INDOOR AIR QUALITY MODELING

by

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A thesis submitted in partial fulfillment

of the requirements for the degree of

Master of Science

(Mechanical Engineering)

at the

UNIVERSITY OF WISCONSIN - MADISON

1990

Abstract

The quality of the indoor air is important to both the health and productivity of building occupants. In order to gain a better understanding of how pollutants are distributed in the indoor environment, computer models can be utilized to great advantage. The study of indoor air quality modeling presented in this thesis is divided into two main topics. The first is the investigation of air flow and pollutant distribution. in a ventilated room. The second topic studied is pollutant transport through a multiple-zone ventilation system. This thesis also presents a comparison of several methods aimed at control of indoor pollutants.

The three-dimensional modeling of air and pollutant distribution in a ventilated room was carried out using the finite difference computer program FLUENT. The pollutant removal effectiveness, defined as the room exhaust duct pollutant concentration divided by the room average concentration, was used to describe the degree of pollutant mixing in the room. A parametric study was performed to determine if the value of the removal effectiveness factor changes significantly when certain ventilation and source characteristics are varied. The results from this study indicate that removal effectiveness may not vary significantly for some ventilation inlet and exhaust duct arrangements. Further work in this area using a more detailed model of room air flow is necessary before any definite conclusions can be drawn.

In the second phase of this project, an HVAC system for an office zone and a meeting room zone was modeled by the simulation program TRNSYS. A pollutant transport

model was developed as a new component of this code. The TRNSYS model was used to compare several fixed and automatic outside air flow rate methods for ability to control indoor pollutant levels and for heating and cooling energy use. The automatic outside air flow control methods use the concentration of CO_2 as an indication of the level of occupant-related pollutant in the zones.

The results of this study indicate that an automatic outside air flow control system based on CO_2 concentration can control pollutants as well as the fixed flow of outside air recommended in the 1989 ASHRAE indoor air quality standard. The potential heating and cooling energy savings for the automatic systems as compared to the fixed flow rate recommended by ASHRAE ranged from between 10% for CAV systems to as much as 50% for VAV systems.

Acknowledgements

As I sit here in front of the computer one last time in the wee hours of the morning, my thoughts turn to the friends I have come to know at the Solar Lab and how grateful I am that I had the opportunity to spend some very happy times with them here. To Bill and John, my partners in this project, I am thankful for the knowledge they shared with me, the criticisms that kept me pointed in the right direction, and the encouragement that kept me going at all. To Sandy, I am thankful for encouraging me to come to the Solar Lab in the first place. To all of my fellow students, I am thankful that I was able to share in some of your knowledge and experiences that are so different from my own. Meeting people from around the world is a benefit I didn't expect, but one that I truly cherish.

If nothing else, the graduate school experience has taught me how important friends and family are. To Kelly and Tim, thanks for understanding when Daddy had to spend so much time working at school. Finally, to my closest friend and partner for life Sally, I can never repay all the love and encouragement shown to me during this last year and a half, but believe me, I'm going to try.

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Nomenclature

Symbol	*Meaning*
Roman	
C	volume concentration; constant
C_p	specific heat
DT	circulation air time delay
E	effectiveness
G	generation term
INF	infiltration volume flow
IZF	inter-zone volume flow
K	mixing factor
k	turbulent kinetic energy
$\dot{m}$	mass flow rate
p	pressure
$\dot{Q}$	rate of energy transfer
$\tilde{R}$	normalized residual
S	species mass or volume source strength
t	time
U	directional velocity
V	volume
$\dot{V}$	volume flow rate
X	vector direction; uncorrected fraction of outside air
Y	corrected fraction of outside air

Z	fraction of outside air in the critical zone

Greek

β	fraction of circulation air reaching the occupied zone
Δt	TRNSYS simulation time step
ε	rate of dissipation of turbulent kinetic energy; removal effectiveness; exchange efficiency
η	efficiency
μ	viscosity
ρ	density
σ	Schmidt number
$\langle\tau\rangle$	mean age
ξ	arbitrary variable

Superscripts

n	iteration number

Subscripts

a	air
ahu	air handling unit
C, c	contaminant
circ	circulation
d	displacement
DT	circulation air time delay
E	east

e	east; equilibrium
i	x-direction index, zone index
inf	infiltration
izf	inter-zone flow
j	y-direction index, zone index
k	turbulent kinetic energy; z-direction index
lat	latent
N, n	north
oa	outside air
P	point or node of interest
p	pollutant
r	removal
ret	return air
S, s	south
sens	sensible
sup	supply air
t	time; turbulent
v	ventilation
W, w	west
ε	rate of dissipation of turbulent kinetic energy
μ	viscosity
0	original

Chapter 1
INTRODUCTION

This chapter provides an overview of the history of indoor air quality problems and solutions. Also examined are the circumstances that have led to the current situation, where the desire to provide a healthy indoor environment that enhances worker productivity must be balanced against the energy and monetary cost of providing such an environment. Finally, the objectives for this research project are presented.

1.1 Historical Background

Concern for the quality of the indoor environment can be traced back to the Roman Empire [1]. Shortly after the time of Christ, the Romans constructed buildings that contained the provision for fires to be built in the lower levels with chimneys that traveled through the upper levels to radiate heat to the rest of the building. This primitive form of central heat was augmented by a method to admit fresh air to the building as necessary. The Romans were thus able to use public baths in mid-winter, when temperatures were around 5° C (40° F). Buildings in the northern reaches of the empire were found to have been modified for the colder climate. However, the knowledge of the Romans was apparently lost when barbarians invaded the empire.

In the 1700's, people again began to look at the benefits of proper ventilation. This renewed interest was precipitated by concern for workplace conditions of the era. Mine

workers had been asphyxiated by high concentrations of carbon dioxide in the mine shaft. Stone cutters and chimney sweeps developed respiratory ailments. The French scientist Lavoisier studied human metabolism and discovered that the body requires oxygen and generates CO_2. He reasoned that ventilation is required to remove CO_2 and metabolic moisture.

Figure 1.1 traces the progression of ventilation standards over nearly two hundred years. During the 1800's, increasing amounts of outdoor air ventilation per building occupant were recommended by various scientists. In 1893, the new American Society of Heating and Ventilating Engineers (ASHVE) adopted an outdoor air ventilation standard of 15 liters per second (30 cfm) per occupant. In 1936, a study of human response to odors and methods to control odors was published by C. Yaglou, *et. al.* [2]. This study revealed that odor intensity was linearly related to the logarithm of the outdoor air ventilation rate. Also, odor was found to be reduced significantly by washing recirculated air. Based on this study, in 1946 ASHVE adopted a new standard for outdoor air ventilation of 5 liters per second (10 cfm) per occupant.

In 1973, the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), the successor of ASHVE, adopted an outdoor air ventilation standard that differentiated between areas in a building by their use and whether smoking was allowed. The minimum outdoor air flow rate allowed for any area was 2.5 liters per second (5 cfm) per occupant. This minimum was based on the amount of fresh air required to keep oxygen at safe levels and carbon dioxide below 2500 parts per million [3]. This was the lowest value for an outdoor air ventilation standard since 1824.

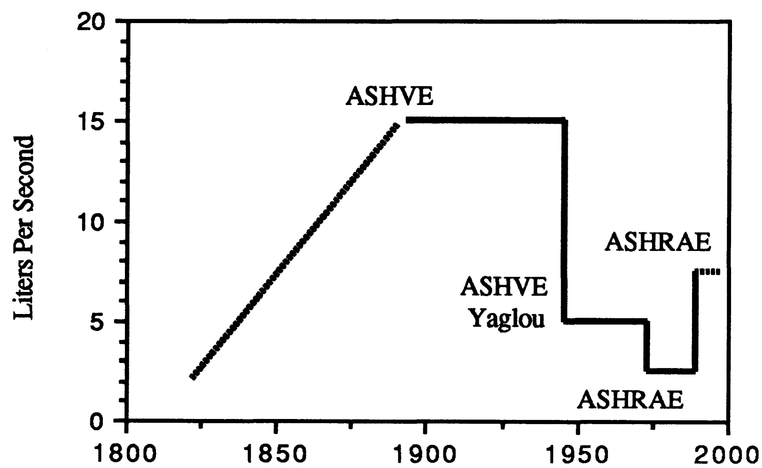

Figure 1.1 *History of Ventilation Standards (adapted from [1])*

During the 1970's, efforts to conserve energy resulted in outdoor ventilation rates in many buildings being reduced to the minimum amounts allowed. A 1981 ASHRAE standard on indoor air quality kept the minimum outdoor air flow rate of the 1973 standard, but also emphasized energy conservation. Also during this period, as energy became more expensive, it became cost-effective to construct buildings with tighter envelopes to reduce the amount of uncontrolled infiltration. Composite building materials using resin glues, foam insulation, and man-made fabrics came into general use in new building construction. All of these factors were later found to contribute to what became known as the "sick building syndrome".

A new ASHRAE standard on indoor air quality was adopted in 1989 to address the problems that surfaced during the preceding two decades [4]. This standard triples the minimum allowable outdoor air ventilation flow to 7.5 liters per second (15 cfm) per person for an area relatively free of pollutant sources. This limit is designed to keep the CO_2 concentration below 1000 parts per million. The minimum amount is increased in the standard for most building areas. For example, an office area is required to have 10 liters per second (20 cfm) per occupant, the additional amount being to offset the pollutants produced by office machines.

1.2 Concepts Relating to Indoor Air Quality Problems

The current emphasis in building ventilation control is a quest for an indoor environment that is energy efficient, healthy for building occupants, and enhances worker productivity. Before solutions of 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 outside. First, there are fewer air changes. A ventilated building may have between 0.4 and 10 air changes per hour, while outdoors, an 8 km per hour (5 mph) breeze will result in 3600 air changes in an hour [1]. 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.

A compilation of 466 building surveys performed by the National Institute for Occupational Safety and Health (NIOSH) was published in 1987 by Stolwijk [5]. Some of the results are summarized in Table 1.1. According to this study, inadequate ventilation was found to be responsible for just over half of the cases of "sick" buildings. It could be argued that problems caused by contamination released inside the building and building material contamination are also an indication of inadequate ventilation. This would bring the total fraction of indoor air quality problems related to insufficient flow of outside air to 72 percent. Since these surveys were completed nearly three years ago, adherence to the new 1989 ASHRAE standard could eliminate many of these problems. In 12 percent of the buildings surveyed, no cause could be pinpointed.

Table 1.1 *Sources of Indoor Air Quality Problems [5]*

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

Adequacy of ventilation is not always indicated by a simple measure of outside air flow rate. A study of the air flow 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 air flow rate [6]. This was largely due to the short-circuiting of the air flow 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. Several different factors have been proposed to describe how efficiently fresh air is delivered to the occupied zone in a room. These factors are discussed further in Chapter 3.

The human perception of indoor air quality is a topic first explored scientifically by Pettenkofer in the 1800's, and Yaglou, whose 1936 paper formed the basis for the 1946 ASHVE ventilation standard. More recently, Fanger [7, 8] has proposed using human perception of the quality of indoor air to both quantify the source level of indoor pollutants and to form the basis for a new air quality standard. The pollution level in an indoor or outdoor space is judged by a panel of unbiased observers as being either satisfactory or unsatisfactory. The percent dissatisfied along with the ventilation flow rate are then converted to a pollutant source level in "olfs". One olf is defined as the rate of pollutant emission (bioeffluents) from a standard person. As an illustration of this concept, pollution sources in several buildings were quantified and reported by Fanger, *et. al.* in 1988 [9]. The air quality standard proposed by Fanger would have a required flow rate of outside air per olf rather than per person, as the standard is now, and would relate directly back to an allowable percentage of persons dissatisfied.

A complementary air quality unit proposed by Fanger is the decipol, which is a measure of the level of pollution actually present in a space. One decipol is equivalent to a pollution source of one olf that is ventilated by one liter per second of unpolluted air. What the panel of observers is actually measuring is the decipol level. A standard based on this idea would take into account pollution sources that are not related to occupancy, such as fabrics and building materials, and could be adjusted to include pollution sources that are not discernable to the human sense of smell. This approach, however, has not yet reached acceptance.

1.3 Proposed Indoor Air Quality Solutions

There have been many solutions proposed for the indoor air quality problems that have surfaced since the early 1970's. 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 an energy cost that most building owners are unwilling to pay. However, there are alternatives that can be attempted first. If a fabric or material used in the construction of the building is giving off objectionable gases, the materials 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 air stream by providing an area exhaust to the outside.

Improper ventilation system maintenance or design can also lead to indoor air pollution problems. Ventilation ducts have been found to contain dirt and biological growth that gave off contaminants. In some buildings, outside air 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 such that a significant fraction of the ventilation air bypasses the occupied zone. Design problems such as these are expensive to remedy after the fact, but they should be anticipated during the design and construction of a new building.

When other air quality solutions are not feasible or did not eliminate the problem completely, the only alternative is to adjust the outdoor air ventilation rate. If this is done carefully, the energy cost of this solution can be minimized.

1.3.1 Using CO_2 to Monitor Indoor Air Quality

The use of CO_2 concentration as a measure of the quality of indoor air is an idea that dates back to the time of Lavoisier. Since the mid 1970's, 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 [10, 11]. Also, since the metabolic production and the outside air concentration of CO_2 are known, measuring the indoor CO_2 concentration provides a means to determine the actual outside air exchange rate for the space. This method has been shown to compare favorably with tracer gas methods using SF_6 and air flow measurements [12]. Figure 1.2 shows how occupancy and CO_2 concentration varied during the course of a day in an office waiting room.

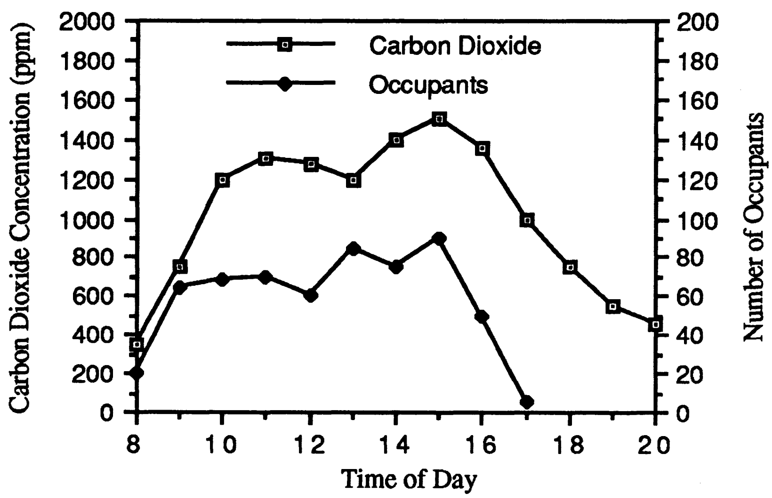

Figure 1.2 *Variation of CO_2 Concentration and Occupancy [12]*

In a 1985 paper, however, Sterling and Sterling [13] showed that CO_2 responds differently to changes in ventilation than do hydrocarbons, CO, and particulates, which are all determined more by outdoor concentrations. It therefore appears that CO_2 can be used as an indicator of the relationship between outdoor air ventilation rate and occupancy, but not as an overall indicator of air quality since CO_2 level will not respond to changes in indoor pollutant sources unrelated to occupancy or outdoor pollutant sources.

1.3.2 CO_2 Based Automatic Systems

With the above limitations in mind, several systems have been proposed to control the amount of outside air delivered to a ventilated space based on the CO_2 concentration present [11, 14, 15]. The controlling value could be either a set concentration limit or the rate of change of the concentration. The system could respond by increasing or decreasing the outside air flow in proportion to the concentration or by switching to 100% outside air once the control conditions are exceeded. A minimum flow of outside air could also be provided to account for indoor pollution sources that are not related to occupancy.

Although the control logic of specific proposed systems varies somewhat, the general idea of a CO_2 based outside air control system would be to vary the outside air flow rate in response to a measured indoor level of CO_2. 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 outside air flow 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 outside air flow rate system based on 10 liters per second (20 cfm) of outside air per person [14]. One of the goals of this research is to determine if such estimates of energy savings are representative.

1.4 Project Objectives

This project provides a general overview of the theory and methods for computer modeling of pollutant dispersal in the indoor environment. The research work performed for this project is divided into two main areas. The first topic examined is air flow and pollutant dispersal in a ventilated room. The objective of this research is to determine if a general term can be used to describe the air and pollutant mixing characteristics of a ventilated room and what factors affect these characteristics. Chapter 2 presents the theory behind three-dimensional air flow modeling, and a parametric study of room air flow and pollutant removal characteristics is described in Chapter 3.

The second phase of this research project involves the dispersal of indoor pollutants by a ventilation system through a multiple-zone building. As a part of this work, various methods of controlling the amount of outside air introduced into a building ventilation system are explored and the corresponding annual energy use of these methods is compared. These automatic methods for controlling outside air flow all use the level of CO_2 as an indicator of the amount of occupant-related pollutants in an indoor environment. The objective of this part of the project is to determine if an automatic system to control outside air flow based on the actual level of pollutants present can result in meaningful energy savings when compared to a constant air flow system based on the ASHRAE indoor air quality standard. The theory and computer model for multiple-zone pollutant transport is discussed in Chapter 4. In Chapter 5, the various methods of CO_2-based automatic outside air flow control are compared.

1.5 Terminology

As is the case in many technical disciplines, the terminology used in discussions of indoor air quality are sometimes ambiguous and given to interpretation. In an effort to diminish such ambiguity, the following definitions will be used consistently in the remainder of this thesis:

Circulation	the forced movement of air through a zone by means of a mechanical system such as a fan and associated ductwork.
Ventilation	the forced movement of air into a zone from the outside of the building (will sometimes be referred to as outside or outdoor air ventilation).
Recirculation	the reintroduction of zone air back into the zone by means of the mechanical system.
Ventilation System	a mechanical system consisting of a fan and associated ductwork to circulate air through a building.
Concentration	volume fraction of a gaseous species mixed in air

With the above definitions in mind, the following equation can be written:

$$\text{Circulation Air} = \text{Ventilation Air} + \text{Recirculation Air} \tag{1.1}$$

The circulation air could be either all ventilation air (100% outside air), all recirculation air (100% recirculation), or a mixture of the two.

Concentrations are presented as volume fractions since this is the most common way of discussing the composition of a gaseous mixture. For this project, the only indoor pollutants considered were gases.

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Chapter 2
MODELING ROOM AIR FLOW

Mathematical and computer modeling of room air flow has been possible since the early 1970's. However, given the capabilities of computers of that time, only two-dimensional, laminar, isothermal, steady-state cases were practical and affordable. By the early 1980's, advancements in computer speed and storage capacity enabled the elimination of most of the simplifications of early models, and full three-dimensional transient simulations became possible. Today, with supercomputers becoming more available, several complete models have been written and compared to actual room measurements for validation [1, 2, 3, 4].

2.1 Numerical Methods for Modeling Turbulent Fluid Flow

Most present-day models for room air flow are based on the Navier-Stokes equations and the Reynolds stress turbulence model or a simplification thereof. A second turbulence model, the large eddy simulation (LES) method, has also been developed [3]. This method employs Navier-Stokes equations that are "spatially filtered" over the grid scale and an eddy viscosity model to approximate the interaction between the large-scale and sub-grid scale motion. This eddy viscosity model requires only one empirical constant and thus has a more solid theoretical foundation than the Reynolds stress model. However, since the solution to these equations is never stationary (as actual turbulent flow is not), a time average of values is required. This makes it computationally expensive, so this newer method will have to await even more powerful supercomputers before it can be applied to practical room flow situations.

The basis for modeling room air flows that is used most often is the two-equation k-ε turbulence model. It is a simplification of the Reynolds stress model that was developed and refined primarily by Launder and Spaulding in the early 1970's [5]. It is based on the solution of transport equations for the turbulent kinetic energy k (equation 2.3) and its rate of dissipation ε (equation 2.4) simultaneously with those for continuity, momentum and concentration (equations 2.1, 2.2, and 2.6). Equation 2.5 serves as a definition of turbulent viscosity. Near a wall where molecular forces are important, the turbulent viscosity μ_t is replaced by effective viscosity μ_{eff}, which is the sum of turbulent and molecular viscosities.

$$\frac{\partial U_i}{\partial X_i} = 0 \qquad \textit{Continuity} \quad (2.1)$$

$$\frac{DU_i}{Dt} = -\frac{\partial}{\partial X_i}\frac{p}{\rho} + \frac{\partial}{\partial X_j}\left[\frac{\mu_t}{\rho}\left(\frac{\partial U_i}{\partial X_j} + \frac{\partial U_j}{\partial X_i}\right)\right] \qquad \textit{Momentum} \quad (2.2)$$

$$\frac{Dk}{Dt} = \frac{\partial}{\partial X_j}\left(\frac{\mu_t}{\rho\sigma_k}\frac{\partial k}{\partial X_j}\right) + \frac{\mu_t}{\rho}G - \varepsilon \qquad \textit{k - Transport} \quad (2.3)$$

$$\frac{D\varepsilon}{Dt} = \frac{\partial}{\partial X_j}\left(\frac{\mu_t}{\rho\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial X_j}\right) + C_1\frac{\varepsilon}{k}\frac{\mu_t}{\rho}G - C_2\frac{\varepsilon^2}{k} \qquad \textit{ε - Transport} \quad (2.4)$$

$$\mu_t = C_\mu \rho \frac{k^2}{\varepsilon} \qquad \textit{Turbulent Viscosity} \quad (2.5)$$

$$\frac{DC}{Dt} = \frac{\partial}{\partial X_j}\left(\frac{\mu_t}{\rho\sigma_C}\frac{\partial C}{\partial X_j}\right) + S_C \qquad \textit{Concentration} \quad (2.6)$$

In the above equations, the substantial derivative is given by

$$\frac{D\xi}{Dt} = \frac{\partial \xi}{\partial t} + \frac{\partial \xi U_i}{\partial X_i} \qquad (2.7)$$

where ξ represents the variable of interest, and the generation term G is equal to

$$G = \left(\frac{\partial U_i}{\partial X_j} + \frac{\partial U_j}{\partial X_i}\right)\frac{\partial U_i}{\partial X_j} \qquad (2.8)$$

Also in the equations above,

U_i	is the velocity in the x, y, or z direction,
X_i	is the distance in the x, y, or z direction,
C	is the mass concentration of a species mixed in the air,
S_C	is a species mass source rate,
p	is the air pressure,
ρ	is the air density,

and C_1, C_2, C_μ, σ_k, σ_ε, and σ_C are all constants with values as in Table 2.1.

Table 2.1 *Values for Constants in the k-ε Equations [5, 6]*

C_1	C_2	C_μ	σ_k	σ_ε	σ_C
1.44	1.92	0.09	1.0	1.3	0.7

The boundary conditions required for solution of these equations as they are applied to room flows generally include inlet velocities and a concentration value in the inlet air stream.

2.2 FLUENT Fluid Mechanics Program

Three-dimensional room air flow and contaminant concentration were modeled for this project using an educational version of a commercially available fluid mechanics program named FLUENT [6]. This program was used to calculate the steady-state three-dimensional velocity and species concentration arrays for the hypothetical room flow and pollutant source situations.

FLUENT is a finite difference fluid dynamics program that uses the mass, momentum, and species conservation equations, and the k-ε turbulence modeling method presented in Section 2.1. In addition, it is capable of solving the energy conservation equation and can handle buoyancy effects and chemically reacting species. FLUENT is also capable of producing three-dimensional species concentration arrays as a function of time. However, a time-dependent FLUENT run is a very time consuming process, and the output is in a format that does not lend itself easily to further data reduction. Since a large number of runs was anticipated for a parametric study of pollutant concentrations, time-dependent cases were not considered to be practical for this research. A study of pollutant distribution as a function of time after the introduction of a pollutant source or after ventilation system startup would be interesting and could be considered for an extension of this project.

2.2.1 FLUENT Method

In FLUENT, the governing partial differential equations for mass and momentum conservation are reduced to finite difference form by integration over the computational grid cell. Scalar quantities such as k, ε, ρ, p, enthalpy, and species concentration are defined at the center of these cells. The vector components of velocity are defined on a staggered grid formed by the surfaces of the cell. Each of the six individual velocity components associated with a cell is evaluated on the cell surface perpendicular to it, as shown in Figure 2.1. The staggered grid method enables the differential equations involving velocities to be discretized using central differencing, making them second-order accurate. Discretizing the governing differential equations for the problem results in a system of coupled algebraic equations for each node.

The solution method employed by FLUENT is fully implicit between time steps and semi-implicit within time steps. In each time step, the momentum equations are solved using guessed pressures. Pressure corrections are then calculated using a modified continuity equation, and the velocities are adjusted using the corrected pressures. The k and ε equations are solved next to calculate an updated distribution of effective viscosity. Any additional equations, such as energy or species conservation, are solved last. These steps are then repeated until the sum of the relative or absolute changes in each value is less than the required tolerance. For time-dependent cases, when convergence is achieved for one time step, the calculations for the next time step may proceed.

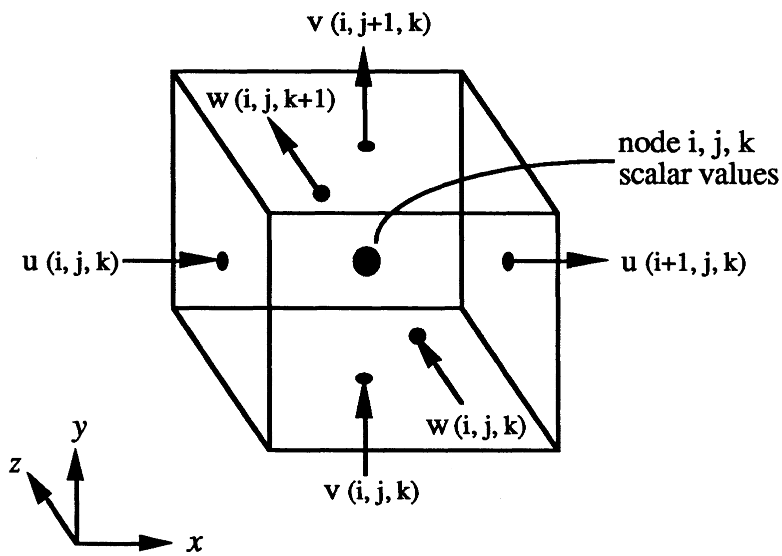

Figure 2.1 *Staggered Grid Method for Velocities Used by FLUENT*

Boundary and inlet conditions need to be specified for all boundary nodes of the domain. Conditions at interior and exit nodes are calculated by FLUENT.

2.2.2 Calculation Procedure

The physical data necessary to describe the room are input to FLUENT by means of an input case file created during a FLUENT interactive run. This input data includes fluid physical properties, room dimensions, finite difference grid dimensions, wall locations, ventilation air inlet and outlet locations, source locations, and inlet velocities and species concentrations. A FLUENT calculation run will then store all converged flow and concentration information in a data file that can be used to create a user-readable list file or as a starting point for further calculations with slightly modified input conditions. This feature can greatly reduce total computation time when several runs with identical geometry but slightly different boundary conditions are required.

For this project, the list files containing the converged species concentration arrays were used to calculate a room average concentration, an average ventilation exhaust duct concentration, and the pollutant removal effectiveness. The pollutant removal effectiveness is the exhaust concentration divided by the room average concentration and is discussed further in section 3.1.

2.2.3 Accuracy Issues

As with any numerical approximation to an actual fluid flow situation, the manner by which the problem is set up and solved will determine the accuracy of the solution.

As described in Section 2.2.1, FLUENT solves the algebraic system of equations that describe the flow problem in a stepwise manner until a converged solution is achieved. In the case of FLUENT, the term converged solution means that the sum of the residuals after the last iteration was less than or equal to the convergence criterion.

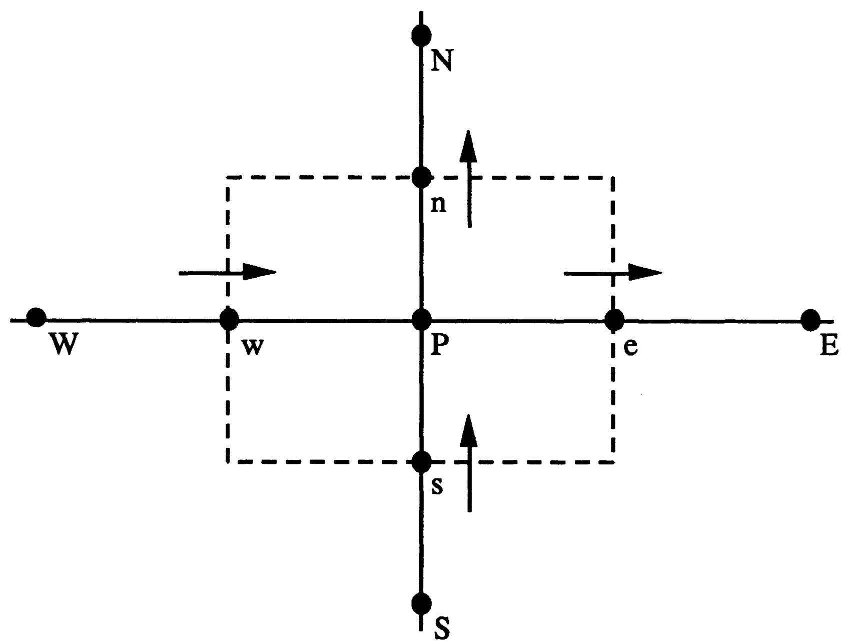

Figure 2.2 *Computational Cell Surrounding Node P [6]*

The discretized conservation equation for an arbitrary dependent variable ξ in two dimensions can be written as

$$A_P\,\xi_P = A_E\,\xi_E + A_W\,\xi_W + A_N\,\xi_N + A_S\,\xi_S + S_u \qquad (2.9)$$

where

$$A_P = A_E + A_W + A_N + A_S + S_\xi \qquad (2.10)$$

In equations 2.9 and 2.10, the As represent the finite difference coefficients for convection and diffusion through a control volume surrounding node P. The subscripts E, W, N, S, and P are locations defined in Figure 2.2. S_u and S_ξ are source terms than include any terms in the conservation equation that are not either convection or diffusion.

FLUENT calculates the sum of residuals after each iteration by first summing the imbalance in equation 2.9 for a dependent variable ξ being solved (u-velocity, for example) over all of the nodes in the problem. This sum of residuals is then divided by the left hand side of equation 2.9 summed over all of the nodes in the problem to arrive at a normalized residual $\widetilde{R}$ for each dependent variable, as in equation 2.11. Finally, the residuals for all dependent variables are summed to compare with the convergence criterion.

$$\widetilde{R} = \frac{\sum_{P} A_E\,\xi_E + A_W\,\xi_W + A_N\,\xi_N + A_S\,\xi_S + S_u - A_P\,\xi_P}{\sum_{P} A_P\,\xi_P} \tag{2.11}$$

For all problems solved by FLUENT for this project, the default value for convergence of 1.0E-3 was used.

The spacing of the finite difference grid is one of the most important choices in setting up a problem that will affect the accuracy of the final results. Due to the relatively large physical size of the room flow problems modeled for this project, it was necessary to use a grid spacing of 0.5 meters in all three directions to keep the computer run times reasonable. It was felt that the large grid size would not be a factor in this case since the main objective of the calculations was to arrive at a value for the room average pollutant concentration. To test the effect of the large grid size, two identical problems were modeled in two dimensions, one with a 0.5m grid spacing, and one with a 0.25m grid spacing. The results are summarized in Table 2.2. The largest difference was for the average concentration at 2.5% which is considered to be an acceptable amount of error.

Table 2.2 *Comparison of Identical Problems with Different Grid Sizes*

	0.25 m GRID	0.5 m GRID	PERCENT DIFFERENCE
EXIT CONCENTRATION	0.001176	0.001189	1.1
AVERAGE CONCENTRATION	0.000577	0.000591	2.5
REMOVAL EFFECTIVENESS	2.038	2.012	1.3

2.3 References

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Chapter 3
POLLUTANT REMOVAL EFFECTIVENESS

Often, the pollutant source and circulation air flow characteristics in a room will result in the pollutant being not uniformly mixed throughout the room air space. The pollutant removal effectiveness, ε_c, as defined by Seppänen [1] is used in this project to describe the non-fully mixed condition of pollutant in a room. In this chapter, a comparison is made between ε_c and other terms described in the literature, and the results of a parametric study are presented in an effort to determine what ventilation and source characteristics will affect the value of removal effectiveness.

3.1 Description

Pollutant removal effectiveness is defined as the ratio of the pollutant concentration in the ventilation exhaust duct to the room average concentration, or:

$$\varepsilon_c = \frac{C_{exhaust}}{C_{average}} \tag{3.1}$$

The value of ε_c can range anywhere between zero and infinity. In the case of complete mixing of the pollutant in a room, the removal effectiveness is equal to one. If a pollutant source is located close to a ventilation system exhaust duct, as it is with a local exhaust in industrial applications, the removal effectiveness can be much greater than one. Conversely, if the source is located in an area of the room where there is very little air movement from the ventilation system, the removal effectiveness will be much less than one.

Pollutant removal effectiveness can be determined experimentally when the pollutant volumetric source strength is known and the concentration in the exhaust duct is measured. The equilibrium average concentration in the room is calculated as in equation 3.2.

$$C_{average} = \frac{\left(\dot{V}_p \, t_e\right) - \int_0^{t_e} C_{exhaust} \, \dot{V}_a \, dt}{V_a} \tag{3.2}$$

The amount of pollutant entering the room is the pollutant volume source strength $\dot{V}_p$ multiplied by the time that equilibrium conditions are achieved t_e. The amount of pollutant leaving the room is determined by integrating the flow of pollutant in the exhaust duct over the same time interval. The difference between the two values is the steady-state amount of pollutant present in the room. Dividing this by the room air volume yields the room average concentration.

An advantage of using pollutant removal effectiveness to characterize the pollutant in the room is that the room can be treated as a single lump rather than having to use a multiple-lump description. A disadvantage is that ε_c will vary with time after a source is introduced or changes in strength.

3.2 Comparison with Other Mixing Parameters

A search through literature revealed more than a dozen models used to account for air and pollutant distribution in a ventilated room. There are basically two types of factors or scales based on these models. The first type measures the efficiency of air distribution in the room. The second type measures the efficiency of the air movement at removing pollutants generated in the room. Pollutant removal effectiveness, ε_c, is of this second type. Some of the other well known factors of both types are described in this section.

The simplest and oldest factor of the first type is the mixing factor, K. It was first described in a *Journal of Hygiene* article from 1946 [2] and is still referenced in the 1987 ASHRAE Systems Handbook [3]. The mixing factor is equal to the effective number of air changes divided by the nominal number of air changes derived from air flow rates. The effective number of air changes is generally used to account for local concentrations of pollutant, short-circuiting of ventilation air, and even the relative toxicity of the pollutant and safety factors. The theoretical basis for K comes from equation 3.3 which describes the concentration, C, of a pollutant in a ventilated room initially at concentration C_0 as a function of time.

$$C(t) = C_0\, e^{-K\left(\dot{V}_a / V_a\right) t} \tag{3.3}$$

Solving for K results in equation 3.4:

$$K = \frac{\ln (C_0 / C(t))}{(\dot{V}_a / V_a)\, t} \tag{3.4}$$

In the above equations, $\dot{V}_a$ is the volume flow rate of air circulated through the room by the ventilation system, and V_a is the volume of the room air space. Values for K are usually estimated to be between 0.1 and 0.3 [3, 4, 5], however experimental work by Drivas, *et al.* [5], and West [6] have determined values as high as 0.85. Like pollutant removal effectiveness, the mixing factor is a direct result of measured concentrations, and it therefore includes air movement effects on concentration. Although the mixing factor has been used to describe general air flow and pollutant source situations, the theory on which it is based does not take into account the imperfect mixing caused by the presence of a pollutant source after time zero.

Most of the more recently derived models for air distribution and pollutant removal scales fall into two categories; those based on the age of the air in the room and those based on a two-zone description of the air flow in the room. The models in these two categories can be quite mathematically involved, so they will only be described briefly.

Seppänen [1] and Sandberg [7] both describe similar models for air distribution based on the age of the air in the room. Seppänen's air exchange efficiency, ε_a, like pollutant removal effectiveness, is a simple ratio. It is equal to the shortest possible age of air in the room divided by the actual mean age:

$$\varepsilon_a = \frac{(V_a / \dot{V}_a) / 2}{\langle \tau \rangle} \tag{3.5}$$

The room air volume V_a divided by the circulation air volume flow rate $\dot{V}_a$ is the nominal time constant of flow through the room and is equal to the average age of air in the room for the completely mixed flow situation. This value is twice the average age of air in the room in the case of piston air flow, where the air sweeps through the room like the top of a piston from the inlet to the exhaust duct. Piston air flow results in the shortest possible age. The actual mean age of air in the room $\langle\tau\rangle$ is determined experimentally using a tracer gas and measurements of exhaust duct concentrations. Sandberg's relative air diffusion efficiency is identical to ε_a except that the divisor 2 in the numerator is not present. Sandberg's factor thus has a maximum value of 200%.

ASHRAE [8], Anderson [9], and Skaret and Mathisen [10] each describe different factors based on a two-zone model of room air flow. An example of a two-zone model is shown in Figure 3.1. ASHRAE and Anderson divide the room into a well mixed flow region and a bypass or short circuit flow region where the circulation air flows from the inlet directly to the exhaust without mixing with the room air. In Skaret and Mathisen, the model is somewhat more generalized with two well-mixed zones and inlet and exhaust flow paths in each zone. The idea of an inter-zone flow that is a fraction (β in Figure 3.1) of the circulation air flow is the same in both models.

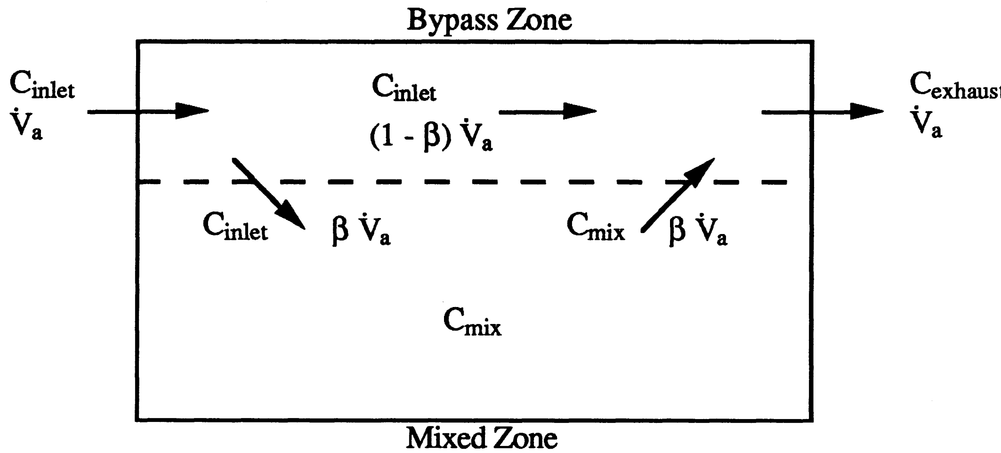

Figure 3.1 *Example of a Two-Zone Air Flow Model*

In Skaret and Mathisen, an air circulation efficiency scale, ε_{st}, is defined as the apparent ventilation air flow rate in the occupied zone divided by the completely mixed ventilation air flow rate. The ASHRAE ventilation effectiveness, E_V, is equal to the fraction of circulation air reaching the mixed zone which is β in figure 3.1. E_V is used in Appendix E of the ASHRAE air quality standard to adjust the required flow of outdoor air so that the actual flow of outdoor air reaching occupants in a space is determined. This is a practical use of an air circulation efficiency scale such as E_V or ε_{st}.

Anderson defines a scale of each type based on the two-zone room model. The displacement efficiency, η_d, is the fraction of room air displaced by the circulation air flow during the time that one air volume is supplied to the room. The removal efficiency, η_r, is the fraction of pollutant released into a uniformly mixed zone that is removed after one air change is supplied to the zone.

Each of the scales for air circulation or pollutant removal efficiency has its own characteristic dependence on various air flow and pollutant source parameters. Seppänen's pollutant removal effectiveness, ε_c, was chosen for use with this project because once ε_c and the room average concentration are determined, the room exit concentration can be found. This is exactly the information required for a multiple-zone pollutant transport model where incomplete mixing in the zones is to be considered. The dependence of ε_c on various parameters is explored in the next section.

3.3 Parametric Study

In order to determine what air flow and pollutant source characteristics might affect the value of pollutant removal effectiveness in a zone, a factorial study was performed.

3.3.1 Objectives and Limitations

The main objective of this factorial study was to determine which parameters will have a statistically significant effect on the value of removal effectiveness ε_c. The data for the study was generated using the three-dimensional fluid dynamics code FLUENT discussed in Chapter 2. Since the study would require a large number of FLUENT runs, each with slightly different inputs, the biggest limitation was computation time. This constraint limited both the number of parameters that could be tested for an effect (the number of runs) and the size of the computational grid into which the room was divided (the time for each run to be completed). While the size of the grid did not affect the accuracy of the computations for the problems modeled as shown in Section 2.2.3, it did affect the accuracy (or correctness) of the models. For example, the ventilation system inlets and outlets were only a single node across, so diffusers to distribute the air could not be modeled.

The number of FLUENT options that could be employed was also limited by computation time. An important option that was left out of the model was utilization of the energy equation. A more complete room air flow model would include a temperature difference between the circulation inlet air and the room air and walls to model temperature-induced buoyancy. An attempt was made to complete the first run with the energy equation turned on, but convergence had not been achieved after twice the number of iterations required for reasonable convergence of the run without the energy equation. Temperature-dependent runs were therefore considered to be impractical for this study.

3.3.2 Experiment Design and Factor Descriptions

Since the number of runs would be limited by computational time, it was decided that a two-level factorial design would provide the most information. Also, since it was only desired to know which parameters significantly affected pollutant removal effectiveness but not how much they affected its value, the two-level factorial would provide all the information required.

The list of parameters that could have an effect on the pollutant removal effectiveness value is probably endless. It was decided at the beginning of the project to keep the study as simple as possible. Since room geometry and placement of objects in the room have unlimited variations, only rectangular, empty rooms were considered. Also, the grid size used would have made anything but an empty room very difficult to model with any accuracy.

Three arrangements for the ventilation inlet and exhaust ducts that were felt to be representative of those in actual use were chosen for the study. The three arrangements are shown in Figures 3.2, 3.3, and 3.4. The bottom-up displacement air flow arrangement in Figure 3.2 has been shown in European studies to be efficient at pollutant removal if the inlet air is at least 3° C colder than the room air [11].

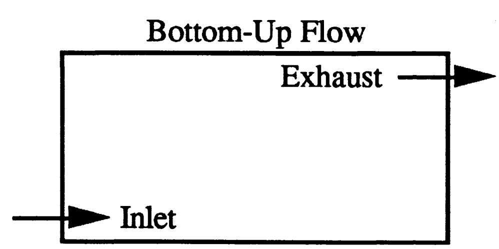

Figure 3.2 *Placement of Inlet and Exhaust for Bottom-Up Displacement Air Flow*

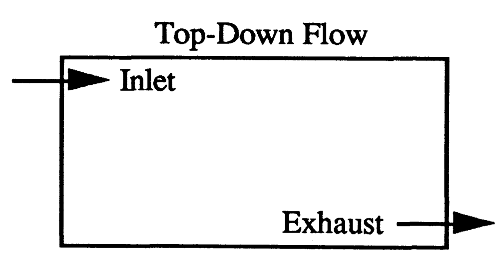

Figure 3.3 *Placement of Inlet and Exhaust for Top Down Displacement Air Flow*

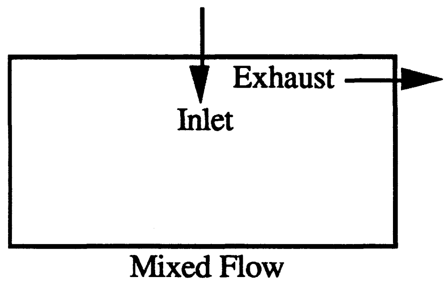

Figure 3.4 *Placement of Inlet and Exhaust for Mixed Air Flow*

Of the many other parameters that could have been chosen, three were expected to have a significant effect on pollutant removal effectiveness. The three were pollutant source strength, circulation air flow rate, and room size. High and low values for each parameter were chosen based on what might reasonably be found in an office setting. A complete list of the values is shown in Table 3.1.

Table 3.1 *Values for Low and High Factor Levels*

Factor	Low Value	High Value
Pollutant Source Strength	$5.0 \times 10^{-6}\ m^3 / s$	$1.5 \times 10^{-5}\ m^3 / s$
Air Flow Rate	1 air change / hr	5 air changes / hr
Room Size (L x H x W)	6m x 3m x 4m	9m x 3m x ∞

In table 3.1, the pollutant source strength shown is per source location. The small room had four locations and the large room had six. The low value is equal to the volume of CO_2 produced per second by a sedentary person. For the high value for room size, the room was actually reflected infinitely along the horizontal axis perpendicular to the plane of Figures 3.2, 3.3, and 3.4.

The resulting experiment design is a 2^3 complete factorial for a total of eight FLUENT runs for each of the three ventilation system arrangements. The eight runs result from taking all combinations of the low and high values for each of the three factors. The order that the runs were completed was not randomized since random noise does not occur in these computer simulations. It was decided not to compare results between the ventilation system arrangements due to the simple way that they were modeled in FLUENT. It seems safe to assume, however, that the value of removal effectiveness will be significantly affected by the inlet and exhaust duct placement.

3.3.3 Data Analysis

The resulting values for pollutant removal effectiveness, ε_c, from the eight runs for each of the ventilation system arrangements were analyzed using the method of Yates' Algorithm to arrive at the estimated effects for the main effects and interactions. Once the significant effects and interactions were decided upon, the expected values and residuals were calculated using Reverse Yates' Algorithm. These techniques are described in Chapter 10 of Box, Hunter, and Hunter [12].

A few words can be said at this point about normal probability plots and the use of residuals in computer simulation experiments. Data that matches a normal distribution will appear as a straight line on a normal probability plot. Data that falls off of the straight line on a normal probability plot is not normally distributed (random) and could therefore be statistically significant. The art in such an analysis is to know where to draw the straight line. Although there is no random noise in a computer simulation, a plot of residuals where statistically significant effects are removed can appear to be random. This is because the residuals are from effects that are not being taken into account, just as random noise is from variables that are not being controlled and therefore not being taken into account.

The estimated effects for the three ventilation arrangements are shown in Tables 3.2, 3.3, and 3.4. By just examining the numbers, none of the effects seem significant for the bottom-up displacement flow type. This is confirmed by the normal probability plots of effects and residuals in Figures 3.5 and 3.8, respectively.

Table 3.2 *Estimated Effects for Bottom-Up Displacement Air Flow*

Effect Label	Estimate of Effect
Mean Value of ε_c	0.929
Main Effects:	
Pollutant Source Rate	-0.013
Air Flow Rate	-0.023
Room Size	0.038
Two-Factor Interactions:	
Pollutant Source Rate and Air Flow Rate	-0.023
Pollutant Source Rate and Room Size	0.008
Air Flow Rate and Room Size	0.028
Three-Factor Interaction:	
Pollutant Source Rate, Air Flow Rate, and Room Size	0.008

In Table 3.3, the estimated values for main effects and interactions for top-down displacement flow are not as revealing as they were for the bottom-up displacement flow. Air flow rate, room size, and the interaction between the two could possibly be significant. The normal probability plot of these effects in Figure 3.6 confirms the ambiguity. Residuals were calculated and plotted for the cases where no effects are significant (Figure 3.9), air flow and room size are significant (Figure 3.10), and all three suspected effects are significant (Figure 3.11). The residual plot most resembling a straight line is the case where air flow and room size are significant. This was the preliminary conclusion.

Table 3.3 *Estimated Effects for Top-Down Displacement Air Flow*

Effect Label	Estimate of Effect
Mean Value of ε_c	1.084
Main Effects:	
Pollutant Source Rate	-0.003
Air Flow Rate	0.078
Room Size	-0.063
Two-Factor Interactions:	
Pollutant Source Rate and Air Flow Rate	0.008
Pollutant Source Rate and Room Size	0.008
Air Flow Rate and Room Size	-0.043
Three-Factor Interaction:	
Pollutant Source Rate, Air Flow Rate, and Room Size	0.008

The estimated effects values for mixed flow shown in Table 3.4 are once again quite easily interpreted. It appears that air flow rate and room size are statistically significant. This is supported by the appearance of the plot of effects in Figure 3.7, and the plot of residuals in Figure 3.12. It is interesting to note that although air flow rate and room size appear to be significant for top-down displacement and mixed flow types, they have the opposite effect on the value of ε_c for the two flow types. For example, increasing air flow rate in top-down flow increases the value of ε_c but decreases the ε_c value in mixed air flow.

Table 3.4 *Estimated Effects for Mixed Air Flow*

Effect Label	Estimate of Effect
Mean Value of ε_c	0.784
Main Effects:	
Pollutant Source Rate	0.013
Air Flow Rate	-0.133
Room Size	0.103
Two-Factor Interactions:	
Pollutant Source Rate and Air Flow Rate	0.003
Pollutant Source Rate and Room Size	-0.003
Air Flow Rate and Room Size	0.003
Three-Factor Interaction:	
Pollutant Source Rate, Air Flow Rate, and Room Size	-0.013

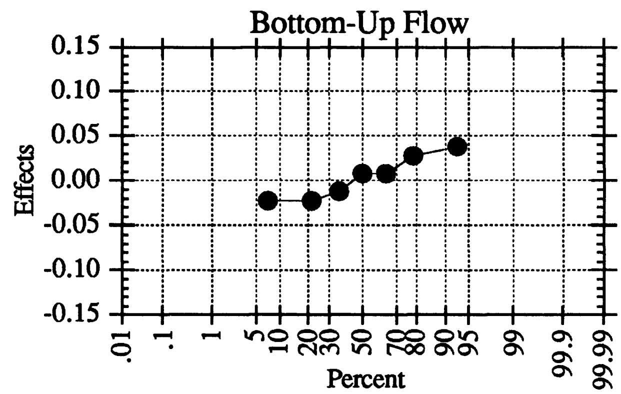

Figure 3.5 *Normal Probability Plot of Effects for Bottom-Up Displacement Air Flow*

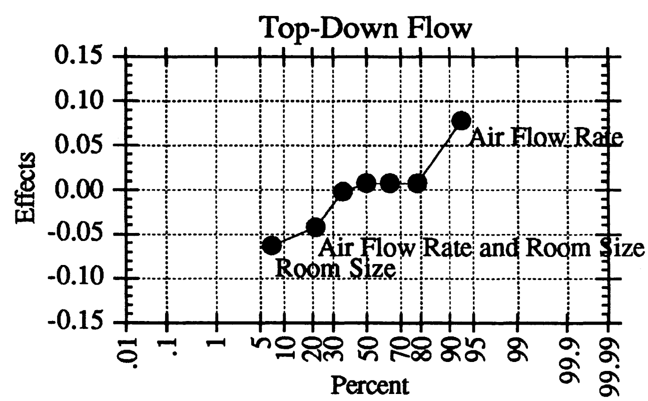

Figure 3.6 *Normal Probability Plot of Effects for Top-Down Displacement Air Flow*

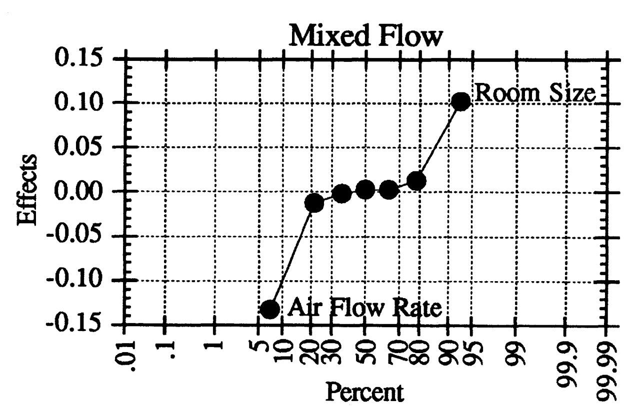

Figure 3.7 *Normal Probability Plot of Effects for Mixed Air Flow*

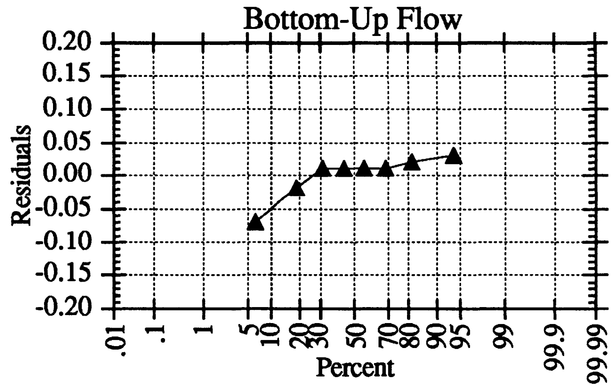

Figure 3.8 *Normal Probability Plot of Residuals for Bottom-Up Displacement Air Flow*

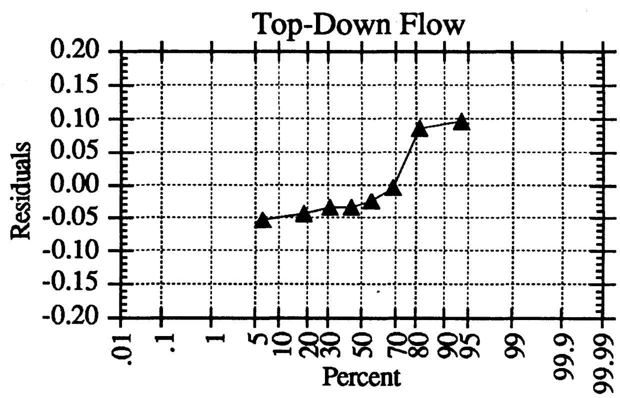

Figure 3.9 *Normal Probability Plot of Residuals for Top-Down Displacement Air Flow*

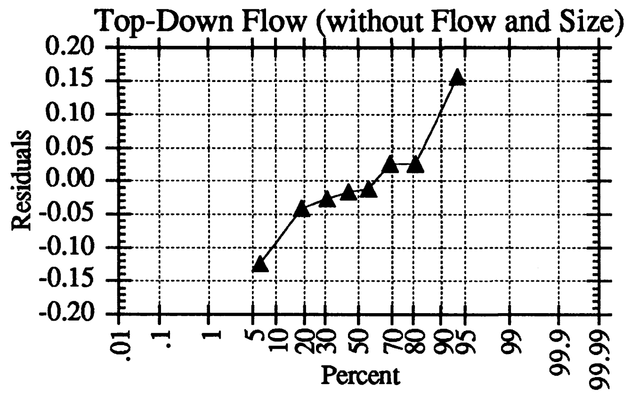

Figure 3.10 *Normal Probability Plot of Residuals for Top-Down Displacement Air Flow Without Air Flow Rate and Room Size Effects*

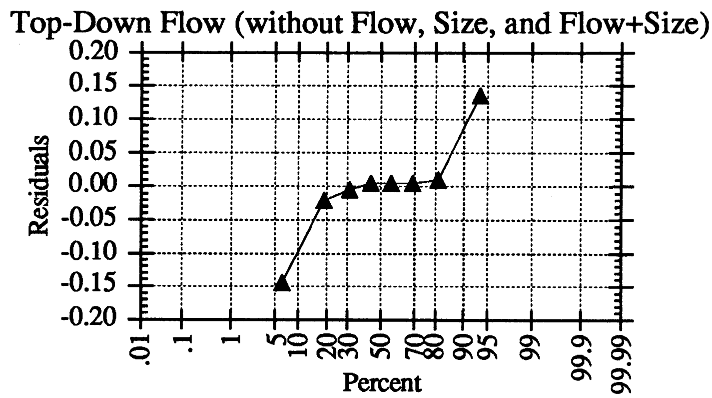

Figure 3.11 *Normal Probability Plot of Residuals for Top-Down Displacement Air Flow Without Air Flow Rate, Room Size and the Air Flow Rate + Room Size Interaction Effects*

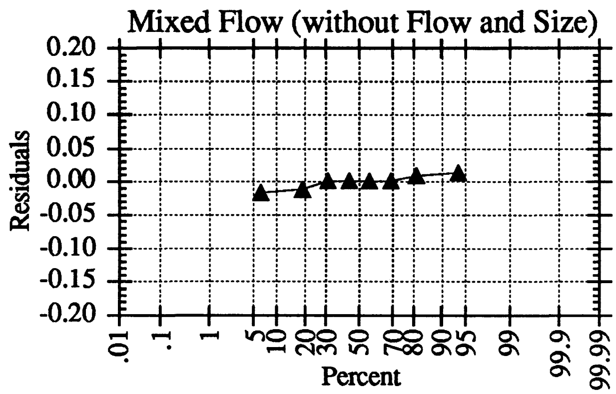

Figure 3.12 *Normal Probability Plot of Residuals for Mixed Air Flow Without Air Flow Rate and Room Size Effects*

One last test of significance can be applied to the estimated effects calculated above. Calculations of pollutant concentration versus time after a source was introduced were made for five values of pollutant removal effectiveness and otherwise identical inputs. The results are plotted in Figure 3.13. From the plot, it appears that a change in removal effectiveness value of less than 0.1 will not significantly affect the concentration curve. On this basis, air flow rate and room size are not significant for the top-down displacement flow type.

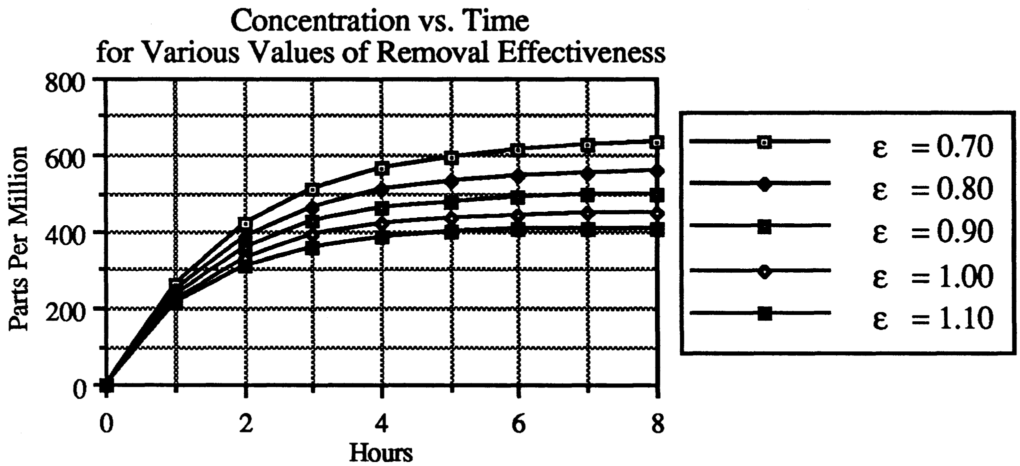

Figure 3.13 *Significance of Removal Effectiveness Value*

Although they will not be compared statistically, the mean values of removal effectiveness for the three ventilation flow arrangements are still of some interest. The highest mean was for top-down displacement flow at 1.08. The next highest was for bottom-up displacement flow with a value of 0.93. The lowest value was for mixed flow at 0.78. This last value is quite a bit lower than the theoretical value for completely mixed flow, which is 1.0. A diffuser added to the inlet may have improved this value somewhat. The performance of the bottom-up displacement flow arrangement would probably have improved if the effects of temperature and buoyancy could have been included in the model.

3.3.4 Conclusions

From the data analysis in the above section, a few conclusions can be drawn. First, for the two displacement air flow types, changing the pollutant source rate, air flow rate, or room size do not appear to significantly affect the value of pollutant removal effectiveness ε_c. Second, for the mixed air flow type, changing the pollutant source rate does not appear to significantly affect the value of ε_c. Third, for the mixed air flow type, increasing the room size appears to increase the value of ε_c. Fourth, for the mixed air flow type, increasing the air flow rate appears to decrease the value of ε_c.

The fourth conclusion above is similar to the one arrived at by West in his 1977 experimental study of factors that affect the mixing factor. West concluded that increasing the air flow rate in a mixed flow type room actually decreases the pollutant removal ability of the ventilation system. This somewhat counter-intuitive conclusion appears to have been supported.

3.4 References

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Chapter 4

MODELING HVAC TRANSPORT OF POLLUTANTS

This chapter will describe a model for the transport of indoor pollutants through a multiple-zone building by the heating, ventilating, and air-conditioning (HVAC) system. The model is developed and then validated through comparison to an existing computer model.

4.1 Multiple Zone Pollutant Transport

Although room air flow models have been in existence for nearly twenty years, the modeling of multiple-zone air flows is a more recent activity, the earliest documentation only having been published about ten 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. A search through literature revealed two basic types of air quality models.

The first type of multiple-zone air flow model uses a network technique similar to the one Kirchoff applied to electrical engineering problems. In this approach, the air flow 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. The nodes are connected by air flow paths such as windows, doors, ducts, and infiltration paths which are represented by the network elements. The nodal equations specify that the sum of the mass or energy flows at each node is zero. Each element equation is a resistance that relates the mass flow rate to the pressure drop across it. A model of this type will generally include the driving forces of stack effect, wind pressure, and forced (circulation) air flow. The node and element equations are formed into a matrix and solved simultaneously using an iterative method until converged, steady-state mass flows and pressures are achieved. 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 [1, 2]. In his 1989 survey of air infiltration and indoor air quality models, Haghighat [3] 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 inter-zone air flows 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. In this form, the equations can be solved as an initial-value problem resulting in concentration values as a function of time. This allows the determination of pollutant concentrations throughout a period of time during which the pollutant source and inter-zone flows may be varying. Since the ability to handle transient conditions was necessary for the objectives of this project, a model of this type was chosen. The Environmental Protection Agency (EPA) program INDOOR also uses this model [4].

4.2 Program Description

The following sections describe the multiple-zone indoor pollutant transport computer program that was written for this project.

4.2.1 Objectives

One of the major objectives of this project was to develop an indoor pollutant transport module for the TRNSYS transient system simulation program from the University of Wisconsin Solar Energy Laboratory (UW-SEL). The nodal pollutant mass balance model was chosen over the network model for two reasons. First, since transient simulations were required, the nodal mass balance was the most practical model to use from the standpoint of minimizing computational time. Second, the ventilation system air flows would not need to be calculated in the pollutant transport module because they were to be calculated by an external HVAC flow control module. Also, infiltration and inter-zone air flows could be estimated as they are for the Type 56 multiple-zone heating and cooling load calculation module in TRNSYS.

In addition to a variable ventilation system air flow, the pollutant transport module needed to be capable of varying the pollutant source level in each zone during the simulation to simulate a changing occupation level in the zones throughout the day. It was also desirable to include in the model a mechanism to deal with a zone where the pollutant source and air flow characteristics result in a non-uniform distribution of the pollutant. Lastly, as this was to be a transient simulation program for large office building ventilation systems, the circulation time of the air through the ventilation system could be important and had to be taken into account (see Section 4.2.4 for the reasoning behind including the circulation time).

4.2.2 Model Development

The general concepts regarding pollutant transport by a ventilation system are shown in Figure 4.1. Air flows into and out of each zone, carrying with it the pollutants in various concentrations. For an individual zone, the possible air flow paths are the circulation flow in and out, infiltration flow in, and inter-zone flow in and out. The flow of pollutant can be obtained from the air flows by multiplying by the concentration of pollutant in the air stream. The zone may also contain a pollutant source.

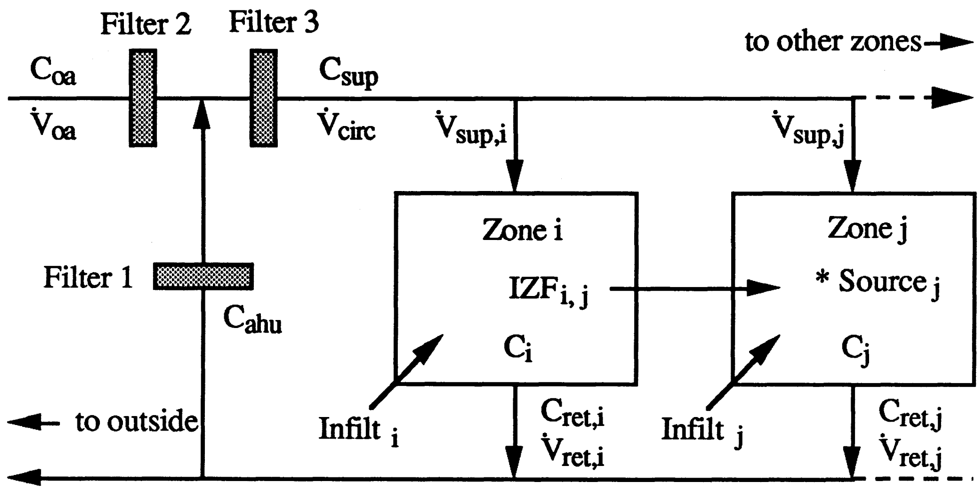

Figure 4.1 *Ventilation System Modeled by Type 60*

The pollutant transport model is based on a mass balance of pollutant in each zone. For now, only mass flow in, mass flow out, and the change in mass present in the zone will be considered. The relationship between these is shown in equation 4.1. The subscript "p" in this and the following equations indicates pollutant quantities.

$$\frac{dm_p}{dt} = \dot{m}_{p,in} - \dot{m}_{p,out} \tag{4.1}$$

For this model, the density of the pollutant is assumed to be a constant. Note that the density is the mass of the pollutant present divided by the *partial* volume occupied by the pollutant, not the entire zone volume. If both sides of equation 4.1 are divided by the pollutant density, a volume balance is the result as in equation 4.2. Again, the volumes are partial volumes occupied by the pollutant.

$$\frac{dV_p}{dt} = \dot{V}_{p,in} - \dot{V}_{p,out} \tag{4.2}$$

The volume flow rate of pollutant is equal to the air volume flow rate times the volume fraction (or volume concentration) of pollutant. This is shown in equations 4.3 and 4.4. The subscript "a" indicates quantities for air.

$$\frac{dV_p}{dt} = \dot{V}_{a,in}\left(\frac{V_p}{V_a}\right)_{in} - \dot{V}_{a,out}\left(\frac{V_p}{V_a}\right)_{out} \tag{4.3}$$

$$\frac{dV_p}{dt} = \dot{V}_{a,in}\, C_{in} - \dot{V}_{a,out}\, C_{out} \tag{4.4}$$

If both sides of equation 4.4 are now divided by the air volume in the zone (essentially equal to the nominal zone volume for dilute pollutant concentrations), the result is equation 4.5. For dilute concentrations, the air volume is nearly constant, so it can be moved inside the differential on the left-hand side of the equation.

$$\frac{d\left(\frac{V_p}{V_a}\right)}{dt} = \frac{\dot{V}_{a,in}}{V_a} C_{in} - \frac{\dot{V}_{a,out}}{V_a} C_{out} \tag{4.5}$$

The left-hand side of equation 4.5 is really the time derivative of the overall volume fraction or volume concentration for the zone. The final form of the general pollutant balance equation for a zone as used in the pollutant transport model is shown in equation 4.6.

$$\frac{dC}{dt} = \frac{\dot{V}_{a,in}}{V_a} C_{in} - \frac{\dot{V}_{a,out}}{V_a} C_{out} \tag{4.6}$$

If equation 4.6 is now expanded to include all of the possible air flow paths into or out of zone "i" as shown in Figure 4.1, the result, equation 4.7, has four in-flow terms and two out-flow terms. The subscript "i" has been added to denote quantities specific to zone i. The volume flows are now all for air, so the subscript "a" has been dropped.

$$\frac{dC_i}{dt} = \frac{S_i}{V_{a,i}} + \frac{\dot{V}_{inf,i}}{V_{a,i}} C_{oa} + \frac{\dot{V}_{sup,i}}{V_{a,i}} C_{sup} - \frac{\dot{V}_{ret,i}}{V_{a,i}} \varepsilon_{c,i} C_i + \frac{\sum_j \left(\dot{V}_{izf,j,i} C_j\right)}{V_{a,i}} - \frac{\left(\sum_j \dot{V}_{izf,i,j}\right) C_i}{V_{a,i}} \tag{4.7}$$

The first term on the right-hand side of equation 4.7 is a pollutant source located in zone i. The source S_i has units of volume per unit time. The second term is the air infiltration volume flow rate into the zone multiplied by the constant outside air concentration of the pollutant.

The third and fourth terms in equation 4.7 are for the air circulation flow through the zone. The supply air flow is multiplied by a variable supply air concentration to arrive at the pollutant flow into the zone. The return air flow is multiplied by the zone average concentration as modified by the pollutant removal effectiveness, ε_c, for the zone. The removal effectiveness is the zone return duct pollutant concentration divided by the zone average concentration as described in Chapter 3, and it accounts for the fact that the pollutant may not be fully mixed in the zone air.

The last two terms in equation 4.7 are for the inter-zone flows of pollutant. The next to the last term is the flow into zone "i" from all other zones "j". The air flow $\dot{V}_{izf,j,i}$ represents the volume air flow 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 inter-zone pollutant flow into zone i. The last term in equation 4.7, the inter-zone pollutant flow out of zone i, is similar except that the air flow is multiplied by the pollutant concentration in zone i.

The air volume flow rate returning to the ventilation system from zone i is calculated by a balance of all the other air flows for the zone as in equation 4.8.

$$\dot{V}_{ret,i} = \dot{V}_{sup,i} + \dot{V}_{inf,i} + \dot{V}_{izf,i,j} - \dot{V}_{izf,j,i} \tag{4.8}$$

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 a flow normalized summation of the pollutant flows from all of the zones. This is shown in equation 4.9. In the equation, t is the present time. 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. The importance of DT is explained in Section 4.2.5.

$$C_{ahu} = \frac{\sum_i \left(\dot{V}_{ret,i}\, \varepsilon_{c,i}\, C_{i,t\text{-}DT}\right)}{\sum_i \dot{V}_{ret,i}} \tag{4.9}$$

The second step in calculating the supply air pollutant concentration C_{sup} accounts for the fact that some of the circulation air may be exhausted to the outside and replaced with an equal volume of outdoor air. The concentrations are again flow-normalized. This process is shown in equation 4.10.

$$C_{sup} = \frac{\left(\sum_i \dot{V}_{sup,i} - \sum_i \dot{V}_{oa,i}\right) C_{ahu} + \left(\sum_i \dot{V}_{oa,i}\right) C_{oa}}{\sum_i \dot{V}_{sup,i}} \tag{4.10}$$

4.2.3 Program Implementation of the Model

The pollutant transport model represented by equations 4.7 through 4.10 was implemented as subroutine Type 60 for the TRNSYS simulation program. Equation 4.7, the concentration differential equation, is solved in Type 60 by the same iterative modified Euler method used in other TRNSYS components [5]. The concentrations for each zone at time t+Δt are first estimated using the simple Euler method using the zone concentrations at time t as shown in equation 4.11. The superscript on concentration represents the iteration number. The function f is the right-hand side of equation 4.7, and Δt is the Type 60 time step.

$$C^1_{t+\Delta t} = C_t + f(C_t)\,\Delta t \tag{4.11}$$

The second and subsequent estimates of concentration in each zone are calculated using the estimated slope at the midpoint of the time interval as in equation 4.12.

$$C_{t+\Delta t}^{n+1} = C_t + f\left(C_{t+\Delta t/2}^{n}\right)\Delta t \tag{4.12}$$

The concentration at the midpoint of the interval is an average of the concentration at time t and the most recent estimate of the concentration at time t+Δt. This is shown in equation 4.13.

$$C_{t+\Delta t/2}^{n} = \frac{C_t + C_{t+\Delta t}^{n}}{2} \tag{4.13}$$

The iterations continue until two successive estimates of concentration for each zone differ by less than a specified tolerance. For this project, the tolerance for convergence was set to 1.0E-8 (10 parts per billion).

The time step for the modified Euler method used in this program is set independently of the TRNSYS simulation time step. This was necessary because the concentration transients in response to changing circulation and outside air flows and changing source levels require a much shorter time step than is practical for TRNSYS simulations. The Type 60 time step is restricted to not smaller than 1/60th of the TRNSYS simulation time step. Also, the simulation time step should be an integer multiple of the Type 60 time step. The Type 60 time step used for most of the simulations in this project was 30 seconds, and the TRNSYS simulation time step was 15 minutes.

The circulation and outside air flows are calculated external to the Type 60 program and input at each simulation time step. The building information necessary for Type 60 calculations such as zone volumes, zone removal effectiveness, infiltration and inter-zone flows are read in from a data file at the beginning of the simulation. The variable pollutant source in Type 60 was accomplished by reading in a unit source strength and a schedule of source level multipliers. The current simulation time is compared to the schedule time at each simulation time step to adjust the source level accordingly.

Type 60 outputs the average and return duct concentration for each zone, and the concentrations at the air handling unit and in the supply air. Also available is the volume flow of outdoor air divided by the source level multiplier for each zone. This is useful since if the unit source strength is equivalent to the pollution generated by one person, then the source level multipliers are equivalent to the number of people present at the schedule time. The result is the volume flow rate of outdoor air per person for each zone at each simulation time step.

4.2.4 Additional Features

The Type 60 pollutant transport module includes a few other features that were not mentioned in the above description. Figure 4.1 shows the location of three filters in the ventilation air stream. These are modeled as ideal, non-saturating filters that change the pollutant concentration in the air stream by a filter efficiency factor as in equation 4.14. The filter model could quite easily be modified to include saturation effects by using a variable efficiency. The filter capability was not used for this project since the pollutant modeled was CO_2 gas.

$$C_{after} = C_{before}\, \varepsilon_{filter} \tag{4.14}$$

Although the inter-zone flows used in the pollutant transport model are input as constants, it was reasoned that this flow would decrease considerably at times when the ventilation system is shut down. Therefore, at times when there is no ventilation system air flow, the inter-zone flows are set equal to zero.

The zone pollutant removal effectiveness, ε_c, is also a constant in the model. Although there are several variables that could change the value of the removal effectiveness as shown in Chapter 3, there isn't 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 model accounts for this by setting ε_c for a zone equal to 1.0 when there is no pollutant source in the zone.

4.2.5 Importance of the Air Circulation Delay Time

From the beginning of work on this project, it was recognized that the time that the air and pollutants take to circulate through the ventilation system could affect the transient pollutant concentration in a ventilated room. Measurements of air flow rates were performed in the Engineering Research Building at the University of Wisconsin in order to quantify the time delay that could exist in a large ventilation system. The time for air to travel through the supply air ducts was very short, less than 30 seconds in all cases. Depending on the location of the room, however, the time for air to travel through the return air path was much longer, up to several minutes. The difference in times is because the supply duct volume is small compared to the return air path volume, which is usually the ceiling plenum or a corridor. With a Variable Air Volume (VAV) system at its low flow setting, these delay times could be lengthened by a factor of two or three.

The effect of various air circulation delay times on the transient pollutant concentration in a room with a typical air flow rate is shown in Figure 4.2.

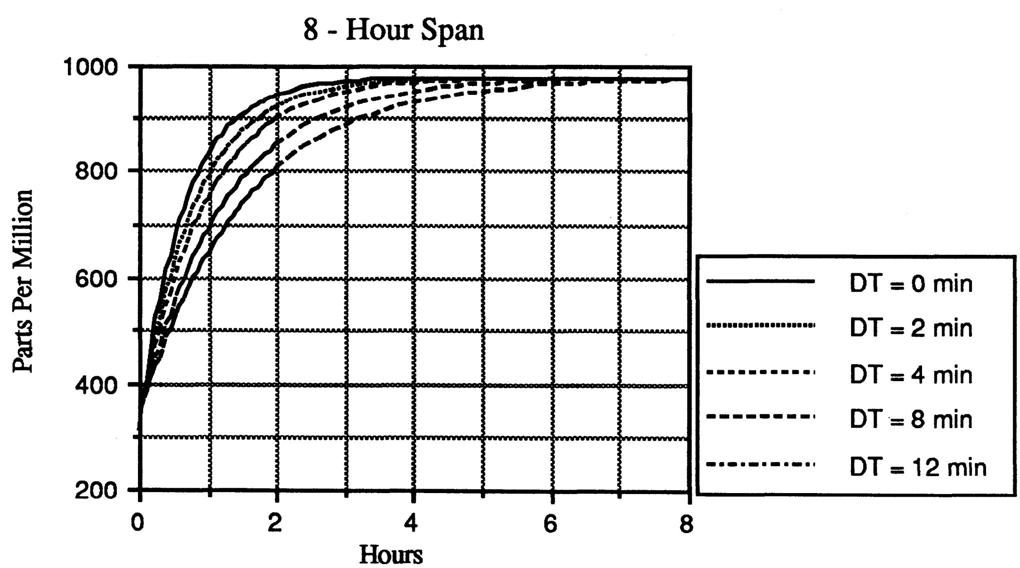

Figure 4.2 *Concentration as a Function of Air Circulation Delay Time, DT*

The equilibrium pollutant concentration is not affected by the delay time, but the time required to reach equilibrium (or any other specified concentration level) is affected. The difference that this delay time can have upon a control system is illustrated in Figure 4.3. For example, suppose a system has an air circulation delay time of 4 minutes, and a control action was to take place at a concentration of 800 parts per million. The action would take place 1 hour and 12 minutes into the transient. Without the delay time, the control action would take place over 20 minutes sooner.

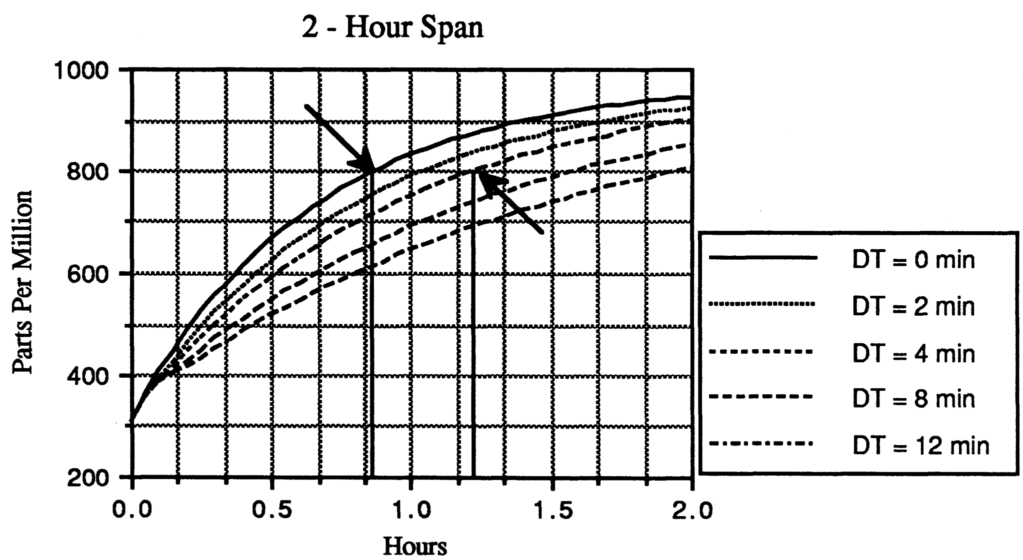

Figure 4.3 *Effect of Air Circulation Delay Time DT on Time to Reach a Specified Concentration Value*

The effect of the delay time on the energy use by an automatic system that adjusts outside air flow when a certain concentration setpoint is reached was evaluated for a VAV system with a ceiling plenum return air system. The total difference in annual

HVAC energy use between simulations with and without the delay time was less than 1 percent for the system modeled in this project. However, a comparison of the concentration profiles for the two year-long simulations shows a shift in the maximum concentration value of 100 parts per million higher when the delay time is not considered. The complete histogram of the fraction of time at various concentration values is displayed graphically in Figure 4.4

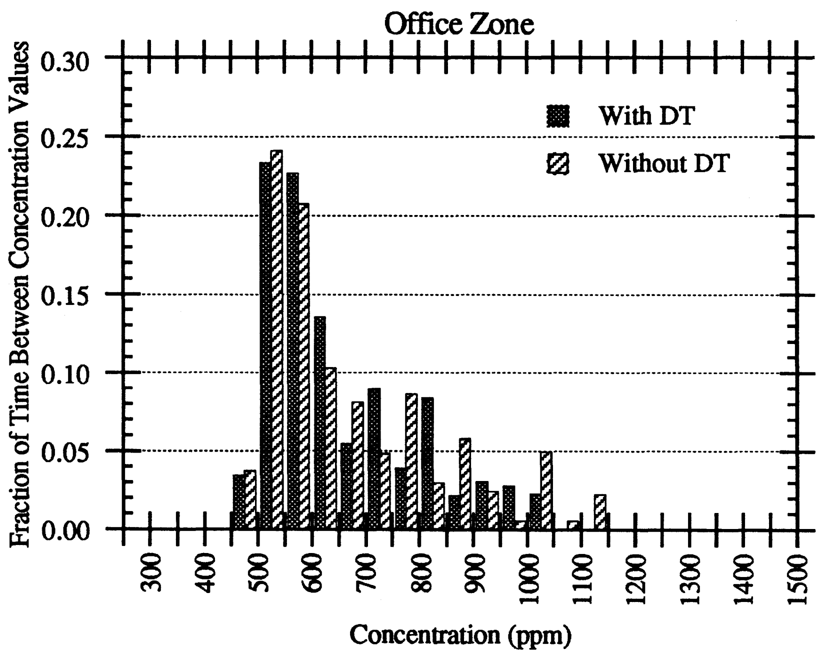

Figure 4.4 *Effect of Air Circulation Delay Time on Concentration Profile for One Year Simulation*

4.3 Model Validation

The comparison of a new model with some accepted standard is a necessary step to ensure its accuracy and validity. Due to constraints of time and resources, it was not possible to perform controlled experimental pollutant measurements to compare with predicted concentration values from the Type 60 model. There is also, at present, a lack of such measurement data published in the literature that provides sufficiently detailed information to be of use for validation purposes. However, the EPA indoor air quality model INDOOR was available for comparison runs with identical inputs [6]. The calculations of this model have been compared with test measurements at the EPA test house and with the NIST indoor air quality model CONTAM with good results [7, 8].

Five different runs were made with identical inputs to compare the Type 60 pollutant transport model predictions to those of INDOOR. The results of these comparisons are shown in Figures 4.5 through 4.9. The first two comparisons were made with two zones of different volumes and a pollutant source in one of the zones. The first run had 100% recirculated air, and the second run had an air flow consisting of 20% outside air. The third comparison run had only one zone and 100% outside air. The fourth comparison was made with 20% outside air and a pollutant source located in each of two zones. The last comparison was for two equal size rooms with a source in one, and included infiltration and inter-zone flows. As can be seen from the graphs, the predicted concentration values from the two programs are essentially identical.

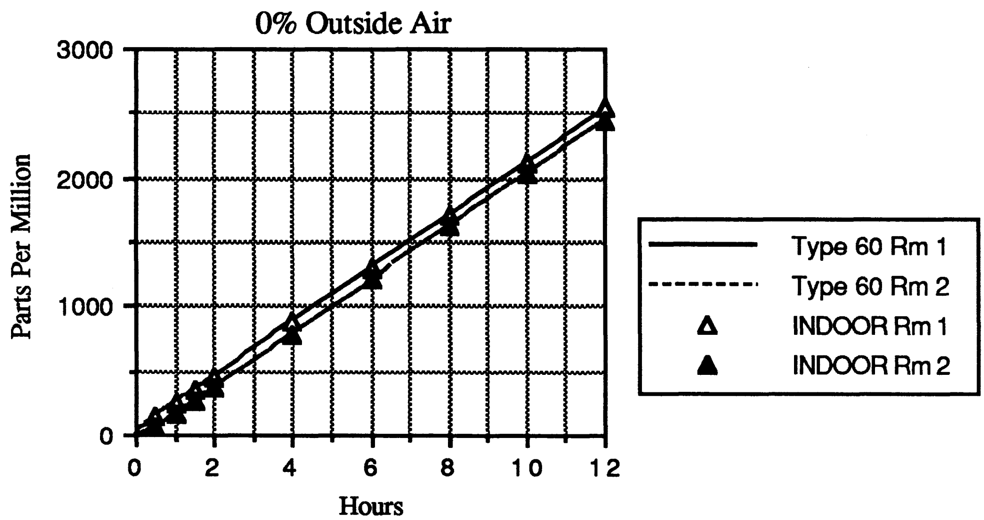

Figure 4.5 *Comparison of Type 60 and INDOOR Predicted Concentrations for 100% Recirculated Air*

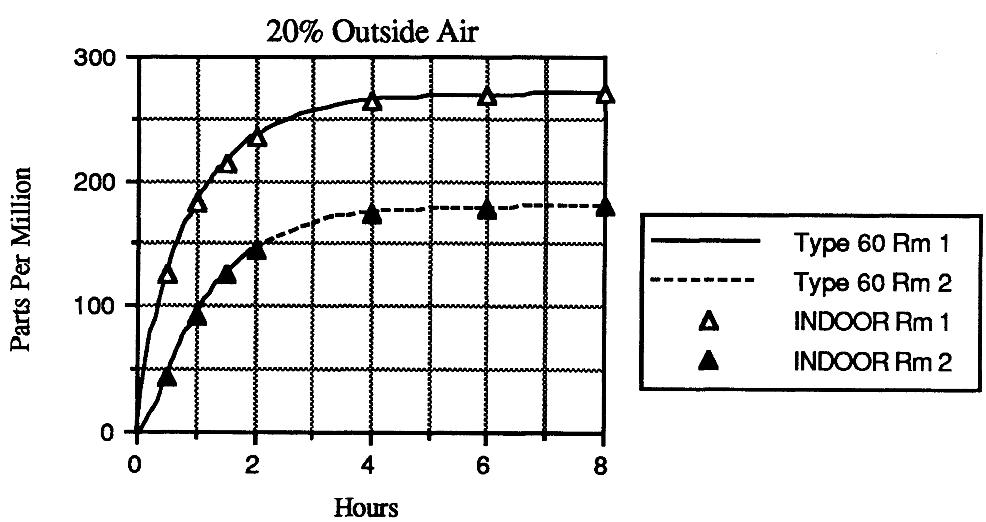

Figure 4.6 *Comparison of Type 60 and INDOOR Predicted Concentrations for 20% Outside Air*

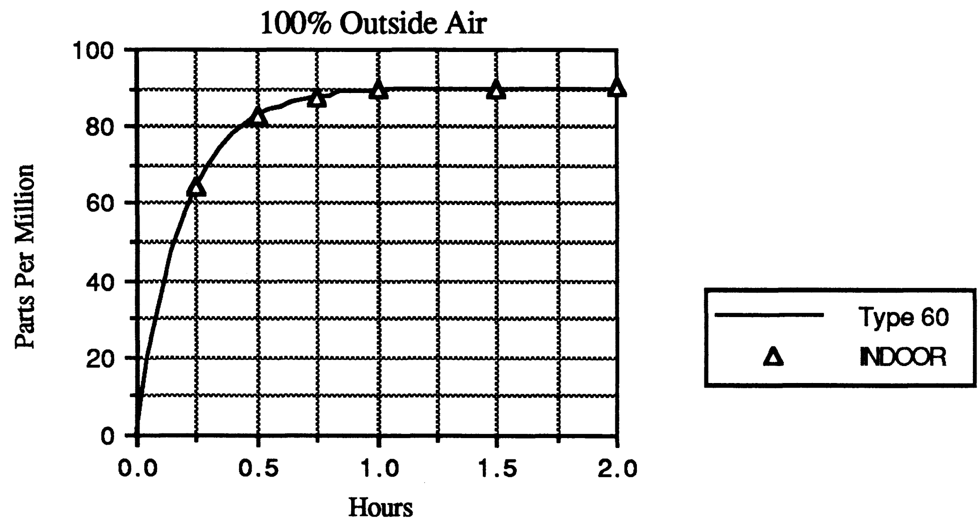

Figure 4.7 *Comparison of Type 60 and INDOOR Predicted Concentrations for 100% Outside Air*

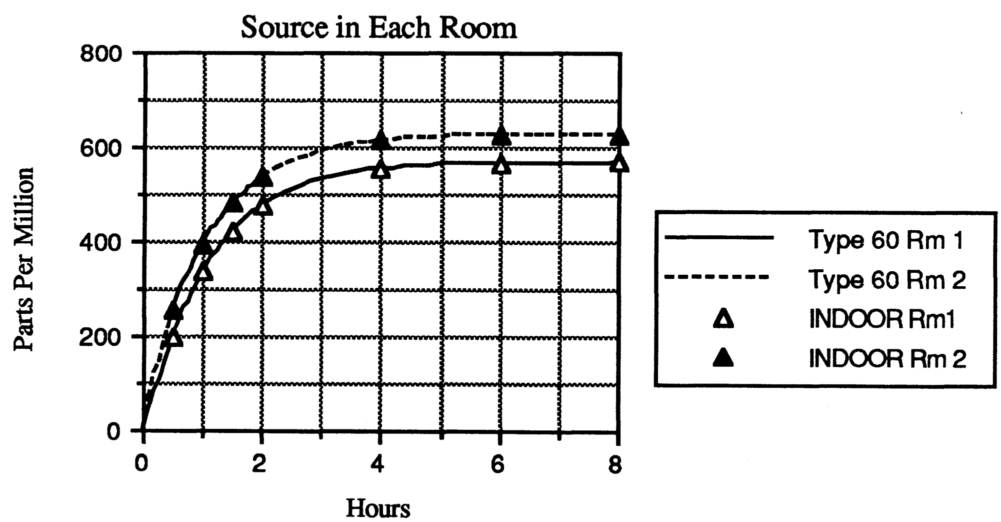

Figure 4.8 *Comparison of Type 60 and INDOOR Predicted Concentrations for a Source in Each Zone*

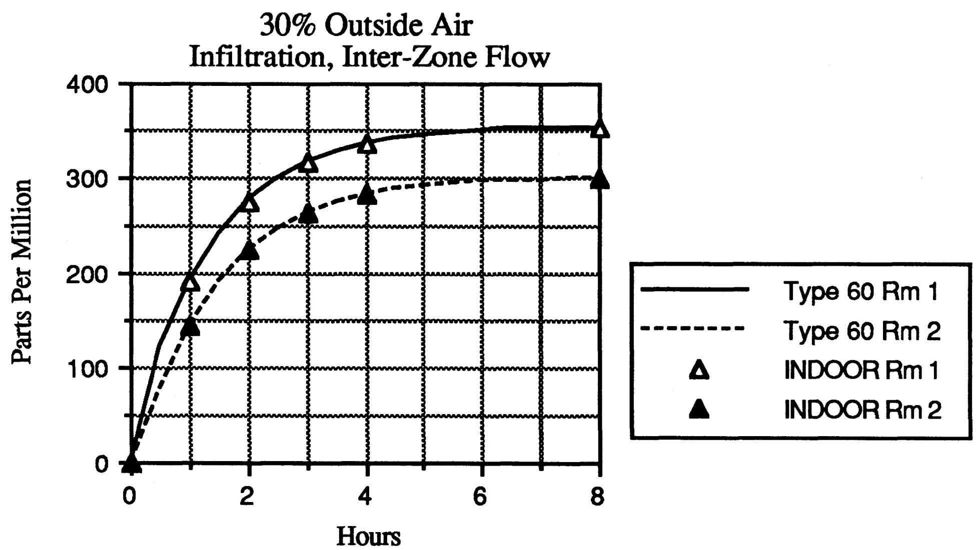

Figure 4.9 *Comparison of Type 60 and INDOOR Predicted Concentrations for 30% Outside Air with Infiltration and Inter-Zone Flows*

4.4 References for Chapter 4

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Chapter 5
INDOOR AIR QUALITY CONTROL SCHEMES

The model for pollutant transport through a multiple-zone building described in Chapter 4 can be used for a number of different indoor air quality investigations. Some of these were explored as a part of this project and are examined in this chapter. Another new component for TRNSYS, the circulation and ventilation air flow controller, Type 65, is also described.

5.1 Ventilation Air Flow Control

To enable testing various proposed schemes for indoor air quality control, the Type 65 component was written for TRNSYS to control the flow of outside air based on the pollutant concentration calculated by the Type 60 pollutant transport component. There are four varieties of flow controller in Type 65: a proportional controller, a purge controller, a scheduled purge controller, and a temperature-based economizer controller. These are explained in Sections 5.1.3 to 5.1.6. For a variable air volume (VAV) system, Type 65 also calculates the circulation air flow based on the zone sensible load calculated by the Type 56 multiple zone building load component [1]. This process is explained in Section 5.1.7.

5.1.1 Pollutant Sensor

Type 65 takes action based on one of three pollutant concentration values from Type 60. The controlling concentration can be either the maximum zone return air duct concentration, the mixed return air concentration at the air handling unit inlet, 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. All four controller types in Type 65 increase the flow of outside air when the controlling concentration value exceeds a high limit setpoint. The flow of outside air is decreased when the controlling concentration value falls below a low limit setpoint.

Commercially available carbon dioxide sensors operate on the principle of absorption of infrared light by CO_2 [2]. The detector cell consists of a chamber for the gas sample with a source of infrared light at one end and an infrared light sensor at the other. The infrared light from the source is absorbed by the intervening gas in proportion to the CO_2 concentration present. The level of infrared light reaching the sensor is then converted to an electrical signal. Models are available that will sample four points sequentially and provide a trip signal based on either the highest or average concentration. These units also provide an automatic periodic zero check and have a claimed accuracy of $\pm$ 5%. A sensor of this type could be located such that the length of the sample lines required would be minimized. This is important both to minimize cost and because the length of sample line will determine how quickly the sensor system can respond to changing conditions in a zone.

5.1.2 Flow Limitations

The outside air flows output by Type 65 are limited by high and low values as specified in the TRNSYS input deck. The low limit for outside air flow is the initial value for outside air flow set in the input deck. This enables setting a base or minimum value for an outside air flow controller and can be used to specify a constant value for outside air flow to model non-controlling ventilation systems. The maximum value for outside air flow is the initial value for circulation flow in the input deck. This value is also the constant circulation flow rate of a constant air volume (CAV) system and the maximum circulation flow rate for a VAV system. In Type 65, priority is always given to providing the outside air flow rate required to control the pollutant level. Therefore, the outside air flow rate determined by the controller based on the pollutant concentration is also the minimum circulation air flow rate for the VAV system and the minimum outside air flow rate for the temperature-based economizer system.

Type 65 allows the ventilation system to be started up and shut down once each day at a scheduled time. The times for this are specified in a separate input file read by Type 65 at the beginning of the TRNSYS simulation.

5.1.3 Proportional Control

This type of controller increases the flow of outside air by 20% of the maximum circulation air flow 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 outside air is decreased by 20% of the maximum circulation air flow rate at each time step. These decreases continue at each time step until the minimum outside air flow is reached.

5.1.4 100% Purge Control

The purge controller operates basically the same way as the proportional controller, except that when the high concentration limit setpoint is exceeded, the system switches to 100% outside air at the maximum circulation air flow rate. The flow drops to the minimum outside air flow rate when the concentration falls below the low limit setpoint.

5.1.5 Proportional and Scheduled Purge Control

Although the flexibility of this controller allows quite a few different simulation scenarios, the original idea behind it was to purge the indoor space of pollutants during the morning hours of a warm day in an attempt to minimize the amount of outside air that would be required during the hottest part of the afternoon.

The controller operates first as a proportional controller. It will also increase the outside air flow rate to a specified fraction of the maximum circulation air flow rate when certain conditions are met. First, the controlling pollutant concentration level must be above a specified fraction of the high limit setpoint. Next, the outside air temperature must be above a specified value. Lastly, the time of day must match a schedule time for the purge to take place. Up to six times and corresponding fractions of the maximum circulation flow rate can be specified. The limits and schedule information are read in by Type 65 from a data file at the beginning of the simulation.

5.1.6 Proportional and Temperature-Based Economizer Control

Again, this controller operates first as a proportional controller. The temperature-based economizer control is used at times when there is a cooling load in the building and the outside air temperature is less than the indoor air temperature. The idea is to minimize the amount of energy required by the cooling coil to meet the building load. When the outside air temperature is less than the indoor air temperature but greater than the specified coil outlet temperature, the outside air flow rate is set equal to the circulation air flow rate (100% outside air). When the outside air temperature is less than the specified coil outlet temperature, outside air is mixed with recirculated room air in an attempt to achieve the specified coil outlet temperature without requiring any energy removal by the coil. For this case, the ratio of outside air flow to circulation air flow is calculated as shown in equation 5.1.

$$\dot{V}_{oa} = \dot{V}_{circ} \frac{(T_{zone} - T_{coil})}{(T_{zone} - T_{oa})} \tag{5.1}$$

In the above equation, $\dot{V}_{oa}$ is the outside 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 outside air temperature.

5.1.7 Variable Air Volume System Modeling

In a VAV system, the circulation air is at the coil outlet temperature. The flow rate is adjusted so that the flow of air reaching a zone will be just enough to carry the cooling or heating load of that zone. The zone sensible loads are input to Type 65 from the Type 56 building load component and are then used to calculate the circulation air flow rate using equation 5.2.

$$\dot{V}_{circ} = \frac{\dot{Q}_{sens}}{\rho_a \, C_p \, (T_{zone} - T_{coil})} \tag{5.2}$$

In equation 5.2, $\dot{V}_{circ}$ 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.

As was mentioned in Section 5.1.2, the circulation air flow rate output by Type 65 is never less than the outside air flow rate required to control the level of pollutant. The circulation air flow rate may therefore 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 over-cooling the zone. In heating situations, the temperature of the zone inlet air is lowered.

5.2 Parametric Study Description

This section will describe a comparison study of various methods to achieve an acceptable indoor air quality using outside air flow. The performance of the fixed outside air flow rate method recommended in ASHRAE Standard 62-1989 [3] is compared to methods that provide automatic control of the outside air flow rate. The automatic control systems use the level of occupant-generated carbon dioxide present in the ventilation system as an indicator of air quality.

5.2.1 Objectives

The main objective of this comparison is to find out if an automatic system for controlling the flow rate of outside air can provide a level of protection from occupant-generated pollutants equivalent to that provided by the 1989 ASHRAE indoor air quality standard and still result in energy savings. A second objective is to determine which automatic systems have the best potential to provide energy savings and pollutant control.

5.2.2 Office Model

The office area that was simulated for this comparison study is based on a TRNSYS model of the ninth floor of the Independent Life Insurance Building located in Jacksonville, Florida. The single zone model developed by Ruud [4] for a study of building thermal storage was modified for this project by adding 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 1300 m^2 (14000 ft^2) in area and has an air volume of 3370 m^3 (119000 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 (2850 ft^3). The meeting room zone is surrounded by the office zone. The maximum occupancy of the office zone in the model is 100 persons and of the meeting room is 10 persons. These occupancy levels are varied somewhat through the business day to simulate actual conditions. A carbon dioxide generation rate of 5.0×10^{-6} m^3/s (1.77×10^{-4} ft^3/s) per occupant was used as the source strength. This is the CO_2 generation rate given in Appendix D of the ASHRAE air quality standard for an activity level of 1.2 met.

The HVAC system was operated between the hours of 5:00 am and 9:00 pm, 7 days a week. The CAV (or maximum VAV) circulation air flow rate used in the simulations was 6 volume air changes per hour (ach). When the HVAC system was on, an inter-zone flow from the meeting room to the office of 0.5 meeting room volume air changes per hour 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 air flow path. A return air volume equal to 1 m times the floor area was assumed for both zones. The complete thermal description of the office and meeting room zones is given in the building input description (BID) file for Type 56 listed in Appendix E.

5.2.3 Parameter Descriptions

Many of the comparisons made in this study were based on a set of seven year-long simulations that were run for both a CAV and a VAV system using weather data from Madison, Wisconsin, and Miami, Florida. This resulted in a total of 28 base simulations. The set of seven simulations was made up of three constant outside air flow rate situations and one simulation of each of the four methods of automatic outside air flow control described in Section 5.1. The constant outside air flow rate simulations were accomplished by setting the high concentration limit setpoint to 1.0. For the automatic control simulations, the high limit setpoint was 1000 parts per million (ppm) by volume (0.001), and the low limit setpoint was 800 ppm (0.0008).

The first constant outside air flow simulation used the flow rate recommended in the 1989 ASHRAE indoor air quality standard. For a multiple-zone system, the ASHRAE standard provides equation 5.3 to correct for anticipated uneven pollutant loads in the two 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.

$$Y = \frac{X}{(1 + X - Z)} \tag{5.3}$$

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

The required outside air fractions are calculated by multiplying the occupancy level by the required outside air flow rate per person from the ASHRAE standard and dividing by the nominal circulation air flow rate. For an office area, the required flow rate is 10 l/s (20 cfm) of outside 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 outside air fraction for this situation was 0.38 which translates to an outside air flow rate of 2.3 ach using a circulation air flow rate of 6 ach. If the uncorrected fraction of outside air had been applied, the outside air flow rate would have been only 1.1 ach. This would have been an incorrect application of the ASHRAE standard.

The second constant outside air flow rate scenario that was simulated used a "typical" value for outside air flow of 0.7 ach. This typical value is an average of over 3000 measured outside air flows from 14 different office buildings reported by Persily [5] in 1989.

The last "constant" outside air flow scenario was really a temperature-based economizer simulation with a constant minimum outside air flow of 0.7 ach.

The four automatic outside air flow control scenarios did not include a minimum value for outside air flow. Control of pollutant concentration was to be accomplished by the controller alone. This was felt to be the most challenging test of the controllers ability to keep the pollutant concentration at a reasonable level. In actual practice, a base minimum outside air flow would be required to dilute the indoor pollutants that are not related to human occupancy. Comparison simulations with a 1 ach minimum outside air flow rate were performed and are discussed in Section 5.3.7.

5.3 Results and Comparisons

The number of simulation runs that were completed for this study made quite a few comparisons possible. Some of the more interesting ones are summarized in this section.

5.3.1 Energy Use

One of the major objectives of this study was to compare the energy use of the various methods of controlling the amount of outside air delivered to the indoor zones. Tables 5.1 and 5.2 list the estimated annual energy use for the seven scenarios described in the previous section for Madison 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 air conditioning condenser. In all cases, the automatic flow control systems have an energy advantage over the ASHRAE constant air flow rate. 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.

Table 5.1 *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 + Morning Purge	1688	25	2527	13
Proportional + Temperature	1811	20	2648	9
100% Purge Control	1626	28	2466	15

Table 5.2 *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 + Morning Purge	1049	41	1125	42
Proportional + Temperature	1004	44	1058	46
100% Purge Control	972	45	1056	46

There are two more interesting observations to note about the energy use tables. First, for the VAV systems, the energy advantage of automatic flow control over the ASHRAE constant flow is substantially greater than it was for the CAV system. This is explored further in Section 5.3.3. Also, the VAV systems are all shown to use substantially less energy overall The second observation that can be made is that the automatic flow control systems use about the same amount of energy as the constant typical outside air flow scenario. This implies that the automatic systems could be installed to provide control of pollutants with little or no energy penalty as compared to the typical office building outside air flow rate which does not provide adequate pollutant control.

Figures 5.1 through 5.4 show graphically the coil plus reheat energy use for the 28 base scenario simulations. The percentages listed on the graphs are the energy savings as compared to ASHRAE.

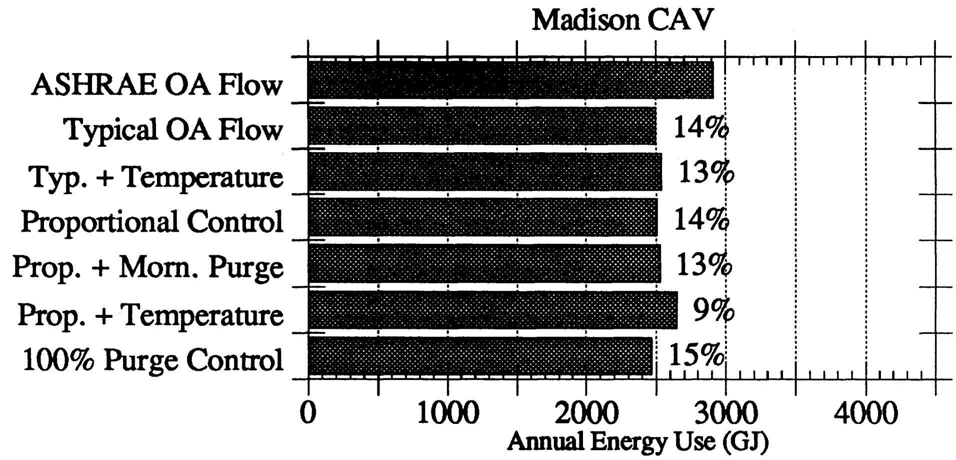

Figure 5.1 *Estimated Energy Use By Various CAV Systems in Madison*

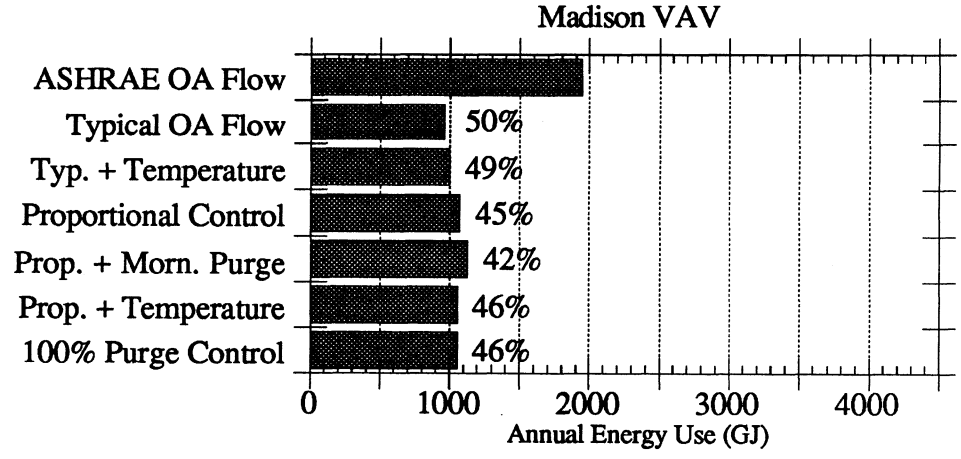

Figure 5.2 *Estimated Energy Use By Various VAV Systems in Madison*

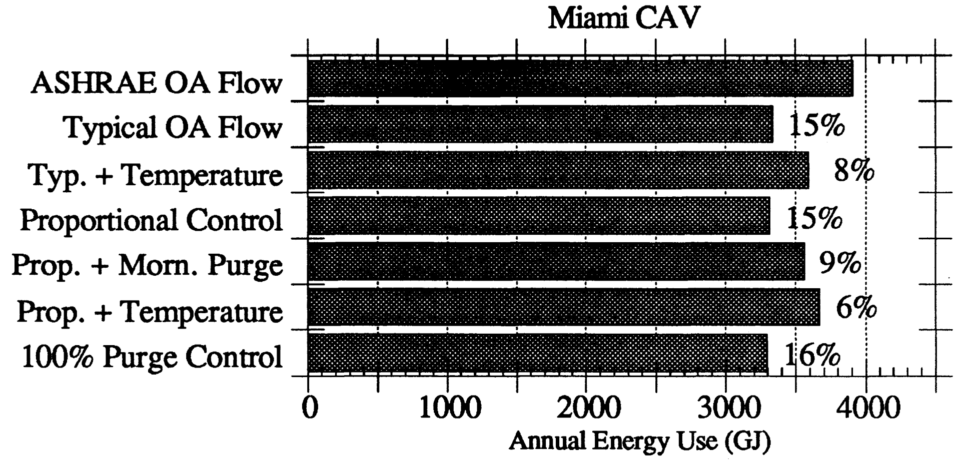

Figure 5.3 *Estimated Energy Use By Various CAV Systems in Miami*

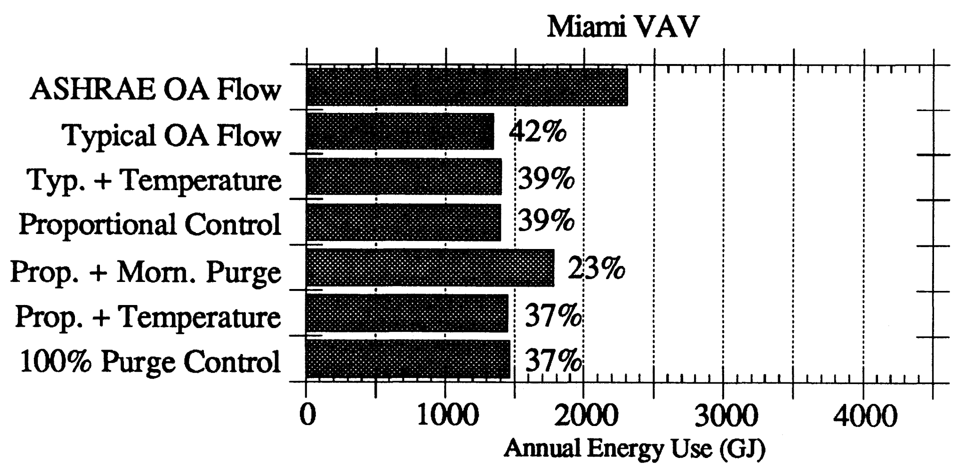

Figure 5.4 *Estimated Energy Use By Various VAV Systems in Miami*

5.3.2 Pollutant Removal Ability Compared to ASHRAE

The energy saving potential of the automatic flow control systems is apparent from the above comparisons. In this section, the ability of the various outside air flow schemes to remove pollutants will be compared.

Table 5.3 lists the average outside air flow rates in air changes per hour and in liters per second per person for the Madison simulations. Again, the nominal flow rate of outside air per person recommended by ASHRAE is 10 l/s.

Table 5.3 *Comparison of Average Outside Air Flow Rates for Madison*

	CAV System		VAV System	
Control Scheme	Average Outside Air Flow Rate (ach)	Per Person Outside Air Flow Rate (liters/s)	Average Outside Air Flow Rate (ach)	Per Person Outside Air Flow Rate (liters/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 + Morning Purge	0.73	6.3	1.01	8.8
Proportional + Temperature	1.92	16.6	1.07	9.3
100% Purge Control	0.54	4.7	0.83	7.2

The typical and automatic flow scenarios use only one-half to one-third as much outside air as the ASHRAE scenario. The two exceptions are the temperature-based flow control scenarios for the CAV system, where the outside air flow rates are only one-third less than ASHRAE. Since the energy advantage of these two scenarios is still present, the energy saving ability of a temperature-based economizer is apparent.

Figures 5.5 through 5.7 compare the concentration of carbon dioxide in the meeting room that results from the ASHRAE outside air flow rate to that resulting from the typical flow, proportional flow control, and 100% purge control, respectively. The data is from the 10th of July for the Miami VAV system simulations.

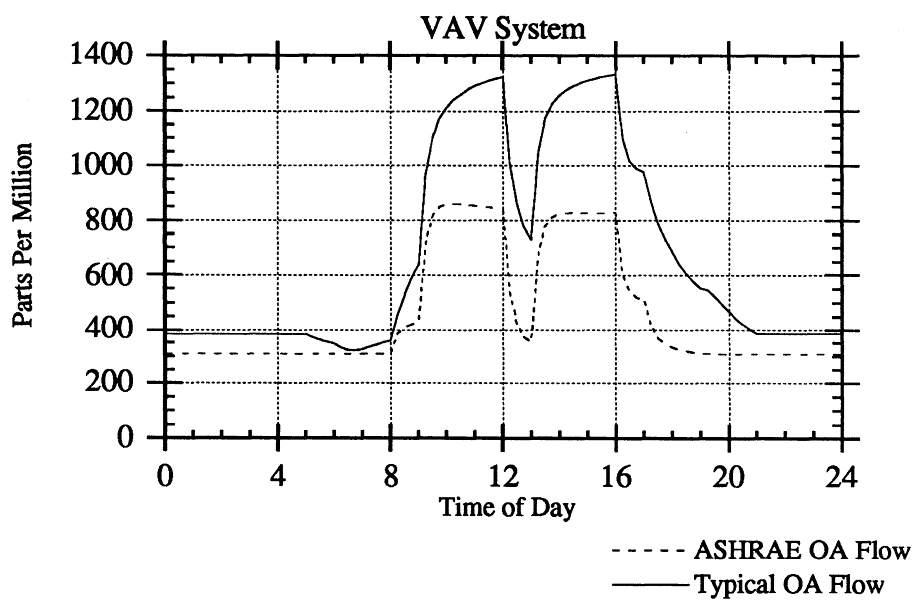

Figure 5.5 *Comparison of Carbon Dioxide Concentration Levels Versus Time of Day For ASHRAE and Typical Outside Air Flow Rates*

The relative inadequacy of the typical outside air flow rate to control pollutants is apparent form Figure 5.5. The two automatic systems control the pollutant concentration to levels comparable to those of the ASHRAE method during the hours that the building is occupied.

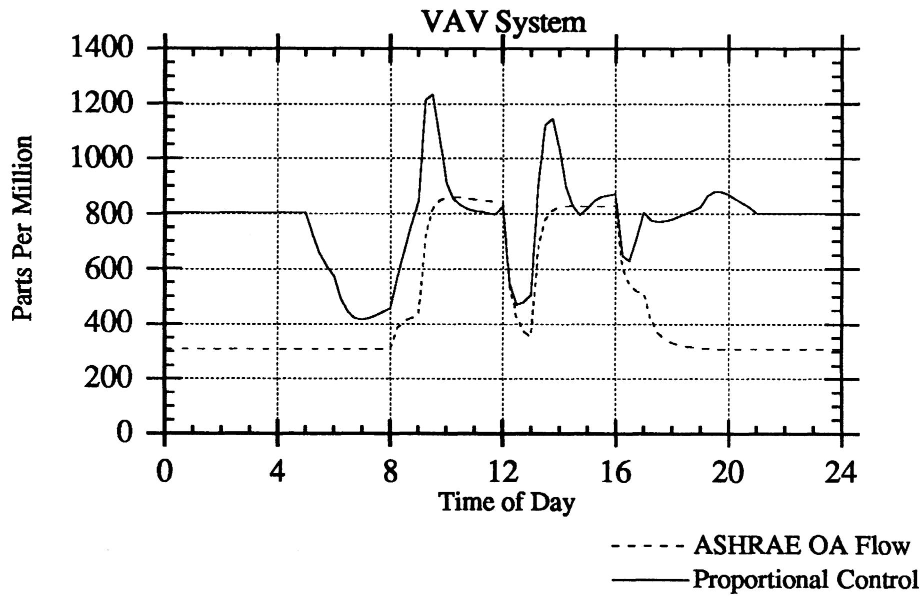

Figure 5.6 *Comparison of Carbon Dioxide Concentration Levels Versus Time of Day For ASHRAE Outside Air Flow Rate and Proportional Control*

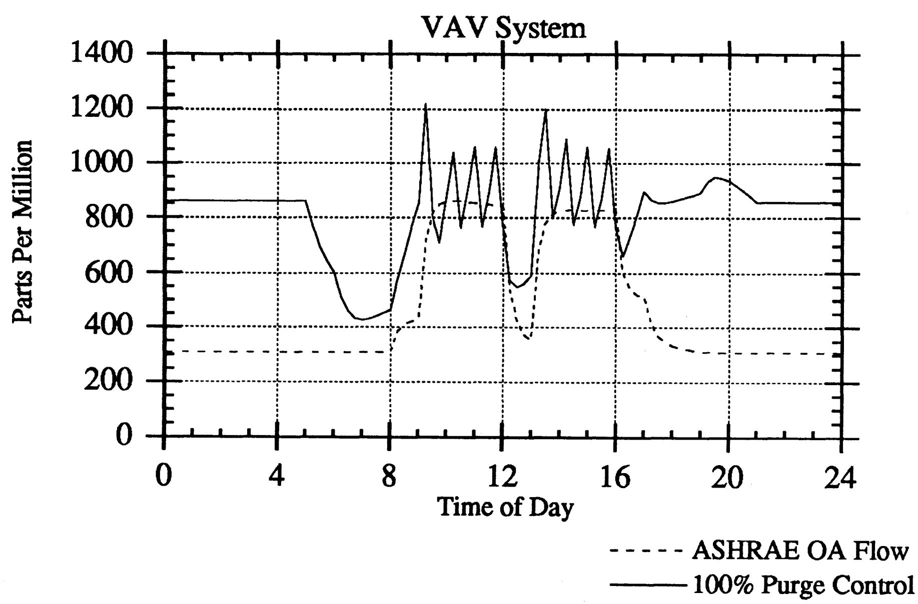

Figure 5.7 *Comparison of Carbon Dioxide Concentration Levels Versus Time of Day For ASHRAE Outside Air Flow Rate and Purge Control*

In Figures 5.6 and 5.7, the dip in the concentration levels for the two automatic control systems at 5:00 am 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 proportional control 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 made this rapid increase difficult to control properly. A real control system should be able to react more rapidly to provide better control in this situation. The effect of a shorter simulation time step is discussed in section 5.3.8.

5.3.3 Relative Pollutant Removal Ability of CAV and VAV Systems

The next six figures are pollutant concentration histogram summaries for the ASHRAE, proportional control, and purge control simulations using VAV and CAV systems. All plots shown are for Madison. The histograms are constructed by adding up the time during the year that the CO_2 concentration fell into one of the 50 ppm concentration intervals. Only the times between the hours of 8:00 am and 5:00 pm on weekdays were counted. This was done because the concentration during unoccupied times is of little interest and would only confound the important information. There is some interesting information to note in these graphs. First, the pollutant concentrations in the office zone are less than in the meeting room zone. This is because of the lower occupant density in the office area. Second, the overall concentrations in both zones are less for the CAV systems than they are for the VAV systems. Third, in the VAV

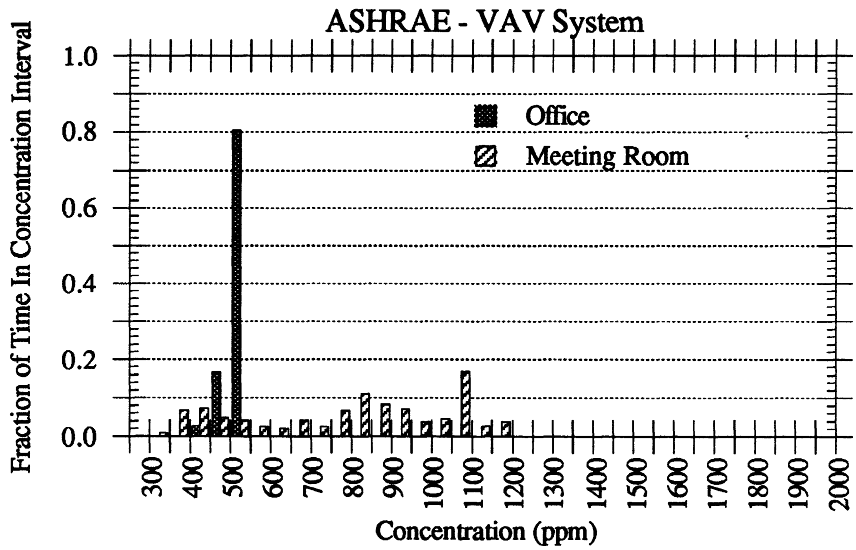

Figure 5.8 *Year Summary Histogram of Carbon Dioxide Concentration For a VAV System and ASHRAE Outside Air Flow Rate*

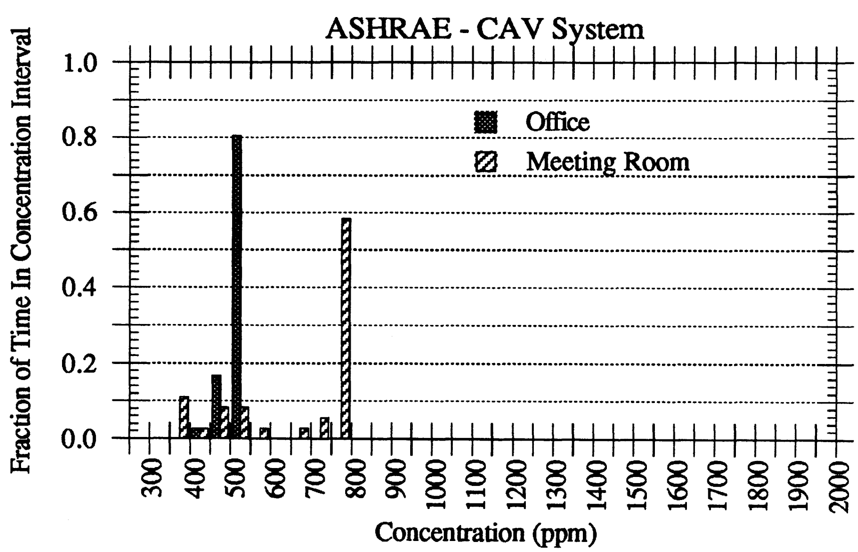

Figure 5.9 *Year Summary Histogram of Carbon Dioxide Concentration For a CAV System and ASHRAE Outside Air Flow Rate*

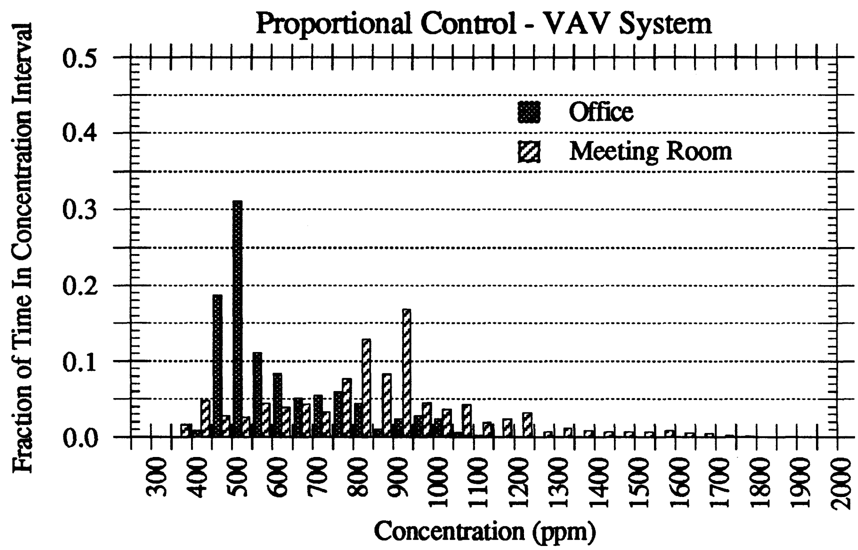

Figure 5.10 *Year Summary Histogram of Carbon Dioxide Concentration For a VAV System and Proportional Control*

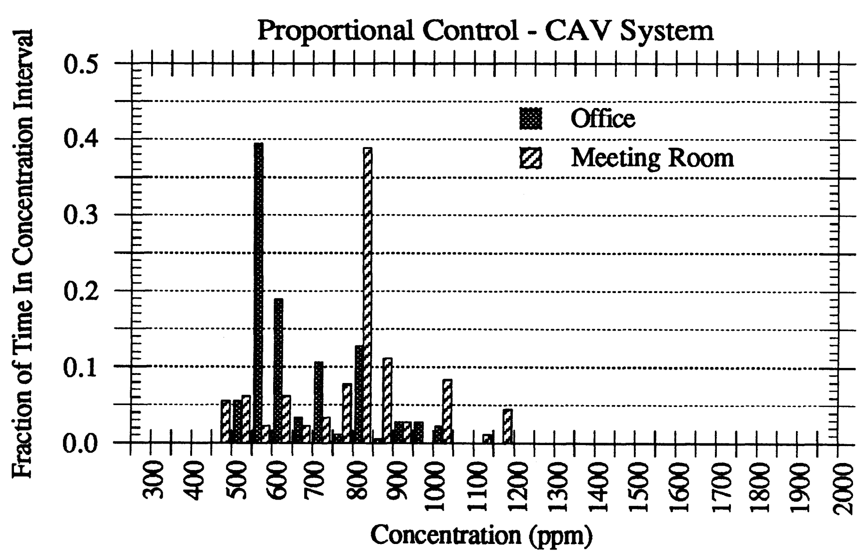

Figure 5.11 *Year Summary Histogram of Carbon Dioxide Concentration For a CAV System and Proportional Control*

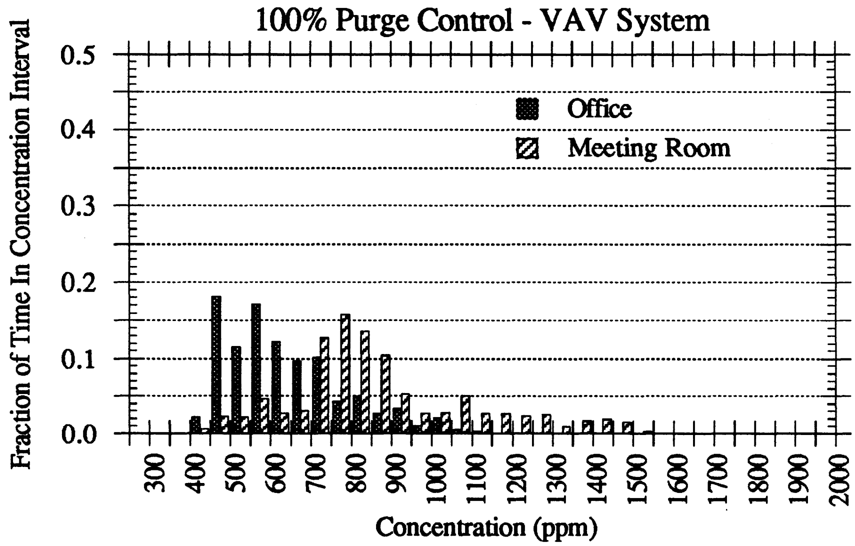

Figure 5.12 *Year Summary Histogram of Carbon Dioxide Concentration For a VAV System and Purge Control*

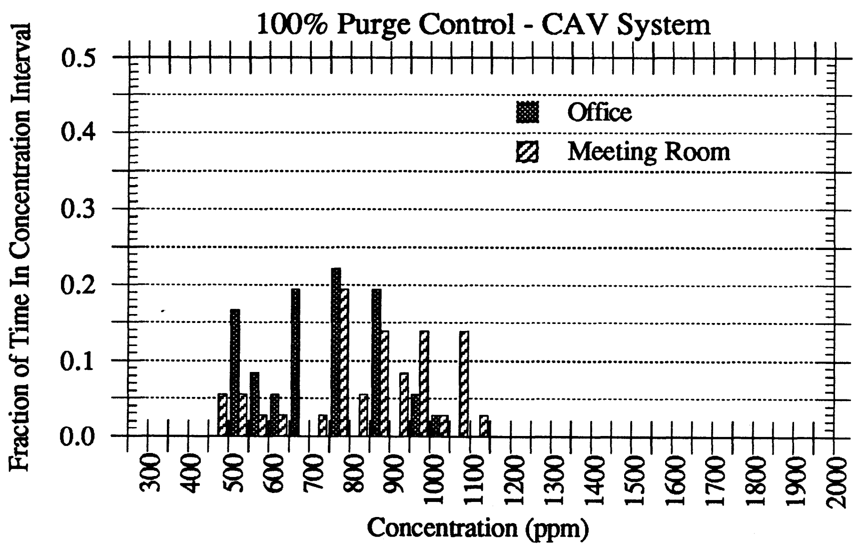

Figure 5.12 *Year Summary Histogram of Carbon Dioxide Concentration For a CAV System and Purge Control*

system plots, the concentrations in the meeting room are generally higher than they are in the office zone. In the CAV system plots, the concentrations in the two zones are much closer. It appears that the lower circulation air flow 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. Lastly, with a VAV system, the ASHRAE constant outside air flow rate still allows concentrations to reach above the 1000 ppm maximum that the air flow rate was designed to achieve.

5.3.4 Proportional Control versus Purge Control

In this section, two automatic control schemes are compared for their ability to remove pollutants. Figures 5.14 and 5.15 show pollutant concentration in the meeting room and outside air flow rate versus time of day for proportional control and purge control. From these graphs, it appears that the purge control may result in less time above the 1000 ppm high concentration limit setpoint. However, when the complete concentration histogram is plotted in Figures 5.16 and 5.17, the purge control system does not seem to provide any real advantage in removing pollutants. Also, from Figures 5.1 to 5.4, the energy consumption of the two methods are very close. Therefore, neither method appears to provide any real advantage over the other.

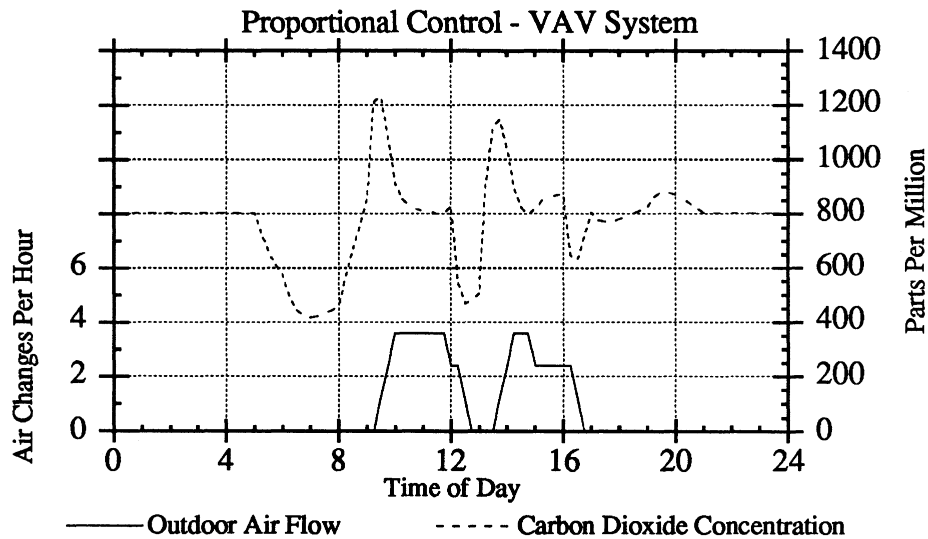

Figure 5.14 *Carbon Dioxide Concentration Levels and Outside Air Flow Versus Time of Day For Proportional Control*

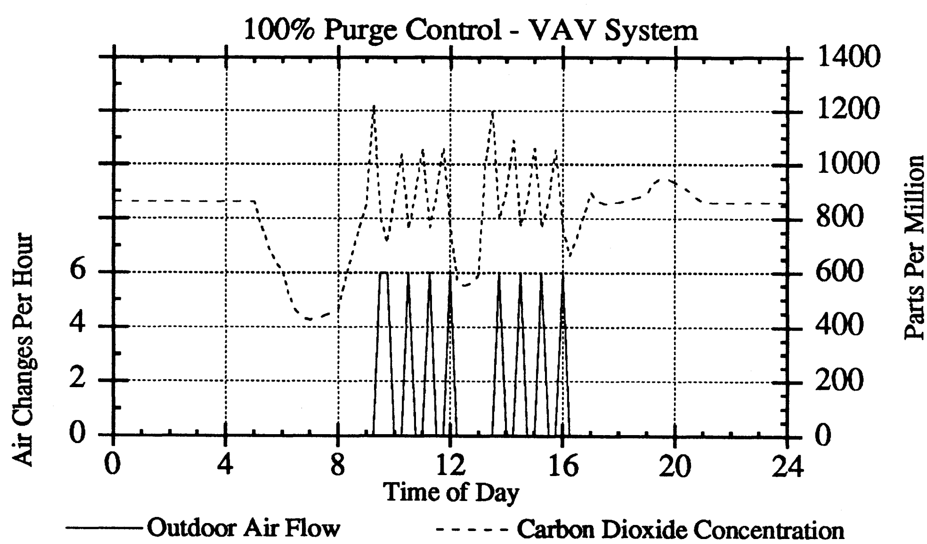

Figure 5.15 *Carbon Dioxide Concentration Levels and Outside Air Flow Versus Time of Day For Purge Control*

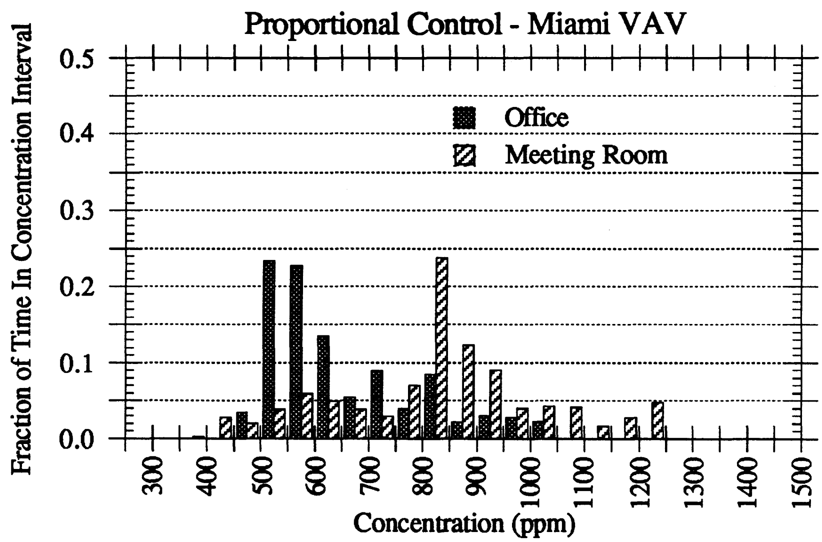

Figure 5.16 *Year Summary Histogram of Carbon Dioxide Concentration For a VAV System with Proportional Control*

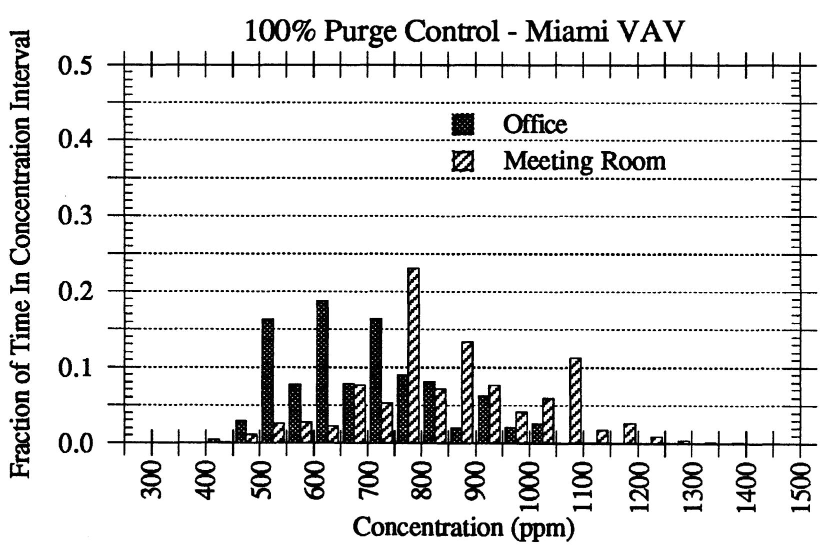

Figure 5.17 *Year Summary Histogram of Carbon Dioxide Concentration For a VAV System with Purge Control*

5.3.5 Pollutant Removal with a Temperature-Based Economizer

The two temperature-based economizer scenarios, typical outside air flow plus temperature-based control and proportional plus temperature-based control, both used approximately the same amount of energy. The difference in pollutant control, however, was quite different between the two methods, as is shown in Figure 5.18.

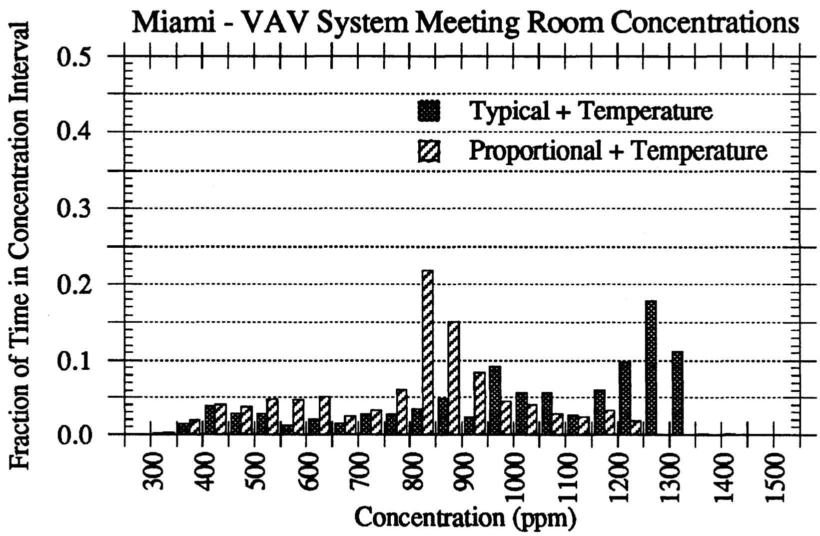

Figure 5.18 *Comparison of Year Summary Histograms of Carbon Dioxide Concentration for Typical Plus Temperature Control and Proportional Plus Temperature Control*

5.3.6 Changing the Value of Pollutant Removal Effectiveness, ε_c

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. The pollutant removal advantages of increasing the value of ε_c in a zone are quite obvious. 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 1/3 in both directions.

The result was somewhat surprising. Increasing the removal effectiveness resulted in an increase in energy use by about 5%, and decreasing it lowered the energy use by about the same amount. This is shown in Figure 5.19. For an automatic control system, this is reasonable. With a higher removal effectiveness, the sensor in the return air duct detects a higher concentration of pollutant and therefore calls for more outside air flow than what is necessary to control the room concentration. The opposite

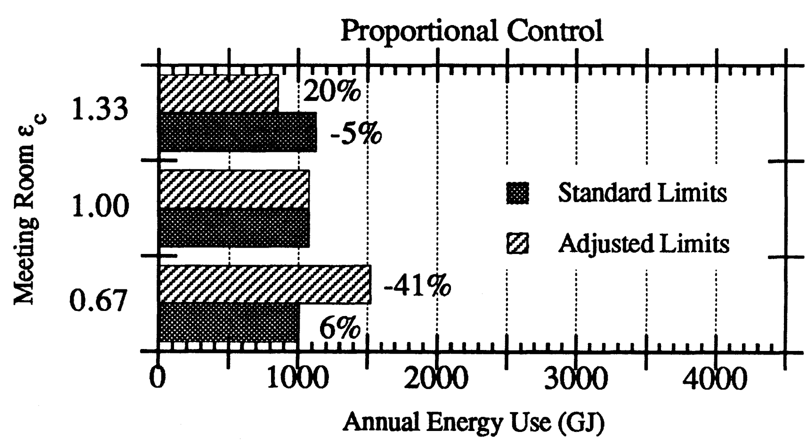

Figure 5.19 *Effect of Changing Pollutant Removal Effectiveness on Annual Energy Use of Proportional Control*

is true for a lower removal effectiveness. This results in over control when ε_c is raised and under control when ε_c is lowered, as can be seen in Figure 5.20.

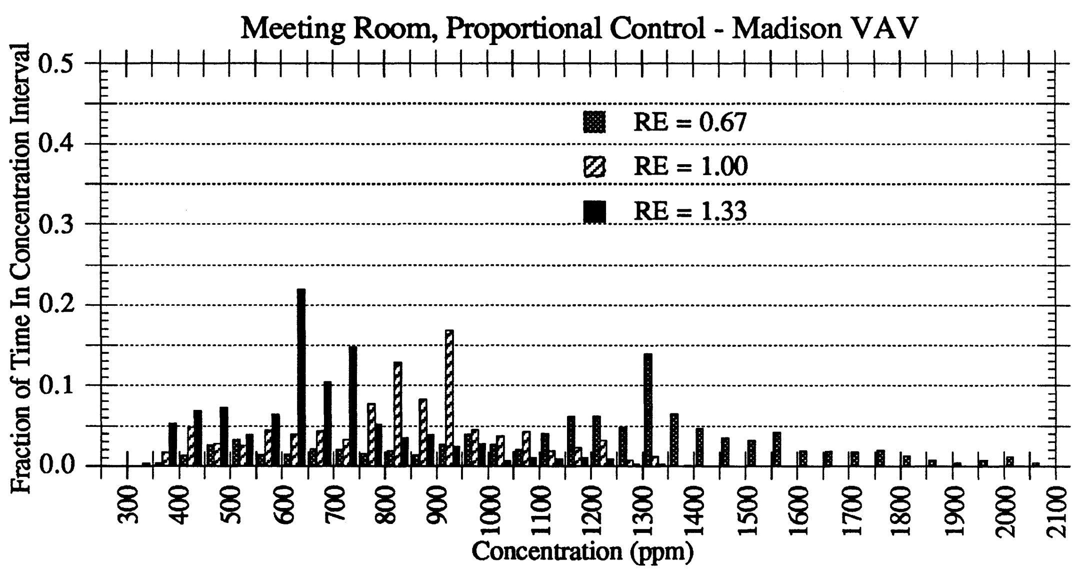

Figure 5.20 *Effect of Changing Pollutant Removal Effectiveness on Concentration Level Histogram*

Now, if the control setpoints are adjusted 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 outside air flow 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. Figure 5.21 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. The lesson to be learned here is that, 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.

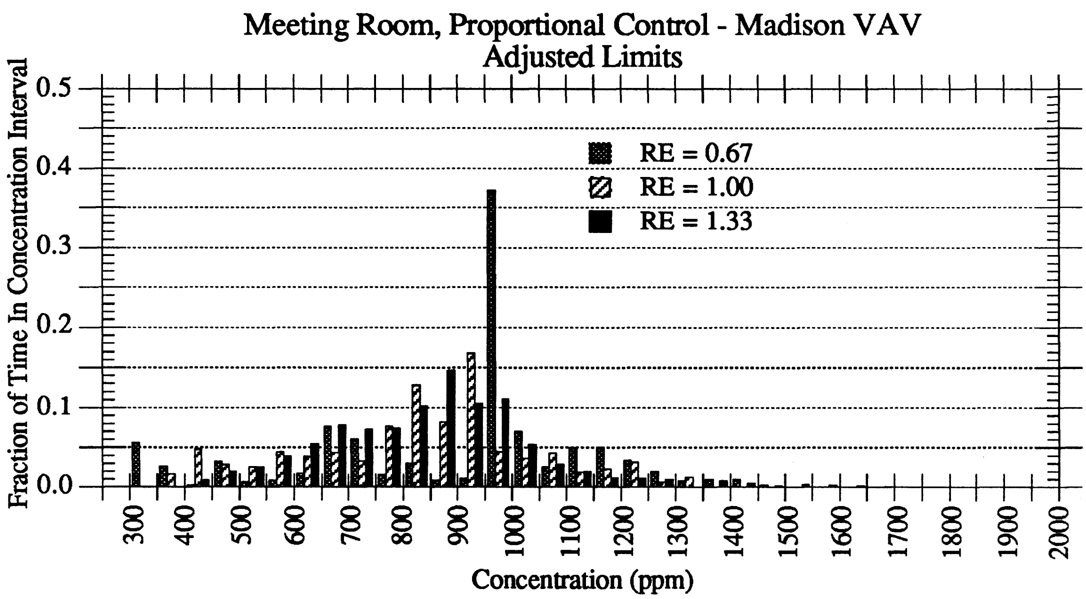

Figure 5.21 *Effect of Changing Pollutant Removal Effectiveness on Concentration Level Histogram with Adjusted Concentration Limits*

5.3.7 Adding a Minimum Flow of Outside Air

In Section 5.2.3, it was mentioned that a real CO_2-based automatic outside air flow control system would provide a minimum flow of outside 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 minimum outside air flow rate. The results are shown in Figures 5.22 and 5.23.

For a VAV system in Madison and Miami, the savings over the ASHRAE fixed air flow method of pollutant control were still 22% and 27% for the two cities, respectively. This is, however, substantially less than the savings of proportional control without a base air flow.

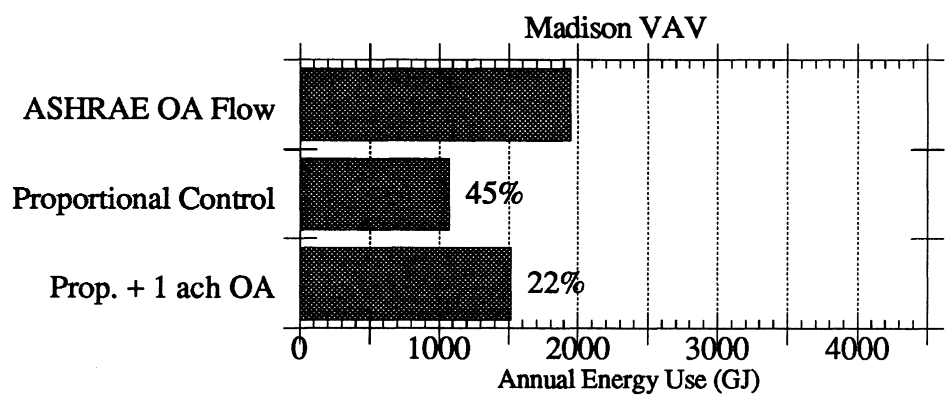

Figure 5.22 *Effect on Annual Energy Use in Madison of Adding Base Outside Air Flow*

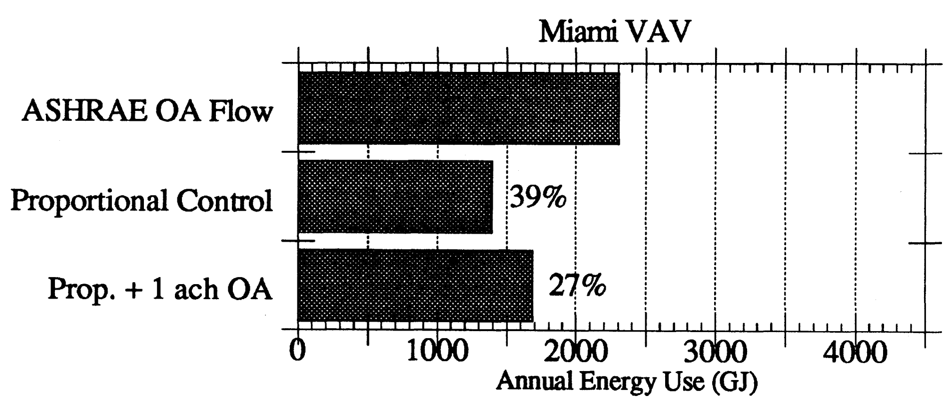

Figure 5.23 *Effect on Annual Energy Use in Miami of Adding Base Outside Air Flow*

5.3.8 Effect of a Shorter Simulation Time Step

The time step used for all of the simulations discussed up to this point was 15 minutes. This means that if the concentration in a zone was just under the high limit setpoint at the end of a time step, it would be another 15 minutes before the controller could increase the flow of outside air to remedy the situation, during which time the concentration in the zone would continue to rise. This effect can be observed in the plots of concentration versus time of day for the automatic controllers (Figure 5.14, for example). To test the idea that a shorter time step might result in better pollutant control, two simulations were run with a time step of 3.75 minutes. Figure 5.24 is the resulting concentration histogram for proportional control. It does appear that using a smaller time step would result in substantially fewer spikes above the high limit setpoint and would better simulate a realistic response time for the controllers.

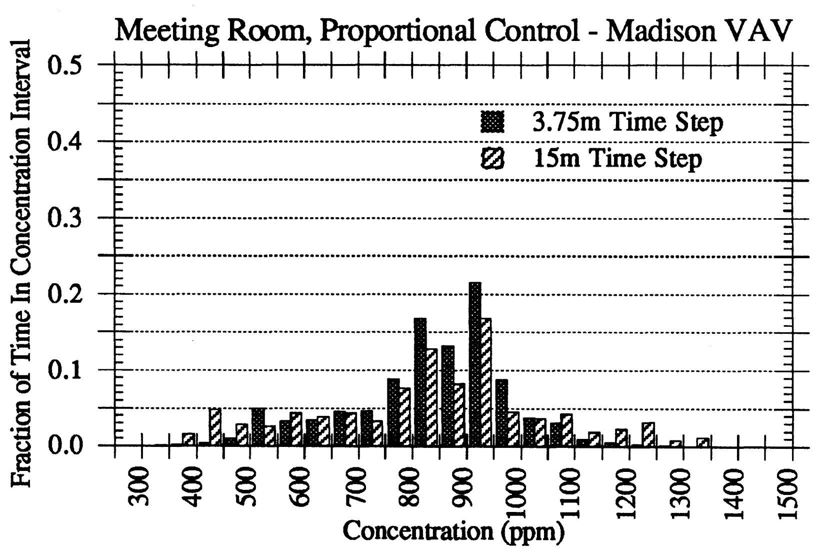

Figure 5.24 *Effect of Changing the Simulation Time Step*

The penalty for using a shorter simulation time step is that the time to complete a simulation would be increased proportionally. With a 15 minute time step, the run time for a year simulation was 1-1/2 hours. The 3.75 minute time step resulted in 6 hour runs.

5.4 Summary of Results

Some general conclusions can be drawn from the information presented in the previous section.

Four systems for automatic control of outside air based on CO_2 concentration were examined for this project. All of these systems can provide equivalent control of the occupant-generated CO_2 concentration to that afforded by the fixed outside air flow rate recommended in ASHRAE Standard 62-1989 for indoor air quality. The heating and cooling energy saved from using an automatic control for outside air flow ranges from 10% in a CAV system to as much as 50% for a VAV system. When comparing energy use, however, it must be remembered that in this model, the constant outside air flow 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 are CAV systems due to their lower circulation air flow rates. Even the ASHRAE recommended fixed outside air flow rate does not keep occupant-generated CO_2 below the 1000 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.

5.5 References

1. *TRNSYS, A Transient Simulation Program*, Version 12.2 Users Manual, Solar Energy Laboratory, University of Wisconsin - Madison, 1988.

2. Acme Engineering Products, Incorporated, product bulletin "Integrated Control Systems Based on Carbon Dioxide Concentration", Moorers, New York, 1982.

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Chapter 6

CONCLUSIONS AND RECOMMENDATIONS FOR FURTHER RESEARCH

This project was intended to be a broad overview of topics relating to the dispersion and dilution of pollutants in the indoor environment. In many respects, only the surface of this topic has been breached, and much remains to be accomplished. The use of computer simulations to explore various alternative strategies for control of indoor pollutants is an area where there is great potential for original contribution. In this chapter, a summary of the major conclusions based on the work performed for this project is presented, and some recommendations for further work that could be done to extend this project are given.

6.1 Room Air Flow Modeling

Three-dimensional room air flow computer modeling is difficult to accomplish with any degree of accuracy. Good finite-difference and finite-element computer codes are available, but a fast computer is needed for a reasonable run time with even relatively simple cases. A realistic computer model of room air flow and pollutant distribution should include the effects of air and surface temperatures and buoyancy. The ventilation air inlet diffuser should be modeled with several nodes across its face so that different flow directions at the inlet can be included.

The pollutant removal effectiveness, ε_c, is defined as the room exhaust duct concentration divided by the room average concentration. It can be a useful tool in describing the interaction of the room ventilation air flow and the pollutant source, especially when used with a multiple-zone pollutant transport model.

It is intuitive that the arrangement of the ventilation inlet and outlet ducts and the source location will affect the value of ε_c. From the parametric study performed for this project, the value of ε_c is not significantly affected by changes in ventilation air flow rate or room size for the top-down or bottom-up displacement flow arrangements. For the mixed flow arrangement, an increase in room size will significantly increase the value of ε_c, while an increase in ventilation air flow rate will significantly decrease the value of ε_c. Pollutant source strength did not affect the value of ε_c in any of the three ventilation system flow arrangements.

Further studies of parameters that affect pollutant removal effectiveness could be carried out using a more accurate room air flow model. An attempt could be made to quantify the effect that a change in a parameter value has on the value of ε_c.

6.2 Multiple-Zone Pollutant Transport

In Chapter 5, an office zone and a meeting room zone joined by a common HVAC system were modeled using the TRNSYS program. Several conclusions were reached regarding automatic control of the outside air flow rate based on CO_2 concentration.

An automatic control system can provide the same degree of pollutant control as the fixed outside air flow rate recommended in the current ASHRAE indoor air quality standard. Heating and cooling energy use for the automatic systems is about the same as for the "typical" outside air flow rate scenarios. Energy savings for the automatic systems as compared to the ASHRAE outside air flow rate range from about 10% for CAV systems to as much as 50% for VAV systems. A VAV system is less efficient at pollutant dilution than a CAV system under the same conditions due to the lower circulation air flow rate for the VAV system.

Work is in progress under the auspices of the International Energy Agency to compile a database of experimental data sets for the validation of multiple-zone pollutant transport models. When this data becomes available, the Type 60 pollutant transport model could be validated against it. The importance of pollutant adsorption and re-emission by sink materials in the indoor environment was not addressed in this project. A study of this topic could lead to a sink model being incorporated into Type 60.

Finally, the automatic outside air flow control methods used in Type 65 were chosen somewhat arbitrarily. Considerable work has been done recently on the optimization of HVAC control systems. The same optimization ideas could be applied to setpoints and methods for automatic control of the outside air flow to improve their performance.

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Appendix A
TYPE 60 FORTRAN LISTING

```
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C  TYPE 60 - MULTIPLE-ZONE INDOOR CONTAMINANT TRANSPORT
C
C  CALCULATES THE CONCENTRATION OF A CONTAMINANT IN A MULTIPLE-ZONE
C  VENTILATED VOLUME AS A FUNCTION OF SIMULATION TIME
C
C  CREATED BY:
C      PAUL KNOESPEL
C
C  CREATED ON:
C      MAR  2, 1990
C
C  MODIFIED ON / BY / REASON:
C
C
C  SUBROUTINE TYPE60 CALLS:
C      SUBROUTINE SHORTCALC
C      SUBROUTINE CALC
C
C  VARIABLE LISTING
C
C  I - INPUT VALUE
C  C - CALCULATED VALUE
C
C      A1           C   ZONE SOURCES INDEPENDENT OF ZONE CONCENTRATION
C      A2           C   ZONE SOURCES DEPENDENT ON ZONE CONCENTRATION
C      A3           C   ZONE SINKS DEPENDENT ON ROOM RETURN DUCT CONCENTRATION
C      C            C   ZONE AVERAGE CONTAMINANT VOLUME CONCENTRATION
C      C1           C   ITERATIVE CONCENTRATION VALUE AT TIME STEP I+1
C      CAHU         C   COMBINED RETURN CONTAMINANT CONCENTRATION AT THE AHU
C      COA          I   CONTAMINANT CONCENTRATION IN OUTSIDE AIR
C      CRET         C   RETURN DUCT CONTAMINANT CONCENTRATION AT THE ZONE
C      CSUP         C   SUPPLY DUCT CONTAMINANT CONCENTRATION
C      DELTAT       I   TRNSYS SIMULATION TIME STEP
C      DT           I   TIME STEP FOR EACH CALCULATION (S)
C      DTDT         C   TRNSYS VARIABLE - NOT USED
C      ERR          C   ERROR TOLERANCE FOR CHANGE BETWEEN ITERATION STEPS
C      ERRFLAG      C   ERROR FLAG SET WHEN ERROR TOLERANCE IS NOT MET
C      F            C   FUNCTION FOR RHS OF CONCENTRATION DIFFERENTIAL EQUATION
C      FILTEFF      I   FRACTIONAL FILTER EFFICIENCY
C      HOD          C   CURRENT HOUR OF THE DAY BASED ON SIMULATION TIME
C      HOW          C   CURRENT HOUR OF THE WEEK BASED ON SIMULATION TIME
C      I            C   TIME STEP INDEX
C      IDELAY       C   NUMBER OF TIME STEPS FOR AIR TO FLOW THROUGH RETURN PATH
C      INFO         C   TRNSYS TYPE INFORMATION
C      IZF          I   INTER-ZONE FLOW (KG/HR, LBM/HR)
C      J            C   ZONE NUMBER INDEX
C      K            C   SOURCE SCHEDULE INDEX
C      L            C   ZONE NUMBER INDEX IN 2D IZF(J,L) ARRAY
```

```
C       LUB           I   FORTRAN LOGICAL UNIT FOR BUILDING DATA FILE
C       LUS           I   FORTRAN LOGICAL UNIT FOR SOURCE DATA FILE
C       ND            C   NUMBER OF DERIVATIVES
C       NI            C   NUMBER OF INPUTS
C       NP            C   NUMBER OF PARAMETERS
C       NSTEPS        C   NUMBER OF TIME STEPS
C       NZONES        I   NUMBER OF ZONES
C       OUT           C   OUTPUT ARRAY TO TRNSYS
C       PAR           I   CONSTANT INPUT ARRAY FROM TRNSYS
C       QCIZF         C   TOTAL INTER-ZONE CONTAMINANT FLOW INTO A ZONE (ACS)
C       QIZF          C   INTER-ZONE AIR FLOW (M3/S, FT3/S)
C       QIZFNET       C   NET INTER-ZONE FLOW FOR A ZONE (M3/S, FT3/S)
C       QZIFOUT       C   INTER-ZONE FLOW LEAVING A ZONE (M3/S, FT3/S)
C       QOA           I   OUTSIDE AIR VOLUME FLOW RATE INTO A ZONE (M3/S, FT3/S)
C       QOAPU         C   ZONE FLOW RATE OF OUTSIDE AIR PER SOURCE UNIT (L3/S-U,
C                         FT3/MIN-U)
C       QOATOT        C   TOTAL OUTSIDE AIR VOLUME FLOW RATE (M3/S, FT3/S)
C       QRET          C   VENTILATION VOLUME FLOW RATE LEAVING A ZONE (M3/S, FT3/S)
C       QRETTOT       C   TOTAL RETURN DUCT VOLUME FLOW RATE (M3/S, FT3/S)
C       QSUP          I   CIRCULATION AIR VOLUME FLOW RATE INTO A ZONE (M3/S, FT3/S)
C       QCIRC         C   TOTAL CIRCULATION AIR VOLUME FLOW RATE (M3/S, FT3/S)
C       RE            I   ZONE POLLUTANT REMOVAL EFFECTIVENESS WITH HVAC ON
C       REMEFF        C   ZONE REMOVAL EFFECTIVENESS USED IN CALCULATIONS
C       RETVOL        I   VOLUME OF THE ZONE'S RETURN AIR PATH TO THE AHU (M3, FT3)
C       RHOAIR        C   CONSTANT AIR DENSITY (KG/M3, LBM/FT3)
C       SOURCE        I   CONTAMINANT SOURCE VOLUME FLOW RATE IN A ZONE (M3/S, FT3/S)
C       SMULT         C   SOURCE MULTIPLIER FOR THE CURRENT TIME STEP
C       SMULTWD       I   WEEKDAY SCHEDULED SOURCE MULTIPLIER
C       STIMEWD       I   WEEKDAY SCHEDULE TIME THAT SOURCE MULTIPLIER IS APPLIED
C       SMULTWE       I   WEEKEND SCHEDULED SOURCE MULTIPLIER
C       STIMEWE       I   WEEKEND SCHEDULE TIME THAT SOURCE MULTIPLIER IS APPLIED
C       SUNIT         I   UNIT CONTAMINANT SOURCE VOLUME FLOW RATE (M3/S, FT3/S)
C       T             C   TRNSYS VARIABLE - NOT USED
C       TIME          C   SIMULATION TIME (HR)
C       TIMEPREV      C   TIME OF THE PREVIOUS SIMULATION STEP
C       UNITFLAG      I   FLAG TO INDICATE SI OR ENGLISH UNITS (1=SI, 2=ENGLISH)
C       XIN           I   INPUT ARRAY FROM TRNSYS
C       ZONEVOL       C   ZONE VOLUME (M3, FT3)
C

        SUBROUTINE TYPE60(TIME,XIN,OUT,T,DTDT,PAR,INFO)

        IMPLICIT NONE

        INTEGER INFO(10), NI, NP, ND, LUB, LUS
        INTEGER NSTEPS, NZONES, IDELAY(25), I, J, K, L, UNITFLAG

        REAL TIME, XIN(50), OUT(77), T, DTDT, PAR(10)
        REAL C(-60:60,25), CAHU, CSUP, COA, FILTEFF(3)
        REAL SOURCE(25), SUNIT(25), SMULT(25), RHOAIR
        REAL STIMEWD(25,6), SMULTWD(25,6), STIMEWE(25,6), SMULTWE(25,6)
        REAL ZONEVOL(25), RE(25), REMEFF(25), RETVOL(25), DT, DELTAT
        REAL QSUP(25), QRET(25), QOA(25), INFILT(25), IZF(25,25)
        REAL QCIRC, QOATOT, QRETTOT, QCRETTOT, QOAPU
        REAL QIZF(25,25), QIZFNET, QIZFOUT, TIMEPREV, HOW, HOD
        REAL F, A1(25), A2(25), A3(25)

        COMMON A1, A2, A3
```

```
C
C  SET PARAMETERS, DO CHECKS, SET ALL INITIAL ZONE CONCENTRATIONS EQUAL
C  TO THE OUTSIDE AIR CONCENTRATION, AND READ IN SOURCE AND ZONE INFORMATION
C  ON THE FIRST CALL OF THE SIMULATION
C

      IF(INFO(7).EQ.-1)THEN
        NZONES = NINT(PAR(1))
        DT = PAR(2)
        DELTAT = PAR(3)
        COA = PAR(4)
        LUS = NINT(PAR(5))
        LUB = NINT(PAR(6))
        FILTEFF(1) = PAR(7)
        FILTEFF(2) = PAR(8)
        FILTEFF(3) = PAR(9)
        UNITFLAG = NINT(PAR(10))

        NSTEPS = NINT(DELTAT*3600.0/DT)
        TIMEPREV = -1.0

C
C  USE ENGLISH UNITS IF THE UNITS FLAG IS SET; OTHERWISE USE SI UNITS
C

        IF (UNITFLAG .EQ. 2) THEN
          RHOAIR = 0.07517
        ELSE
          RHOAIR = 1.204
        ENDIF

C
C  STOP THE SIMULATION IF NSTEPS IS GREATER THAN 60
C

        IF ((NSTEPS .LT. 1) .OR. (NSTEPS .GT. 60)) THEN
          WRITE (6,*) 'THE TRNSYS SIMULATION TIME STEP PAR(3) IN'
          WRITE (6,*) 'TYPE 60 MUST BE BETWEEN 1 AND 60 TIMES THE'
          WRITE (6,*) 'TYPE 60 TIME STEP PAR(2)'
          WRITE (6,*) '-- SIMULATION STOPPED --'
          STOP
        ENDIF

        NI = 2*NZONES
        NP = 10
        ND = 0
        INFO(6) = 3*NZONES+2
        INFO(9) = 1
        CALL TYPECK(1,INFO,NI,NP,ND)

        DO 5 J = 1, NZONES
          DO 5 I = -60, 60
            C(I,J) = COA
5       CONTINUE
```

```
C
C  READ INPUT VALUES FOR THE SOURCE AND ZONE PHYSICAL DATA
C

         OPEN (LUS, STATUS = 'OLD')
         REWIND (LUS)
         DO 10 J = 1, NZONES
           READ (LUS,*) SUNIT(J)
           READ (LUS,*) (STIMEWD(J,K), K = 1, 6)
           READ (LUS,*) (SMULTWD(J,K), K = 1, 6)
           READ (LUS,*) (STIMEWE(J,K), K = 1, 6)
           READ (LUS,*) (SMULTWE(J,K), K = 1, 6)
10       CONTINUE
         CLOSE (LUS)

         OPEN (LUB, STATUS = 'OLD')
         REWIND (LUB)
         READ (LUB,*) (ZONEVOL(J), J = 1, NZONES)
         READ (LUB,*) (RE(J), J = 1, NZONES)
         READ (LUB,*) (RETVOL(J), J = 1, NZONES)
         READ (LUB,*) (INFILT(J), J = 1, NZONES)

         DO 20 J = 1, NZONES
         READ (LUB,*) (IZF(J,L), L = 1, NZONES)
           DO 15 L = 1, NZONES
             QIZF(J,L) = IZF(J,L) / (3600.0 * RHOAIR)
15         CONTINUE
20       CONTINUE
         CLOSE (LUB)

         ENDIF

C
C  IF THE TIME HAS NOT CHANGED SINCE THE PREVIOUS SUBROUTINE CALL, RETURN TO
C  TRNSYS TO PREVENT CHANGING THE CONCENTRATIONS TWICE DURING A SINGLE TIME STEP
C

       IF (TIME .EQ. TIMEPREV) THEN
         RETURN
       ELSE
         TIMEPREV = TIME
       ENDIF

C
C  FIND THE CURRENT SOURCE VALUE BASED ON THE SOURCE MULTIPLIER APPROPRIATE
C  FOR THE CURRENT SIMULATION TIME; IF THERE IS NO SOURCE IN A ZONE, SET THE
C  ZONE REMOVAL EFFECTIVENESS TO 1.0
C

       HOW = AMOD(TIME,168.0)
       HOD = AMOD(HOW,24.0)
```

```
      DO 25 J = 1, NZONES

        IF (HOW .LE. 120.0) THEN
          SMULT(J) = SMULTWD(J,6)
          DO 22 K = 1, 5
            IF ((HOD .GT. STIMEWD(J,K)) . AND.
     .      (HOD .LE. STIMEWD(J,K+1))) THEN
              SMULT(J) = SMULTWD(J,K)
              GOTO 24
            ENDIF
22        CONTINUE

        ELSE
          SMULT(J) = SMULTWE(J,6)
          DO 23 K = 1, 5
            IF ((HOD .GT. STIMEWE(J,K)) .AND.
     .      (HOD .LE. STIMEWE(J,K+1))) THEN
              SMULT(J) = SMULTWE(J,K)
              GOTO 24
            ENDIF
23        CONTINUE

        ENDIF

24      SOURCE(J) = SUNIT(J)*SMULT(J)
        IF (SOURCE(J) .EQ. 0.0) THEN
          REMEFF(J) = 1.0
        ELSE
          REMEFF(J) = RE(J)
        ENDIF

25    CONTINUE

      QCIRC = 0.0
      QOATOT = 0.0

      DO 30 J = 1, NZONES

C
C  READ INPUT DATA; AIR FLOWS ARE CONVERTED FROM AIR CHANGES PER HOUR
C  TO M3/S (OR FT3/S)
C

        QSUP(J) = XIN(J)*ZONEVOL(J) / 3600.0
        QCIRC = QCIRC + QSUP(J)
        QOA(J) = XIN(J+NZONES)*ZONEVOL(J) / 3600.0
        QOATOT = QOATOT + QOA(J)

C
C  CALCULATE THE RETURN AIR TIME STEP DELAY
C

        IF (QSUP(J) .EQ. 0.0) THEN
          IDELAY(J) = 60
        ELSE
          IDELAY(J) = NINT( RETVOL(J) / (QSUP(J)*DT) )
        ENDIF
```

```
          IF (IDELAY(J) .GT. 60) IDELAY(J) = 60

30      CONTINUE

        QRETTOT = 0.0

        DO 35 J = 1, NZONES

C
C   CALCULATE THE RETURN AIR FLOW BY A FLOW BALANCE; IF THE VENTILATION
C   SYSTEM IS OFF, INFILTRATION CONTINUES BUT INTER-ZONE FLOW IS ASSUMED
C   TO BE ZERO
C

          QIZFNET = 0.0
          QIZFOUT = 0.0

          IF (QCIRC .EQ. 0.0) THEN
            QRET(J) = INFILT(J)*ZONEVOL(J) / 3600.0
          ELSE
            DO 33 L = 1, NZONES
              QIZFNET = QIZFNET + QIZF(L,J) - QIZF(J,L)
              QIZFOUT = QIZFOUT + QIZF(J,L)
33          CONTINUE
            QRET(J) = QSUP(J) + QIZFNET + INFILT(J)*ZONEVOL(J) / 3600.0
          ENDIF

          QRETTOT = QRETTOT + QRET(J)

C
C   CALCULATE CONSTANTS FOR THE dC/dT FUNCTION F
C

          A1(J) = SOURCE(J) / ZONEVOL(J) + ( INFILT(J) * COA ) / 3600.0
          A2(J) = QSUP(J) / ZONEVOL(J)

          IF (QCIRC .EQ. 0.0) THEN
            A3(J) = QRET(J) * REMEFF(J) / ZONEVOL(J)
          ELSE
            A3(J) = ( QRET(J) * REMEFF(J) + QIZFOUT ) / ZONEVOL(J)
          ENDIF

35      CONTINUE

C
C   CALL THE MAIN CALCULATION SUBROUTINE; IF THE VENTILATION SYSTEM IS OFF,
C   A SHORTENED CALCULATION WITHOUT RECIRCULATION IS PERFORMED
C

        IF (QCIRC .EQ. 0.0) THEN
          CALL SHORTCALC (C,COA,DT,NSTEPS,NZONES)
        ELSE
          CALL CALC (C,COA,DT,IDELAY,QRET,QRETTOT,QCIRC,QOATOT,
     .               QIZF,REMEFF,NSTEPS,NZONES,ZONEVOL,FILTEFF)
        ENDIF
```

```
C
C  CALCULATE VALUES FOR OUTPUT; AIR FLOWS ARE CONVERTED BACK TO AIR
C  CHANGES PER HOUR
C
      QCRETTOT = 0.0

      DO 40 J = 1, NZONES

C
C  CALCULATE THE VOLUME FLOW RATE OF OUTSIDE AIR PER SOURCE UNIT;
C  ENGLISH UNITS IN FT3/MIN, SI UNITS IN L/S
C
        IF (SMULT(J) .LE. 0.0) THEN
          QOAPU = 0.0
        ELSEIF (UNITFLAG .EQ. 2) THEN
          QOAPU = QOA(J)*60.0/SMULT(J)
        ELSE
          QOAPU = QOA(J)*1000.0/SMULT(J)
        ENDIF

        QCRETTOT = QCRETTOT + QRET(J)*C(-IDELAY(J),J)*REMEFF(J)

        OUT(J) = QOAPU
        OUT(NZONES+J) = C(0,J)
        OUT(2*NZONES+J) = C(0,J)*REMEFF(J)
40    CONTINUE

      CAHU = QCRETTOT/QRETTOT

      OUT(3*NZONES+1) = CAHU
      IF (QCIRC .NE. 0.0) THEN
        OUT(3*NZONES+2) = ((QCIRC-QOATOT)*CAHU*(1-FILTEFF(1))+
     .                    QOATOT*COA*(1-FILTEFF(2)))*
     .                    (1-FILTEFF(3))/QCIRC
      ENDIF

      RETURN
      END
```

```
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C  SUBROUTINE SHORTCALC
C
C  USES AN ITERATIVE MODIFIED EULER (SECOND-ORDER RUNGE-KUTTA)
C  METHOD FOR SOLVING THE CONCENTRATION DIFFERENTIAL EQUATION WHEN
C  THE VENTILATION SYSTEM IS OFF
C
C  SUBROUTINE SHORTCALC IS CALLED BY:
C      SUBROUTINE TYPE60
C
C  SUBROUTINE SHORTCALC CALLS:
C      FUNCTION F
C

      SUBROUTINE SHORTCALC (C,COA,DT,NSTEPS,NZONES)

      IMPLICIT NONE
      INTEGER I, J, NSTEPS, NZONES, ERRFLAG
      REAL F, C(-60:60,NZONES), COA, DT, C1, ERR


      DO 30 I = 0, NSTEPS - 1

C
C  ESTIMATE CONCENTRATION AT STEP I+1
C

        DO 10 J = 1, NZONES
          C(I+1,J) = C(I,J) + DT*F(0.0,C(I,J),0.0,J)
10      CONTINUE

C
C  CALCULATE CONCENTRATION AT STEP I+1 USING THE SLOPE ESTIMATE AT STEP I+1/2
C

15      ERRFLAG = 0
        DO 20 J = 1, NZONES
          C1 = C(I,J) + DT*F(0.0,(C(I,J)+C(I+1,J))/2.0,0.0,J)
          ERR = ABS( C1 - C(I+1,J) )
          IF (ERR .GE. 1.0E-8) ERRFLAG = 1
          C(I+1,J) = C1
20      CONTINUE
        IF (ERRFLAG .EQ. 1) GOTO 15

30    CONTINUE

C
C  RESET THE CURRENT CONCENTRATION TIME STEP TO 0 TO GET READY FOR THE NEXT
C  ENTRY INTO THIS SUBROUTINE
C

      DO 40 J = 1, NZONES
        DO 40 I = -60+NSTEPS, NSTEPS
          C(I-NSTEPS,J) = C(I,J)
40    CONTINUE

      RETURN
      END
```

```
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C  SUBROUTINE CALC
C
C  USES AN ITERATIVE MODIFIED EULER (SECOND-ORDER RUNGE-KUTTA)
C  METHOD FOR SOLVING THE CONCENTRATION DIFFERENTIAL EQUATION AND
C  INCLUDES A TIME DELAY IN THE HVAC AIR PATH FOR EACH ZONE.
C
C  SUBROUTINE CALC IS CALLED BY:
C      SUBROUTINE TYPE60
C
C  SUBROUTINE CALC CALLS:
C      FUNCTION F
C

       SUBROUTINE CALC (C,COA,DT,IDELAY,QRET,QRETTOT,QCIRC,QOATOT,
     .                  QIZF,REMEFF,NSTEPS,NZONES,ZONEVOL,FILTEFF)

       IMPLICIT NONE

       INTEGER I, J, L, NSTEPS, NZONES
       INTEGER IDELAY(NZONES), ERRFLAG

       REAL F, C(-60:60,NZONES), COA, DT, QRET(NZONES), QRETTOT, QCRETTOT
       REAL QCIRC, QOATOT, QIZF(25,NZONES), REMEFF(NZONES)
       REAL ZONEVOL(NZONES), FILTEFF(3)
       REAL QCIZF(25), CSUP, CAHU, C1, ERR

C
C  BEGIN THE MAIN CALCULATION LOOP BY CALCULATING A NEW CONCENTRATION
C  FOR EACH ROOM AND THEN PROCEEDING TO THE NEXT TIME STEP
C

       DO 50 I = 0, NSTEPS-1

C
C  CALCULATE THE SUPPLY AIR CONCENTRATION AND THE INTER-ZONE
C  CONTAMINANT FLOWS FOR TIME STEP I
C

         QCRETTOT = 0.0

         DO 10 J = 1, NZONES
           QCRETTOT = QCRETTOT + QRET(J)*C(I-IDELAY(J),J)*REMEFF(J)
           QCIZF(J) = 0.0

           DO 5 L = 1, NZONES
             QCIZF(J) = QCIZF(J) + QIZF(L,J)*C(I,L)/ZONEVOL(J)
5          CONTINUE
10       CONTINUE

         CAHU = QCRETTOT / QRETTOT
         CSUP = ((QCIRC-QOATOT)*CAHU*(1-FILTEFF(1))+
     .                   QOATOT*COA*(1-FILTEFF(2)))*(1-FILTEFF(3))/QCIRC

C
C  ESTIMATE CONCENTRATION AT STEP I+1
C
```

```
        DO 20 J = 1, NZONES
          C(I+1,J) = C(I,J)  + DT*F(CSUP,C(I,J),QCIZF(J),J)
20      CONTINUE

C
C  CALCULATE THE SUPPLY AIR CONCENTRATION AND THE INTER-ZONE
C  CONTAMINANT FLOWS FOR TIME STEP I+1/2
C

22      QCRETTOT = 0.0

        DO 30 J = 1, NZONES
          IF (IDELAY(J) .LE. 0 ) THEN
            QCRETTOT = QCRETTOT + QRET(J)*REMEFF(J)*
     .                                            (C(I,J)+C(I+1,J))/2.0
          ELSE
            QCRETTOT = QCRETTOT + QRET(J)*REMEFF(J)*
     .                        (C(I-IDELAY(J),J)+C(I-IDELAY(J)+1,J))/2.0
          ENDIF

          QCIZF(J) = 0.0

          DO 25 L = 1, NZONES
            QCIZF(J) = QCIZF(J) + QIZF(L,J)*(C(I,L)+C(I+1,L))/
     .                                                (2.0*ZONEVOL(J))
25        CONTINUE
30      CONTINUE

        CAHU = QCRETTOT / QRETTOT
        CSUP = ((QCIRC-QOATOT)*CAHU*(1-FILTEFF(1))+
     .               QOATOT*COA*(1-FILTEFF(2)))*(1-FILTEFF(3))/QCIRC

C
C  CALCULATE CONCENTRATION AT STEP I+1 USING THE SLOPE ESTIMATE AT STEP I+1/2
C

        ERRFLAG = 0
        DO 40 J = 1, NZONES
          C1 = C(I,J) + DT*F(CSUP,(C(I,J)+C(I+1,J))/2.0,QCIZF(J),J)
          ERR = ABS( C1 - C(I+1,J) )
          IF (ERR .GE. 1.0E-8) ERRFLAG = 1
          C(I+1,J) = C1
40      CONTINUE
        IF (ERRFLAG .EQ. 1) GOTO 22

50    CONTINUE

C
C  RESET THE CURRENT CONCENTRATION TIME STEP TO 0 TO GET READY FOR THE NEXT
C  ENTRY INTO THIS SUBROUTINE
C

      DO 60 J = 1, NZONES
        DO 60 I = -60+NSTEPS, NSTEPS
          C(I-NSTEPS,J) = C(I,J)
60    CONTINUE

      RETURN
      END
```

```
cccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccc
C
C  FUNCTION F
C
C  THIS FUNCTION IS FOR A ZONE WITH A SOURCE TERM, A VENTILATION
C  AIR TERM, AN INTER-ZONE AIR FLOW TERM, AND A REMOVAL TERM
C

      FUNCTION F (CSUP, C, QCIZF, J)

      IMPLICIT NONE

      INTEGER J, NZONES
      REAL F, A1(25), A2(25), A3(25), CSUP, C, QCIZF

      COMMON A1, A2, A3

      F = A1(J) + A2(J)*CSUP + QCIZF - A3(J)*C

      RETURN
      END
```

Appendix B

TYPE 65 FORTRAN LISTING

```
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C  TYPE 65 - CIRCULATION / OUTSIDE AIR FLOW CONTROLLER
C
C  THIS COMPONENT IS TO BE USED WITH TYPE 60 TO CONTROL THE AMOUNT OF OUTSIDE
C  AIR DELIVERED TO A ZONE BASED ON THE CONCENTRATION OF CONTAMINANTS IN THE
C  HVAC SYSTEM AS MEASURED BY AN IDEALIZED SENSOR
C
C  CREATED BY:
C      PAUL KNOESPEL
C
C  CREATED ON:
C      MARCH 10, 1990
C
C  MODIFIED ON / BY / REASON:
C
C
C  VARIABLE LISTING
C
C  I - INPUT VALUE
C  C - CALCULATED VALUE
C
C      CAHU        I  COMBINED RETURN CONTAMINANT CONCENTRATION AT THE AHU
C      CCONT       C  CONCENTRATION USED TO CONTROL OUTSIDE AIR
C      CPAIR       C  CONSTANT SPECIFIC HEAT OF AIR (KJ/KG-C, BTU/LBM-F)
C      CRET        I  RETURN DUCT CONTAMINANT CONCENTRATION AT THE ZONE
C      CSUP        I  SUPPLY DUCT CONTAMINANT CONCENTRATION
C      CLIMITH     I  CONCENTRATION LIMIT TO INITIATE ACTION
C      CLIMITL     I  CONCENTRATION LIMIT TO RESET ACTION
C      CLIMITV     I  CONCENTRATION LIMIT TO ALLOW VENTILATION SHUTDOWN
C      DTDT        I  TRNSYS VARIABLE - NOT USED
C      FCLH        I  FRACTION OF CLIMITH THAT INITIATES SCHEDULE IN FMODE 3
C      FOA         I  FRACTION OF OUTSIDE AIR APPLIED IN FMODE 3
C      FMODE       I  FLAG TO INDICATE THE MODE OF FLOW CONTROL TO BE USED
C      HCONT       C  ENTHALPY USED TO CONTROL OUTSIDE AIR IN FMODE 5 (KJ/KG,
C                     BTU/LBM)
C      HOA         I  ENTHALPY OF OUTSIDE AIR (KJ/KG, BTU/LBM)
C      HOD         C  CURRENT HOUR OF THE DAY BASED ON SIMULATION TIME
C      HZONE       I  ZONE AVERAGE ENTHALPY (KJ/KG, BTU/LBM)
C      INFO        I  TRNSYS TYPE INFORMATION
C      J           C  ZONE NUMBER INDEX
C      K           C  OUTSIDE AIR FLOW SCHEDULE INDEX
C      LUB         I  FORTRAN LOGICAL UNIT OF THE BUILDING DATA FILE
C      LUC         I  FORTRAN LOGICAL UNIT OF THE CONTROL SCHEDULE INPUT FILE
C      ND          C  NUMBER OF DERIVATIVES
C      NI          C  NUMBER OF INPUTS
C      NP          C  NUMBER OF PARAMETERS
C      NZONES      I  NUMBER OF ZONES
C      OUT         C  OUTPUT ARRAY TO TRNSYS
C      PAR         I  CONSTANT INPUT ARRAY FROM TRNSYS
C      QCIRC       C  TOTAL CIRCULATION AIR VOLUME FLOW RATE (M3/S, FT3/S)
```

```
C       QOA          C  OUTSIDE AIR VOLUME FLOW RATE INTO A ZONE (M3/S, FT3/S)
C       QOAIN        C  QOA AS ORIGINALLY INPUT TO THE SUBROUTINE (M3/S, FT3/S)
C       QOATEMP      C  QOA BASED ON TEMPERATURE IN FMODE 4 (M3/S, FT3/S)
C       QSUP         C  CIRCULATION AIR VOLUME FLOW RATE INTO A ZONE (M3/S, FT3/S)
C       QSUPIN       C  QSUP AS ORIGINALLY INPUT TO THE SUBROUTINE (M3/S, FT3/S)
C       RHOAIR       C  CONSTANT AIR DENSITY (KG/M3, LBM/FT3)
C       SMODE        I  FLAG TO INDICATE THE SENSOR LOCATION TO BE USED
C       STIME        I  DAILY SCHEDULE TIME THAT FOA IS APPLIED IN FMODE 4
C       T            I  TRNSYS VARIABLE - NOT USED
C       TCONT        C  TEMPERATURE USED TO CONTROL OUTSIDE AIR IN FMODE 4 (C, F)
C       TCOOL        I  TEMPERATURE OF SUPPLY AIR FOR VAV COOLING (C, F)
C       THEAT        I  TEMPERATURE OF SUPPLY AIR FOR VAV HEATING (C, F)
C       TIME         C  SIMULATION TIME (HR)
C       TIMEOFF      I  TIME OF THE DAY THAT VENTILATION SYSTEM IS TURNED OFF
C       TIMEON       I  TIME OF THE DAY THAT VENTILATION SYSTEM IS TURNED ON
C       TIMEPREV     C  TIME OF THE PREVIOUS SIMULATION STEP
C       TOA          I  OUTSIDE AIR TEMPERATURE (C, F)
C       TOALH        I  TOA ABOVE WHICH SCHEDULED FLOW IS ALLOWED IN FMODE 4 (C, F)
C       TSUP         C  SUPPLY AIR TEMPERATURE BASED ON THEAT OR TCOOL (C, F)
C       TZONE        I  ZONE AVERAGE TEMPERATURE (C, F)
C       UNITFLAG     I  FLAG TO INDICATE SI OR ENGLISH UNITS (1=SI, 2=ENGLISH)
C       VMODE        I  FLAG TO INDICATE VENTILATION TYPE (1=CAV, 2=VAV)
C       XIN          I  VARIABLE INPUT ARRAY FROM TRNSYS
C       ZONEP        I  ZONE SENSIBLE HEATING OR COOLING POWER (KJ/HR, BTU/HR)
C       ZONEVOL      I  ZONE VOLUME (M3, FT3)
C

        SUBROUTINE TYPE65(TIME,XIN,OUT,T,DTDT,PAR,INFO)

        IMPLICIT NONE

        INTEGER NI, NP, ND, INFO(10)
        INTEGER J, K, FMODE, SMODE, VMODE, NZONES, LUB, LUC, UNITFLAG

        REAL TIME, XIN(154), OUT(53), T, DTDT, PAR(11)
        REAL THEAT, TCOOL, CLIMITH, CLIMITL
        REAL ZONEVOL(25), CRET(25), CAHU, CSUP, QSUPIN(25), QOAIN(25)
        REAL ZONEP(25), TZONE(25), TOA, HZONE(25), HOA
        REAL TSUP, CPAIR, RHOAIR, CCONT, TCONT, HCONT
        REAL HOD, FCLH, TOALH, STIME(6), FOA(6), TIMEON, TIMEOFF, CLIMITV
        REAL QSUP(25), QOA(25), QOATEMP(25), QCIRC, TIMEPREV

C
C   SET PARAMETERS AND PERFORM CHECKS ON THE FIRST CALL OF THE SIMULATION
C

        IF(INFO(7).EQ.-1)THEN
          FMODE = NINT(PAR(1))
          SMODE = NINT(PAR(2))
          VMODE = NINT(PAR(3))
          NZONES = NINT(PAR(4))
          THEAT = PAR(5)
          TCOOL = PAR(6)
          LUB = NINT(PAR(7))
          LUC = NINT(PAR(8))
          CLIMITH = PAR(9)
          CLIMITL = PAR(10)
          UNITFLAG = PAR(11)
```

```
C
C  USE ENGLISH UNITS IF THE UNITS FLAG IS SET; OTHERWISE USE SI UNITS
C

        IF (UNITFLAG .EQ. 2) THEN
          CPAIR = 0.2418
          RHOAIR = 0.07517
        ELSE
          CPAIR = 1.012
          RHOAIR = 1.204
        ENDIF

        TIMEPREV = -1.0
        QCIRC = 0.0

        NI = 6*NZONES+4
        NP = 11
        ND = 0
        INFO(6) = 2*NZONES+3
        INFO(9) = 1
        CALL TYPECK(1,INFO,NI,NP,ND)

C
C  READ INPUTS THAT DO NOT CHANGE DURING THE SIMULATION - AIR FLOWS ARE
C  CONVERTED FROM AIR CHANGES PER HOUR TO M3/S (OR FT3/S)
C

        OPEN (LUB, STATUS = 'OLD')
        REWIND (LUB)
        READ(LUB,*) (ZONEVOL(J), J = 1, NZONES)
        CLOSE (LUB)

        DO 5 J = 1, NZONES
          QSUPIN(J) = XIN(J)*ZONEVOL(J)/3600.0
          QOAIN(J) = XIN(NZONES+J)*ZONEVOL(J)/3600.0
5       CONTINUE

C
C  READ CONTROL SCHEDULE DATA
C

        OPEN (LUC, STATUS = 'OLD')
        REWIND (LUC)
        READ (LUC,*) TIMEON
        READ (LUC,*) TIMEOFF
        READ (LUC,*) CLIMITV
        IF (FMODE .EQ. 3) THEN
          READ (LUC,*) FCLH
          READ (LUC,*) TOALH
          READ (LUC,*) (STIME(K), K = 1, 6)
          READ (LUC,*) (FOA(K), K = 1, 6)
        ENDIF
        CLOSE (LUC)
        ENDIF

C
C  IF THE TIME HAS NOT CHANGED SINCE THE PREVIOUS SUBROUTINE CALL, RETURN TO
C  TRNSYS TO PREVENT CHANGING THE FLOWS TWICE DURING A SINGLE TIME STEP
C
```

```
      IF (TIME .EQ. TIMEPREV) THEN
        RETURN
      ELSE
        TIMEPREV = TIME
      ENDIF

C
C  READ INPUT VALUES AT EACH TIME STEP
C

      DO 10 J = 1, NZONES
        TZONE(J)  = XIN(2*NZONES+J)
        ZONEP(J)  = XIN(3*NZONES+J)
        HZONE(J)  = XIN(4*NZONES+J)
        CRET(J)  = XIN(5*NZONES+J)
10    CONTINUE

      CAHU = XIN(6*NZONES+1)
      CSUP = XIN(6*NZONES+2)
      TOA = XIN(6*NZONES+3)
      HOA = XIN(6*NZONES+4)

C
C  USING THE SENSOR LOCATION FLAG, DETERMINE WHICH CONCENTRATION VALUE
C  WILL BE CONTROLLING OUTSIDE AIR FLOW
C

      IF (SMODE .EQ. 1) THEN
        CCONT = 0.0
        DO 15 J = 1, NZONES
          IF (CRET(J) .GT. CCONT) CCONT = CRET(J)
15      CONTINUE
      ELSEIF (SMODE .EQ. 2) THEN
        CCONT = CAHU
      ELSE
        CCONT = CSUP
      ENDIF

C
C  DETERMINE THE HOUR OF THE DAY; IF THE VENTILATION SYSTEM IS
C  SCHEDULED TO BE OFF AND THE CONCENTRATION IS BELOW CLIMITV, SET
C  THE VENTILATION FLOW RATES TO 0.0 AND SKIP TO THE OUTPUT SECTION;
C  IF THE VENTILATION SYSTEM IS TO BE TURNED ON, RESET THE VENTILATION
C  FLOW RATES TO THE ORIGINAL INPUT VALUES
C

      HOD = AMOD(TIME,24.0)
      IF (((HOD .LE. TIMEON) .OR. (HOD .GT. TIMEOFF)) .AND.
     . (CCONT .LT. CLIMITV)) THEN
        DO 20 J = 1, NZONES
          QSUP(J)  = 0.0
          QOA(J)  = 0.0
20      CONTINUE
        GOTO 55
```

```
      ELSEIF (QCIRC .EQ. 0.0) THEN
        DO 25 J = 1, NZONES
          QSUP(J) = QSUPIN(J)
          QCIRC = QCIRC + QSUP(J)
          QOA(J) = QOAIN(J)
25      CONTINUE
      ENDIF

C
C  DETERMINE THE CONTROL TEMPERATURE FOR FLOW CONTROL MODE 4
C

      TCONT = 0.0
      IF (FMODE .EQ. 4) THEN
        DO 30 J = 1, NZONES
          TCONT = TCONT + TZONE(J)*QSUP(J)/QCIRC
30      CONTINUE
      ENDIF

C
C  DETERMINE THE CONTROL ENTHALPY FOR FLOW CONTROL MODE 5
C

      HCONT = 0.0
      IF (FMODE .EQ. 5) THEN
        DO 35 J = 1, NZONES
          HCONT = HCONT + HZONE(J)*QSUP(J)/QCIRC
35      CONTINUE
      ENDIF


C
C  BEGIN CALCULATION LOOP FOR EACH ZONE
C

      DO 50 J = 1, NZONES

C
C  FOR VAV SYSTEM, CALCULATE A SUPPLY AIR FLOW RATE BASED ON THE HEATING
C  OR COOLING POWER CALCULATED BY TYPE 56, WITH A MAXIMUM VALUE OF QSUPIN
C

        IF (VMODE .EQ. 2) THEN
          IF (ZONEP(J) .GT. 0.0) THEN
            TSUP = TCOOL
          ELSE
            TSUP = THEAT
          ENDIF

          QSUP(J) = ZONEP(J)/(CPAIR*RHOAIR*3600.0*(TZONE(J)-TSUP))

          IF (QSUP(J) .GT. QSUPIN(J)) QSUP(J) = QSUPIN(J)
          IF (QSUP(J) .LT. 0.05*QSUPIN(J)) QSUP(J) = 0.0
        ENDIF
```

```
C
C  FLOW CONTROL MODE 1 - CONTROL OUTSIDE AIR FLOW BASED ON CONTAMINANT
C  CONCENTRATION WITH A MINIMUM VALUE OF QOAIN
C

        IF (FMODE .EQ. 1) THEN
          IF (CCONT .GE. CLIMITH) THEN
            QOA(J) = QOA(J) + 0.2*QSUPIN(J)
            IF (QOA(J) .GT. QSUPIN(J)) QOA(J) = QSUPIN(J)
          ELSEIF (CCONT .LE. CLIMITL) THEN
            QOA(J) = QOA(J) - 0.2*QSUPIN(J)
            IF (QOA(J) .LT. QOAIN(J)) QOA(J) = QOAIN(J)
          ENDIF

C
C  FLOW CONTROL MODE 2 - SWITCH FROM QOAIN TO 100% OUTSIDE AIR FLOW BASED
C  ON CONTAMINANT CONCENTRATION
C

        ELSEIF (FMODE .EQ. 2) THEN
          IF (CCONT .GE. CLIMITH) THEN
            QOA(J) = QSUPIN(J)
          ELSEIF (CCONT .LT. CLIMITL) THEN
            QOA(J) = QOAIN(J)
          ENDIF

C
C  FLOW CONTROL MODE 3 - CONTROL OUTSIDE AIR FLOW BASED ON A DAILY SCHEDULE
C  AND CONTAMINANT CONCENTRATION WITH A MINIMUM VALUE OF QOAIN
C

        ELSEIF (FMODE .EQ. 3) THEN
          IF (CCONT .GE. CLIMITH) THEN
            QOA(J) = QOA(J) + 0.2*QSUPIN(J)
            IF (QOA(J) .GT. QSUPIN(J)) QOA(J) = QSUPIN(J)
          ELSEIF (CCONT .LE. CLIMITL) THEN
            QOA(J) = QOA(J) - 0.2*QSUPIN(J)
            IF (QOA(J) .LT. QOAIN(J)) QOA(J) = QOAIN(J)
          ENDIF

          IF ((TOA .GT. TOALH) .AND. (CCONT .GE. FCLH*CLIMITH)) THEN
            IF (((HOD .LE. STIME(1)) .OR. (HOD .GT. STIME(6))) .AND.
     .        (QOA(J) .LT. FOA(6)*QSUPIN(J))) THEN
              QOA(J) = FOA(6)*QSUPIN(J)
              GOTO 45
            ENDIF

            DO 40 K = 1, 5
              IF ((HOD .GT. STIME(K)) .AND. (HOD .LE. STIME(K+1)) .AND.
     .          (QOA(J) .LT. FOA(K)*QSUPIN(J))) THEN
                QOA(J) = FOA(K)*QSUPIN(J)
                GOTO 45
              ENDIF
40          CONTINUE

          ENDIF
```

```
C
C  FLOW CONTROL MODE 4 - CONTROL OUTSIDE AIR FLOW RATE BASED ON CONCENTRATION
C  WITH A MINIMUM VALUE OF QOAIN; WHEN THERE IS A COOLING LOAD, ADJUST OUTSIDE
C  AIR FLOW BASED ON OUTSIDE AIR TEMPERATURE
C

        ELSEIF (FMODE .EQ. 4) THEN
          IF (CCONT .GE. CLIMITH) THEN
            QOA(J) = QOA(J) + 0.2*QSUPIN(J)
            IF (QOA(J) .GT. QSUPIN(J)) QOA(J) = QSUPIN(J)
          ELSEIF (CCONT .LE. CLIMITL) THEN
            QOA(J) = QOA(J) - 0.2*QSUPIN(J)
            IF (QOA(J) .LT. QOAIN(J)) QOA(J) = QOAIN(J)
            IF (CLIMITH .GE. 0.9) QOA(J) = QOAIN(J)
          ENDIF

          IF (ZONEP(J) .GT.0.0) THEN
            QOATEMP(J) = 0.0

            IF ((TOA .LE. TCONT) .AND. (TOA .GE. TCOOL)) THEN
              QOATEMP(J) = QSUP(J)
            ELSEIF (TOA .LT. TCOOL) THEN
              QOATEMP(J) = QSUP(J)*((TCONT-TCOOL)/(TCONT-TOA))
            ENDIF

            IF (QOA(J) .LT. QOATEMP(J)) QOA(J) = QOATEMP(J)
          ENDIF

C
C  FLOW CONTROL MODE 5 - CONTROL OUTSIDE AIR FLOW RATE BASED ON CONCENTRATION,
C  WITH A MINIMUM VALUE OF QOAIN; WHEN THERE IS A COOLING LOAD, ADJUST OUTSIDE
C  AIR FLOW BASED ON OUTSIDE AIR ENTHALPY
C

        ELSE
          IF (CCONT .GE. CLIMITH) THEN
            QOA(J) = QOA(J) + 0.2*QSUPIN(J)
            IF (QOA(J) .GT. QSUPIN(J)) QOA(J) = QSUPIN(J)
          ELSEIF (CCONT .LE. CLIMITL) THEN
            QOA(J) = QOA(J) - 0.2*QSUPIN(J)
            IF (QOA(J) .LT. QOAIN(J)) QOA(J) = QOAIN(J)
            IF (CLIMITH .GE. 0.9) QOA(J) = QOAIN(J)
          ENDIF

          IF (ZONEP(J) .GT.0.0) THEN
            IF ((HOA .LE. HCONT) .AND. (TOA .GE. TCOOL)) THEN
              QOA(J) = QSUP(J)
            ENDIF
          ENDIF

        ENDIF

45      IF (QOA(J) .LT. 0.05*QSUPIN(J)) QOA(J) = 0.0
        IF (QSUP(J) .LT. QOA(J)) QSUP(J) = QOA(J)

50    CONTINUE
```

```
C
C  CALCULATE VALUES FOR OUTPUT - AIR FLOWS ARE CONVERTED TO AIR CHANGES
C  OR VOLUME PER HOUR
C

55      OUT(2*NZONES+1) = 0.0
        OUT(2*NZONES+2) = 0.0
        QCIRC = 0.0

        DO 60 J = 1, NZONES
          OUT(J) = QSUP(J)*3600.0/ZONEVOL(J)
          OUT(NZONES+J) = QOA(J)*3600.0/ZONEVOL(J)
          OUT(2*NZONES+1) = OUT(2*NZONES+1)+QSUP(J)*3600.0
          OUT(2*NZONES+2) = OUT(2*NZONES+2)+QOA(J)*3600.0
          QCIRC = QCIRC + QSUP(J)
60      CONTINUE

        IF (OUT(2*NZONES+1) .EQ. 0.0) THEN
          OUT(2*NZONES+3) = 0.0
        ELSE
          OUT(2*NZONES+3) = OUT(2*NZONES+2) / OUT(2*NZONES+1)
        ENDIF

        RETURN
        END
```

Appendix C

TYPE 17 FORTRAN LISTING

```
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C  TYPE 17 - MODIFIED BIN SORTER
C
C  THIS COMPONENT WILL SORT DATA INTO BINS AND CALCULATE THE TOTAL TIME THAT
C  A VARIABLE VALUE IS BETWEEN THE LOW AND HIGH LIMITS OF EACH BIN
C
C  CREATED BY:
C      PAUL KNOESPEL
C
C  CREATED ON:
C      MAY 9, 1990
C
C  MODIFIED ON / BY / REASON:
C
C
C  VARIABLE LISTING
C
C  I - INPUT VALUE
C  C - CALCULATED VALUE
C
C      BIN          C  ACCUMULATED TIME VARIABLE VALUE IS BETWEEN BIN LIMITS
C      BINLIMIT     C  DIVISION BETWEEN BINS (VARIABLE UNITS)
C      DBIN         C  WIDTH OF A BIN (VARIABLE UNITS)
C      DTDT         C  TRNSYS VARIABLE - NOT USED
C      HILIMIT      I  HIGH VALUE OF THE LAST BIN (VARIABLE UNITS)
C      HODOFF       I  HOUR OF THE DAY TO END TRACKING (HR)
C      HODON        I  HOUR OF THE DAY TO START TRACKING (HR)
C      HOWOFF       I  HOUR OF THE WEEK TO END TRACKING (HR)
C      HOWON        I  HOUR OF THE WEEK TO START TRACKING (HR)
C      I            C  BIN NUMBER INDEX
C      INFO         C  TRNSYS TYPE INFORMATION
C      J            C  VARIABLE NUMBER INDEX
C      LOLIMIT      I  LOW VALUE OF THE FIRST BIN (VARIABLE UNITS)
C      LU           I  FORTRAN LOGICAL UNIT FOR OUTPUT
C      NBIN         I  NUMBER OF TIME BINS INTO WHICH THE VARIABLES ARE SORTED
C      ND           C  NUMBER OF DERIVATIVES
C      NI           C  NUMBER OF INPUTS
C      NP           C  NUMBER OF PARAMETERS
C      NVAR         I  NUMBER OF VARIABLES TO BE TRACKED
C      OUT          C  OUTPUT ARRAY TO TRNSYS
C      PAR          I  CONSTANT INPUT ARRAY FROM TRNSYS
C      T            C  TRNSYS VARIABLE - NOT USED
C      TEND         I  SIMULATION TIME TO END TRACKING (HR)
C      TIME         I  SIMULATION TIME (HR)
C      TIMEPREV     C  SIMULATION TIME OF THE PREVIOUS SUBROUTINE CALL (HR)
C      TSTART       I  SIMULATION TIME TO START TRACKING (HR)
C      XIN          I  INPUT ARRAY FROM TRNSYS
C
C
```

```
      SUBROUTINE TYPE17(TIME,XIN,OUT,T,DTDT,PAR,INFO)

      IMPLICIT NONE

      INTEGER NI, NP, ND, INFO(10)
      INTEGER LU, NVAR, NBIN, I, J

      REAL TIME, XIN(25), OUT(20), T, DTDT, PAR(11)
      REAL TSTART, TEND, HOWON, HOWOFF, HODON, HODOFF, HOW, HOD
      REAL LOLIMIT, HILIMIT, TIMEPREV
      REAL BIN(0:51,25), BINLIMIT(0:50), DBIN

C
C  SET PARAMETERS AND PERFORM CHECKS ON THE FIRST CALL OF THE SIMULATION
C

      IF(INFO(7).EQ.-1)THEN
        TSTART = PAR(1)
        TEND = PAR(2)
        HOWON = PAR(3)
        HOWOFF = PAR(4)
        HODON = PAR(5)
        HODOFF = PAR(6)
        LU = NINT(PAR(7))
        NVAR = NINT(PAR(8))
        LOLIMIT = PAR(9)
        HILIMIT = PAR(10)
        NBIN = NINT(PAR(11))

        IF (NBIN .GT. 50) NBIN = 50

        NI = NVAR
        NP = 11
        ND = 0
        INFO(6) = 0
        INFO(9) = 2
        CALL TYPECK(1,INFO,NI,NP,ND)

        DO 5 I = 0, NBIN+1
          DO 5 J = 1, NVAR
            BIN(I,J) = 0.0
5       CONTINUE

        TIMEPREV = 0.0
        BINLIMIT(0) = LOLIMIT
        DBIN = (HILIMIT - LOLIMIT) / FLOAT(NBIN)

        DO 10 I = 1, NBIN
          BINLIMIT(I) = BINLIMIT(I-1)+DBIN
10      CONTINUE

      ENDIF

      HOW = AMOD(TIME,168.0)
      HOD = AMOD(TIME,24.0)
```

```
C
C  SORT INPUT VALUES AT EACH TIME STEP THAT MEETS THE WEEKLY AND DAILY
C  TIME REQUIREMENTS
C
      IF (((TIME .GT. TSTART) .AND. (TIME .LE. TEND))   .AND.
     . ((HOW  .GT. HOWON)  .AND. (HOW  .LE. HOWOFF)) .AND.
     . ((HOD  .GT. HODON)  .AND. (HOD  .LE. HODOFF))) THEN
        DO 20 J = 1, NVAR
          IF (XIN(J) .LT. BINLIMIT(0)) BIN(0,J) = BIN(0,J) + (TIME-TIMEPREV)
          DO 15 I = 1, NBIN
            IF ((XIN(J) .GT. BINLIMIT(I-1)) .AND.
     .          (XIN(J) .LE. BINLIMIT(I))) THEN
              BIN(I,J) = BIN(I,J) + (TIME-TIMEPREV)
            ENDIF
15        CONTINUE
          IF (XIN(J) .GT. BINLIMIT(NBIN)) BIN(NBIN+1,J) =
     .                                BIN(NBIN+1,J) + (TIME-TIMEPREV)
20      CONTINUE
      ENDIF
      TIMEPREV = TIME

      IF (TIME .EQ. TEND) THEN
        DO 30 J = 1, NVAR
          WRITE(LU,200) J
          WRITE(LU,210) BINLIMIT(0), BIN(0,J)
          DO 25 I = 1, NBIN
            WRITE(LU,220) BINLIMIT(I-1), BINLIMIT(I), BIN(I,J)
25        CONTINUE
          WRITE(LU,230) BINLIMIT(NBIN), BIN(NBIN+1,J)
30      CONTINUE
      ENDIF

200   FORMAT('1', 9X, 'INPUT NUMBER ', I2)
210   FORMAT(10X, 'BELOW', 8X, F9.6, ' - ', F8.2)
220   FORMAT(10X, F9.6, ' TO ', F9.6, ' - ', F8.2)
230   FORMAT(10X, 'ABOVE', 8X, F9.6, ' - ', F8.2)

      RETURN
      END
```

Appendix D
EXAMPLE TRNSYS INPUT DECK

```
NOLIST
**************************************************
*                                                *
*      INDEPENDENCE LIFE INSURANCE BUILDING      *
*                 FLOOR MODEL                    *
*               April 10,  1990                  *
*                                                *
**************************************************

SIM 0 8760 0.25
WIDTH 132
*
*************************
UNIT 9 TYPE 9 DATA READER
*************************
*  Convert to English Units
*  1 - DN Solar, 2 - Horz Solar, 3 - Dry Bulb, 4 - Humidity, 5 - Wind
PAR 19
5 1  -1 0.08816 0 -2 0.08816 0 3 0.18 32.0 4 0.0001 0 5 3.281 0 10 1
(9X,F4.0,1X,F4.0,1X,F4.0,1X,F6.0,1X,F2.0)
*
******************************************
UNIT 20 TYPE 33 PSYCHROMETRICS OUTSIDE AIR
******************************************
*
   PAR 2
*  Dry Bulb & Humidity Ratio, English Units
   4 2
   INP 2
*  Toa  Woa
   9,3  9,4
   80.0 0.01
*
***********************************
UNIT 16 TYPE 16 RADIATION PROCESSOR
***********************************
*
   PAR 9
*  Rndl Fix Rndl Day Lat  Solc  Shft Eng IE
   5    1   3    1   43.1 428.0 0    2   -1
   INP 14
*  Ih  Toa  RH   Td1  Td2  rho  N         S         E         W
   9,2 9,3  20,1 9,19 9,20 0,0  0,0  0,0  0,0  0,0  0,0  0,0  0,0  0,0
   0.0 71.6 0.5  0.0  0.0  0.2  90. -165. 90.  15.  90. -75.  90.  105.
*
```

```
****************************************
UNIT 21 TYPE 33 PSYCHROMETRICS FLOOR9 AIR
****************************************
*
   PAR 2
*  Dry Bulb & Humidity Ratio, English Units
   4 2
   INP 2
*  Tz1  Wz1
   56,1 56,7
   72.0 0.008
*
*****************************************
UNIT 22 TYPE 33 PSYCHROMETRICS MTGROOM AIR
*****************************************
*
   PAR 2
*  Dry Bulb & Humidity Ratio, English Units
   4 2
   INP 2
*  Tz2  Wz2
   56,2 56,8
   72.0 0.008
*
******************************
UNIT 56 TYPE 56 TWO-ZONE OFFICE
******************************
*
   PAR 2
*  LU LU
   11 12
   INP 9
*  Toa Woa Ih   N    S     E     W     Flow1 Flow2
   9,3 9,4 16,4 16,6 16,11 16,14 16,17 65,3  65,4
   0.0 0.0 0.0  0.0  0.0   0.0   0.0   0.0   0.0
*
***********************************
UNIT 65 TYPE 65 HVAC FLOW CONTROLLER
***********************************
   PAR 11
*  FMode SMode VMode NZones THeat TCool LUB LUC CLimitH CLimitL Units
   1     1     2     2      85.0  50.0  14  15  0.001   0.0008  2
   INP 16
   0,0   0,0   0,0   0,0   56,1  56,2  56,3  56,4  21,3  22,3
   60,5  60,6  60,7  60,8  9,3   20,3
   6.0   6.0   0.0   0.0   0.0   0.0   71.6  71.6  26.0  26.0
   0.0   0.0   0.0   0.0   71.6  26.0
*
**********************************
UNIT 60 TYPE 60 CONTAMINANT MODULE
**********************************
   PAR 10
*  NZones DT(s) DeltaT(hr) COA      LUS LUB Filt1 Filt2 Filt3 Units
   2      30.0  0.25       0.000310 13  14  0.0   0.0   0.0   2
   INP 4
   65,1  65,2  65,3  65,4
   0.0   0.0   0.0   0.0
*
```

```
**********************************************
UNIT 25 TYPE 25 OUTPUT FOR FLOW DATA TRACKING
**********************************************
   PAR 4
*  DTPr(hr)  TOn  TOff  LUnit
   0.25      4440 4464  21
   INP 9
   65,1   65,2   65,3    65,4    60,1   60,2   60,3 60,4 65,7
   NChgs1 NChgs2 OAChgs1 OAChgs2 QOAPU1 QOAPU2 C1   C2   OAFrac
*
*******************************************************
UNIT 51 TYPE 15 ZONE 1 CLG/HTG/VENT ENERGY CALCULATION
*******************************************************
   PAR 14
   0 8 -4 -11 7 8 -4 -12 8 -4 -12 7 8 -4
   INP 2
*  PZone1  QVent1
   56,3    56,5
   0.0     0.0    0.0
*  Outputs  +PZone1  -PZone1  +QVent1  -QVent1
*
*******************************************************
UNIT 52 TYPE 15 ZONE 2 CLG/HTG/VENT ENERGY CALCULATION
*******************************************************
   PAR 14
   0 8 -4 -11 7 8 -4 -12 8 -4 -12 7 8 -4
   INP 2
*  PZone2  QVent2
   56,4    56,6
   0.0     0.0    0.0
*  Outputs  +PZone2  -PZone2  +QVent2  -QVent2
*
********************************************
UNIT 50 TYPE 28 CLG/HTG/VENT ENERGY SUMMARY
********************************************
   PAR 30
*  Monthly Totals
*  DTPr(Mo) TOn   TOff   LUnit Output
   -1       0     8760   21    2
   0 0 3 -3 0 0 3 -3 3 0 0 3 -3 3 -4 0 0 3 -3 0 0 3 -3 3 -4
   INP 10
*  +PZone1  +PZone2  -PZone1  -PZone2  QLat1  QLat2
   51,1     52,1     51,2     52,2     56,9   56,10
*  +QVent1  +QVent2  -QVent1  -QVent2
   51,3     52,3     51,4     52,4
   LABELS 7
   ClgEng HtgEng LatEng TotCHL +QVent -QVent TotQVt
*
*****************************************************
UNIT 41 TYPE 15 ZONE 1 OUTSIDE AIR ENERGY CALCULATION
*****************************************************
   PAR 19
   0 -1 119000 1 -1 0.075 1 0 0 4 1 -21 -31 8 -4 -31 7 8 -4
   INP 3
*  OAChg1  Hoa     HZone1
   65,3    20,3    21,3
   0.0     25.0    25.0
*  Outputs  +Qoa1  -Qoa1
*
```

```
*****************************************************
UNIT 42 TYPE 15 ZONE 2 OUTSIDE AIR ENERGY CALCULATION
*****************************************************
   PAR 19
   0 -1 2850 1 -1 0.075 1 0 0 4 1 -21 -31 8 -4 -31 7 8 -4
   INP 3
*  OAChg2  Hoa     HZone2
   65,4    20,3    22,3
   0.0     25.0    25.0
*  Outputs  +Qoa2  -Qoa2
*
******************************************
UNIT 40 TYPE 28 OUTSIDE AIR ENERGY SUMMARY
******************************************
   PAR 19
*  Monthly Totals
*  DTPr(Mo) TOn   TOff   LUnit Output
   -1       0     8760   21    2
   0 -4 0 -4 0 0 3 -3 0 0 3 -3 3 -4
   INP 6
*  CirTot OATot  +Qoa1  +Qoa2  -Qoa1  -Qoa2
   65,5   65,6   41,1   42,1   41,2   42,2
   LABELS 5
   CircFl OAirFl +QOAir -QOAir TotQOA
*
************************************************
UNIT 31 TYPE 15 ZONE 1 REHEAT ENERGY CALCULATION
************************************************
   PAR 24
   0 -1 119000 1 -1 0.075 1 -1 0.241 1 0 -1 50.0 4 1 0 4 8
   -13 -1 0.0 9 1 -4
   INP 3
*  NChgs1  TZone1  PZone1
   65,1    56,1    56,3
   0.0     72.0    0.0
*
************************************************
UNIT 32 TYPE 15 ZONE 2 REHEAT ENERGY CALCULATION
************************************************
   PAR 24
   0 -1 2850 1 -1 0.075 1 -1 0.241 1 0 -1 50.0 4 1 0 4 8
   -13 -1 0.0 9 1 -4
   INP 3
*  NChgs2  TZone2  PZone2
   65,2    56,2    56,4
   0.0     72.0    0.0
*
```

```
*************************************
UNIT 30 TYPE 28 REHEAT ENERGY SUMMARY
*************************************
   PAR 11
*  Monthly Totals
*  DTPr(Mo) TOn   TOff   LUnit Output
   -1       0     8760   21    2
   0 -3 0 -3 3 -4
   INP 2
*  ClgRH1  ClgRH2
   31,1    32,1
   LABELS 3
   ClgRH1 ClgRH2 RHTot
*
*********************************************
UNIT 27 TYPE 17 ZONE CONCENTRATION BIN SORTER
*********************************************
   PAR 11
*  TStart TEnd   HOWOn  HOWOff HODOn  HODOff LUnit
   0      8760   0      120    8      17     21
*  NVar   LoLim  HiLim  NBin
   2      3.0E-4 2.0E-3 34
   INP 2
*  CZone1  CZone2
   60,3    60,4
   0.0     0.0
*
END
```

Appendix E
DATA FILES

E.1 Building Data File

```
119000.0        2850.0
0.9             1.0
21750.0         507.0
0.2             0.0

0.0             0.0
107.0           0.0

Building Data File for TYPE60 and TYPE65
Enter the data above for each variable for J = 1, NZONES as follows:

ZONEVOL(1)...ZONEVOL(NZONES)
REMEFF(1)....REMEFF(NZONES)
RETVOL(1)....RETVOL(NZONES)
INFILT(1)....INFILT(NZONES)

IZF(1,1)........IZF(1,NZONES)
   .               .
   .               .
   .               .
IZF(NZONES,1)...IZF(NZONES,NZONES)

Where -
      ZONEVOL  is the zone volume in m3 or ft3
      REMEFF   is the zone pollutant removal effectiveness
      RETVOL   is the zone HVAC return air path volume in m3 or ft3
      INFILT   is the zone infiltration in air changes per hour
      IZF(A,B) is the inter-zone flow from zone A to zone B in kg/hr or lbm/hr
```

E.2 Source Data File

```
1.77E-4
6.0     8.0     12.0    13.0    17.0    19.0
20.0    100.0   50.0    100.0   30.0    0.0
6.0     8.0     12.0    13.0    17.0    19.0
0.0     10.0    5.0     10.0    0.0     0.0

1.77E-4
8.0     9.0     12.0    13.0    16.0    17.0
2.0     10.0    0.0     10.0    3.0     0.0
6.0     8.0     12.0    13.0    17.0    19.0
0.0     0.0     0.0     0.0     0.0     0.0
```

Source Schedule Data File for TYPE60
Enter the data above for each variable for J = 1, NZONES as follows:

```
SUNIT(J)
STIMEWD(J,1)...STIMEWD(J,6)
SMULTWD(J,1)...SMULTWD(J,6)
STIMEWE(J,1)...STIMEWE(J,6)
SMULTWE(J,1)...SMULTWE(J,6)
```

Where -

```
      SUNIT   is the unit source volume flow rate in m3/s or ft3/s
      STIMEWD is the schedule hour of the day for days 1 through 5
      SMULTWD is the source multiplier for weekdays
      STIMEWE is the schedule hour of the day for days 6 and 7
      SMULTWE is the source multiplier for weekend days
```

E.3 Control Data File

```
5.0
21.0
1.0

0.50
70.0

5.0    11.0    12.0    16.0    18.0    20.0
1.0    0.0     0.0     0.0     0.0     0.0

Control Schedule Data File for use with TYPE65
Enter the data above for each variable as follows:

TIMEON
TIMEOFF
CLIMITV

FCLH
TOALH

STIME(1)...STIME(6)
FOA(1).....FOA(6)

Where -
      TIMEON   is the hour of the day that the ventilation system is turned on
      TIMEOFF  is the hour of the day that the ventilation system is turned off
      CLIMITV  is the concentration limit below which turning off the
               ventilation system is permitted
      FCLH     is the fraction of the high concentration limit above
               which the scheduled flow will be used
      TOALH    is the outside air temperature above which the scheduled flow
               will be used in C or F
      STIME    is the schedule hour of the day
      FOA      is the fraction of outside air to be used
```

Appendix F
BUILDING INPUT DESCRIPTION

```
****************************************************
*                                                  *
*   INPUT DATA FOR INDEPENDENCE INSURANCE OFFICE   *
*        9th FLOOR MODEL WITH MEETING ROOM         *
*                  ENGLISH UNITS                   *
*                                                  *
*                  APRIL 5, 1990                   *
*                                                  *
****************************************************

*********************
PROPERTIES
*********************
*
  DENSITY =0.075      : CAPACITY =0.241 : HVAPOR =1055.0
  SIGMA   =1.7122E-09 : RTEMP    =527.67
*
*********************
TYPES
*********************
*
*--------------------LAYERS
*
* THICKNESS       (FT)
* CONDUCTIVITY    (BTU/HR-FT-F)
* CAPACITY        (BTU/LBm-F)
* DENSITY         (LBm/FT**3)
* RESISTANCE      (HR-FT**2-F/BTU)
*
* Values From ASHRAE Fundamentals 1981
*
  LAYER CONCRETE
* Interpolated Values, Average Thickness 4.75 Inch
*
    THICKNESS = 0.396  : CONDUCTIVITY =   0.45
    CAPACITY  = 0.22   : DENSITY      = 110.0
*
  LAYER FIREPROOF
* Modeled as resistance
*
    RESISTANCE = 2.5
*
  LAYER CARPET
* Table 3.1A Carpet and Rubber Pad
* Experiment 2 R=0.77
*
    RESISTANCE = 0.77
*
```

```
  LAYER CORNERS
* Model as 4 Inch Concrete
*
    THICKNESS = 0.33    : CONDUCTIVITY =   0.45
    CAPACITY  = 0.22    : DENSITY      = 110.0
*
  LAYER GYPWALL
* Interior Walls - 1/2 Inch Thick Gypsum
*
    THICKNESS = 0.0417 : CONDUCTIVITY =   0.093
    CAPACITY  = 0.26   : DENSITY      =  50.0
*
  LAYER GYPART
* Interior Partition - 3/8 Inch Thick Gypsum
*
    THICKNESS = 0.0312 : CONDUCTIVITY =   0.097
    CAPACITY  = 0.26   : DENSITY      =  50.0
*
  LAYER HOLLOW
* Interior Walls - Non Reflective Airspace 2.5 Inch
*
    RESISTANCE = 1.02
*
*
*--------------------INPUTS
*
* Input Ventilation Flow Rate From Controller
*
    INPUTS FLOW1, FLOW2
*
*--------------------WALLS
*
* CONVECTION (H)  (BTU/HR-FT**2-F)
* ASHRAE HFRONT = 1.46 Inside Still Air
*        HBACK  = 4.00 Outside 7 MPH Wind
*
  WALL OUTSIDE
    LAYERS    = GYPWALL, HOLLOW, CORNERS
    ABS-FRONT = 0.8  : ABS-BACK = 0.8
    HFRONT    = 1.46 : HBACK    = 4.00
  WALL INSIDE
    LAYERS    = GYPWALL, HOLLOW, GYPWALL
    ABS-FRONT = 0.8  : ABS-BACK = 0.8
    HFRONT    = 1.46 : HBACK    = 1.46
  WALL PARTITION
    LAYERS    = GYPART, HOLLOW, GYPART
    ABS-FRONT = 0.8  : ABS-BACK = 0.8
    HFRONT    = 1.46 : HBACK    = 1.46
  WALL FLOOR
    LAYERS    = CARPET, CONCRETE, FIREPROOF
    ABS-FRONT = 0.8  : ABS-BACK = 0.8
    HFRONT    = 1.35 : HBACK    = 1.35
  WALL ROOF
    LAYERS    = FIREPROOF, CONCRETE, CARPET
    ABS-FRONT = 0.8  : ABS-BACK = 0.8
    HFRONT    = 1.35 : HBACK    = 1.35
```

```
*
*--------------------WINDOWS
*
* Curtain Wall Modeled As Glass Adjusted For Framing
* ASHRAE HOUTSIDE = 1.46 Inside Still Air
*        HINSIDE  = 4.00 Outside 7.5 MPH Wind

  WINDOW DOUBLE
    UGLASS      = 0.972 : HINSIDE     = 1.46 : HOUTSIDE = 4.00
    ABSORBTANCE = 0.580 : REFLECTANCE = 0.356
*
*--------------------GAINS
*
* Add Gains and Schedules
* Computer Equipment PC's
* Energy in BTU/Hr, Humidity in LBM/HR
*
  GAIN PEOPLE
* Table 4.5, 26.24 - 255 BTU/Hr Sensible, 255 BTU/Hr Latent
* Gain / Person  Convective 68% Radiative 32%
    CONVECTIVE = 173.0 : RADIATIVE = 82.0 : HUMIDITY = 0.0024
*
  GAIN LIGHTS
* Estimated at 92 Watts/Fixture * 3.413 = 314.0 Btu/Hr-Fixture
* Convective 15% Radiative 85%
    CONVECTIVE = 47.0 : RADIATIVE = 267.0 : HUMIDITY = 0.0
*
  GAIN EQUIPMENT
* Typewriters, Computers, Printers, Copiers
* Total Load 10,762 Watts * 3.413 = 36,730 Btu/Hr
* Convective 20% Radiative 80%
    CONVECTIVE = 7246.0 : RADIATIVE = 29384.0 : HUMIDITY = 0.0
*
*--------------------SCHEDULES
*
* Weekday & Weekend Schedules

* Occupancy
  SCHEDULE OCCWKD
    HOURS  = 0,   6,   8,   12,  13,  17,  19
    VALUES = 0.0, 0.2, 1.0, 0.5, 1.0, 0.3, 0.0
  SCHEDULE OCCWKE
    HOURS  = 0,   6,   8,   12,  13,  17,  19
    VALUES = 0.0, 0.0, 0.1, 0.0, 0.1, 0.0, 0.0
  SCHEDULE OCCUPY
    DAYS   = 1,6
    HOURLY = OCCWKD,OCCWKE
  SCHEDULE MTGWKD
    HOURS  = 0,   8,   9,   12,  13,  16,  17
    VALUES = 0.0, 0.2, 1.0, 0.0, 1.0, 0.3, 0.0
  SCHEDULE MEETING
    DAYS   = 1,6
    HOURLY = MTGWKD,OCCWKE
```

```
* Lighting
  SCHEDULE LITEWKD
    HOURS  = 0,   6,   21
    VALUES = 0.1, 1.0, 0.1
  SCHEDULE LITEWKE
    HOURS  = 0,   6,   21
    VALUES = 0.1  0.2  0.1
  SCHEDULE LIGHT
    DAYS   = 1,6
    HOURLY = LITEWKD, LITEWKE

* Day Cooling & Heating
  SCHEDULE COOLWKD
    HOURS  = 0,  5,  21
    VALUES = 0.0 1.0 0.0
  SCHEDULE COOLWKE
    HOURS  = 0,  5,  21
    VALUES = 0.0 1.0 0.0
  SCHEDULE AMCOOL
    DAYS   = 1,6
    HOURLY = COOLWKD, COOLWKE

* Humidity Limit
  SCHEDULE LATWKD
    HOURS  = 0,  5,     21
    VALUES = 1.0 0.0085 1.0
  SCHEDULE LATWKE
    HOURS  = 0,  5,     21
    VALUES = 1.0 0.0085 1.0
  SCHEDULE LATENT
    DAYS   = 1,6
    HOURLY = LATWKD, LATWKE

* Equipment
  SCHEDULE EQPWKD
    HOURS  = 0,   7,   17
    VALUES = 0.6, 1.0, 0.6
  SCHEDULE EQPWKE
    HOURS  = 0,   7,   17
    VALUES = 0.6  0.6  0.6
  SCHEDULE EQUIP
    DAYS   = 1,6
    HOURLY = EQPWKD, EQPWKE

*
*--------------------INFILTRATION
*
  INFILTRATION LEAK
    AIRCHANGE = 0.2
*
*--------------------VENTILATION
*
  VENTILATION OAVENT1
    TEMPERATURE = OUTSIDE
    AIRCHANGE   = INPUT FLOW1
    HUMIDITY    = OUTSIDE
```

```
  VENTILATION OAVENT2
    TEMPERATURE = OUTSIDE
    AIRCHANGE   = INPUT FLOW2
    HUMIDITY    = OUTSIDE
*
*------------------HEATING & COOLING
*
  COOLING CHILL1
    ON       = 74.0
    POWER    = SCHEDULE 450000.0*AMCOOL
    HUMIDITY = SCHEDULE 1.0*LATENT
  HEATING HOT1
    ON       = 68.0
    POWER    = SCHEDULE 220000.0*AMCOOL
    HUMIDITY = SCHEDULE 0.0*LATENT
*
  COOLING CHILL2
    ON       = 74.0
    POWER    = SCHEDULE 8000.0*AMCOOL
    HUMIDITY = SCHEDULE 1.0*LATENT
  HEATING HOT2
    ON       = 68.0
    POWER    = SCHEDULE 5000.0*AMCOOL
    HUMIDITY = SCHEDULE 0.0*LATENT
*
*-------------------ORIENTATIONS
*
  ORIENTATIONS HORIZONTAL NORTH SOUTH EAST WEST
*
*-------------------ZONES
*
  ZONES FLOOR9, MTGROOM
*
*********************
BUILDING
*********************
*
* ASSUMPTIONS:
*   Identical boundary floor and ceiling
*
  ZONE FLOOR9

    WINDOW = DOUBLE
      AREA = 1440.0  : ORIENTATION = NORTH : TRANSMITTANCE = 0.064
    WINDOW = DOUBLE
      AREA = 810.0   : ORIENTATION = EAST  : TRANSMITTANCE = 0.064
    WINDOW = DOUBLE
      AREA = 1440.0  : ORIENTATION = SOUTH : TRANSMITTANCE = 0.064
    WINDOW = DOUBLE
      AREA = 810.0   : ORIENTATION = WEST  : TRANSMITTANCE = 0.064

    WALL = OUTSIDE
      AREA =   108.0  : EXTERNAL          : ORIENTATION = NORTH
    WALL = OUTSIDE
      AREA =   108.0  : EXTERNAL          : ORIENTATION = EAST
    WALL = OUTSIDE
      AREA =   108.0  : EXTERNAL          : ORIENTATION = SOUTH
    WALL = OUTSIDE
      AREA =   108.0  : EXTERNAL          : ORIENTATION = WEST
```

```
    WALL = INSIDE
      AREA =  5000.0   : INTERNAL
    WALL = INSIDE
      AREA = 660.0   : ADJACENT = MTGROOM
      FRONT         : COUPLING = SCHEDULE 107.0*AMCOOL
    WALL = PARTITION
      AREA = 10000.0   : INTERNAL
    WALL = ROOF
      AREA =  14500.0  : INTERNAL
    WALL = FLOOR
      AREA =  14500.0  : INTERNAL

    REGIME
      GAIN = PEOPLE     : SCALE = SCHEDULE 100.0*OCCUPY
      GAIN = LIGHTS     : SCALE = SCHEDULE 240.0*LIGHT
      GAIN = EQUIPMENT  : SCALE = SCHEDULE EQUIP
      INFILTRATION = LEAK
      VENTILATION  = OAVENT1
      COOLING      = CHILL1
      HEATING      = HOT1

      CAPACITANCE  = 1.0  : VOLUME   = 119000.0
      TINITIAL     = 71.6 : WINITIAL =      0.008
      WCAPR        = 1.0
*
*
  ZONE MTGROOM

    WALL = INSIDE
      AREA = 660.0   : ADJACENT = FLOOR9
      BACK          : COUPLING = 0.0
    WALL = ROOF
      AREA =  338.0  : INTERNAL
    WALL = FLOOR
      AREA =  338.0  : INTERNAL

    REGIME
      GAIN = PEOPLE     : SCALE = SCHEDULE 10.0*OCCUPY
      GAIN = LIGHTS     : SCALE = SCHEDULE 6.0*LIGHT
      GAIN = EQUIPMENT  : SCALE = SCHEDULE 0.01*EQUIP
      VENTILATION  = OAVENT2
      COOLING      = CHILL2
      HEATING      = HOT2

      CAPACITANCE  = 1.0  : VOLUME   = 2850.0
      TINITIAL     = 71.6 : WINITIAL =      0.008
      WCAPR        = 1.0
*
*********************
OUTPUTS
*********************
*
TRANSFER : TIMEBASE = 0.25
ZONES  = FLOOR9, MTGROOM
NTYPES = 1, 2, 5, 9, 10, 3, 4, 6, 7, 8
*
END
```

Appendix G
COIL AND REHEAT ENERGY CALCULATION

The Type 56 building load component was used to calculate coil and reheat energy required for the office and meeting room. Since the method is not straightforward, it is presented here. Figure G.1 shows the layout of a basic HVAC system and serves to define the variables used in the derivation of the necessary equations.

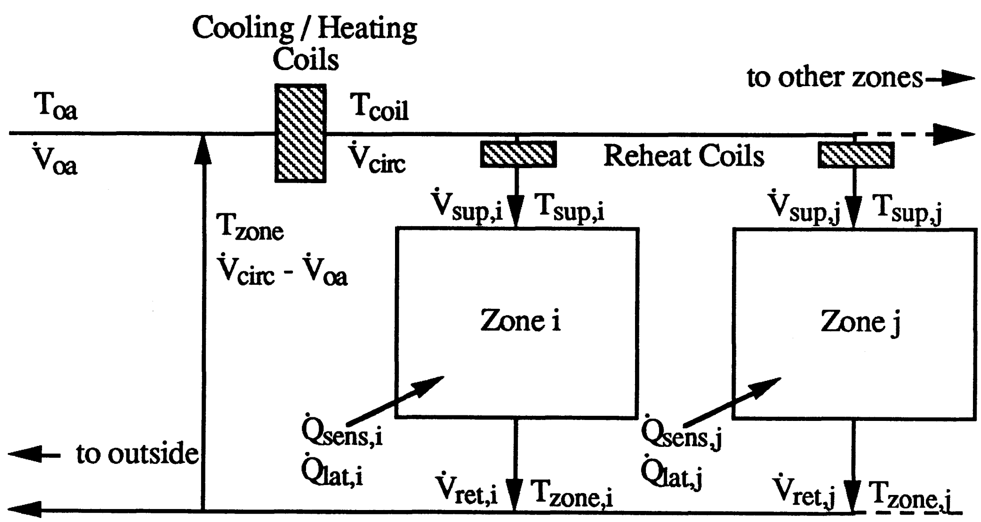

Figure G.1 *Ventilation System Definitions for Coil and Reheat Energy Calculations*

If the volume flow rates are integrated over time resulting in a volume of air that passed through the coil, the energy removed by the coil is:

$$Q_{coil} = V_{circ}\, \rho_{air}\, C_p\, (T_{mix} - T_{coil}) + Q_{lat} \qquad (G.1)$$

where T_{mix} is the temperature of the mixed recirculated and outside air streams. Separating these air streams yields equation G.2.

$$Q_{coil} = (V_{circ} - V_{oa})\, \rho_{air}\, C_p\, (T_{zone} - T_{coil}) + V_{oa}\, \rho_{air}\, C_p\, (T_{oa} - T_{coil}) + Q_{lat} \qquad \text{(G.2)}$$

Collecting the outside air terms together gives:

$$Q_{coil} = V_{circ}\, \rho_{air}\, C_p\, (T_{zone} - T_{coil}) + V_{oa}\, \rho_{air}\, C_p\, (T_{oa} - T_{coil}) - V_{oa}\, \rho_{air}\, C_p\, (T_{zone} - T_{coil}) + Q_{lat} \qquad \text{(G.3)}$$

If the outside air terms are now combined, equation G.4 results.

$$Q_{coil} = V_{circ}\, \rho_{air}\, C_p\, (T_{zone} - T_{coil}) + V_{oa}\, \rho_{air}\, C_p\, (T_{oa} - T_{zone}) + Q_{lat} \qquad \text{(G.4)}$$

The outside air term is now equivalent to what is called Q_{vent} in Type 56.

$$Q_{coil} = V_{circ}\, \rho_{air}\, C_p\, (T_{zone} - T_{coil}) + Q_{vent} + Q_{lat} \qquad \text{(G.5)}$$

The circulation air term can be split into two terms. The first is the energy to raise the circulation air from the supply temperature to the zone temperature. The second term is the energy to raise the circulation air from the coil temperature to the supply temperature.

$$Q_{coil} = V_{circ}\, \rho_{air}\, C_p\, (T_{zone} - T_{sup}) + V_{circ}\, \rho_{air}\, C_p\, (T_{sup} - T_{coil}) + Q_{vent} + Q_{lat} \qquad \text{(G.6)}$$

The first term in equation G.6 is the sensible heat load of the building. The sensible load as calculated by Type 56 is positive for cooling and negative for heating. The second term in equation G.6 is the reheat energy, which is explained further below. The final form of the equation for coil energy in terms of values available from Type 56 is:

$$Q_{coil} = Q_{sens} + Q_{rh} + Q_{vent} + Q_{lat} \tag{G.7}$$

The reheat energy term Q_{rh} is non-zero in cooling load conditions when the circulation air flow needs to be more than what is necessary to carry the building thermal load at T_{coil}. This will occur almost all the time for a CAV system and when the minimum circulation flow is reached for a VAV system. Under these conditions, the supply air temperature T_{sup} must be greater than the coil outlet temperature T_{coil}. The energy to raise the temperature of the air stream from T_{coil} to T_{sup} is the reheat energy.

$$Q_{rh} = V_{circ}\, \rho_{air}\, C_p\, (T_{sup} - T_{coil}) \tag{G.8}$$

Since the supply temperature is unknown to Type 56, another term can be added so that all information in the equation is known.

$$Q_{rh} = V_{circ}\, \rho_{air}\, C_p\, (T_{zone} - T_{coil}) - V_{circ}\, \rho_{air}\, C_p\, (T_{zone} - T_{sup}) \tag{G.9}$$

In the first term, both the zone temperature and the coil temperature are known. In the second term, the supply air temperature is still present, but the term is equal to the zone sensible load calculated by Type 56. The final form of the reheat energy equation is equation G.10.

$$Q_{rh} = V_{circ}\,\rho_{air}\,C_p\,(T_{zone} - T_{coil}) - Q_{sens} \tag{G.10}$$