone are less than in the meeting room zone. The overall concentrations in both zones are less for the CAV systems (Figures 7 and 9) than they are for the VAV systems (Figures 6 and 8). For the VAV system plots, the concentrations in the meeting room are generally higher than they are in the office zone, while for the CAV system, the concentrations in the two zones are much closer. It appears that the lower circulation airflow rates inherent to the VAV system result in less mixing of the pollutants between the two zones. The meeting room pollutant concentration thus reaches higher concentrations more quickly, and the automatic control systems are less able to keep the peak concentrations down. With a VAV system, the ASHRAE constant outdoor airflow rate still allows concentrations to reach above the 1,000 ppm maximum that the airflow rate was designed to achieve. It is only with a CAV system that the flow rates of the ASHRAE standard maintain levels below 800 ppm.

The histograms for the two temperature-based economizer systems, which employ typical outdoor airflow plus temperature-based control, are shown in Figure 10. Both use approximately the same amount of energy. The difference in pollutant control is quite apparent, with the proportional plus temperature controller performing much better. However, both strategies produce a significant number of hours with high concentration levels.

For most of the simulations, pollutant-removal effectiveness values of 0.9 for the office zone and 1.0 for the meeting room were used. There is a pollutant-removal advantage in increasing the value of ϵ_c in a zone. In order to determine if there could be an energy advantage to doing so, the value of ϵ_c in the meeting room (the critical zone) was varied by one-third in both directions for the VAV system. Increasing the removal effectiveness resulted in an increase in energy use of about 5%, and decreasing it

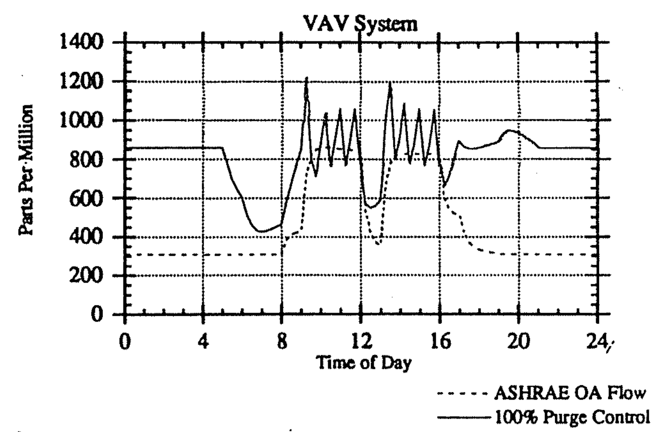

Figure 5 *Comparison of carbon dioxide concentration levels vs. time of day for ASHRAE outside airflow rate and purge control*

lowered the energy use by about the same amount, as shown in Table 5. For an automatic control system, with a higher removal effectiveness, the sensor in the return air duct detects a higher concentration of pollutant and therefore calls for more outdoor airflow than what is necessary to control the room concentration. The opposite is true for a lower removal effectiveness.

TABLE 5
Effect of Changing Pollutant-Removal Effectiveness on Annual Energy for Proportional Control

	ϵ_c	Annual Energy Use
Base	1.00	1012
Standard limits	1.33	1065
	0.67	950
Adjusted limits	1.33	810
	0.67	1425

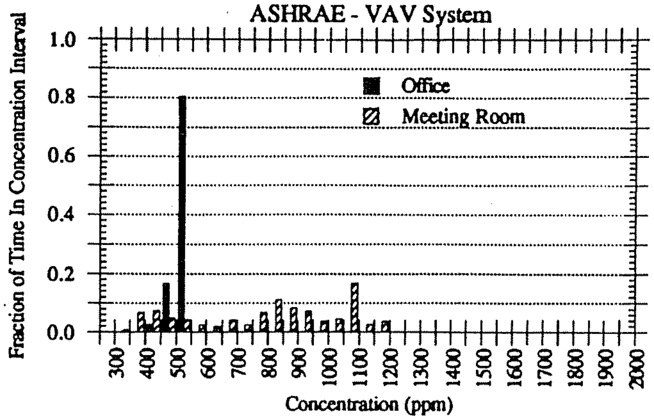

Figure 6 *Year summary histogram of carbon dioxide concentration for a VAV system and ASHRAE outside airflow rate*

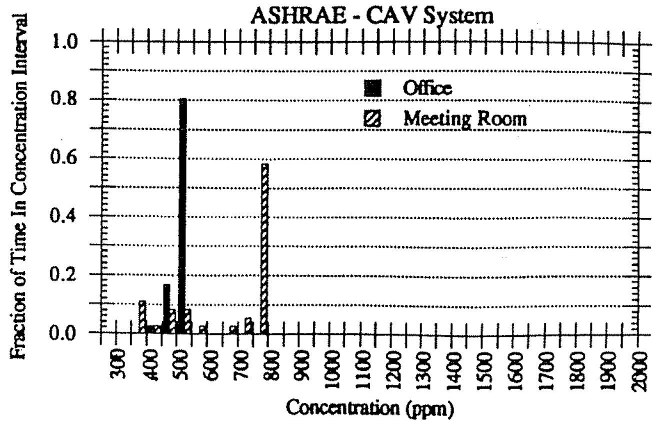

Figure 7 *Year summary histogram of carbon dioxide concentration for a CAV system and ASHRAE outside airflow rate*

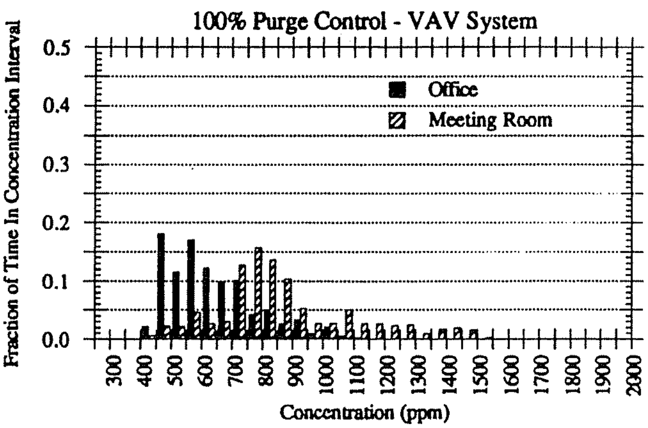

Figure 8 *Year summary histogram of carbon dioxide concentration for a VAV system and purge control*

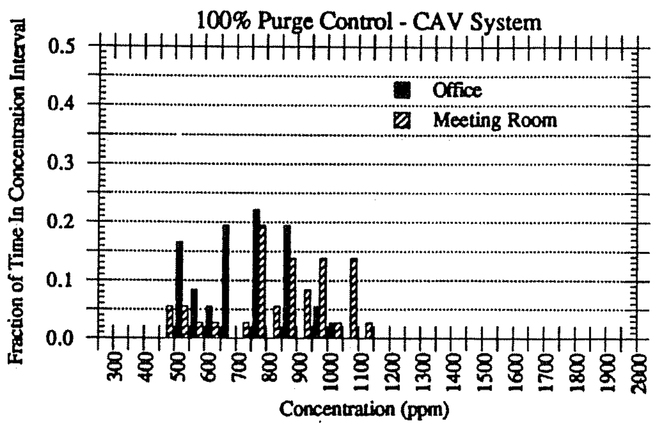

Figure 9 *Year summary histogram of carbon dioxide concentration for a CAV system and purge control*

By adjusting the control setpoints in the same proportion that ϵ_c is raised or lowered, the sensor in the return duct will not call for an increase in the outdoor airflow until the room concentration reaches the desired maximum pollutant concentration. Simulations run with the adjusted limits resulted in energy savings of about 20% for increasing ϵ_c and an increased energy use of about 41% for a lower ϵ_c value, also given in Table 5. Figures 11 and 12 show the histograms for the different pollutant-removal effectivenesses. Without adjusting the limits, proportional control results in over- and undercontrol, as shown in Figure 11. Figure 12 shows that the pollutant control ability for the two simulations with adjusted limits is nearly equivalent to the case with ϵ_c equal to 1.0. With an automatic flow control system, increasing the removal effectiveness of a critical zone can result in energy savings only if the control setpoint for that zone is adjusted to compensate.

An actual CO_2-based automatic outdoor airflow control system would probably provide a minimum flow of outdoor air to dilute the indoor pollutants that are not related to human occupancy. To determine how this would affect the energy use of an automatic flow control system, simulations were done for the proportional controller with a 1.0 ach

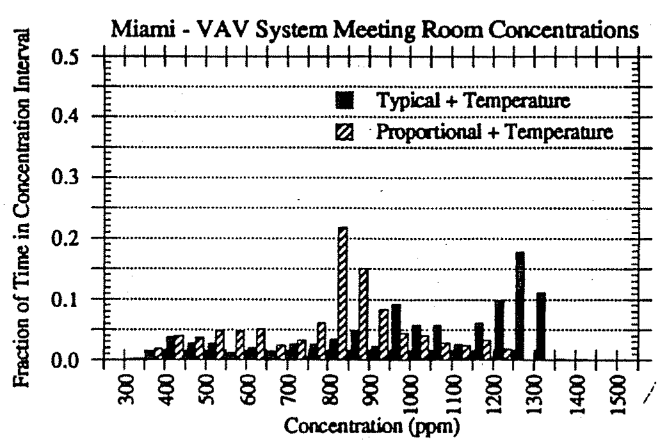

Figure 10 *Comparison of year summary histograms of carbon dioxide concentration for typical plus temperature control and proportional plus temperature control*

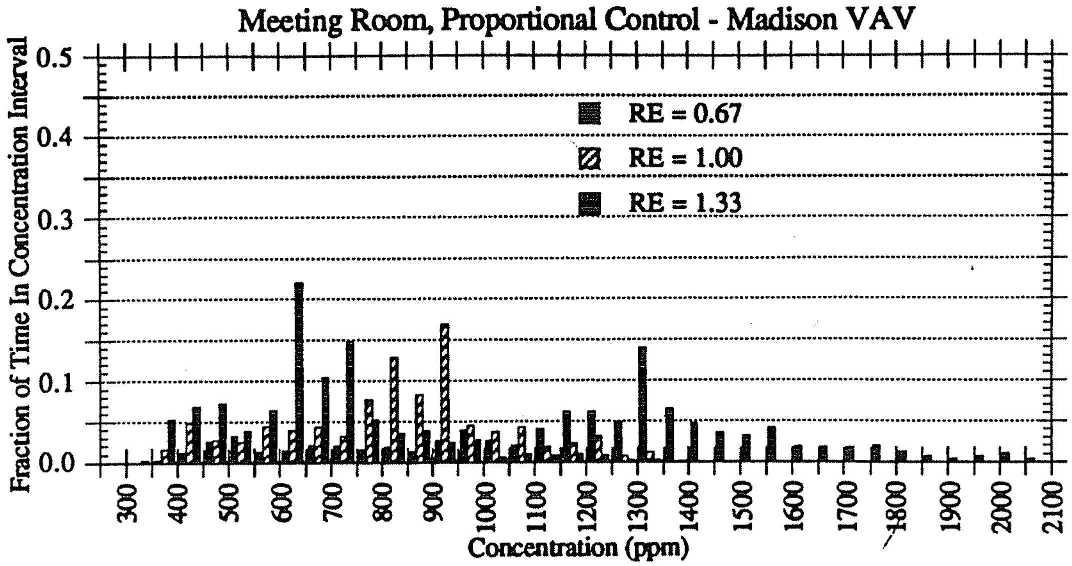

Figure 11 *Effect of changing pollutant-removal effectiveness on concentration level histogram*

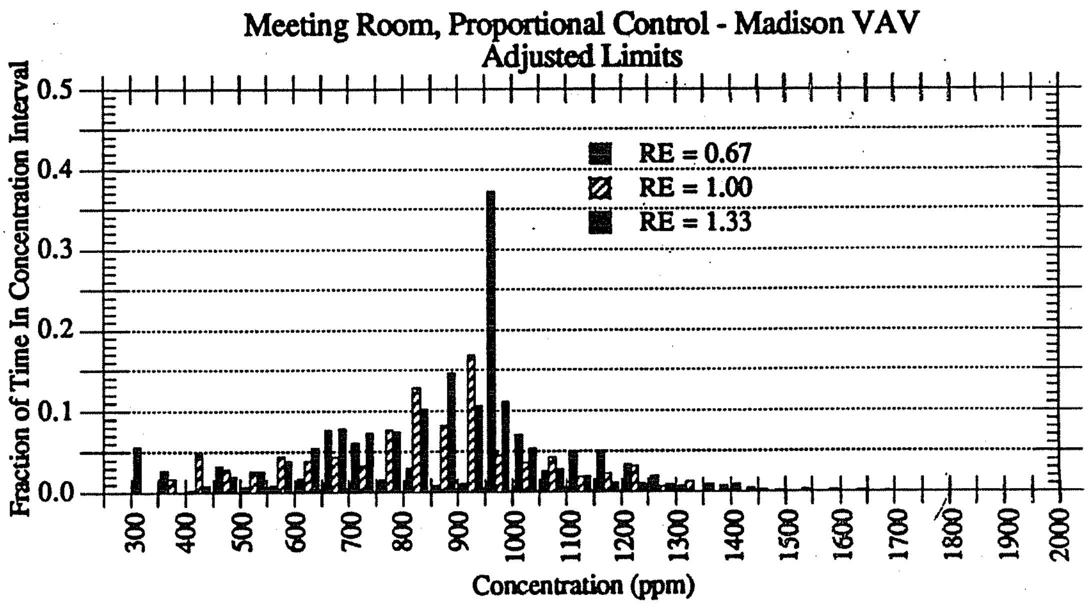

Figure 12 *Effect of changing pollutant-removal effectiveness on concentration level histogram with adjusted concentration limits*

minimum outdoor airflow rate. The results are given in Table 6. For a VAV system in Madison and Miami, the savings over the ASHRAE standard for fixed airflow were 22% and 27% for the two cities, respectively. This is, however, substantially less than the savings of proportional control without a base airflow.

SUMMARY OF RESULTS

Some general conclusions can be drawn from this study. All three systems studied can provide essentially equivalent control of the occupant-generated CO_2 to that afforded by the fixed outdoor airflow rate recommended in *ASHRAE Standard 62-1989* for indoor air quality. The heating and cooling energy saved from using an automatic

TABLE 6
Effect of Addition of Base Outdoor Airflow on Energy Use, VAV Systems

	Energy Use (GJ)	
	Madison	Miami
ASHRAE OA Flow	1950	2300
Proportional Control	1100	1400
Proportional Control (plus 1.0 ach.)	1500	1700

control for outdoor airflow ranges from 10% in a CAV system to as much as 50% for a VAV system. When comparing energy use, the constant outdoor airflow required by the ASHRAE standard was biased toward providing the meeting room with sufficient ventilation air, and the office zone was therefore overcontrolled.

VAV systems are not as efficient at pollutant dilution as CAV systems are due to their lower circulation airflow rates. Even the ASHRAE-recommended fixed outdoor airflow rate does not keep occupant-generated CO_2 below the 1,000 ppm target concentration in a VAV system. A lower setpoint for an automatic flow control system (such as 800 ppm) could be used to counter this effect.

Altering the pollutant-removal effectiveness of a critical zone does affect the energy use of an automatic flow control system. If the removal effectiveness is increased, the high-limit setpoint can be proportionally increased and still provide an equivalent dilution of the pollutant. Conversely, if the removal effectiveness is lowered, the setpoint must also be lowered.

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