Experimental and Theoretical Evaluation of the Effect of CO_2-Based Demand-Controlled Ventilation Strategies on Energy Consumption

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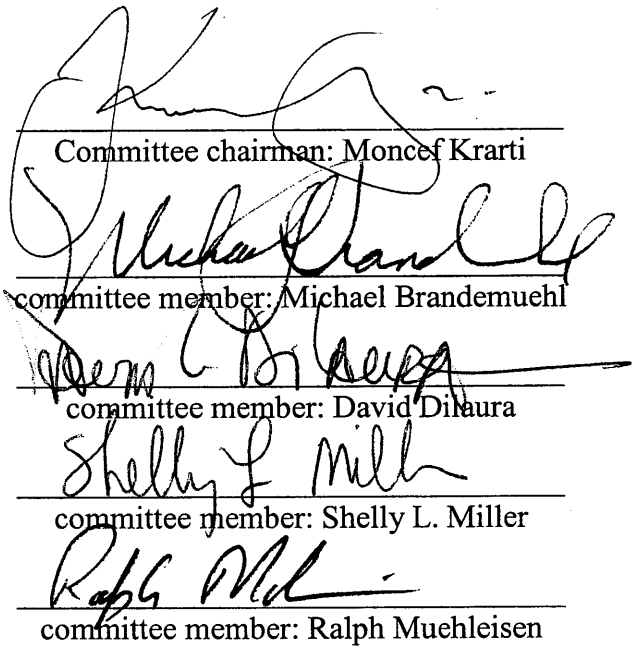
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Experimental and Theoretical Evaluation of the Effect of CO_2-Based Demand-Controlled Ventilation Strategies on Energy Consumption

Thesis directed by Associate professor Moncef Krarti

The ASHRAE Standard 62-1989 recommends two approaches for determining ventilation rates for buildings. The first, the Ventilation Rate Procedure, specifies a fixed rate for each building type based on its function. The second, the Air Quality Procedure, allows the use of lower ventilation rates than those mentioned in the Ventilation Rate Procedure, as long as the indoor air quality in the building's zones is maintained at the minimum required level.

This research investigates how the energy consumption in HVAC systems can be reduced based on the ASHRAE's Air Quality Procedure approach. Both the experimental and the theoretical parts of this research are based on the concept that carbon dioxide is a surrogate of indoor air quality that is easily sensed, easily controlled, and has a concentration level directly related to occupancy density.

All experiments were performed in the Joint Center for Energy Management (JCEM) at the Larson Building Systems Laboratory. A full set of experiments was performed to ensure that the setup would operate as expected and to correct all the problems related to laboratory equipment and control. In addition to the base case control strategy (20 cfm/person for office buildings), three other control strategies were evaluated: ON/OFF, linear, and proportional-integral-

derivative (PID). During these experiments, both the CO_2 setpoint and the temperature were controlled at 900 ppm and 70 °F, respectively. The analysis of the experiments revealed that these control strategies consume less outside air, and thus, less energy. An uncertainty analysis was also performed for all the experimental results.

In the theoretical part of this research, a multi-zone contaminant transport model and a contaminant control model were developed. These models were validated using the experimental work. Variable-air volume (VAV) and HVAC equipment models were also developed to determine the energy performance. All four models have been integrated as an IAQ/HVAC model to determine the energy consumption of the control strategies. The energy consumption of each control strategy has been calculated for a number of US cities to reveal savings or penalties.

The sensitivity analysis revealed that the outside air leakage from the outside damper, the use of a temperature economizer, the proportional gain constant (K_P) in the PID control equation, and the CO_2 setpoint can have an impact energy on consumption.

I dedicate this work to

My mother, my father, and my wife.

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CHAPTER ONE

Introduction

1.1 Definition of Indoor Air Quality

Indoor air quality (IAQ) refers to the quality of air being circulated, conditioned, and breathed within the confines of a building. There are three major sources of IAQ problems: building materials, building occupants, and ambient air.

Emissions from contaminant sources within a building are the primary determinants of IAQ. They include building materials containing formaldehyde, consumer products, cleaners, and furnishings that contain a number of volatile organic compounds (VOC); combustion and food preparation appliances and other processes; and biological growth from standing water and damp surfaces. In addition, building occupants and their activities produce carbon dioxide, water vapor, biological particulate matters, biological aerosols, environmental tobacco smoke, and other contaminants. In recent years, a number of these indoor contaminants have received increased attention. Depending on the location of the building, fresh outdoor air taken in by the air-handling unit (AHU) may actually contain high concentrations of pollutants dispersed by other buildings, industrial and utility factories, and automobile traffic.

Having adequate IAQ in commercial buildings was once a fortunate result of construction methods and materials of the era. Envelope construction, although not necessarily shoddy, allowed for much uncontrolled infiltration. In addition, most windows were operable. If occupants needed more fresh air than was provided by infiltration, they could simply open the nearest window. As the fresh air entered the room, it was conditioned by the HVAC system. If windows remained open during the hot summer days and cold winter nights, the system would use significant amounts of energy. The oil embargo in 1973 changed this concept dramatically, and thus, changed the whole context of indoor air quality. The higher cost of energy made uncontrolled infiltration unacceptable. So, a greater emphasis was placed on constructing extremely tight buildings and reducing ventilation rates.

Inoperable windows became a commonplace in the late 1970's and early 1980's and HVAC systems became the only controllable source of fresh air. It is estimated now that conditioning and distribution costs associated with moving air in buildings account for 50 to 60 percent of America's total energy consumption, more than $80 billion per year (Meckler, 1991). In addition to the ventilation problems caused by tighter building construction, the increased use of toxic materials found in modern office equipment, cleaning agents, and synthetic building materials added more problems. Productivity-limiting acute building sickness like Sick Building Syndrome (SBS), Building Related Illness (BRI), and Multiple Chemical Sensitivity (MCS) are significant concerns of today's building industry (ASHRAE, 1989).

1.2 Sick Building Syndrome

Sick building Syndrome (SBS) is a term associated with poor indoor air quality and is defined as the condition in which at least 20 percent of the building occupants display symptoms of illness of an unknown source. The symptoms go away when people leave the building. Building Related Illness (BRI) refers to an illness that can be directly traced to specific building sources. Two examples of BRI are Legionnaire's disease and asthma. Once identified, the contaminants inducing SBS can develop into BRI, and if proper remedial steps are not taken, they can threaten the health of the building occupants.

The types of pollutants relevant to SBS and BRI can be divided into two broad categories: biological and chemical contaminants, which originate from dead and living organisms, including bacteria, fungi, molds, and to a lesser extent, protozoa and viruses. Chemical pollutants that arise from non-biological sources include cigarette smoke, carbon monoxide emissions from outside air, formaldehyde from building materials, and volatile organic compounds (VOCs). Symptoms from exposure to such contaminants include nose and eye irritation, respiratory tract disorders, headaches, fever, nausea, skin irritation, and general lethargy (Hays and Ganick, 1992).

Multiple Chemical Sensitivity (MCS) is another acute building sickness and refers to people who become sensitive to chemicals in the environment. Although exposure to levels commonly encountered in buildings do not effect most people, some individuals appear to suffer repeated acute headaches when exposed to these chemicals. Such individuals report significant adverse effects, but the lack

of a clear diagnostic criteria and data or tools to evaluate the extent of the effects often leads to some skepticism. While no widely accepted test of physiological function has been shown to correlate with the symptoms, the sheer mass of anecdotal data is reasonable cause for concern (Meckler, 1991).

According to estimates prepared by Minneapolis-based Honeywell Inc., up to 20 percent of the 4 million commercial buildings in the United States may face indoor air quality problems that could harm the health of their occupants (Kim, 1990). The National Institute of Occupational Safety and Health (NIOSH) claimed that the number of structures afflicted with Sick Building Syndrome was expected to rise as the population of office workers increased in the United States economy of the 1990s (Kim, 1990).

1.3 Carbon Dioxide Control Principle

Carbon dioxide is produced by people in their respiratory process, so that the concentration of this gas generally rises in proportion to the number of people present in a given space and declines with the introduction of outside air, which has low concentration of CO_2 (350-400 ppm). The ASHRAE standard 62-1989 adopts this concept and specifies that if the indoor air concentration of CO_2 is below a certain limit (set-point), then acceptable indoor air quality has been achieved provided that other contaminants are also within acceptable limits. It should be noted that this assumption is not always valid, but is generally acceptable in office buildings.

Assuming that the CO_2 concentration is a good indicator of indoor air quality, a very simple approach to meet this criterion would be to monitor the CO_2 concentration at an appropriate location in the building and to introduce outside air to make sure that the indoor air is maintained at good quality and within the specified limits. In order to save energy, outdoor air should not be introduced to the space if people are not present.

1.4 History of Ventilation and ASHRAE Standard 62-1989

In 1895, the American Society of Heating and Ventilating Engineers (ASHVE) recommended a rate of 30 cfm/person. By 1925, 23 states required buildings to be ventilated at this rate. From 1934 to 1936, ASHVE conducted research which showed that appropriate ventilation rates depend on the age, activity level, cleanness of the room occupants, and room temperature and humidity (Janssen, 1994).

After changing the name of ASHVE to ASHRAE, and through Standard 62, the first attempt to define acceptable indoor air quality was made. This was through the first version of the "Standard for Natural and Mechanical Ventilation" (Standard 62-1973). This standard provided a perspective approach that specified both minimum and recommended outdoor air flow rates for a variety of indoor spaces. The absolute minimum rate was 5 cfm/person. There were many questions as to whether the recommended air rates could be applied to all variations of buildings environments. In particular, HVAC engineers

questioned whether the same ventilation rates mentioned in the Standard could be applied to smoking and non-smoking environments.

Standard 62-1981 was revised to the current version 62-1989, accepting 15 cfm per person as the lowest permissible ventilating rate under the Ventilation Rate Procedure, without discriminating between smoking-permitted or smoking-prohibited environments. This rate, based on measurements of odor level versus ventilating rate, was required to dilute the odor in a room to a concentration that 80% of the visitors entering the space would find acceptable. Supporting studies in which room CO_2 levels were controlled showed that no negative human response was found when the CO_2 level was maintained below 1,000 ppm by increasing the outdoor air flow rate to 15 cfm per person.

ASHRAE's revised Standard 62-1989 answered these questions. The proposed version recommended different outdoor airflow rates for smoking-permitted and for smoking-prohibited environments. In addition, the Standard introduced an alternative air quality procedure to permit innovative, energy-conserving ventilation practices. This alternative method allowed engineers to use lower rates as long as the indoor air contaminants would be held below the limits.

1.5 ASHRAE Standard 62-1989R and its Impact on CO_2-Based Demand Controlled Ventilation (DCV)

The CO_2-based demand controlled ventilation is discussed in the Revised Version of ASHRAE Standard 62-1989. This revised version is not officially an accepted standard. It is presented as optional ventilation controls that vary

ventilation rates as a function of the number of occupants in the space. This means that CO_2 is a surrogate for the concentration occupant related contaminants such as bio-effluents, thus allowing the rate of outdoor air supplied to the spaces to be modulated below the design levels established in section 6 of the revised standard. However, the standard requires the maintenance of a minimum outdoor air intake rate whenever the system is operated, regardless of the CO_2 concentration. The minimum shall be the minimum outdoor air rate required to control building sources adjusted for system ventilation efficiency. The Standard also mentions that ventilation systems designed using the Simple Systems Procedure of Section 6 may not use CO_2-based demand controlled ventilation.

Section 6 in the revised Standard presents three procedures for calculating ventilation rates for a ventilating system. These procedures are briefly described below:

The simple systems procedure can be used for single zone systems and systems that supply 100% outdoor air to each space served. The ventilation rates prescribed in this procedure are intended to dilute and exhaust odorous bioeffluents from occupants and a moderate amount of odorous and sensory irritant contaminants from other types of sources common to that type of space. Systems designed for using this procedure may not use CO_2-based demand controlled ventilation because credit for occupancy diversity is implicit in the ventilation rates, and such credit is not allowed when demand controls are used. Also, use of demand controls requires knowledge of the individual components

(building and people) of the outdoor air ventilation rate which are not explicitly calculated with this procedure. Minimum total outdoor air rate (V_{ot}) required by this procedure is determined in accordance with equation 6.1 in the Standard. The minimum supply rate (MSR) served by the system is determined in accordance with equation 6.2.

The Prescriptive procedure may be used for any ventilation system type. The ventilation rates prescribed by this procedure are intended to control indoor air contaminants in properly designed and maintained spaces used for the intended occupancy at levels that will meet the sensory satisfaction of a substantial majority of adapted occupants. Table 6.1 in the Standard lists the minimum prescribed ventilation requirements for this procedure. Equations 6.5 and 6.7 calculate the MSR and V_{ot} in accordance with this procedure, respectively.

The analytical procedure may also be used for any ventilation system type. This procedure, however, is recommended for spaces where there are strong sources that cannot be eliminated or controlled, for spaces designed for especially low concentrations of contaminants, or for spaces where other levels of acceptance are the design goal. This procedure satisfies both the health and the comfort criteria. The steps for calculating design ventilation rates according to this procedure are outlined in the Standard.

1.6 How ASHRAE 62-1989R Accounts for Pollutants in CO_2 DCV

Many activities can effect the indoor air quality of a building. These may include occupant activities, building maintenance activities, past control, housekeeping, carpet cleaning, renovation and remodeling activities, new furnishings, and building materials. While CO_2 concentrations can be appropriate means of characterizing the acceptability of a space in terms of body odor, they do not provide information on the control of contaminants from other pollutant sources, such as those mentioned above. While maintaining CO_2 concentrations within 700 ppm of outdoors should provide acceptable perceived air quality in terms of human body odor, it does not necessarily imply adequate control of other pollutant sources.

To account for these pollutants in the CO_2-based demand controlled ventilation strategies, the Revised Version of the Standard determines the amount of minimum design ventilation rate that should be supplied to the space regardless of the concentration of CO_2. ***The revised standard adjusts the equations of the minimum outdoor air rate (Vot) in section 6 to account for those other pollutants.*** The Standard discusses the specifics for three types of systems and adjusts the design ventilation rate (DVR) equation to be implemented in the CO_2-based demand controlled ventilation. These systems, as discussed in appendix F-2 of the standard, are as follows:

1. Systems that supply 100% outdoor air to each space with no re-circulation.

2. Systems that supply a mixture of outdoor air and recirculated air.
3. Systems serving spaces with high occupancy densities.

The revised standard also specifies the requirements for any system that implements CO_2-based demand controlled ventilation. These minimum requirements are as follow:

1. The control algorithm must be designed to result in ventilation rates that are at least 90% of the rates calculated in accordance with equation 6-3 using the actual space occupancy.
2. The control system must measure the room CO_2 concentration in the occupied zone. The sensor for a given space may be placed in the return air duct within 1.5 m of the return air grille. A single CO_2 sensor may be used to represent a group of spaces that are expected to exhibit similar occupancy patterns, provided that the floor area of the group does not exceed 2500 m^2.
3. For systems serving multiple spaces, not all spaces served by the system need to be monitored. However, in spaces for which the CO_2 concentration is not measured, peak occupancy conditions must be assumed during all times that the system is operating.
4. The control system must measure outdoor CO_2 concentration at a point within 3 m at the outdoor air intake. Outdoor CO_2 concentration can be assumed to be constant at the annual average and determined based on historical data collected at or near the building site. It can be assumed to be 350 ppm if no such data are available.

5. If a single sensor is used to measure both room and outdoor concentrations, the sensor must have an accuracy of 100 ppm in the range of 300 ppm to 2000 ppm. If different sensors are used, then each must have an accuracy of 50 ppm in the range of 300 ppm to 2000 ppm.
6. The control algorithm must deliver outdoor air rates of at least 90% of set-point, determined in step 1 above, whenever the area served by the system is occupied.
7. The system must be designed to ensure that the minimum outdoor air intake for the building is maintained during scheduled occupancy periods and during pre-occupancy purge operation.

1.7 Objectives of the Study

This research project has two main objectives:

1. To experimentally and theoretically measure the effectiveness of CO_2-based demand-controlled ventilation strategies in saving energy in HVAC equipment under controlled laboratory conditions. These measurements will assess the impact of demand controlled ventilation systems on both energy use and contaminant (i.e. carbon dioxide) concentration in office buildings.
2. The experimental study will provide a test-bed to validate a computer simulation model that analyzes the effect of CO_2 controlled ventilation. The computer simulation will be used to determine the effectiveness of demand-controlled ventilation for a variety of representative office

building energy systems, building occupancy profiles, climates, and demand controlled ventilation strategies.

This work is the first extensive experimental study of the effectiveness of CO_2-based demand-controlled ventilation strategies in a controlled dynamic environment that provides repeatable test conditions that are otherwise available only in uncontrolled field test sites. Specifically, all the experimental work for this project will be conducted at the University of Colorado's full-scale commercial building HVAC Larson laboratory where extensive HVAC research has been conducted in the past.

1.8 Approach

The approach to the project involved three fundamental phases. The first phase included review and documentation of current practice in ventilation control. The second phase involved actual experimental investigation to quantify the impact of demand-controlled ventilation on the indoor air quality for a representative space of an office building. The third phase used the results of the experimental study to validate an HVAC/IAQ simulation model and to develop a set of guidelines specific to the application of CO_2-based demand controlled ventilation systems. The following sections describe the various tasks that were performed to achieve the objectives set for each phase.

1.8.1 Phase 1: Review of Current Practice in Controlling Ventilation

Task 1.1: Literature Review

The engineering literature was surveyed to review existing methods to control ventilation rates and provide acceptable indoor air quality. In particular, the ventilation procedures recommended by ASHRAE Standard 62-1989 were reviewed and documented. This review helped clarify the rationale of the Standard and diffuse some of the confusion concerning the issues of ventilation, carbon dioxide, and the Standard.

Task 1.2: Identification of CO_2-Based Demand Controlled Ventilation Strategies

The various strategies of controlling ventilation rates that are suitable for office buildings were identified, classified, and documented. The Standard 62-1989 permits controlling ventilation rates based on indoor contaminant measurements. In particular, the performance method of the ASHRAE Standard 62-1989 suggests that if the indoor concentration of CO_2 is below 1000 ppm (on average over a working day), then acceptable indoor air quality has been provided. However, some studies have suggested that the set-point of CO_2 concentration needs to be lowered to account for indoor pollution sources that are not related to occupancy. The control strategies that have already been identified include those ranging from simple approaches, such as purge controls and on-off controls, to more complex methods such as proportional controls. In the proportional control strategies, the outside air damper is modulated (in discrete jumps or continuously) to maintain carbon dioxide concentration below a set-

point level. The inherent advantages and disadvantages of each identified control strategy were documented with emphasis on the practicality of the approach.

1.8.2 Phase 2: Experimental Testing

Using the information collected in Phase 1, an experimental testing plan, including a reliable testing procedure, was developed to determine the impact of a demand controlled ventilation system on both energy use and indoor air quality in a representative office building with multiple zones and under various conditions. The test setup and procedure are described below:

Task 2.1 Test Facility Preparation

The Larson HVAC Laboratory at JCEM was used for the experimental work of this project. The facility is unique in that it permits the study of entire systems in a controlled dynamic environment, providing repeatable test conditions that have been heretofore unavailable. The laboratory has been designed for maximum flexibility to encourage a wide variety of research and testing procedures. In particular, both the HVAC and control systems in the laboratory are re-configurable in that components, subsystems, or entire systems can be readily installed, tested, and modified.

The JCEM laboratory provides an ideal setting in which to perform the proposed project. Specifically, the lab has two full size zones representative of typical spaces in an office building. The zones are insulated from the rest of the laboratory space with R-50 insulation in the ceiling and walls and R-30 insulation in the floor to thermally isolate the zones from the lab environment.

Time-varying loads can be imposed in these zones to simulate internally generated thermal loads, conduction heat transfer through walls and windows, and radiation gains due to solar radiation. In addition, the ventilation air introduced through the air-handling unit from the outdoors is preconditioned to simulate conditions essentially anywhere in the United States.

The laboratory was equipped with a CO_2-monitoring system to measure CO_2 levels inside the two full zones. To implement the CO_2-based demand controlled ventilation strategy, the CO_2 levels were continuously monitored and analyzed by a computer connected to the data acquisition system (DAS) to determine the proper outside air flow rate required to maintain acceptable indoor air quality. The Direct Digital Control (DDC) system adjusts automatically the outside air damper.

Task 2.2: Development and Performance of the Experimental Plan

The testing study was performed to investigate the effectiveness of various demand ventilation control strategies for a representative office space. A series of tests were conducted to determine the impact of these control strategies on outside air consumption. The setpoint level of CO_2 was fixed to 900 ppm as recommended by the ASHRAE Standard 1989. Tests were performed to include the variation of the following parameters:

1. **The ventilation strategy:** The ventilation control strategies that were tested include: fixed minimum outside air flow (base case), a purge on/off control strategy (system provides 100 % outside air whenever $C0_2$ level

exceeds the set-point concentration), and proportional-integral-derivative (PID) control strategy.

2. **The CO_2 sensor location:** The testing evaluated the performance of demand controlled ventilation systems for two cases: one sensor in the return duct, and sensors in both full zones. In particular, the effectiveness of the single sensor approach (located in the return duct) in maintaining acceptable indoor air quality in all zones was analyzed and compared with that approach that calls for several sensors (one sensor for each zone).

The series of tests described above was performed under controllable conditions in the HVAC laboratory at the University of Colorado. The laboratory is a facility unique to the HVAC industry that allows dynamic testing of a complete HVAC and building system.

Task 2.3: Data Analysis

Following the performance of all the tests, a sensitivity analysis depicting the effectiveness of various demand controlled ventilation strategies was conducted. The data analysis took into account the measurement errors for the various parameters. Based on this analysis, data emerged which permited evaluation of the effects of parameters such as pollutant set-point level on the performance of a CO_2-based demand-controlled ventilation strategy in maintaining acceptable indoor air quality while minimizing energy usage in office buildings.

1.8.3 Phase 3: Evaluation of Effectiveness of Ventilation Control Strategies

In this phase, the results of the experimental study were used and combined with a simulation tool to develop a complete set of guidelines for determining the effectiveness of CO_2-based demand-controlled ventilation strategies in saving energy in HVAC equipment. The following tasks were performed:

Task 3.1: Development of a HVAC/IAQ Simulation Environment

The HVAC/IAQ simulation environment was developed using FORTRAN 77 computer language. The algorithm predicts time-varying indoor contaminant levels. Since small time steps are needed, the multi-zone IAQ model used a one-minute time step. A model based on pollutant mass balance rather than on an airflow network is more suitable for this project. The proposed algorithm for modeling multi-zone pollutant transport was validated using the experimental work.

The multi-zone IAQ algorithm uses an iterative method to predict the concentration of CO_2 in each zone, assuming the inter-zone flows and the infiltration rates. The time step for the IAQ algorithm was set so that the concentrations would be accurately estimated. The algorithm was based on the mass conservation of the pollutant (CO_2) in each zone. The change of pollutant mass in each zone is simply the difference of the mass coming in and the mass going out. The pollutant entering the zone is due to infiltration, air supply, local generation (i.e., people), and air flowing from other zones. The pollutant exits the zone through the return duct or to other zones.

Task 3.2: Validation of Simulation Results

In this phase, the integrated energy/IAQ simulation tool was validated using the results of the experimental testing conducted in Phase 2. The predictions of the simulation model were compared with the testing results of the cases treated in Phase 2. During this validation study, the simulation model was adjusted and calibrated to provide accurate results that agree with the experimental data.

Task 3.3: Development of Test-bed of Simulation Runs

Using the calibrated energy/IAQ simulation tool to be developed in Task 3.1 and validated in Task 3.2, the effectiveness of various demand ventilation control strategies for a representative office building was evaluated. Many computer simulations were performed to allow investigation of a wide range of climate, CO_2 set-point levels, economizer cycle controls, and ventilation control strategies. In each simulation run, the following parameters were varied:

1. **The climate:** Four locations were selected for this project to represent various US climates. The locations included Miami (hot and humid), Phoenix (hot and dry), Denver (cold and dry), and Madison (cold and humid).
2. **The CO_2-based demand-control ventilation strategy:** The base case is the fixed minimum outside air flow. This minimum ventilation flow is determined using the ASHRAE 62-1989 Standard for an

office building. For this project, two additional demand controlled ventilation strategies were evaluated: the on/off control strategy, and the PID control strategy. In the on/off control option, the HVAC system switches to 100% outside air whenever the CO_2 concentration exceeds the set-point level. In the PID control approach, the flow of outside air is increased in proportion to the difference between the indoor concentration level and the set-point level. This increase is done continuously until the maximum ventilation flow is reached or the indoor concentration falls below the set-point value.

3. **The economizer cycle:** The performance of the demand controlled ventilation strategies is greatly affected by the economizer cycle control option. For this project, the impact of a temperature economizer cycle was evaluated.
4. **The CO_2 concentration setpoint:** The ASHRAE Standard 1989 suggests that CO_2 levels in office buildings should not exceed 1000 ppm. However, studies indicate that this level may not be enough to maintain an acceptable indoor air quality in some office buildings (Vaculik and Plett, 1993). A more stringent CO_2 level of 900 ppm is generally suggested as an alternative to better control indoor air pollutants.
5. **The CO_2 sensor location:** To determine the ventilation needed by the HVAC system to maintain acceptable CO_2 concentration levels in the building, three monitoring approaches are generally suggested. The

first approach calls for a single sensor located in the return air duct just before the AHU to measure a representative "average" CO_2 level for all the zones. The second approach involves monitoring the CO_2 level in all the zones. The ventilation rate provided by the HVAC system would meet the requirements of the zone with the highest CO_2 concentration. The third approach is to use the supply air concentration.

6. **The base minimum for outside air flow-rate:** Since CO_2 is only one of many pollutants that can cause indoor air quality problems, a minimum base for outside air flow is recommended to help dilute pollutants that are not directly related to CO_2 generation.

Task 3.4 Computer Simulation and Analysis Approach

The results of the computer simulations performed were analyzed and used to develop a set of guidelines to define the benefits of using CO_2-based demand controlled ventilation systems. In particular, the effectiveness of each ventilation control strategy in reducing heating and cooling energy use while maintaining acceptable indoor air quality was evaluated. A sensitivity analysis was conducted as part of the activities of this task. The sensitivity analysis included effects on control strategies of factors such as climate and economizer cycle.

The results of the evaluation study provided useful application guidelines for effective application of CO_2-based demand controlled ventilation systems in office buildings. In particular, it identified the specific system characteristics for which ventilation control strategies can maintain acceptable indoor air quality

with some energy cost savings over the current approach of using fixed minimum ventilation.

CHAPTER TWO

Literature Review

A demand-controlled ventilation (DCV) system attempts to achieve acceptable indoor air quality (IAQ) at reduced energy costs by controlling the outdoor airflow to the building based on measured parameters. These parameters can be values of indoor pollutant concentrations or measures of building occupancy based on occupancy sensors. In the case of DCV, CO_2 is used almost exclusively as an indicator of occupancy. Carbon dioxide sensors that measure the concentration are used to control the ventilation supplied to a particular zone. The main advantage of applying this strategy is that it saves energy by increasing the ventilation when occupancy is high and decreasing it when occupancy is low.

In the last 12 years, interest in CO_2 demand-controlled ventilation has led to numerous research studies that include applications, case studies and field tests, and computer simulations.

2.1 Applications

One of the earliest discussions of using CO_2 to control outdoor air intake as a means of saving energy was presented by Kusuda (1976). In his paper, he presented some of the theoretical equations of how indoor CO_2 concentrations vary as a ventilation system is modulated between on and off positions. The

equations permit the calculations of the maximum and minimum limits of CO_2 in a particular zone when the following ventilation parameters are known: natural air leakage characteristics of the zone, total ventilation rate of the mechanical system, number of occupants, CO_2 generation rate, room volume, and mixing parameters. According to Kosoda's calculations, potential energy savings of 40% for an office space were feasible.

Davidge (1991) and Houghton (1995) described the circumstances under which DCV might be expected to be most effective, including unpredictable variations in occupancy, a climate where heating or cooling is required for most of the year, and low pollutant emissions from non-occupant sources. Davidge pointed out that when such a system is considered, one must address the base ventilation rate that is not controlled by DCV in order to control these non-occupant pollutant sources. The impact of free cooling (economizer control) on DCV systems was also discussed. He pointed out that long periods of free cooling could reduce energy consumption. He also discussed the potential for purge ventilation, both before and after occupancy, to control non-occupant sources.

Vaculik and Plett (1993) presented specific descriptions of control algorithms in DCV. In particular, they discussed the principles of CO_2-based DCV, including set-point and proportional control. They discussed in detail the mathematical representations of both the on-off and the proportional control strategies. They also described a method which accounts for differences between CO_2 concentration at the measurement location and the critical location of the

building in which the control set-point is modified to account for differences between the measured concentration and the set-point.

Houghton (1995) presented information about using CO_2 to control ventilation and described the potential energy savings benefits in applying DCV. Strategies for the use of CO_2-based DCV were also described. These strategies included simple set-point control, where the outdoor air intake damper is either open or closed, depending on the indoor CO_2 concentration, and proportional control in which the intake damper or outdoor air fan flow is proportional to the CO_2 concentration. Recommendations were made on the application of these control strategies based on the occupancy level.

Elovitz (1995) discussed various options for controlling minimum outdoor air flow rates in VAV systems, including sequencing supply and return fans, controlling return or relief fans based on building pressure, measuring outdoor air flow rates directly, fan tracking, controlling the pressure in the outdoor plenum, using an outdoor air injection fan, and using CO_2 control. Advantages and disadvantages of each approach were discussed. Elovitz pointed out that CO_2 control does not necessarily ensure satisfactory indoor air quality. He pointed out that it depends on the existence and strength of other contaminant sources that are not proportional to the number of occupants.

Bjorsell (1996) presented a control algorithm of a DCV along with a simple simulation example. The control algorithm he described attempts to calculate the optimal system flow to minimize a cost function that depends on concentration and ventilation flow. Bjorsell's study did not specify the cost function.

Federspiel (1996) described a control algorithm called on-demand ventilation control (ODCV). This control algorithm attempts to set the ventilation rate proportional to the occupant density by using a well-mixed single-zone model to estimate the current CO_2 generation rate from measured concentrations and airflows. He presented a simple illustration to show how his ODCV strategy controls the CO_2 concentration below a setpoint of 1,000 ppm.

2.2 Field Tests

There have been many case studies in which CO_2-based DCV systems were installed and monitored in buildings. These case studies included different building types such as offices, conference rooms, schools, and auditoriums.

One of the earliest studies of CO_2-based DCV in office buildings took place in Helsinki (Sodergen, 1982). The CO_2 control system was compared to constant outdoor air and timer-based control, and plots of CO_2 concentrations were generated. The outdoor air control algorithm was not described in this study.

Gabel et al. (1986) presented a study that took place in a bank in Pasco, Washington. The project involved the measurement of energy consumption and contaminant levels including nitrogen dioxide, carbon monoxide, and particulates. Occupant responses to the indoor air quality were also surveyed. The indoor air quality was monitored in winter, spring, and summer seasons, with one week under normal operation followed by one week of CO_2-based DCV. At a CO_2 set-point of 1,000 ppm, they found that air leakage through the closed damper was sufficient enough to keep the CO_2 concentration below the

set-point. They also found that all measured contaminants were maintained below the indoor standards. The savings in energy consumption were calculated to be 7.8% when the CO_2 control strategy was implemented. According to the questionnaire, the occupants were not able to detect any differences in air quality at CO_2 levels of 300 and 1,000 ppm.

Davidge (1991) presented a CO_2-based controlled DCV in a 30,000 m^2 office building. According to his study, the system never controlled the ventilation rate because the outdoor temperatures during winter were not low enough to allow free-cooling. During the summer, however, the damper leakage was more than enough to control the CO_2 level.

A comprehensive study of CO_2-based DCV took place in Montreal (Donnini et al. 1991) on two floors of an office building. One floor was equipped with a CO_2 DCV system, while the other floor served as a control. The CO_2 control algorithm was set so that the outdoor damper was closed at concentrations below 600 ppm and opened at concentrations above 600 ppm, with the maximum opening occurring at 1,000 ppm. In addition to measuring the CO_2 concentration, the study measured formaldehyde, volatile organic compounds (VOC's) and particles, ventilation system performance, thermal comfort, and occupant perception, and energy demand. The outdoor air dampers were kept closed most of the year because there were not enough people to raise the CO_2 concentration above the limit. The thermal comfort was generally adequate enough on both floors. Occupants of the DCV floor complained significantly more about the

indoor environment than those occupants of the control floor. There were 12% annual energy savings, however, for the DCV.

Fleury (1992) performed a study on the performance of a CO_2-based DCV in a conference room. According to his study, the fan motor speed was adjusted based on the CO_2 concentration. Occupants rated the air quality in the conference room from good to excellent. The measured CO_2 concentration in the space was between 350 and 850 ppm.

2.3 Computer Simulations

The reported simulation case studies vary widely in both the description of important parameters and the discussion of the results. Most studies have focused on the potential energy savings of the CO_2-based DCV systems, with CO_2 concentrations reported as a measure of IAQ performance. A few studies have calculated concentrations of other pollutants. The majority of this research has involved office buildings, with others examining public spaces and residential buildings. One important issue discussed in computer simulation studies is the treatment of infiltration and inter-zonal air-flows. Most of the studies, however, assumed rates and inter-zonal air-flows.

Knoespel et al. (1991) investigated the application of a CO_2-based DCV system in a two-zone office building. He investigated both constant-air-volume (CAV) and variable-air-volume (VAV) HVAC systems. A multiple-zone pollutant transport model was used and a ventilation airflow controller model was developed as modules for a transient thermal system simulation program.

Knoespel compared the performance of six ventilation strategies: (1) including constant outdoor airflow at the ASHRAE Standard 62-1989 prescribed flow of 20 cfm/person, (2) constant outdoor airflow at a typical rate of 0.7 hr^{-1}, (3) minimum outdoor airflow at the typical rate with temperature-based economizer, (4) DCV with a step-flow control algorithm, (5) DCV with step-flow control and a temperature-based economizer, and (6) DCV with on-off control. In the step-flow control algorithm, the fraction of outdoor air in the circulation flow was changed in 20% steps, depending on whether the measured CO_2 concentration in either zone was above or below the specified limit. On-off control employed an algorithm in which outdoor airflow was set at 100% if the high CO_2 set-point was exceeded, and at 0% if the CO_2 concentration dropped below the low set-point. The set-points used were 800 and 1,000 ppm. Simulations were performed for Miami, Florida and Madison, Wisconsin. In Madison, the DCV strategies provided acceptable control of CO_2 levels with coil energy savings from 9% to 28% for CAV systems and from 43% to 46% for VAV systems compared to the ASHRAE Standard 62-1989 prescribed rate strategy. The savings for Miami were of similar absolute magnitude but of smaller percentages. These results did not include fan energy use. Compared to the economizer and constant outdoor airflow strategies resulted in similar energy use with better control of CO_2 concentrations for both CAV and VAV systems.

Emmerich et al. (1994) used the same model developed by Knoespel et al. (1991) to examine the performance of DCV on non-occupant-generated pollutants. Emmerich used the same building in Madison and the same HVAC

systems described above, but varied the conditions simulated to include a pollutant removal effectiveness as low as 0.5 and an occupant density of 50% greater than design. For all cases examined, the DCV system reduced the annual cooling and heating loads from 4% to 41% while maintaining acceptable CO_2 concentrations. In addition to requiring more energy use, the constant outdoor airflow strategy resulted in CO_2 levels above 1,000 ppm for more than half of the occupied hours for cases with poor pollutant removal effectiveness. Emmerich also examined the impact of DCV on non-occupant-generated pollutants by modeling a constant source of a non-reactive pollutant located in the main office zone. Four ventilation strategies were compared: (1) constant outdoor air at a prescribed rate (based on ASHRAE Standard 62-1989), (2) DCV with step control and set-points of 800 and 1,000 ppm, (3) DCV with a constant minimum outdoor airflow rate of 5 cfm per person (calculated using the multiple space methods of ASHRAE Standard 62-1989), and (4) DCV with scheduled purges of 100% outdoor air from 7:30 a.m. to 8:30 a.m. and 12:30 p.m. to 1:00 p.m. The non-occupant-generated pollutant source strength was specified such that the system with a constant outdoor airflow rate just met a short-term limit of 2 ppm and an eight-hour average limit of 1 ppm. Emmerich found that both the constant DCV and the DCV with minimum outdoor airflow rate failed to meet the pollutant concentration limits for both the CAV and VAV systems, but the DCV with scheduled purge strategy successfully limited the pollutant concentration. The purge strategy increased building heating and cooling loads over the constant DCV strategy, but still reduced the loads by 17% (CAV) and 25% (VAV)

compared to the constant outdoor airflow case. The success of the purge strategy was attributed partially to the ability to schedule the purges when most needed.

Haghighat et al. (1993) investigated the effects of poor ventilation air mixing by simulating the performance of a CO_2-based DCV system in a large office building in Montreal. The baseline ventilation system had a flow rate of 20 cfm per person. A mixing parameter of 0.7 was used in the model. The DCV system used a minimum ventilation rate of 5 cfm per person, and the ventilation rate was adjusted each hour to maintain a CO_2 concentration of 800 ppm. Infiltration was 0.4 per hour. Four cases of occupant density were examined. The DCV system saved from 7% to 15% in energy use, from 2% to 6% in energy costs, and from 7% to 17% in peak demand compared to a fixed ventilation rate strategy. In a follow-up study using the same office model with different infiltration, operating hours, and other assumptions, Zmeureanu and Haghighat (1995) found energy consumption for the DCV system ranging from a 5% decrease to a 2% increase. However, because of peak demand reductions, annual energy cost savings ranging from 3% to 26% were found.

Meckler (1994) simulated the application of CO_2-based DCV in an office building. The energy performance of a DCV system with the ventilation rate varied to maintain 800 ppm and 920 ppm was compared to a baseline system with a constant ventilation rate of 20 cfm per person. The office building had 10 floors with two air-handling units (AHU) for outdoor air on each floor, a central hydronic heating and cooling plant, and an economizer. Both energy and economic impacts were presented for five U.S. cities (Miami, Atlanta,

Washington, D.C., New York, and Chicago). Reported energy savings ranged from less than 1% to 3% for electricity and from 16% to 22% for gas. Payback periods of 1.5 to 2.2 years were estimated for all cities.

Sorensen (1996) described simulations performed for a two-zone office with a conference room. His study focused on examining the short-term dynamics of the system by simulating a 10-hour period with one-second time steps and detailed modeling of the HVAC system. A VAV system with dual-temperature and CO_2 control was simulated. Because a detailed VAV system model was used, the control algorithm was more complex than in most studies reviewed and involved both dampers and fans. When the CO_2 concentration was above an upper limit of 900 ppm, the damper actuator position increased by 1%. If the concentration remained above the upper limit, the position continued to increase until it was fully open or until it dropped below the limit. After the damper was fully open, a concentration above the upper limit increased the fan speed by 5% until the fan reached maximum speed or the concentration fell below the limit. The algorithm also used a lower limit of 700 ppm to decrease fan speed and damper position. According to this study, the VAV system used 31% less energy than the CAV system.

Carpenter (1996) examined both the energy and the IAQ impacts of CO_2-based DCV in a mid-sized commercial building complying with ASHRAE Standard 90.1 in four climatic zones (Chicago, Nashville, Phoenix, and Miami). Simulations were performed using a combination of an energy analysis program (EE 1990) and the multi-zone pollutant transport program CONTAM87 (Axley

1988). Three HVAC systems (single zone, multi-zone, and VAV) and five ventilation control strategies (fixed ventilation rate, DCV with building return air controlled to 1,000 ppm and 800 ppm, DCV with floor return controlled to 1,000 ppm, and DCV with each zone controlled to 1,000 ppm) were analyzed.

For single-zone systems, the DCV strategy reduced heating energy by about 30% for a setpoint of 1,000 ppm and by 20% for a set-point of 800 ppm. The DCV system with a set-point of 800 ppm also reduced average CO_2 concentrations by 50 ppm to 90 ppm compared to the fixed ventilation rate strategy. The DCV strategies had little effect on cooling energy because the DCV system tended to reduce ventilation during the cooler morning and evening hours and increase ventilation during the warmer mid-day periods. For VAV systems, the energy savings were similar to those with single-zone systems. For multi-zone systems, the reduction in heating energy was similar in absolute terms, but was smaller in percentages (5% to 12%) because of a larger total heating load. DCV with a set-point of 1,000 ppm resulted in average CO_2 concentrations 70 to 150 ppm higher than the fixed ventilation strategy, while a set-point of 800 ppm kept concentrations lower than the fixed strategy, and kept the maximum concentration below 1,000 ppm in all zones. Providing additional sensors in return ducts on each floor had little impact on energy use and IAQ. Installing sensors in each zone ensured that the concentration in each zone stayed below 1,000 ppm, but yielded a slightly higher energy use. The performance of DCV with sensors set at 1,000 ppm in each zone was similar to central control with a set-point of 800 ppm. Formaldehyde concentrations were also simulated to

evaluate the impact of DCV strategies on pollutants from non-occupant sources. None of the DCV strategies controlled the formaldehyde concentrations as well as the fixed ventilation strategy. It was suggested that a morning purge should be included in a DCV strategy when non-occupant-generated pollutants are a concern, but this option was not simulated. Different DCV control algorithms, including on-off, linear proportional, proportional-integral-derivative (PID), and the Vaculik method were discussed but not simulated.

Shirey and Rengarajan (1996) simulated the impact of a CO_2-based DCV system in a 400-m^2 office building located in Miami, Orlando, and Jacksonville to examine the impacts of ASHRAE Standard 62-1989 ventilation rates on indoor humidity levels. The baseline system, a conventional direct expansion (DX) air-conditioning system with a sensible heat ratio of 0.78, was unable to keep the indoor humidity below the target of 60% relative humidity (RH) when the ventilation rate was increased from 5 cfm to 20 cfm per person. System modifications considered included a low-SHR DX air conditioner, a high-efficiency low-SHR air conditioner, a conventional air conditioner with CO2-based DCV, a conventional air conditioner with an enthalpy recovery wheel, a heat-pipe-assisted air conditioner, and a conventional air conditioner with a separate 100% outside air DX unit. The operation of the DCV system was simulated by matching ventilation rates to occupancy profiles. Four alternative systems (DCV, enthalpy wheel, heat pipe, and 100% outdoor air DX unit) maintained acceptable humidity levels for more than 97% of the occupied hours. Of the systems with acceptable humidity performance, only the DCV and

enthalpy wheel options did so with less than 5% increases in annual HVAC energy use, compared to the conventional system with a ventilation rate of 5 cfm per person. The DCV system also significantly lowered the peak heating demand in Orlando and Jacksonville. An economic analysis showed that the DCV system resulted in annual HVAC operating cost increases of 7% or less, first cost increases of about 14%, and life-cycle cost increases of about 12% compared to the system with 5 cfm per person.

Davanagere et al. (1997) applied the same methodology with many of the same assumptions as Shirey and Rengarajan (1996) to study HVAC system options, including CO_2 –based DCV in a Florida school. As in the previous study, the baseline for comparison was a conventional system with ventilation as required by ASHRAE Standard 62-1981. In addition to DCV, the options simulated included the conventional system with ASHRAE Standard 62-1989 ventilation rates and various combinations of pre-treating outdoor air, thermal energy storage, enthalpy recovery wheels, gas fired desiccant systems, and cold air distribution systems. Results reported included energy use, humidity levels, first costs, and life-cycle costs. In general, the DCV system resulted in the smallest increases in energy costs and installed first costs compared to the baseline system. The thermal energy storage system options generally resulted in the smallest increases (or even decreases) in peak cooling demands and life-cycle costs. DCV was the only option that reduced peak heating demands. Although the DCV system reduced humidity levels compared to the baseline system, many of the other simulated options controlled humidity even better.

Residential applications have also been simulated by several researchers. Yuill and Jeanson (1990) described a simulation study of DCV in a single-family residence. The simulations were performed using a combination of CONTAM87 and a multizone airflow program (Walton, 1989). The house modeled was based on a house in Winnipeg, which is a single story, three-bedroom with a basement and four occupants. All interior doors were modeled as open, and all exterior doors and windows were modeled as closed. Simulations were performed for weather data for a typical day. They compared the IAQ performance of four ventilation strategies, including a heat recovery ventilator (HRV) with continuous ventilation of 62 cfm, HRV with CO_2 DCV, a laminar air-flow super window (LAFSW) with continuous ventilation of 62 cfm, and a LAFSW with DCV. The LAFSW draws air into the house at the bottom of a window. The air flows between the panes of glass and is discharged into the house at the top of the window. Concentrations of radon, formaldehyde, and an arbitrary pollutant generated from six point sources were also calculated. The DCV systems were modulated such that the CO_2 concentration in the exhaust flow was maintained at a constant 800 ppm. It was concluded that the continuous HRV system provided the best control of radon. The continuous control outperformed the DCV in controlling formaldehyde, and the DCV strategies performed significantly better at controlling human-generated pollutants. Finally, the continuous LAFSW system performed best at minimizing occupant exposure to point source pollutants.

Hamlin and Cooper (1991) used an energy analysis program combined with CONTAM87 to examine the energy impacts of a CO_2-based DCV system in a single-family house located in Toronto, Winnipeg, and Vancouver. Ventilation strategies simulated included continuous exhaust only, continuous balanced flow with heat recovery (HRV), and DCV with the set-point equal to the heating-season average CO_2 concentration with continuous ventilation (matched-peak strategy). Simulations were performed with both three and six occupants, with a natural infiltration constant at 0.1 h^{-1}, and baseline system ventilation plus infiltration rates of about 0.3 air changes per hour (ACH). The results considered impacts on both space heating energy (assuming gas heat) and fan energy. The DCV systems resulted in total energy savings from 4% to 7% for the matched-average strategy and from 14% to 15% for the matched-peak strategy compared to the continuous exhaust strategy. Cost savings per year ranged from about $20 to $70. The HRV resulted in larger space-heating energy savings than the DCV system, but required more fan energy. The net cost savings of the HRV and DCV systems were similar. The peak CO_2 concentrations for the DCV cases ranged from 655 ppm to 823 ppm for the cases with three occupants.

Yuill et al. (1991) compared the performance of four ventilation strategies including central continuous, distributed continuous, central DCV (set-point 632 ppm), and distributed DCV (set-point 658 ppm) for the same houses, weather data, and pollutant sources described above. The DCV strategies showed no clear advantage over continuous ventilation, with reduced exposure to some pollutants and increased exposure to others. In some cases, the relative performance of

DCV depended on whether the supply was central or distributed. As mentioned above, the concentrations at the end of the day did not match the concentrations at the beginning.

In addition to offices and residences, public spaces have also been investigated for DCV. Ogasawara et al. (1979) evaluated the potential energy savings for a DCV system in a 30,000 m^2 department store in Tokyo, Japan. Three ventilation strategies were evaluated, including fixed outdoor air at design rate, manual control with maximum ventilation on Sundays and half of that on weekdays, and DCV. The DCV algorithm used was proportional control with a closed damper at 800 ppm and a fully opened damper at 1,000 ppm. Infiltration assumptions were not specified. Energy use was calculated for four cooling months and four heating months. The DCV system reduced energy use by 40% for the cooling season and by 30% for the heating season. An economic analysis showed an advantage for the DCV system.

Waren and Harper (1991) evaluated the subject of DCV simulation studies. They evaluated the potential heating energy savings for a CO_2-based DCV system applied to an auditorium in London. Energy simulations were performed using a building energy analysis program (Clarke and Mclean 1986), with ventilation rates calculated separately based on occupancy profiles. Three ventilation strategies were compared, including 100% outdoor airflow at a rate of 10,000 cfm, DCV with a minimum outdoor airflow rate of 7,500 cfm, and DCV with no minimum air flow rate. The DCV with a minimum outdoor airflow rate rarely exceeded the minimum rate to maintain CO_2 concentrations below 1,000

ppm and saved 26.4% in heating energy use compared to the 100% outdoor airflow case. The DCV with no minimum air flow rate saved 53.3%.

Brandemuehl and Braun (1999) developed a simulation model to predict the energy requirement associated with alternative ventilation control strategies in a variety of commercial building types. The strategies included different combinations of economizer and demand control ventilation strategies. In their model, no specific control strategy was used to control CO_2 concentration. That is, when demand control ventilation is enabled, a minimum flow rate of ventilation air is determined that will keep the CO_2 concentration in the return duct at or below a specified setpoint (1000 ppm). When the concentration is below the setpoint, a minimum ventilation flow rate is provided as determined using ASHRAE 62-1989. Four different economizer options were analyzed: the minimum fixed outside air with no economizer, with dry bulb economizer, with single enthalpy, and with differential enthalpy. The demand-controlled ventilation options included DCV without economizer, with dry bulb, with single enthalpy, and with differential enthalpy. Four different types of buildings were considered in the study: office, retail store, school, and restaurant. Twenty cities were chosen for the analysis. All runs were based on hourly simulations.

In their results, they indicated that the greatest savings in all cases will occur for the combination of differential enthalpy economizer and demand controlled ventilation when compared to no economizer and demand controlled ventilation. The maximum savings were about 23% for Atlanta and Madison and 14% for Albuquerque. The results showed that savings associated with demand controlled

ventilation is much more significant for heating than for cooling, since economizer operation does not play a role. That is, the reduced ventilation requirements directly decrease the heating loads and lead to very significant savings. The percent reduction in annual heating energy for demand controlled ventilation with no economizer, when compared to the base case (minimum outside air with no economizer), were 27%, 38%, and 42% for Madison, Albuquerque, and Atlanta, respectively.

2.4 Conclusions

In conclusion, the CO_2-based demand controlled ventilation subject has been the interest of a large number of researchers in the last 10 years. The studies performed included field tests, computer simulations, and applications. Many buildings were investigated, such as offices, residences, conference rooms, cinemas, auditoriums, and theaters. Many researchers concentrated on the CO_2 DCV impact on energy savings. However, a small number of studies described their control algorithm. Among the control algorithms found in the literature are the on/off and the proportional control strategies. No experiments on the effectiveness of CO_2-based demand control ventilation have yet been performed in a controlled dynamic environment that provides repeatable test conditions.

CHAPTER THREE

HVAC laboratory Setup and Preliminary Experiments

3.1 Laboratory Setup

3.1.1 Larson Building Systems Laboratory

All experiments were performed in the Larson Building Systems Laboratory at the Joint Center for Energy Management (JCEM). The mechanical systems of this lab consist of the primary plant: chiller, boiler, and primary pumps; the secondary HVAC system: ducts, air handlers, secondary pumps, and coils; and the thermal load generators: two-zone simulators and two full-sized zones. Figure 3.1 is an isometric view of the laboratory showing the major pieces of HVAC equipment and the zones. The main HVAC system consists of a 12,000-cfm air-handling unit (AHU) connected to the four zones by variable-air volume (VAV) fan-powered mixing boxes. The cooling capacity of the air-handling unit (AHU) is 20 tons whereas the heating capacity is 12 tons. Heating or cooling is accomplished by water-to-air heat exchangers.

The AHU is supplied with chilled water via a 42-ton cooling capacity chiller while heat is rejected from the chiller either through the air-cooled condenser located outside the lab or through a heat recovery water condenser attached to the chiller. The heat recovery condenser, with 43 tons heating capacity, supplies the hot water to generate zone loads. Hot water can also be supplied by an electric

boiler. This boiler can provide 308 lb/h steam at 10 psig with 90 kW electric input. The HVAC system controls conditions in four representative building zones. Each zone is provided with a parallel fan-powered mixing box to deliver VAV air distribution. These boxes are also equipped with 10-kW electric heaters for supplemental zone heating. Two zones are full-size spaces, each with 400 ft^2 floor area. To thermally isolate the full-size zones from the lab environment, each zone is insulated using R-50 insulation in the ceiling and walls, and R-32 insulation in the floor. Each full-size zone is equipped with six baseboard heating panels to generate the heating load. The other two zones are zone simulators that each consists of a set of heating and cooling load coils. Zone loads can be programmed for these zone simulators to follow prescribed load profiles. The thermal behavior of these two zones mimics that of 2000 ft^2 floor area zones for an actual commercial building. Ventilation air introduced to the main air-handling unit from outdoors is preconditioned to simulate conditions anywhere in the United States. This outdoor air is conditioned by a separate 12,000-cfm air-handling unit called the outdoor air conditioning station (OACS), which consists of a 30-ton cooling coil, a 12-ton hot water heating coil, and a 10-ton steam preheat coil. Internal face-and-bypass damper is used for the in OACS controls. The OACS is controlled to maintain a prescribed temperature profile independent of the airflow rate.

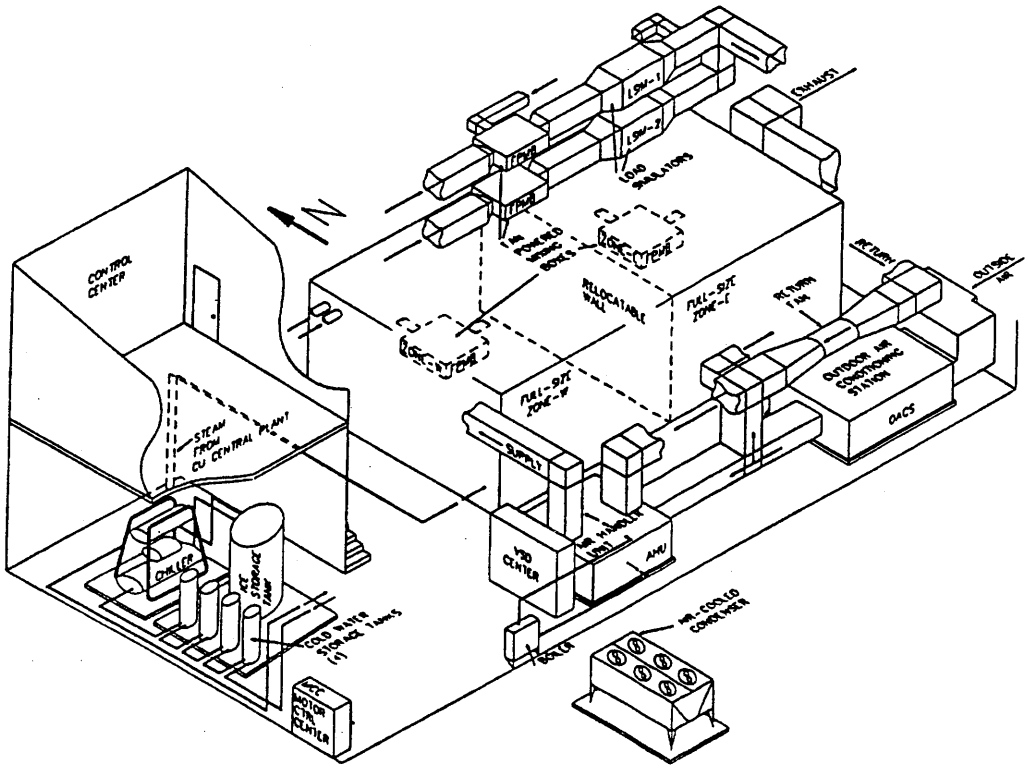


Figure 3.1: Isometric View of the Larson Laboratory Including Full-size Zones and Major Pieces of HVAC Equipment

The HVAC system is controlled using programmable direct digital control (DDC). The DDC system uses electronic actuators for damper and valve control, all of which accept standard analog control signals. Other pneumatic actuators are also available in the lab.

The data acquisition system (DAS) accommodates 360 data channels. Data collected include temperature, humidity, pressure, flow rate, fan speed, electric power consumption, damper position and control, equipment ON/OFF and CO_2 level. The last 60 channels are virtual channels that the experimenter can use to immediately calculate some other parameters. For example, a virtual channel might be created to calculate the outside air fraction based on two fixed channels that are used to measure the recirculated and outside air flows.

3.1.2 Carbon Dioxide (CO_2) Sensors

Three types of CO_2 sensors were selected to measure the carbon dioxide level in the zones. One type is manufactured by Engelhard Corporation and has a range of 0 to 2000 ppm with an accuracy of ±50 ppm. The second type of sensor is manufactured by Honeywell Company and has a range of 0 to 2000 ppm with an accuracy of ±50 ppm. The third type is manufactured by Johnson Controls with an accuracy of ±50 ppm. A preliminary experimental test showed that the first two types of sensors give the same readings when subjected to the same CO_2 generation rate. Problems encountered with the Engelhard sensor lead to their replacement by sensors from Honeywell Company.

3.1.2.1 Carbon Dioxide (CO_2) Sensors Calibration

All CO_2 sensors were calibrated against a known CO_2 concentration contained in prefilled cylinders. The calibration gas was acquired from Air Liquide Company in New Jersey. The sensors were calibrated at four points: 0 ppm, 300 ppm, 700 ppm, and 1000 ppm. The calibration with the 0 ppm was performed using a pure SF_6 gas that contains no concentration of CO_2.

3.1.2.2 Calibration Test Procedure

A total of eight (8) CO_2 sensors were placed in a testing bag that has one inlet for CO_2 calibration gas and one outlet to remove the excess gas and keep pressure in the bag at ambient pressure. The specification of each sensor is shown in Table 3.1. As mentioned earlier, to make the calibration, four calibration points were selected. The calibration at each point was repeated several times to measure the repeatability of the sensors. Specifically and to ensure their repeatability, the sensors were calibrated from one first point to a second point and then from a third point back to the first point. In total, there were nine tests: three at 0 ppm, two at 300 ppm, two at 700 ppm, and two at 1000 ppm.

Table 3.2 shows the calibration results of the sensors. The following observations can be made:

1. The sensor readings in Table 3.2 represent an average value of the sensors over the last 10 minutes after the sensors reached their steady-state conditions.

2. After the first test, sensor E5 was damaged because it read constant high values throughout the remaining tests.
3. The test duration varies from one test to another to allow the sensors to reach their steady-state values and become stable. It was noticed that sensors 7 and 8 reach the reading 0 ppm relatively quickly (within 5 minutes only) compared to the other readings. The system sensor, on the other hand, reached the steady-state value within only one minute for all concentrations used in the calibration test (response time is only one minute). All Honeywell sensors (sensors 1, 2, and 3) reach their steady-state values nearly at the same time. The response time of these sensors is about 6-8 minutes.

Figures 3.2, 3.3, and 3.4 show the calibration for each sensor grouped by the manufacturer. The equations in each graph represent the calibration equation for each sensor whereas the R^2 value is the correlation coefficient.

Table 3.1 Sensors Specifications

	Carbon Dioxide (CO_2) Sensor			
Manufacturer	Engelhard		Honeywell	Johnson
Detection Method	Photoacoustic	Infrared	Photoacoustic	Photoacoustic
Range (ppm)	0-2000	0-2000	0-2000	0-2000
Accuracy	±2.5% of reading or ±50 ppm (E4), ±80 ppm (E5); whichever is greater	±5.0% of reading or ±50 ppm, whichever is greater	±50 ppm	±20 ppm
Drift	Data not available	±20 ppm max per year	±100 ppm max in 5 years	±20 ppm max per year
Operating Temperature (oF)	60-90 F	32-122	59-104 °F	32-122
Operating Humidity	0-95% RH non-condensing	5-95 RH non-condensing	15-95 RH non-condensing	5-95 RH non-condensing
Operating air Velocity (ft/min)	Data not available	Data not available	<200 ft/min	Data not available
Temperature Influence	±0.05% of reading/F	±0.1% per °C or ± 1 ppm per °C, whichever is greater	<3.4 ppm/°F	±0.05% of reading/F
Humidity Influence	Data not available	Data not available	1.2 ppm/%RH	Data not available
Data Collection	Through system	Databear Software	Through system	Through system
Number of Sensors and Designation	2 (E4,E5)	2 (E7,E8)	3 (H1,H2,H3)	1 (J6)

Table 3.2: CO_2 Sensor Calibration

Test	Calibration Point (ppm)	Test Duration (min)	Sensors Readings (ppm)							
			Sensor H1	Sensor H2	Sensor H3	Sensor E4	Sensor E5	sensor J6	Sensor E7	Sensor E8
1	0	50	102	24	32	21	98	8	0	0
2	0	60	109	55	55	29	damaged	16	0	0
3	0	60	101	54	54	30	damaged	13	0	0
4	300	74	305	305	282	223	damaged	258	153	147
5	300	75	336	344	304	258	damaged	258	260	159
6	700	75	672	711	640	513	damaged	531	631	515
7	700	77	664	726	640	523	damaged	524	638	539
8	1000	80	922	1000	890	765	damaged	726	898	825
9	1000	78	929	1008	890	741	damaged	726	950	834

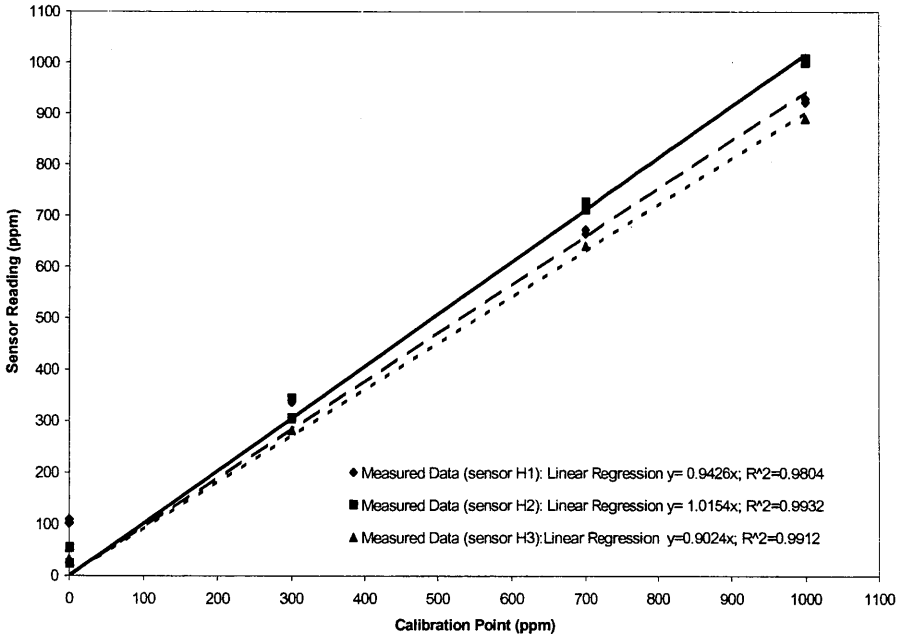


Figure 3.2: Honeywell Sensors (H1, H2, and H3) Calibration Results

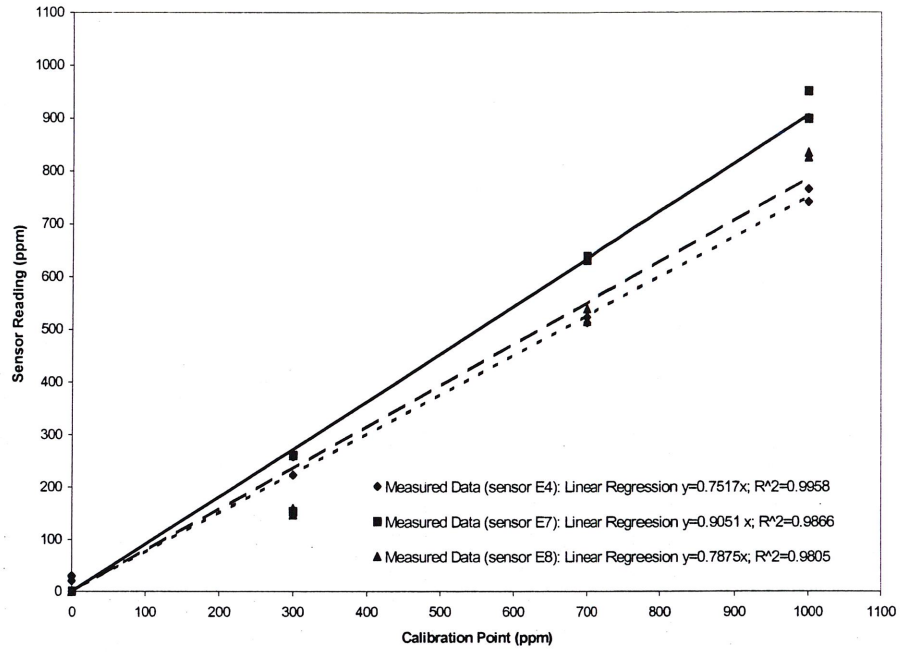


Figure 3.3: Engelhard Sensors (E4, E7, and E8) Calibration Results

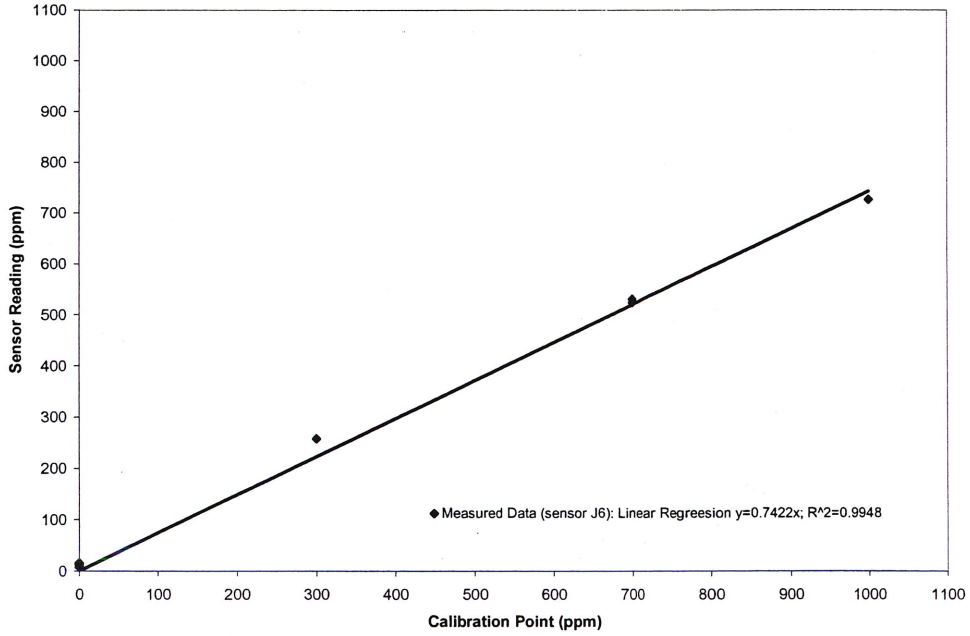


Figure 3.4: Johnson Sensor (J6) Calibration Results

3.1.2.3 Sensors Repeatability

The repeatability of the sensors was measured and compared to the declared manufacturers' values. The repeatability is measured as follows:

1. The average value of sensor reading is calculated for all tests at each calibration concentration level.
2. The percent error is calculated between the average value and the actual reading for a given CO_2 concentration level. This percentage represents the measured repeatability of the sensor at the given calibration level.
3. The measured repeatability is compared to the declared manufacturers' repeatability values and the required repeatability to make a decision of the sensor's repeatability at each calibration point. That is, the sensor is considered repeatable at the calibration point if its measured repeatability is less or equal to 2%.

Table 3.3 shows the error percentage, the manufacturers' repeatability values, and the results of the comparative analysis between measured and declared repeatability values for all the CO2 concentration levels used in the calibration test.

The following conclusions can be drawn from the calibration results for the CO_2 sensors:

Table 3.3: Measurements of Sensors Repeatability

Calibration point (ppm)	Calculated Sensor Repeatability (%)						
	Sensor H1	Sensor H2	Sensor H3	Sensor E4	Sensor J6	Sensor E7	Sensor E8
0	1.4	31.8	21.8	14.9	23.2	0.0	0.0
300	4.8	6.0	3.8	7.3	0.0	0.0	0.0
700	0.6	1.0	0.0	1.0	0.7	0.6	2.3
1000	0.4	0.4	0.0	1.6	0.0	2.8	0.5
Manufacturer's repeatability	2% or ± 40 ppm	2% or ± 40 ppm	2% or ± 40 ppm	2.5%	1%	2.5%	2.5%
Required Rep. At 1000 ppm	2%	2%	2%	2%	2%	2%	2%
Rep. At 0 ppm	YES	NO	NO	NO	NO	YES	YES
REP. AT 300 ppm	NO	NO	NO	NO	YES	YES	YES
REP. AT 700 ppm	YES	YES	YES	YES	YES	YES	NO
REP. AT 1000 ppm	YES	YES	YES	YES	YES	NO	YES

1. Only sensors H1, E7, and E8 are repeatable for the 0 ppm of CO_2 concentration.
2. All sensors are repeatable in ranges above 300 ppm, except sensor E7.
3. Because the range of CO_2 concentrations typically considered in the subsequent tests for demand ventilation controls is between 500 ppm and 1000 ppm, all sensors can be used, except sensors E7, and E8.
4. In general, similar sensors tend to behave similarly in terms of accuracy and repeatability. The Honeywell sensors (H1, H2, and H3) are both repeatable and accurate for CO_2 concentrations between 500 ppm and 1000 ppm. The Johnson sensor (J6) is very repeatable, but not accurate. The infrared Engelhard sensors (E7, and E8) are repeatable only at 0 ppm, while not repeatable at any other point. Their accuracy is bad. The photo-acoustic Engelhard sensor (E4) is repeatable, but not accurate.

Another test was conducted by placing all sensors at one point in the West zone and subjected to a variable amount of CO_2 for a period of about 10.5 hours. The concentration of each was then calibrated against the best sensor (the Honeywell sensor with the highest correlation coefficient (R^2), that is sensor H2). The variation of CO_2 concentration for all sensors is shown in Figure 3.5.

As noted in the Figure 3.5, all sensors follow the same pattern of concentration, however, this is not an indication of the accuracy of the sensor. To measure the accuracy, each has been calibrated against the H2 sensor as shown in the following figures. Two graphs were developed for each sensor: one is for the concentrations at all ranges (that is concentration below and above 1000 ppm), and the other is at concentrations below 1000 ppm, which is the maximum concentration a sensor is expected to read at any experimental test. The calibration equation obtained from this second graph was used as the final calibration equation entered in the system to measure CO_2 concentration.

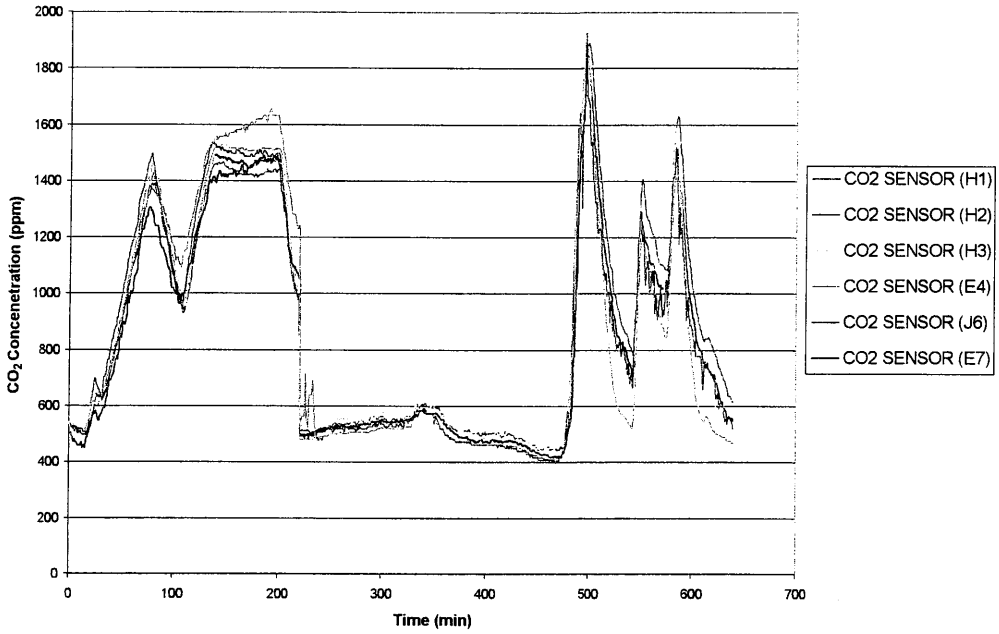


Figure 3.5: Calibration of CO_2 Sensors for Variable Concentration Levels

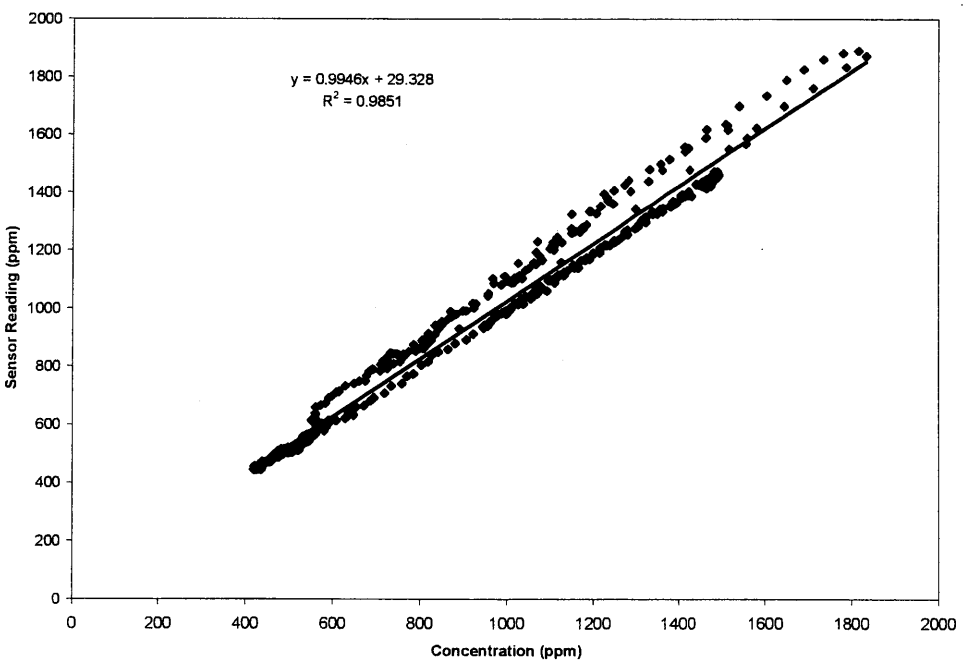


Figure 3.6: Calibration of Honeywell Sensor H1

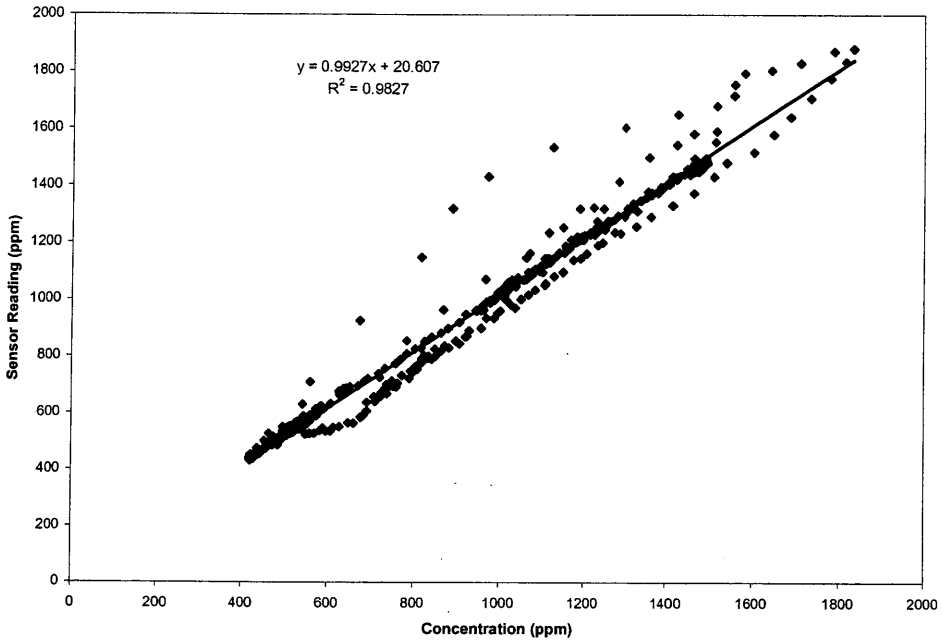


Figure 3.7: Calibration of Honeywell Sensor H3

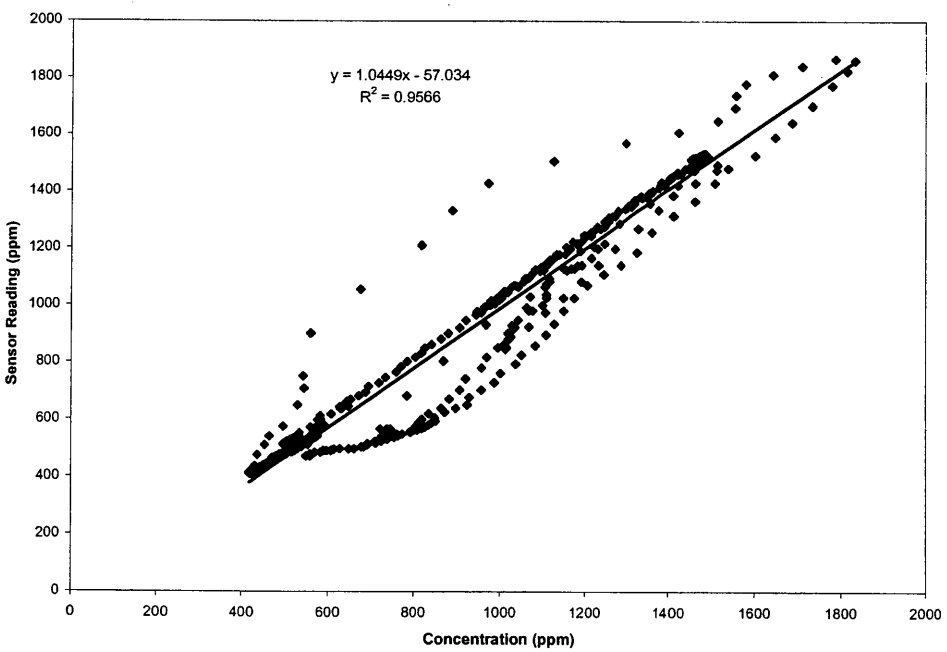


Figure 3.8:Calibration of Engelhard Sensor E4

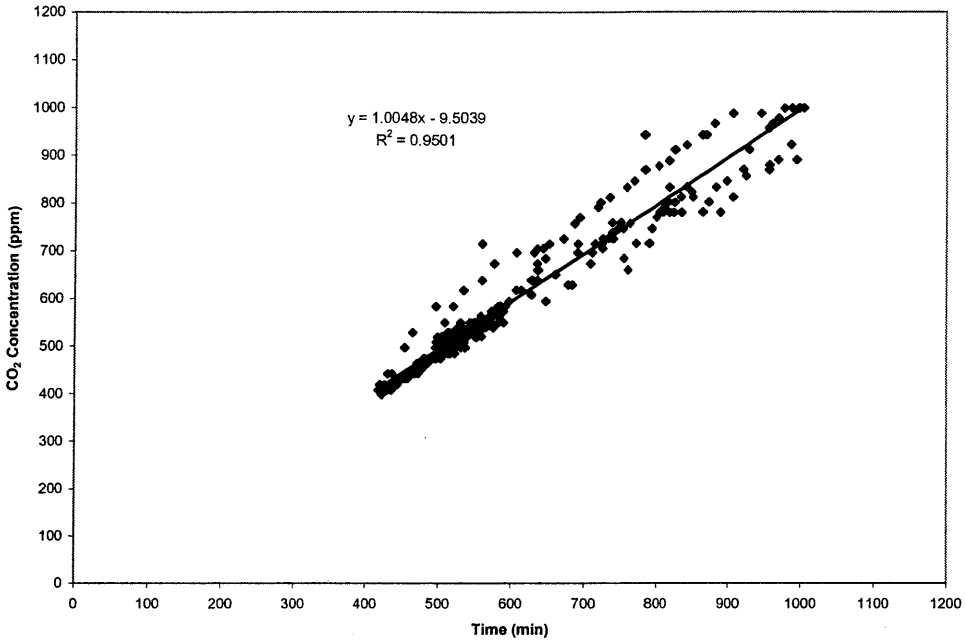


Figure 3.9: Calibration of Johnson Sensor (J6) for CO_2 Concentrations below 1000 ppm

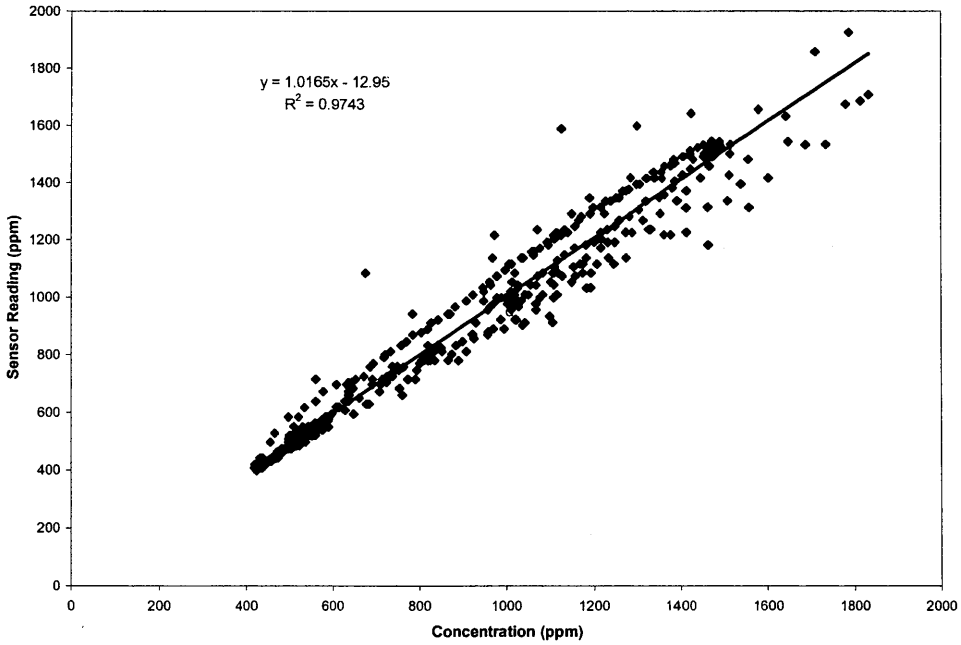


Figure 3.10: Calibration of Johnson System Sensor 6

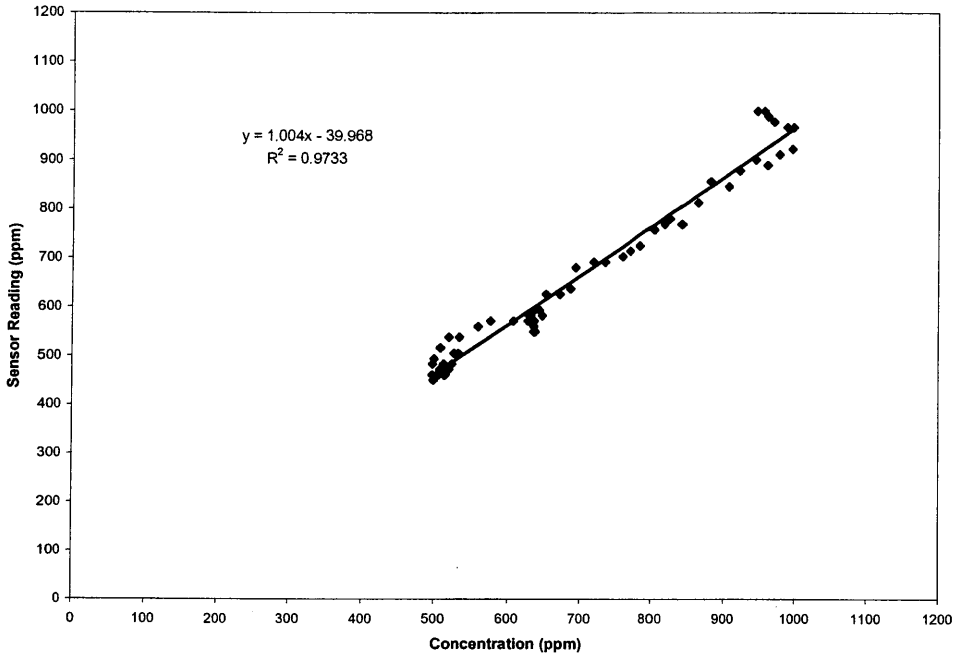


Figure 3.11: Calibration of Engelhard Sensor E7 for CO_2 Concentrations below 1000 ppm

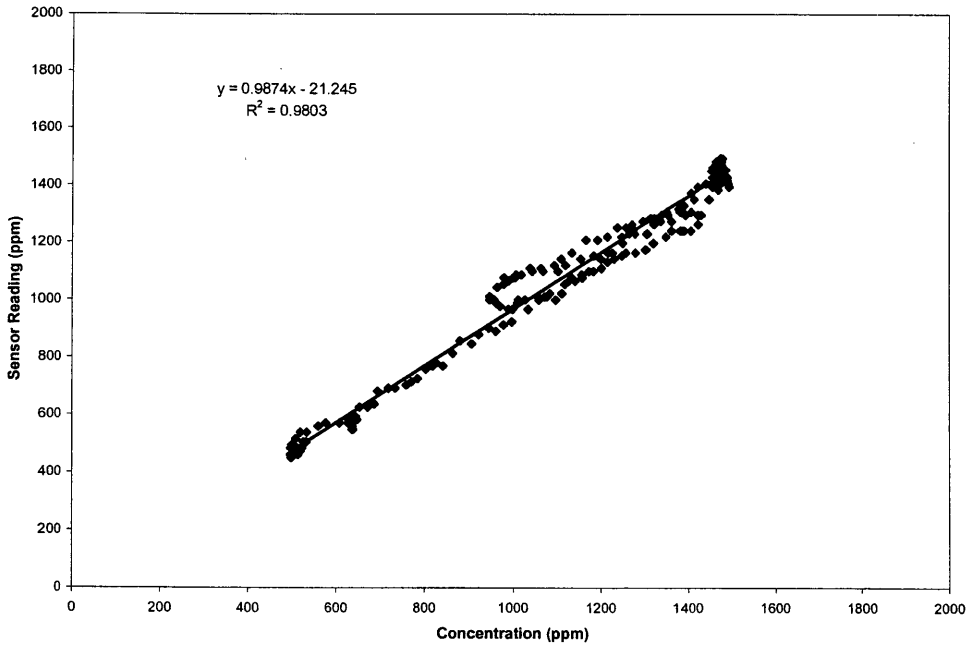


Figure 3.12: Calibration of Engelhard Sensor E7

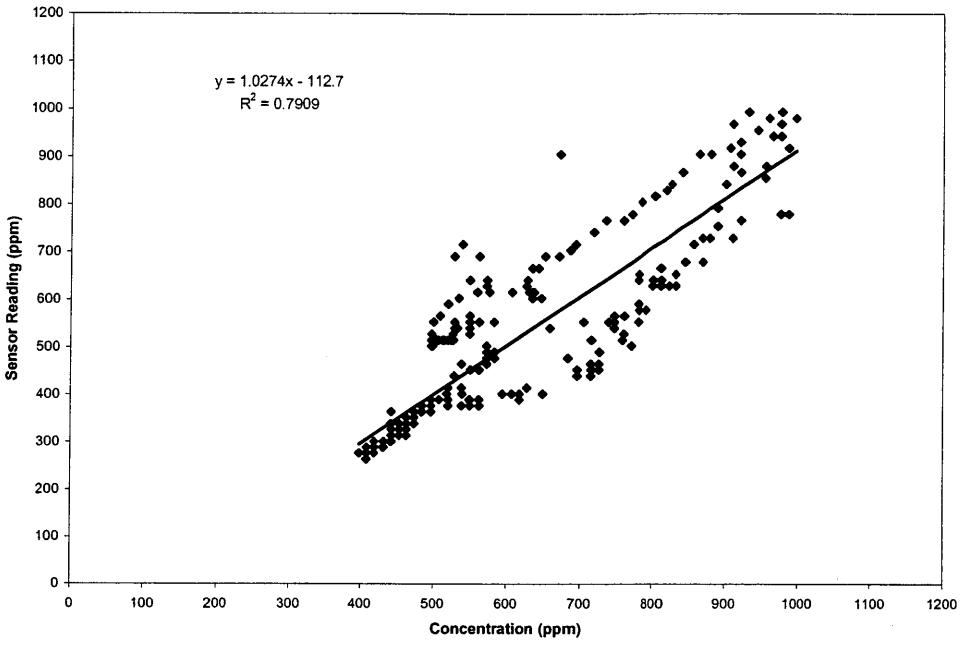


Figure 3.13: Calibration of Engelhard Sensor E8 for CO_2 Concentrations below 1000 ppm

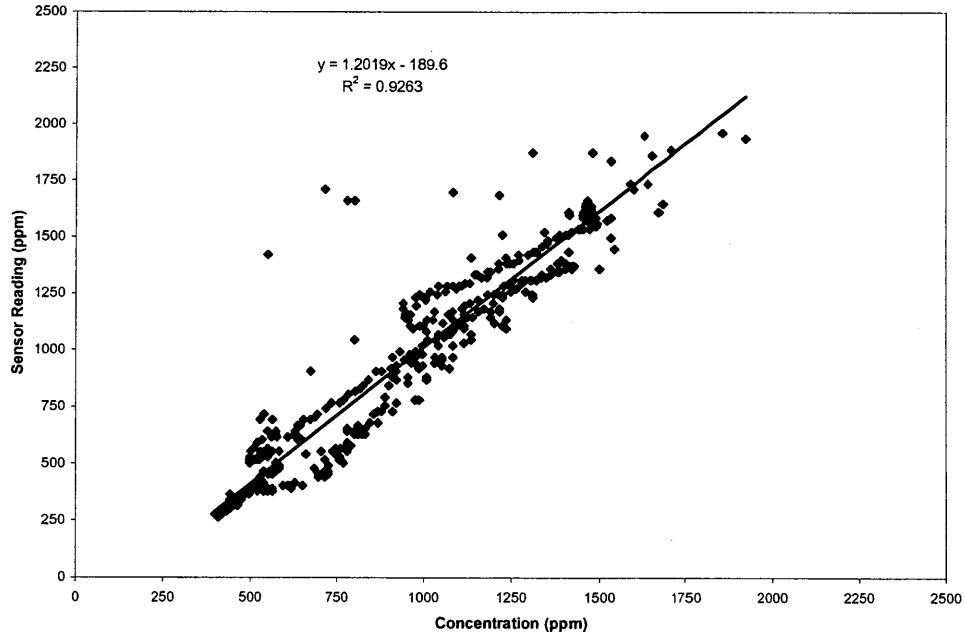


Figure 3.14: Calibration of Engelhard Sensor E8

3.1.3 Carbon Dioxide Flowmeters

Two manual flowmeters are used to inject the required amount of carbon dioxide to the zones; one for each zone. These flowmeters were calibrated by the manufacturer for CO_2 gas. Although both flowmeters are similar in terms of the shape and scale from 0 mm to 150 mm, one is calibrated to supply a maximum of 19 liters/min (1 mm = 0.126 liters/min) and the other is calibrated to supply a maximum of 6 liters/min (1 mm = 0.04 liters/min), which represents a maximum CO_2 generation rate associated with 60 people and 20 people, respectively. The manufacturer of these flowmeters is Advanced Specialty Gas Equipment, California.

3.1.4 Pressure Transducers

Each zone is equipped with a pressure transducer to measure and maintain constant pressure during the tests. A constant value of 0.05 in WG relative to the Larson laboratory is maintained during the tests. The return fan was used to maintain this value of differential pressure by pulling more air out of the zone when the pressure increases or holding the air when the pressure decreases.

3.2 Indoor Air Quality Experiments

At the start of the IAQ experiments, some problems were noticed with the laboratory setup and equipment. These problems included:

1. The outside air conditioning station (OACS) fan could not be used and operated at even low speeds (30%). Indeed, the OACS fan forced fresh air to leak through the outside damper. This leakage was approximately 20-25% of the total supply air when the fan is operating at 30% full-rated speed. This problem was avoided by not using the OACS. However, switching the OACS fan off during the experiments prevented the pre-conditioning of outside air. Consequently, outside air was provided at the existing ambient conditions to the main air-handling unit.
2. The equipment setup and the layout of the duct work in the laboratory made it difficult to run the IAQ experiments. Indeed, the supply fan was on the same line as the outside air duct (refer to Figure 3.15). Thus, whenever the supply fan was operating, undesirable amounts of fresh air were pulled in. Operating the OACS fan made the situation even worse because it pulled the outside air towards the supply fan. Even when the outside damper and the outside damper were fully closed, the outside leakage ranged from 8% to 12 %. To overcome this leakage, the recirculated damper was weather-stripped and the OACS outside air damper was kept closed during the experiments. It is only when outside air was needed that the OACS outside damper was fully opened.

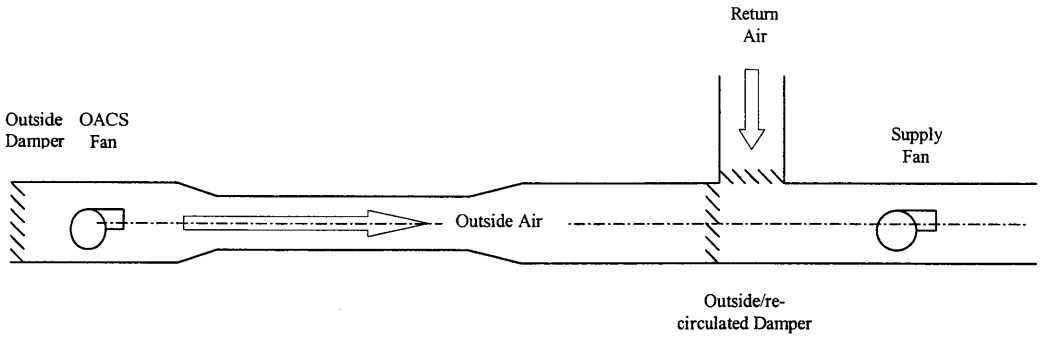


Figure 3.15: Outside Air Duct Layout in the Laboratory

3. Because the maximum supplied cfm from the VAV boxes located in either the East and West full-size zones is 550 cfm, only lighting and people loads were simulated for the zones. Therefore, both zones are considered as internal zones with no significant conduction or solar loads. It is found that the total supply air requirement for a maximum occupancy of 15 people is about 550 cfm using a temperature set-point of 70 °F.
4. Moreover, the air discharge temperature from the main air-handling unit was changed from 50 °F to 55 °F to allow more air supply to the zones and to overcome any incerase in CO_2 level above the setpoint. To allow further increase in the supply air to both zones, the temperature setpoint was decreased by two degrees (from 72 °F to 70 °F).
5. One of the most important problems encountered during the experiments is the measurement of air quantities for the supply, recirculated, outside, and exhaust flow stations in the laboratory. A thorough testing of these flow stations revealed that the exhaust station is more accurate than the other stations. Although all stations were thoroughly calibrated by a pitot tube, they did not show consistent flow rate measurements. The pitot tube itself cannot

accurately measure small flow rates, making the calibration even more difficult. It was decided to replace the flow stations by more accurate flow stations for small airflow rates. Two new flow stations were donated and installed in the supply air and outside air ducts. An existing flow station of the same type was removed from its original location exactly near the main air-handling unit and installed at the recirculated air duct. All flow stations measured accurate and consistent readings after calibration.

6. All CO_2 sensors require factory calibration every six months as recommended by the manufacturer. At first, two sensors were aquired from two different manuafacturers: Engelhard Technologies, and Honeywell Corporation. When placed at the same location, both sensors tend to give the same CO_2 readings, but after some testing, they had different readings that ranged between 5 ppm to 50 ppm. To overcome the two sensors discrepancies, a third honeywell sensor was aquired to replace the Engelhard sensor. For the entire duration of the experiments, the Honeywell sensors provided consistent CO_2 readings.

The indoor air quality experiments were divided into three main phases. Phase 1 consisted of checking the experiment setup, measuring the outdoor CO_2 level, calibrating the CO_2 flowmeters the CO_2 sensors. In addition, Phase 1 involved monitoring CO_2 levels for different ventilation rates and zone pressures.

Phase 2 investigated one-zone controls of CO_2 level for a typical occupancy profile with a controlled CO_2 set-point (1000 ppm). Three different control

strategies were considered, namely: On/Off control, proportional or linear control, and proportional-integral-derivative (PID) control.

Phase 3 investigated one-zone and two-zone controls of CO_2 level and indoor temperature for a typical occupancy profile. For this phase, the CO_2 set-point was selected to be 900 ppm and the indoor temperature set-point was 70 ^{o}F. Phase 3 is discussed in chapter four.

In all experiments, a set of variables is monitored. These variables include zone temperature, zone discharge (supplied) cfm, CO_2 level, zone VAV damper position and control, zone differential pressure, outside damper position and control, fan speeds and power consumption, chiller and pumps power consumption, zone discharge, and return temperatures and humidities.

All experiments performed in phase 2 and 3, and some of the experiments in phase 1, required nine hours of testing time. Depending on the conditions of the experiments, the time needed to setup each test, laboratory initial temperature, and supplied cfm and pressure in the zone, each experiment may have a different start time. However, most of the experiments were started between 7:30 a.m. and 10:00 a.m., depending on how fast the starting conditions (initial supplied cfm, initial CO_2 concentration, and differential pressure in the zone) were attained.

For all indoor air quality experiments, the two full-size zones (FSZ-E and FSZ-W) were used. The two zones setup is shown in Figure 3.16. Depending on the availability of CO_2 sensors, each zone is furnished with at least two sensors. When one-zone tests are performed, all CO_2 sensors are placed in that zone, and the control system calculates an average CO_2 concentration in the zone. Control

is based on this average value. For all tests, the CO_2 gas is injection from one single point located exactly in the middle of the zone.

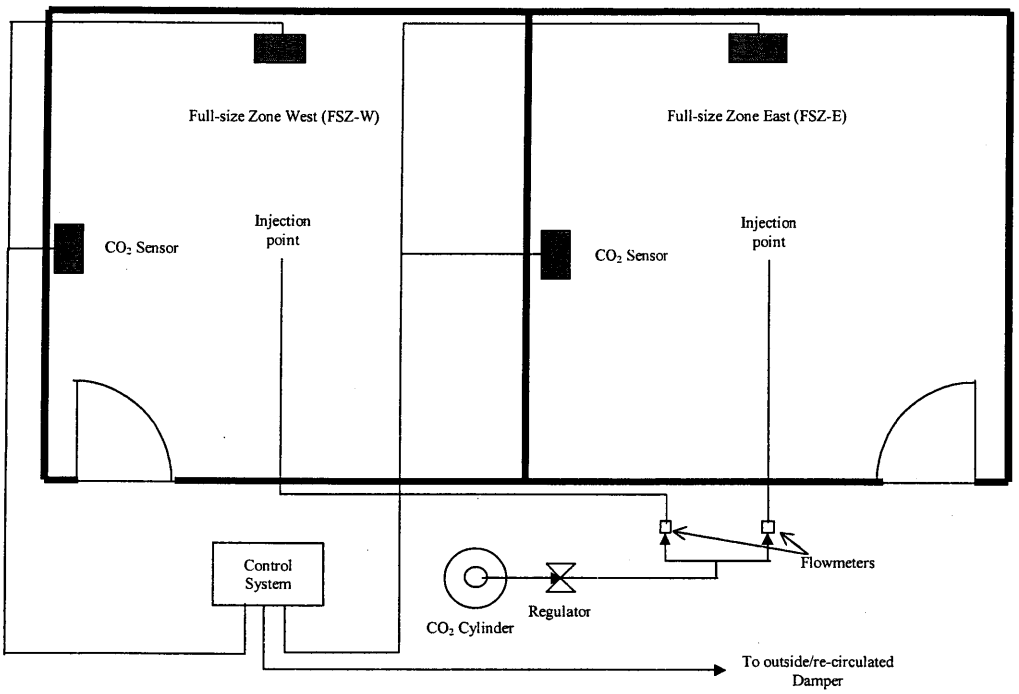


Figure 3.16: Control Setup in the Two Full-size Zones

3.2.1 Measuring and Monitoring CO_2

The main objectives of the experiments conducted for phase 1 are:

1. Verify laboratory and experimental setup.
2. Measure the outdoor CO_2 concentration level for a typical day.
3. Measure the indoor CO_2 concentration in the zone when no CO_2 is injected and no ventilation is provided.
4. Measure and monitor CO_2 concentration level at different ventilation rates for a fixed occupancy profile.

5. Measure and monitor CO_2 concentration level at different zone differential pressures.
6. Measure the ventilation effectiveness in the zones to be used in the model (FSZ-W zone only).

3.2.1.1 Measuring Outdoor and Indoor CO_2 Concentrations

In some experiments, both the outdoor and indoor CO_2 levels were measured. The outdoor CO_2 level tends to range between 350 ppm and 400 ppm. The indoor CO_2 level in the zones, however, tends to range between 420 ppm to 450 ppm when no CO_2 is injected and no ventilation is provided to any of the zones. Figure 3.16 shows the values of indoor and outdoor concentrations when no CO_2 is injected in the zone (during the first 70 minutes of the test). Figure 3.16 shows also the indoor concentration when 1.99 lit/min (0.0703 ft^3/min) is injected in the West zone after 70 minutes since the start of the test. Based on the mass balance calculations, the CO_2 concentration should reach the 2000 ppm level when 6 persons occupy the zone for about one hour without any ventilation supplied to the zone. These calculations are summarized below:

At specific pressure P and temperature T, the concentration of CO_2 is determined as follows (Flagan and Seinfeld, 1988):

$$CO_2 \text{ Concentration (ppm)} = \frac{C_i}{C}(10^6) \qquad \text{Eq. 3.1}$$

where:

$$C = \frac{P}{RT} \qquad \text{Eq. 3.2}$$

and

$$C_i = \frac{m_i}{M_i}(10^6) \qquad \text{Eq. 3.3}$$

where:

C = moles per volume of air

C_i = moles per volume of CO_2

P = pressure, (84.18 kPa)

T = indoor temperature, (298 K)

R = universal gas constant, (8314 J/(kg mol K)

m_i = CO_2 mass concentration (ug/m^3)

M_i = molecular weight of CO_2 (44.01 g/mol)

Substituting for C_i and C in equation 3.3 yields

$$CO_2 \text{ Concentration in ppm} = \frac{RTm_i}{PM_i} \qquad \text{Eq. 3.4}$$

or

$$m_i = \frac{PM_i(CO_2 \text{ Concentrat ion})}{RT}$$
$$= \frac{84.185 \times 10^3\, Pa \times 44.01\, g\, mol\ K \times (2000 - 500)\, ppm \times 10^{-6}}{8.314\, Pa\, m^3\, mol \times 298\, K} \qquad \text{Eq. 3.5}$$
$$= 2.24\, g/m^3$$

The above equation assumes that the initial CO_2 concentration in the zone is 500 ppm (refer to Figure 3.17).

For P = 84.18 kPa, and T =289 K, total mass of CO_2 needed for the zone is 2.24 $g/m^3 \times$ 90.6 m^3 (volume of zone) or 203 g.

An average human being performing light office work (1.2 mets) releases 0.315 liters/min of CO_2 (ASHRAE 1989-62 Standard). Thus, the generation rate of CO_2 is:

CO_2 generated = 3.15×10^{-4} m^3/ min $\times$ 1.83 kg/ m^3 (density of CO_2)

$= 5.76 \times 10^{-4}$ kg/min/person

= 0.0346 kg/hr/person

Therefore, the number of persons needed to reach the 2000 ppm concentration level in the zone is (the initial concentration in the zone is $\approx$ 500 ppm):

Number of persons = 0.203 kg/0.0346 kg/hr/person = 5.86 $\approx$ 6 persons in one hour.

Figure 3.17 agrees with the basic calculations made above when CO_2 is injected at a rate of 1.99 lit/min (0.0702 ft^3/min) for one hour (starting at minute 75 of the test). The concentration inside the zone reached 2000 ppm about 60 minutes after injection.

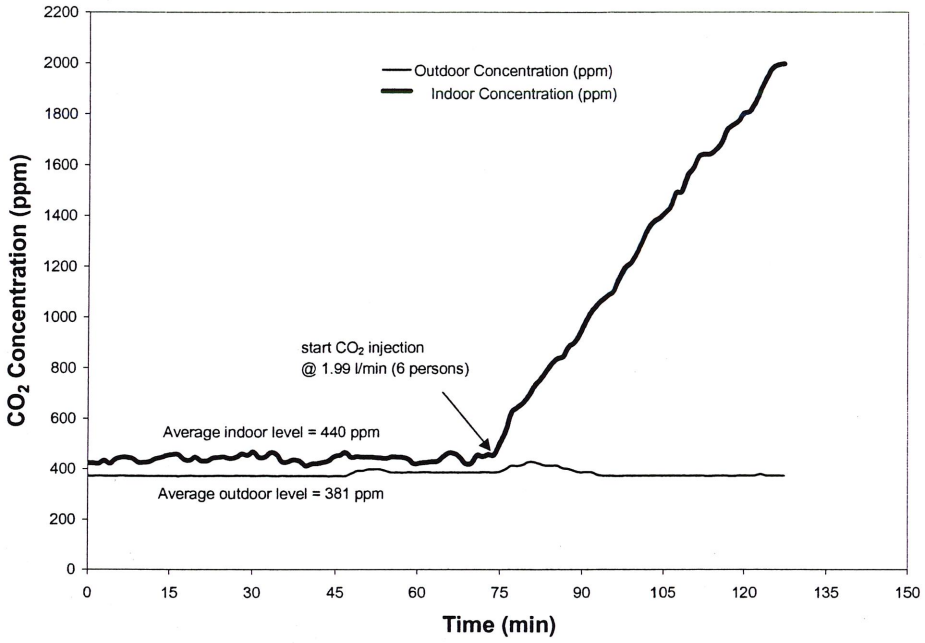


Figure 3.17: Indoor and Outdoor CO_2 Concentration Profiles in the FSZ-W (No Air is Supplied to Zone)

3.2.1.2 Indoor Concentrations at Different Ventilation Rates

The indoor CO_2 concentrations were measured at various ventilation rates supplied to the West zone. The zone is setup for an occupancy of 15 persons throughout the duration of the experiments. The CO_2 concentrations were measured for 5 cfm/person, 10 cfm/person, and 15 cfm/person ventilation rates of 100% outside air. For example, at 5 cfm/person and 15 people, the total outside air required is 75 cfm. Hence, the zone is provided with 75 cfm total supply air. Likewise, for 10 cfm/person and 15 cfm/person, the required total supply air (100% outside air) is 150 cfm and 225 cfm, respectively. The results of these experiments are shown in Figure 3.18. Figure 3.18 shows that at 5 cfm/person,

the CO_2 concentration stayed constant after one hour since the sensor reached its limit (2000 ppm).

A simple mathematical model was developed to compare the results obtained from the experiments described above with an analytical solution for the time variation of CO_2 concentration. This analytical model is based on the mass balance of CO_2 flow in and out of a zone, including CO_2 generation and decay processes (Wadden and Scheff, 1988). Figure 3.20 is a graphical representation of this model. The equations governing this model are provided below:

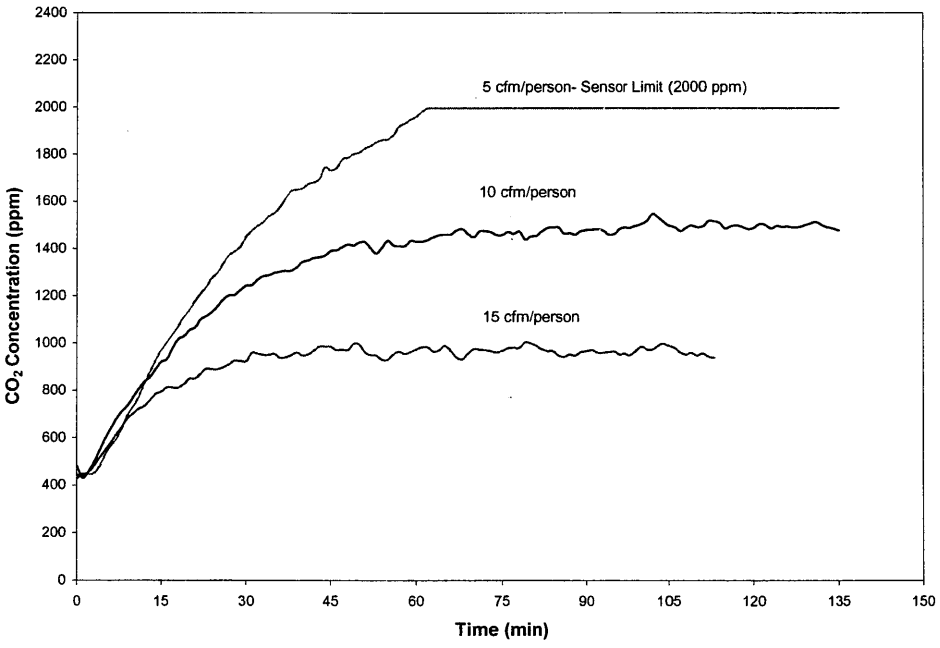


Figure 3.18: Indoor CO_2 Concentration in the FSZ-W at Different Ventilation Rates

Air mass balance:

$$Q_0 + Q_2 = Q_3 + Q_4 \qquad \text{Eq. 3.6}$$

CO_2 mass balance:

$$V\frac{dC_i}{dt} = kQ_0C_0(1-F_0) + kQ_1C_i(1-F_1) + kQ_2C_0 - k(Q_0+Q_1+Q_2)C_i + S - R \qquad \text{Eq. 3.7}$$

Where:

C_i = CO_2 zone concentration, (ppm)

C_0 = CO_2 outdoor concentration, (ppm)

Q_0 = outside air flow rate. (m^3/min)

Q_1 = recirculated air flow rate, (m^3/min)

Q_2 = air flow rate from infiltration, (m^3/min)

Q_3 = air flow rate from exfiltration, (m^3/min)

Q_4 = exhaust air flow rate, (m^3/min)

F_0 = filter efficiency for outside air

F_1 = filter efficiency for recirculated air

V = volume of the zone, (m^3)

S = CO_2 production rate, (g/min)

R = indoor sink removal rate (g/min)

k = ventilation effectiveness

The solution of equation 3.9 for the variation of C_i with time t, using the assumption that all the parameters in the equation except C_i are constant and setting $C_i = C_s$ at $t = 0$ (Wadden and Scheff, 1988) is:

$$C_i = \frac{k[Q_0(1-F_0)+Q_2]C_0 + S - R}{k(Q_0 + Q_1F_1 + Q_2)}[1 - e^{-(k/V)(Q_0+Q_1F_1+Q_2)t}] + C_s e^{-(k/V)(Q_0+Q_1F_1+Q_2)t} \quad \text{Eq. 3.8}$$

The steady-state value for the indoor concentration, $C_{i,ss}$, is obtained by letting t approach ∞ in equation 3.10:

$$C_{i,ss} = \frac{k[(1-F_0)+Q_2]C_o + S}{k(Q_0 + Q_1F_1 + Q_2) + E} \quad \text{Eq. 3.9}$$

where

E = proportionality constant = R/C_i

The results of this model compared with the experimental results are shown in Figure 3.20. The model predictions conform to the results obtained experimentally. At 15 cfm per person, the two curves coincide with each other. At 10 cfm per person, however, the curve of the experimental results has a steeper slope at the beginning of the injection and before reaching steady-state conditions. A possible reason for this is that the mixing is not perfect at this initial stage of the experiment. The ventilation effectiveness in the model is determined based on the minimum error found between the experimental and the analytical CO_2 concentration at the steady-state conditions. Table 3.5 lists the

factors and constants used to calculate the CO_2 concentration using the model of equation 3.10.

Table 3.4: Factors and Constants Used to Apply the Model

Factor	Ventilation rate (10 cfm/person)	Ventilation rate (15 cfm/person)
Ventilation Effectiveness (k)	0.80	1.05
Outdoor Air Flow (cfm)	150	225
CO_2 Decay Rate (R)	0.0	0.0
CO_2 Production Rate (lit/min/person)	0.315	0.315
Filter Efficiency (F)	0.0	0.0
Initial CO_2 Concentration (C_s) (ppm)	450	450
Outdoor CO_2 Concentration (C_o) (ppm)	390	390
Calculated Model $C_{i,ss}$ (ppm)	1444	1091
Calculated Average Experimental C_{ss} (ppm)	1486	1003
Percent Difference Between Average Experimental C_{ss} and Model $C_{i,ss}$ (%)	2.91	-8.10

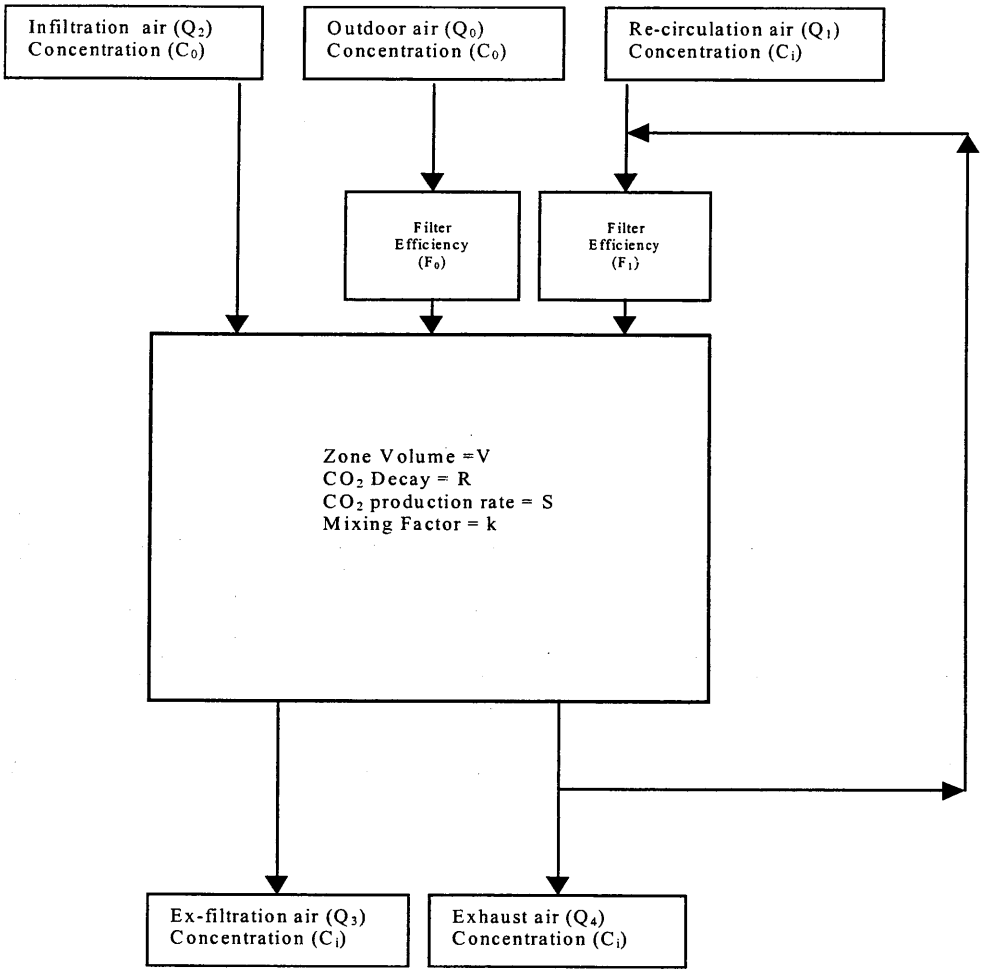


Figure 3.19: Ventilation System for Time-varying Indoor-Outdoor Model

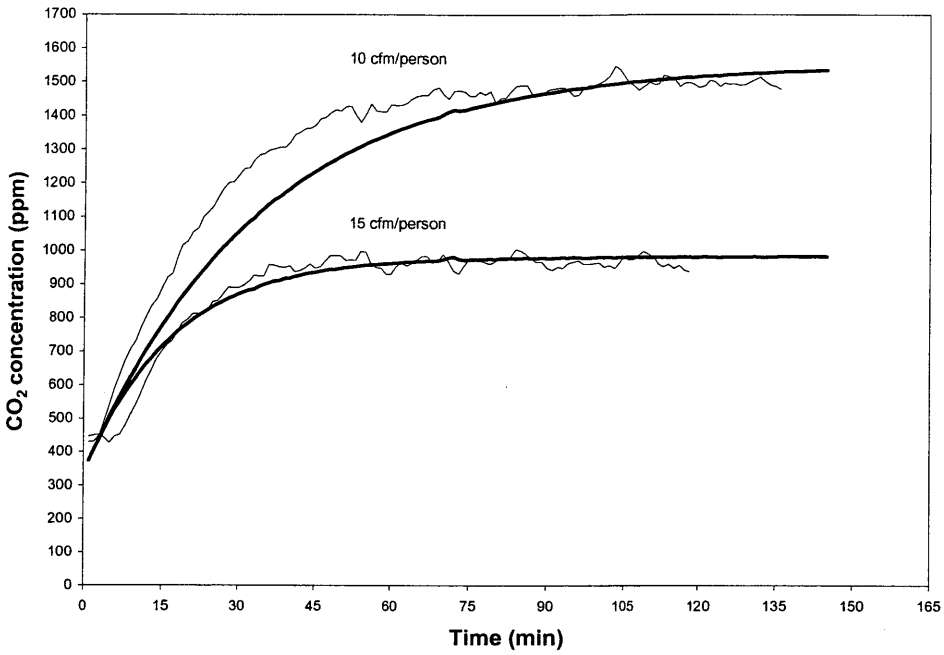


Figure 3.20: Comparison between Analytical Model and Experimental Results

3.2.1.3 Indoor Concentrations at Different Room Static Pressures

Experiments were performed to determine how the concentration of CO_2 would vary with the room static pressure. The zone was setup to receive 225 cfm (for 15 persons at 15 cfm/person). Two different zone pressures were used: 0.02 in WG and 0.08 in WG, which are the operating limits of the zone pressure transducer. The CO_2 concentration in both tests was allowed to reach steady-state conditions. The results showed that the CO_2 concentration was not affected by changing the pressure. Figure 3.21 shows the time variation of CO_2 level within the zone for the two static pressures.

To verify the experimental findings described above, equation 3.10 can be used. This equation states that regardless of the differential pressure, the

steady-state CO_2 concentration in a zone is calculated as follows (ASHRAE Standard 62-1989):

$$C_i = \frac{S * N}{Q} 10^6 + C_o \qquad \text{Eq. 3.10}$$

where:

C_i = steady-state CO_2 concentration in the zone, (ppm)

S = CO_2 production rate per person, (5×10^{-6} m^3/s)

N = number of people in the zone (15 people)

C_o = outside CO_2 concentration, (380 ppm)

Q = volumetric flow rate (0.11 m^3/s)

Using equation 3.10, the steady-state concentration in the zone is found to be 1060 ppm. According to the experiment, the average steady-state CO_2 concentration is 975 ppm. The percent difference is 8%, which can be accounted for errors in CO_2 sensor readings in when measuring the outside CO_2 concentration and airflow sensor reading in the zone (refer to uncertainty analysis in chapter four).

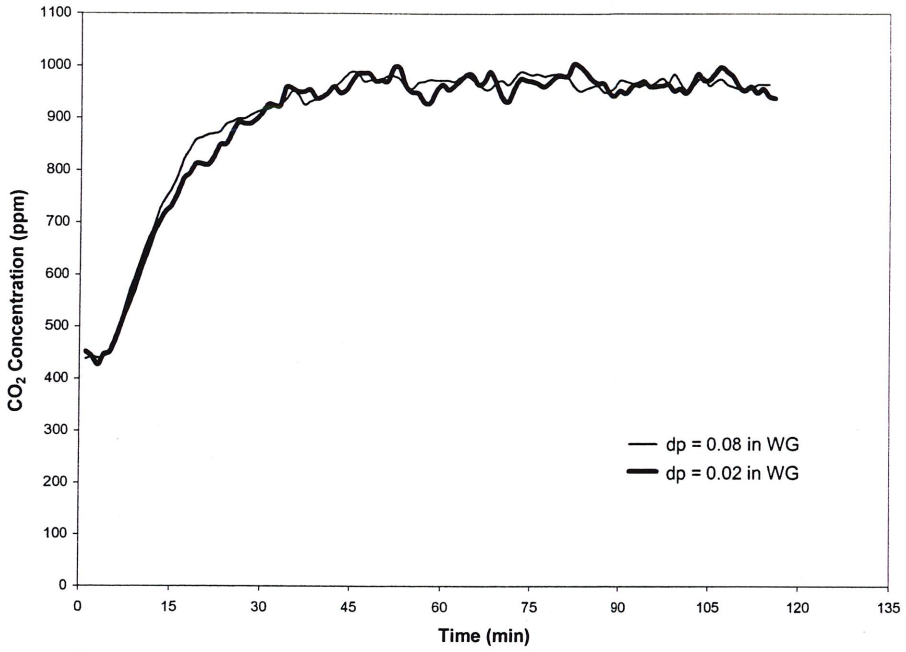


Figure 3.21: Indoor CO_2 Concentration at Different Room Static Pressures

3.2.1.4 Monitoring CO_2 Concentration at Different Ventilation Rates

The last set of experiments performed in the first phase involved monitoring of CO_2 level for one full business day (nine hours). Two ventilation rates were used: 15 cfm/person and 20 cfm/person, which imply outside air supply of 225 cfm and 300 cfm to the zone, respectively. According to ASHRAE 62-1989, the minimum outside air required is 20 cfm/person for office spaces. This minimum value will assure that the concentration will not reach the 1000-ppm level. Figure 3.23 shows that the CO_2 concentration can be above the 1000 ppm limit if 15 cfm/person is used. However, if 20 cfm/person (see Figure 3.23) is supplied, the CO_2 concentration remains below the limit.

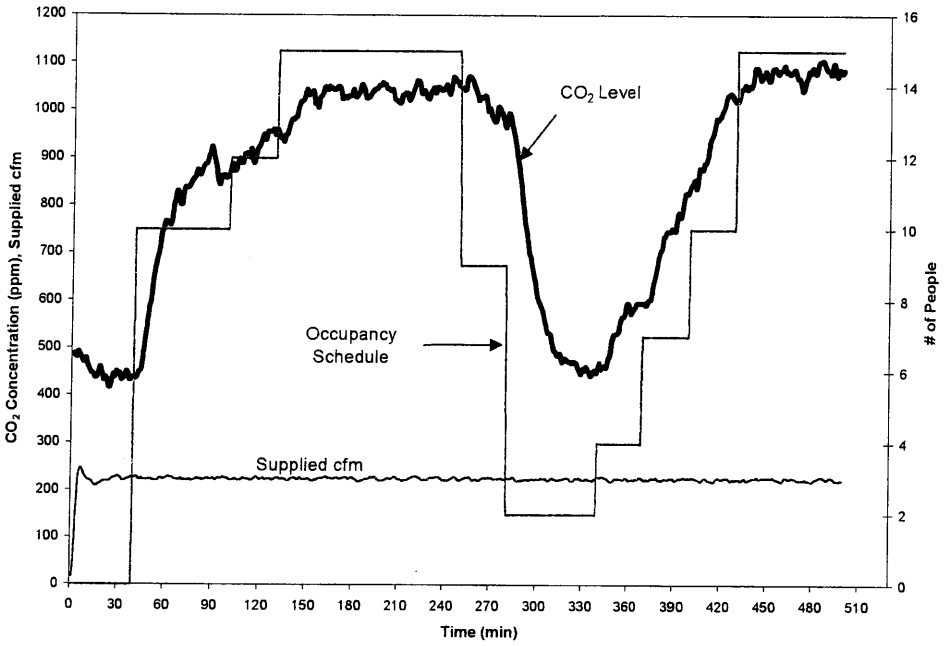


Figure 3.22: CO_2 Concentration Profile for a Ventilation Rate of 15 cfm/person

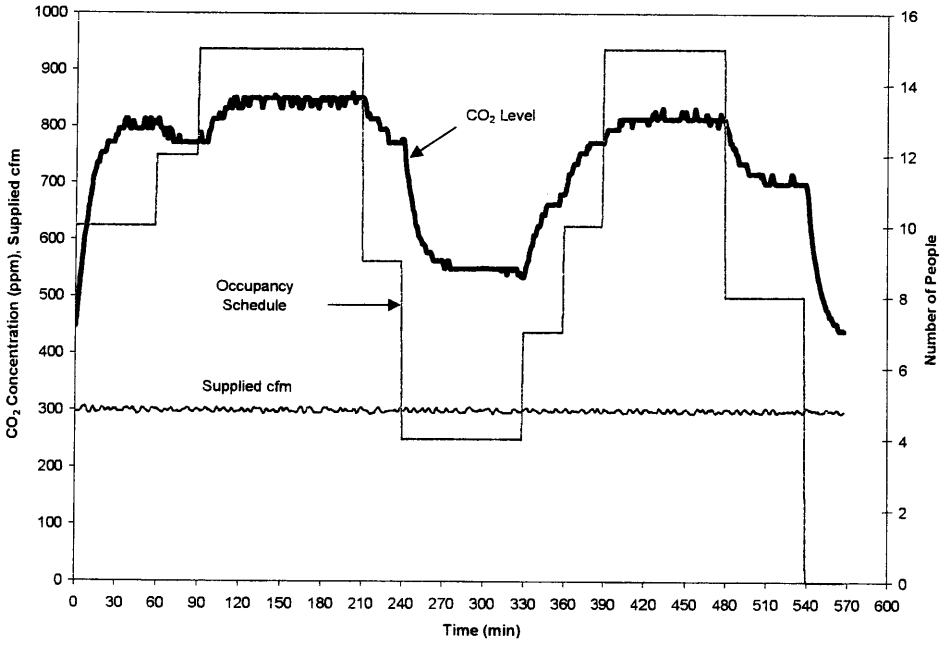


Figure 3.23: CO_2 Concentration Profile for a Ventilation Rate of 20 cfm/person

3.2.1.5 Measurement of Zone Ventilation Effectiveness

To ensure a healthy environment for occupants, an efficient ventilation strategy should provide an adequate supply of fresh air to dilute contaminants to acceptable concentration levels. The ventilation effectiveness can be measured at any location provided that the contaminant is defined and its concentration is measurable.

To achieve an acceptable indoor environment, fresh air should first be distributed in the occupied space, and any contaminant generated in the space should quickly be removed. Therefore, it is important to know the flow rate and pattern of both fresh air and contaminated air. A map of the contaminant concentrations may help determine the flow characteristics. This map can be obtained for the pollutant by measuring the concentrations at various locations within the zone.

Various measurement methods can be used to analyze the flow pattern and the distribution of contaminants in a zone. To assess the performance of the ventilation system, the air exchange efficiency, i.e., how the fresh air is distributed in the room, is determined. The ventilation effectiveness, i.e., how effective the system is in reducing room contaminant concentration, can also be determined.

The ventilation effectiveness ε_c expresses the efficiency in extracting the contaminants generated in the zone, and is generally defined as the ratio of two net concentrations:

$$\varepsilon_c = \frac{C_e - C_o}{C_r - C_o} \qquad \text{Eq. 3.11}$$

Where:

C_e is the contaminant concentration in the exhaust air, ppm

C_o is the contaminant concentration in the outdoor air, ppm

C_r is the contaminant concentration at the location of interest, r, ppm

The location of interest can be:

1. Any particular location, r, in the space.
2. The location within the highest contaminant concentration, which gives the lowest ventilation effectiveness.
3. Representative location of the entire space, by taking the average concentration over the space. The result will then be the average ventilation effectiveness.

The ventilation effectiveness is zero if there is a short circuit between exhaust and inlet diffusers ($C_e = C_o$), and is equal to one for complete mixing ($C_e = C_r$), and is infinity at the locations reached by pure fresh air ($C_r = C_o$).

It must be noted that the definition and the measurement of ε_c specify in most cases the measurement of the exhaust concentration C_e. This, in turn, requires the air to leave the room through a single exhaust duct.

There are three methods that can be used to measure ventilation effectiveness, and air exchange efficiency:

1. The pulse method where the tracer gas is injected into the inlet duct as a short pulse.
2. The step-up method where the tracer gas is injected as a constant flow rate.
3. The decay method where the gas is allowed to decay after injection.

The decay method was used to determine the ventilation effectiveness in this work. The tracer gas is dispersed throughout the zone with the aid of fans to obtain a uniform steady-state initial concentration, at which point the ventilation effectiveness is calculated.

Experiments for the ventilation effectiveness are performed for three different air exchange rates, 2 ACH, 4 ACH and 6 ACH, corresponding to a supply flow rate of 101 cfm, 202 cfm, and 304 cfm, respectively. As shown in Table 3.6, the tracer gas (carbon dioxide) injection rate is equivalent to an amount of cfm supplied for a ventilation rate of 20 cfm/person. The locations of the CO_2 sensors are shown in Figure 3.24. A total of 5 sensors were used. Four were placed in the zone (three Honeywells (H1, H2, and H3) and one Engelhard (E4)), and the fifth was placed in the return duct (Johnson Controls sensor (E6)). Figures 3.25 through 3.27 show the CO_2 concentrations for all sensors in the West zone at air exchange rates of 2 ACH, 4 ACH, and 6 ACH, respectively.

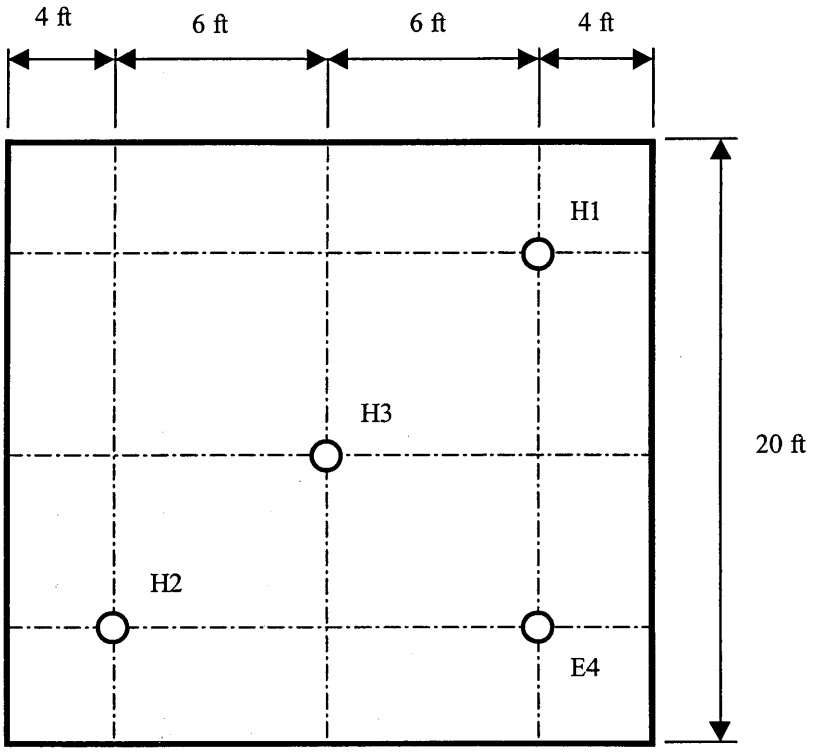


Figure 3.24 Location of CO_2 Sensors in the FSZ-West

Table 3.5: Tests Performed for the Ventilation Effectiveness

Parameter	Air Exchange Rate (ACH)		
	2 ACH	4 ACH	6 ACH
Supplied cfm	101	202	304
Number of people	6.7	13.5	20.3
CO_2 injection rate (l/min)	2.1	4.2	6.4
Scale reading (mm)	22.0	40.0	58.0
Differential pressure in zone (in WG)	0.05	0.05	0.05

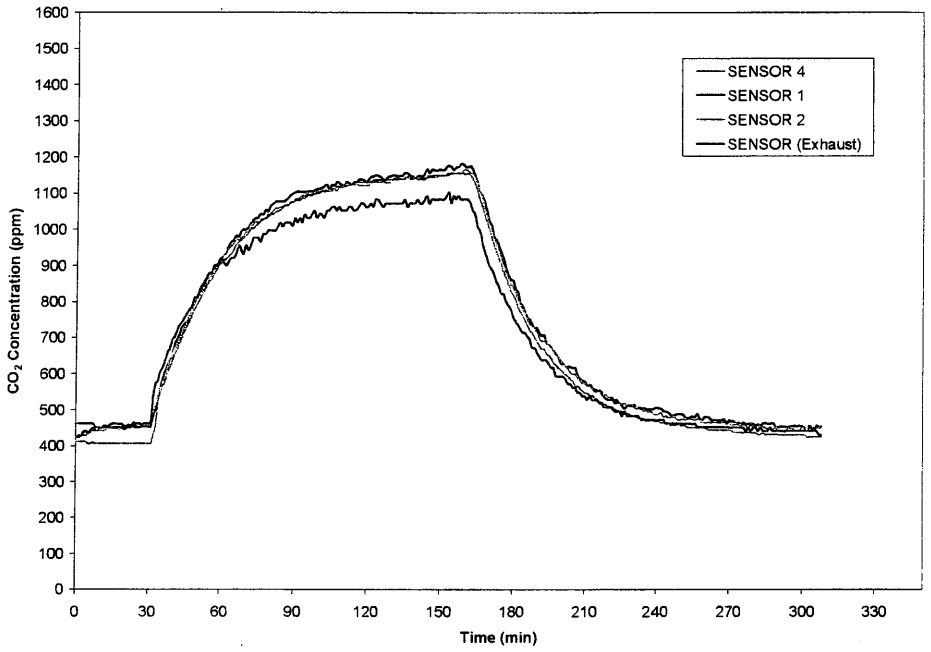


Figure 3.25: CO_2 Concentration Measured during Ventilation Effectiveness Experiment for Air Exchange Rate =2 ACH

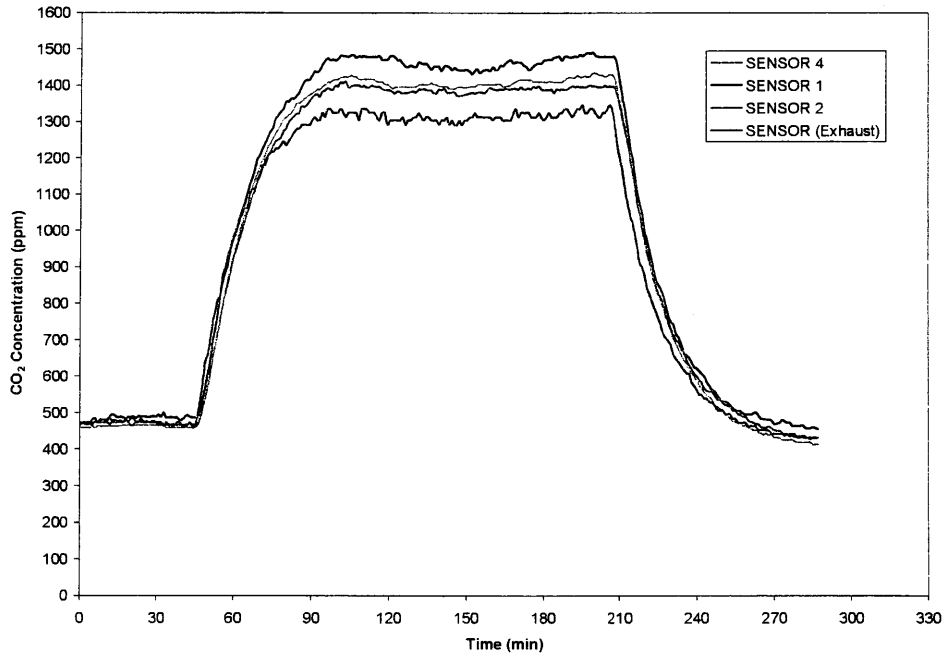


Figure 3.26: O_2 Concentration Measured during the Ventilation Effectiveness Experiment for Air Exchange Rate =4 ACH

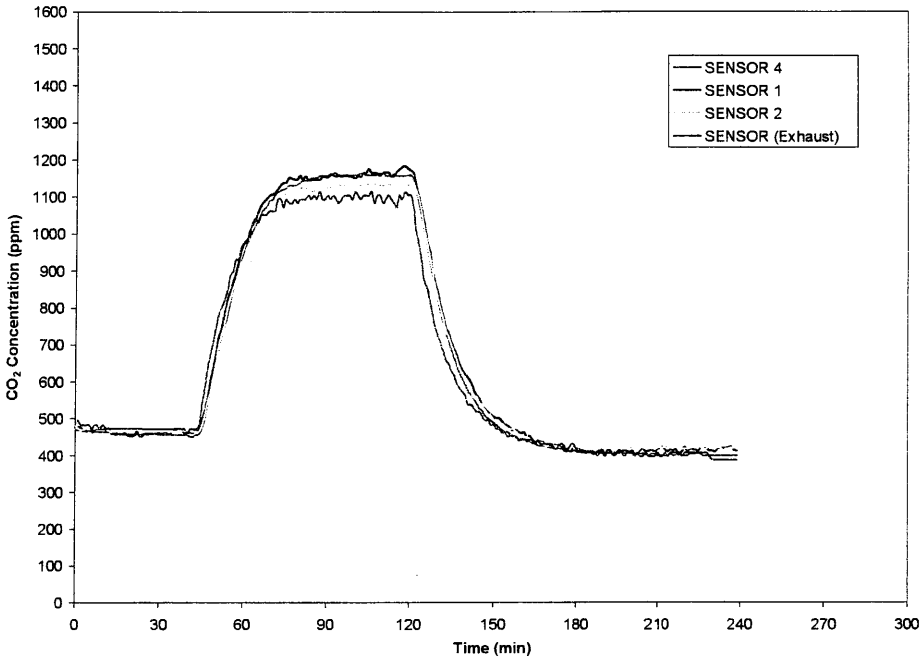


Figure 3.27:CO_2 Concentration Measured during the Ventilation Effectiveness Experiment for Air Exchange Rate = 6 ACH

As shown in Table 3.7, and as expected, the ventilation effectiveness is equal to 1 at the exhaust return duct (sensor J5). Sensor H3 provides different values of ventilation effectiveness because it is located near the CO_2 source, where the concentration is nonuniform and unstable. The ventilation effectiveness calculated by sensor H3 is therefore not used.

In conclusion, the average ventilation effectiveness in the West zone increases with the supply airflow rate. In the simulation model, an average value of 0.90 (90%) is used since the airflow rate supply to the zone is constant at 300 cfm (5.9 ACH).

Table 3.6: Results of Ventilation Effectiveness

	Air Exchange Rate =2 ACH					Air Exchange Rate =4 ACH					Air Exchange Rate =6 ACH				
	SENSOR No.					SENSOR No.					SENSOR No.				
	H1	H2	H3	E4	J5	H1	H2	H3	E4	J5	H1	H2	H3	E4	J5
Initial Concentration (ppm)	452	451	510	427	440	459	433	460	418	432	415	421	515	403	401
Steady-state Concentration (ppm)	1130	1123	819	1108	1009	1463	1389	1857	1408	1315	1166	1133	941	1160	1100
Ventilation Effectiveness	0.82	0.83	1.61	0.85	1.0	0.85	0.92	0.61	0.91	1.0	0.91	0.95	1.37	0.92	1.0
Average Ventilation Effectiveness	0.84					0.89					0.93				

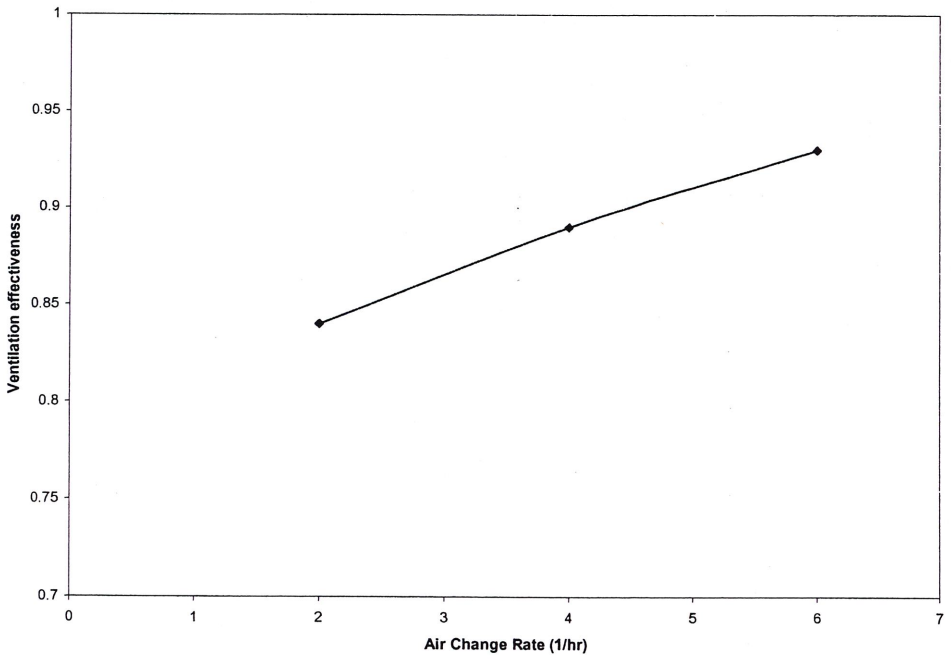


Figure 3.28: The Relationship between the Air Exchange Rate and Ventilation Effectiveness in the West Zone

3.3 Conclusions

This chapter discussed the laboratory setup and the preliminary indoor air quality experiments conducted to ensure proper installation and functioning of equipment.

Calibration of eight (8) sensors was performed revealing that only five of them could be used. The average outdoor CO_2 concentration was measured to be 380 ppm. The average indoor CO_2 concentration in both the East and the West zones, however, was measured to be 440 ppm, when there were no people occupying the zones.

As expected, room differential pressure had no effect on the steady-state CO_2 concentration. Experiments performed at room differential pressures of 0.02 in WG and 0.08 in WG revealed no change in the steady-state CO_2 values in the zone.

The main findings of the first IAQ experiments were that the ASHRAE minimum outdoor air ventilation rate (20 cfm/person) is needed to maintain an acceptable indoor air quality of less than 1000 ppm of CO_2 concentration. The experiments performed at lower ventilation rates showed that these rates do not guarantee that the CO_2 concentration level is kept below the limit (1000 ppm).

The experiments performed to measure the ventilation effectiveness in the FSZ-West revealed that as the supplied cfm increases, the ventilation effectiveness increases. The average ventilation effectiveness in the zone is measured to be 0.9.

CHAPTER FOUR

Indoor Air Quality Experiments

4.1 One-zone Controls of CO_2 Only

The objective of these experiments is to measure the amount of ventilation air required while controlling the CO_2 level in the zone at a 1000-ppm set-point. There is no thermal load involved in these experiments. Three control strategies were considered, namely: ON/OFF, linear or proportional, and the proportional-integral-derivative (PID).

The same occupancy profile is used in all the three control strategies and is given in Figures 4.1, 4.2, and 4.3. To overcome the problem of outside air leakage, the outside damper (refer to Figure 3.15) was kept fully opened while the VAV damper in the zone was controlled based on the CO_2 level within the zone. Thus, the VAV box damper acts as the outside damper supplying fresh air when needed. There was no recirculated air provided to the zone.

4.1.1 ON/OFF Controls

In this experiment, the outside damper (VAV damper for these experiments) is allowed to open and close whenever the CO_2 limits are reached. Specifically, the damper is fully open when the CO_2 level in the zone is above 1050 ppm and is fully closed when the concentration is below 950 ppm (ΔC = 100 ppm). As shown in Figure 4.1, the damper is moving between ON (maximum supply air, 350 cfm) and OFF (minimum supply air, 17 cfm) as the CO_2 concentration

varies. Figure 4.1 depicts that the ON/OFF strategy is not capable of properly controlling the concentration in that it allows CO_2 level to be higher than the upper limit of 1000 ppm and reaches levels as high as 1150 ppm. Consequently, the required amount of outside air has to be high to avoid the high CO_2 concentrations.

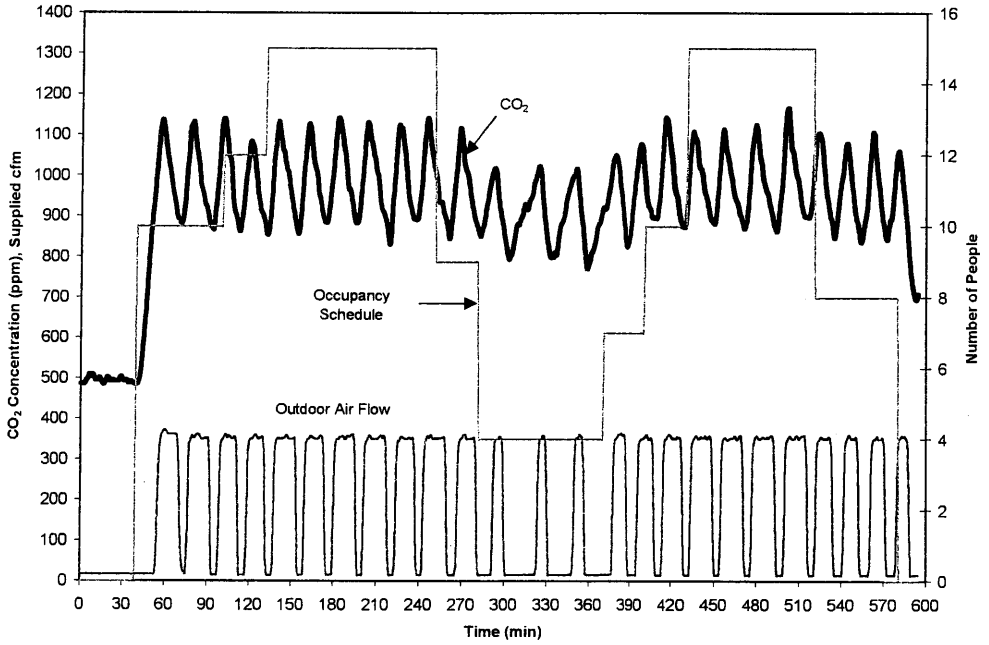


Figure 4.1: One-zone Control of CO_2 Level (ON/OFF Control)

4.1.2 Linear Proportional Controls

In this experiment, the CO_2 set-point is 1000 ppm and the amount of outside air is adjusted based on the measured CO_2 level and the maximum ventilation air required for 15 people is as follows:

$$cfm(t) = \frac{cfm_{\max}}{\Delta C}(C_{zone}(t) - C_{\min}) \qquad \text{Eq. 4.1}$$

where:

cfm_{max} = the maximum amount of cfm required, (cfm_{max} =225 cfm)

ΔC = the concentration range, (ΔC = 50 ppm)

$C_{zone}(t)$ = the concentration in the zone at time t

C_{min}, C_{max} = the minimum and maximum concentrations between which the CO_2 concentration level is acceptable.

The results of this experiment are shown in Figure 4.2. The total amount of ventilation air required to control the CO_2 set-point is lower than that of the previous strategy (ON/OFF control) by 6.4 %(refer to Table 4.1). Consequently, the damper does not have to open in full to control the CO_2 concentration level. The CO_2 level within the zone remains within a narrow band between 950 ppm and 1100 ppm.

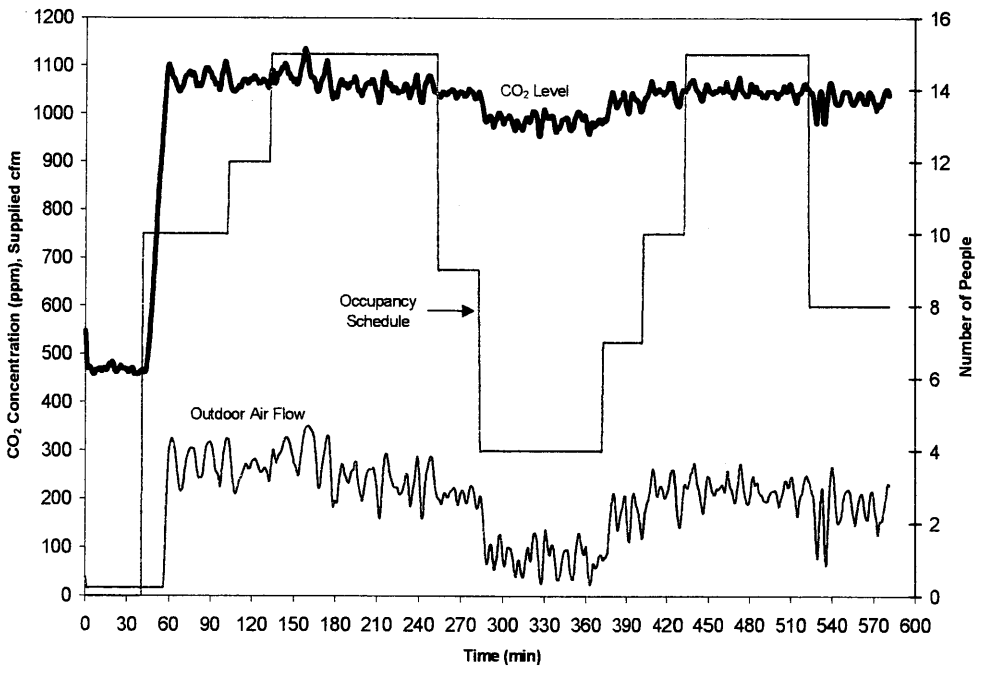


Figure 4.2: One-zone Control of CO_2 Level (Linear Control)

4.1.3 Proportional-Integral-Derivative (PID) Control

The PID control strategy allows the outside damper to be controlled using PID controls. The control process for this strategy depends on the magnitude of the error between the CO_2 concentration in the zone and the CO_2 concentration set-point (the controlled process). The basic PID equation used is:

$$U = U_o + K_p e + K_i \int e dt + K_d \frac{de}{dt} \qquad \text{Eq. 4.2}$$

where:

U = controller output (outside damper position signal), (cfm)

U_o = the mid point of the output when no error exists

K_p = proportional gain (in this experiment, K_c =0.22 cfm/ppm)

K_i = integral gain (in this experiment, K_I =150 cfm/[ppm.min])

K_d = derivative gain (in this experiment, K_d =50 [cfm.min]/ppm)

e = error [CO_2 (set-point) - CO_2 (measured)] (ppm)

t = time, (min)

In this experiment, the PID constants were found by trial and error. Figure 4.3 shows the CO_2 concentration for a PID controller. It is clear that the amount of supplied cfm is lower compared to the other two control strategies. Comparing the ON/OFF control strategy with this control strategy, the CO_2 level within the zone is maintained closer to the set-point than the ON/OFF control strategy. As Figure 4.3 shows, there are high fluctuations in the CO_2 level, which are due to

the time delay of the CO_2 sensors (refer to sensitivity analysis in Chapter Six). It should also be noted that there is inconsistency in the CO_2 level for being constant between minutes 300 and 360 when there were four people in the zone. A possible reason for this inconsistency is an error in CO_2 injection at this time.

Table 4.1 lists a summary of the findings from the testing of the three control strategies.

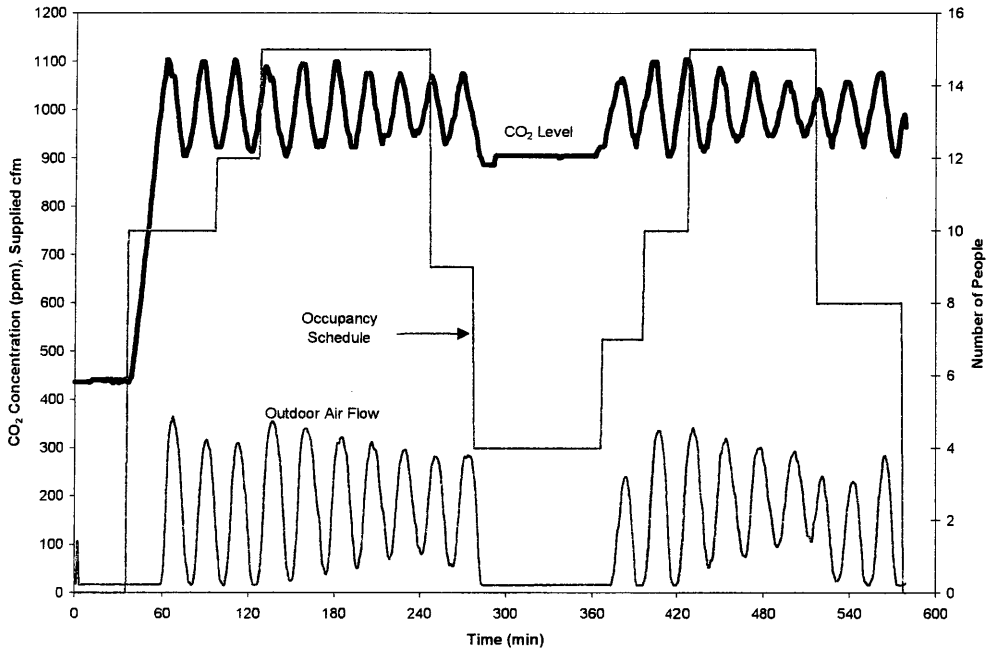


Figure 4.3: One-Zone Control of CO_2 Level (PID Control)

Table 4.1: Outdoor Air Requirements for the Control Strategies

Control Strategy	Average CO_2 Value (ppm)	Minimum CO_2 Value (ppm)	Maximum CO_2 Value (ppm)	Average Supplied Outdoor Air (cfm)	Percent Reduction in Outdoor Air (%)
ON/OFF	966±36	770±36	1165±36	203±10	-
Linear	1040±36	953±36	1134±36	190±10	6.4±6.7
PID	983±36	886±36	1103±36	137±10	32.5±5.9

The low values of minimum CO_2 concentration shown in Table 4.1 occur during the lunchtime when there is minimum number of people. Comparing the values of maximum CO_2 values, the PID control strategy is the best in maintaining the concentration within the acceptable limit. Referring to Table 4.1, the average CO_2 concentration for the PID test is 983 ppm. In addition, the results indicate that the PID strategy has the lowest outside air consumption.

4.2 One-zone and Two-zone Controls of Both CO_2 Level and Temperature

The objective of this phase is to investigate controls that maintain the CO_2 level and the indoor temperature of the zone to specific setpoints. The experimental application of such a concept experimentally led to complications that needed to be considered at the beginning of the experiment and during the experiment. For example, each experiment had to start with the same conditions, including supplied cfm, temperature, and differential pressure, in order to get the same initial CO_2 profile. Thus, the start time for all experiments might not be the same. The outside weather conditions were also a factor that needed to be considered when a set of comparable experiments were conducted.

During the experiments, various parameters were recorded, including the CO_2 flowmeters injection rate to the zones, differential pressure in the zones, air discharge temperature set-point, damper positions, and chiller operation.

In all experiments, only cooling load from people and lighting were considered (conduction or solar loads have not been included). Thus, both zones are representatives of internal zones only. The reason for this assumption is

because the maximum allowable air supplied by the VAV boxes in both zones is limited to 550 cfm. For the lighting load, 18 of 40-watt fluorescent bulbs were simulated that are switched on at the beginning of the day (8:00 a.m.) and are switched off at the end of the day (5:00 p.m.). The cooling load for people, on the other hand, was simulated based on the number of people available at the time, with each person performing light office work and producing 256 Btu/hr (75 W) of heat (ASHRAE Fundamentals, 1993). The heating load from both lighting and people was summed up and provided to a virtual channel that controls the heating produced by the baseboard panels located in the zones. To ensure that the heat is well distributed in the zone, all of the six heating baseboard panels were used.

The East zone was considered a conference room with significant fluctuations in the occupancy schedule. Based on the results of previously performed experiments, the occupancy level ranged between 8 to 12 people.

The West zone was simulated with four people present during the entire day except for one lunch hour when it was assumed that the zone was unoccupied. This zone represented an office space.

Since the two full-size zones were used to run the experiments, the other two zone simulators were blocked off.

The calculation of the outside air fraction for the base case is based on 20 cfm/person delivered to the zone with 12 people as the design maximum number of persons for the East zone and 4 people for the West zone.

4.2.1 Two-Zone Controls of Both CO_2 and Temperature

The experiments for the two-zone control of both CO_2 and temperature are performed to maintain both temperature and CO_2 level in two zones with different cooling loads within specified set-points. The control of the CO_2 level is based on critical zone, that is, the zone with the higher CO_2 concentration determines the amount of outside air to be supplied to both zones. A virtual channel was used to compare between CO_2 concentrations in the two zones, and controls the outside damper according to the highest CO_2 concentration.

The experiments in this phase involved several variables to be set or modified to obtain successful tests. These variables include:

1. Set-point temperature in the zones
2. Occupancy schedule
3. Chilled air discharge temperature set-point
4. Initial conditions of the experiment.
5. Supply duct static pressure

The zone temperature set-point was varied between 70 °F and 74 °F. Decreasing the temperature set-point increases the supplied cfm and decreases the CO_2 level near the set-point.

The set-point of the supply air temperature was increased from 50 °F to 55 °F to allow more supply of air to the zones.

4.2.1.1 Base Case Experiment

This experiment involves the monitoring of the CO_2 level while maintaining the temperature at the set-point during the entire period of the test (70 °F). The outside damper is controlled to provide 20 cfm/person outside ventilated air. The calculation of the outside air fraction for the base case is detailed below:

The maximum load imposed to the East zone is calculated as follows:

People's cooling load = 12 person × 75 W/person

= 900 W

Lighting cooling load = 18 bulbs × 40 W/bulb

= 720 W

Hence, the maximum cooling load = 900 + 720= 1,620 W

(5,530 Btu/hr)

The total cooling load needed is (Kreider and Rabl, 1994):

$$Q_{air} = V\rho c_p \Delta T \qquad \text{Eq. 4.3}$$

where,

ρ = average air density (0.062 lb_m/ft^3)

c_p = air specific heat (0.24 $Btu/(lb_m\ F)$)

V = maximum volumetric flow rate supplied (ft^3/min)

ΔT = difference between zone temperature and supply air temperature (70-56 = 14 °F)

Hence, $V = 26{,}550\ ft^3/hr = 442\ cfm$. The actual experimental value was found to be 460 cfm, with a percent difference of 3.8%.

The outside air fraction is:

The maximum outside cfm = 12 persons × 20 cfm/person =240 cfm

OA Fraction (%) = (maximum outside cfm/ maximum supplied cfm)×100

= 52%

Performing the same calculations for the West zone with a cooling load of 1020 W (3,480 Btu/hr, 4 persons and 18 40-watts bulbs), the supplied cfm is found to be 243 cfm. The actual experimental supplied cfm is 250 cfm, which makes a percent difference of 2.5%. The minimum outside air for this zone is 80 cfm (4 ×20 cfm).

For multiple spaces, the outside air fraction should be calculated according to equation 6.1 in the ASHRAE Standard 62-1989. Based on the equation:

X= (220+80)/(250+460) = 0.4225

Y= 220/460 = 0.48

Z= 0.4225/(1+0.4225-0.48) = 0.45

Therefore, the outside air fraction is 45% when both zones are used at peak occupancy. Setting the damper position at 45% outside air does not guarantee that the required amount of outside air is provided to all zones. Therefore, the damper was controlled to bring the 45% of outside air by creating a virtual channel that uses airflow rates measured for the recirculated air and the outside. In the virtual channel, a ratio of total outside air to total supply air, $OA_{Fraction}$ is calculated as follows:

$$OA_{Fraction} = \frac{Outside\ airflow\ rate}{total\ supply\ airflow\ rate} = \frac{Outside\ airflow\ rate}{(Outside\ airflow\ rate + Recirculated\ airflow\ rate)} = 45\% \qquad \text{Eq. 4.4}$$

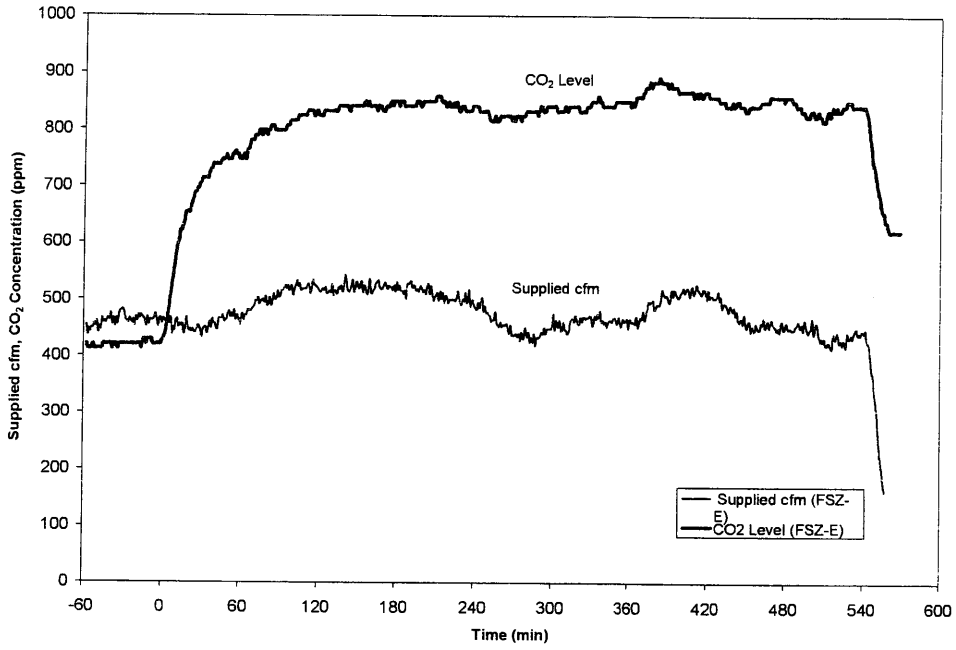


Figure 4.4a: Two-Zone Control of CO_2 and Temperature in the FSZ-East (Base Case, OA Fraction = 45%)

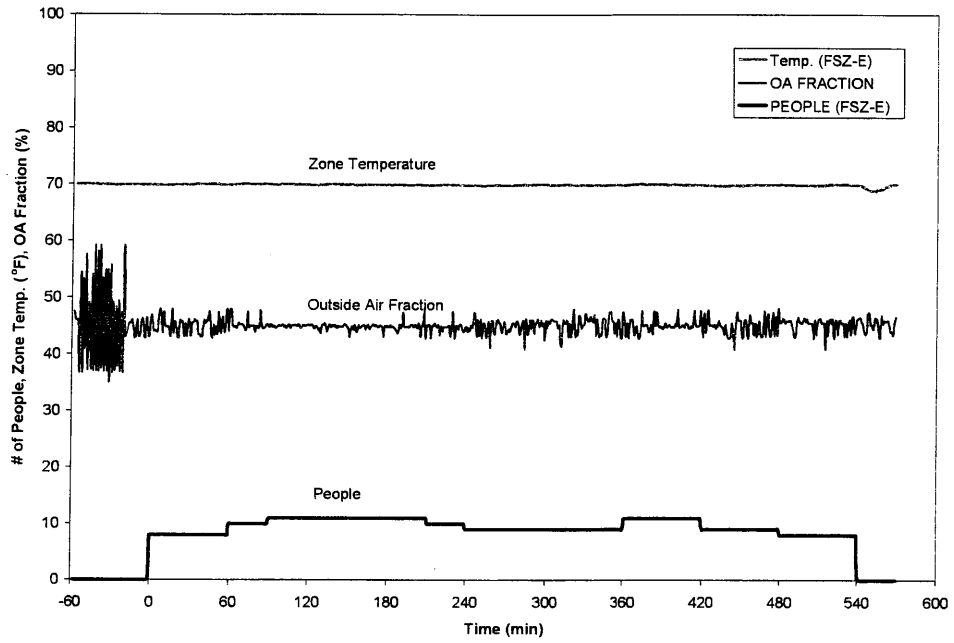


Figure 4.4b: Two-Zone Control of CO_2 and Temperature in the FSZ-East (Base Case, OA Fraction =45%)

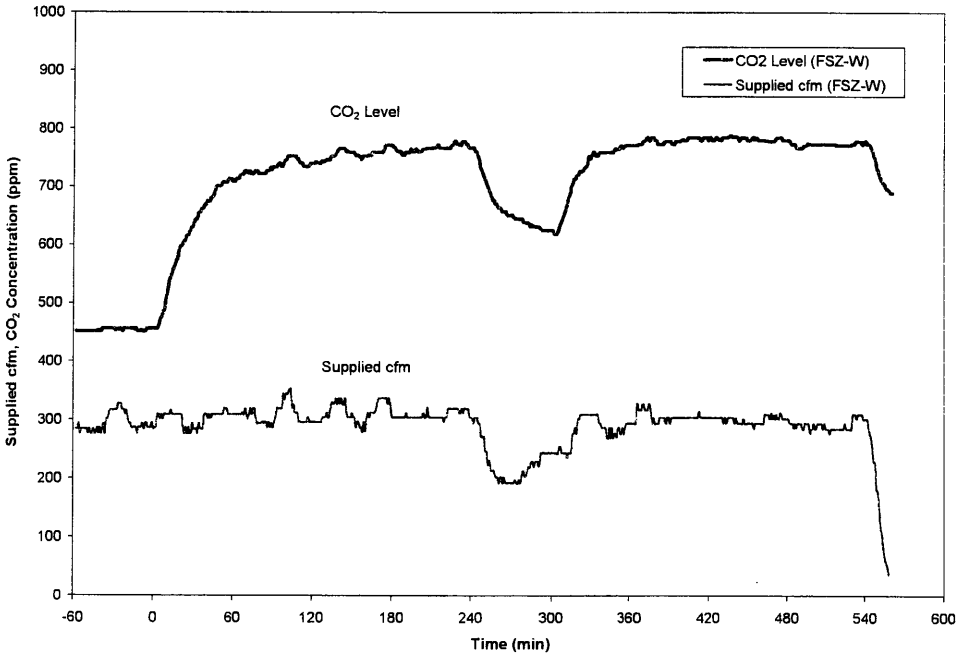


Figure 4.5a: Two-zone Control of CO_2 and Temperature in the FSZ-West (Base Case, OA Fraction =45%)

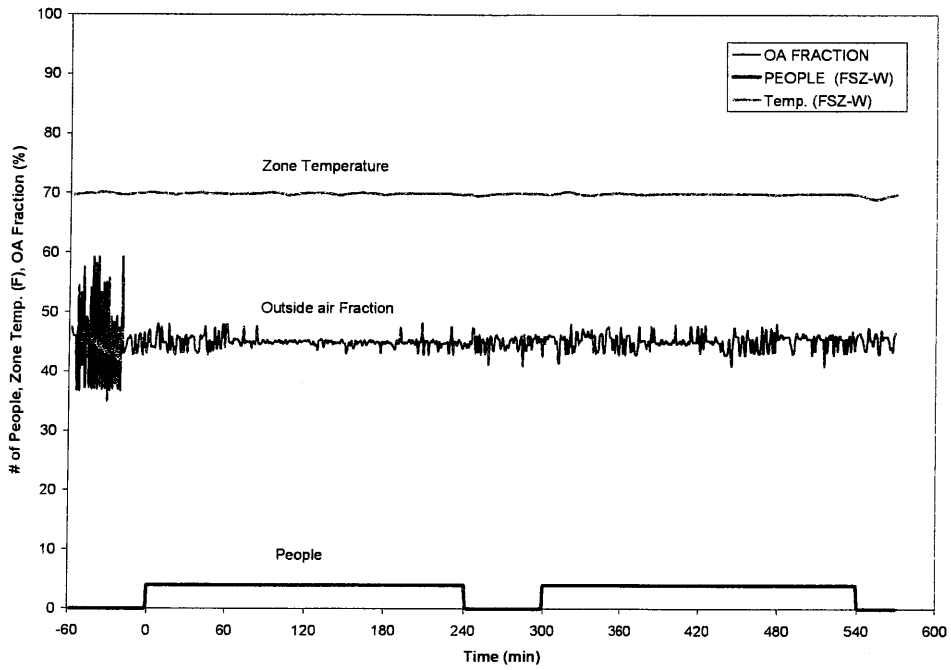


Figure 4.5b: Two-zone Control of CO_2 and Temperature in the FSZ-West (Base Case, OA Fraction =45%)

The following can be noted from the base case control strategy experiment:

1. The outside damper was well controlled at the 45% outside air. Just before the beginning of the experiment, the damper was fluctuating around the setpoint. To bring the damper to setpoint, the gain constants that were used (base case control strategy) are $K_p = 1.6$, $K_i = 0.05$ and $K_d = 0.19$.
2. The cfm supplied to the East zone is fluctuating to satisfy the cooling load with varying occupancy. In the West zone, however, the supplied cfm is almost constant because the occupancy profile is constant (four people). The time when there is no people (between 12:00 and 13:00), the cfm dropped to satisfy lighting load only.

3. Because the VAV system varies the amount of supplied cfm according to the cooling load imposed, the CO_2 concentration does not seem to fluctuate with the cfm in the base case experiment. That is, a low amount of supplied cfm to the zone implies that the number of people is low, too, and vice versa. The CO_2 concentration dropped to the minimum in the West zone when there were no people.
4. The average concentration in the East zone is 819 ppm. This means that there is a chance of outdoor air savings to bring the CO_2 level to the 900-ppm setpoint. These savings can be accomplished by the control strategies.
5. In both zones, the temperature was well maintained at setpoint (70 °F) through the variation of supplied cfm at 55 °F discharge air temperature.

4.2.1.2 ON/OFF Control Strategy

The ON/OFF control strategy resets the outside damper to allow outside airflow based on the CO_2 concentration level for the critical zone. The damper opens and closes based on the following conditions:

- CO_2 set-point = 900 ppm
- Outside damper 100% open at CO_2 Level > 910 ppm
- Outside damper 100% close at CO_2 Level < 890ppm

As the case with the base case, a virtual channel was used to control the damper. Figures 4.6a and 4.6b show the indoor CO_2 concentration and the temperature profiles for the East zone (critical zone). The West zone CO_2

concentration and the supplied cfm profiles are shown in Figure 4.7a and 4.7b. The temperature set-point is kept constant at 70 °F for both zones. As shown in Figure 4.6 b, the on/off control strategy opens the damper to a 100% when the CO_2 concentration reaches the upper CO_2 limit. When the concentration drops below the lower limit, it shuts off. In both positions, however, 100% open or 100% close can not be fulfilled. That is because when the damper is 100% open, still there is leakage of return air (about 13%), and when the damper is close, still there is a leakage of outside air. This problem is a known characteristic of outside dampers. The average concentration in the East critical zone is 897 ppm, hence maintaining the setpoint at 900 ppm.

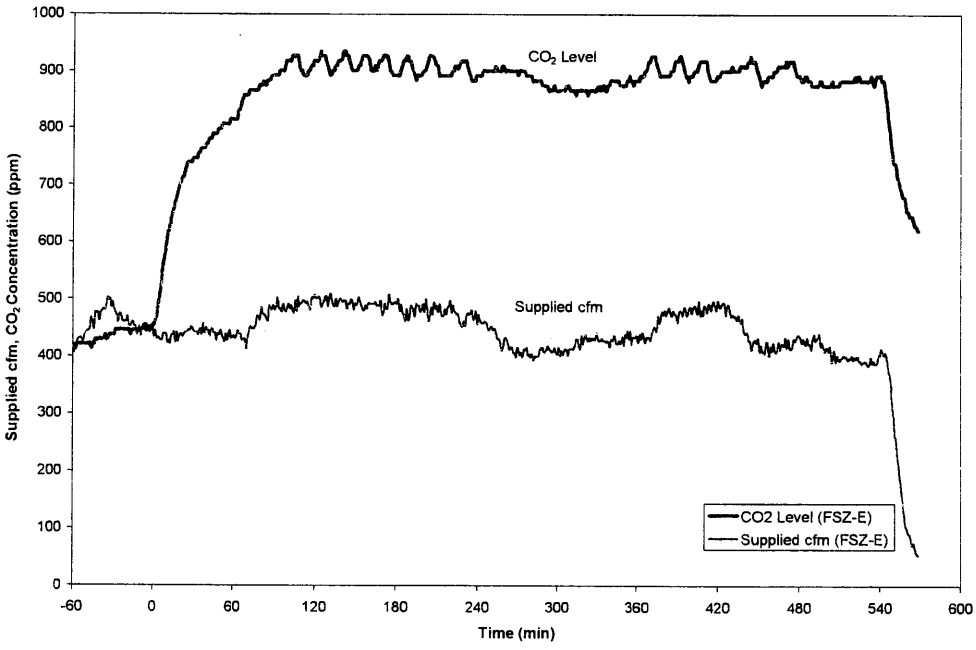


Figure 4.6a: Two-zone Controls of CO_2 and Temperature in the FSZ-E (ON/OFF Control)

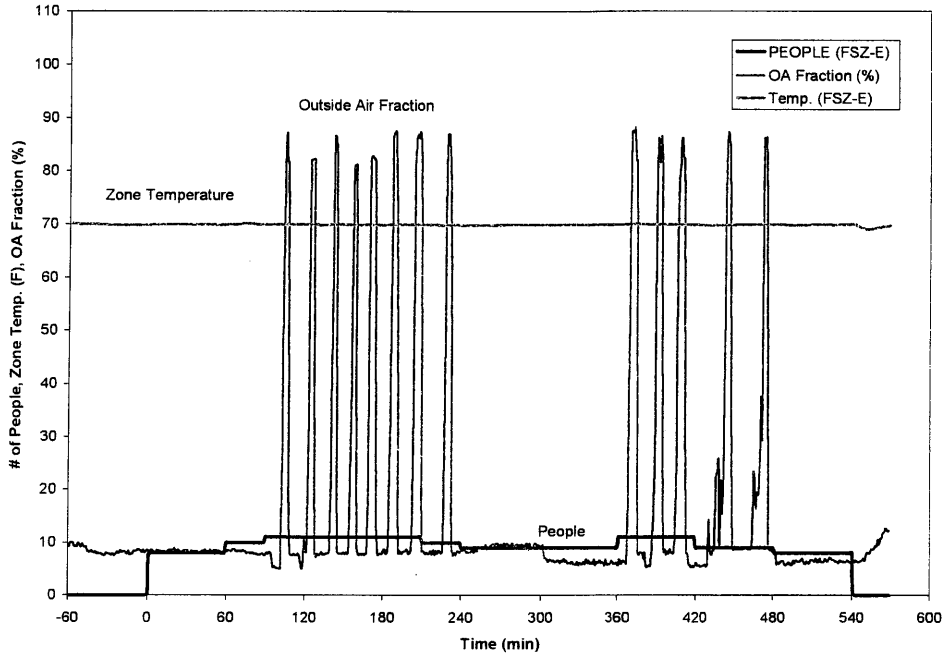


Figure 4.6b: Two-zone Control of CO_2 and Temperature in the FSZ-E (ON/OFF Control)

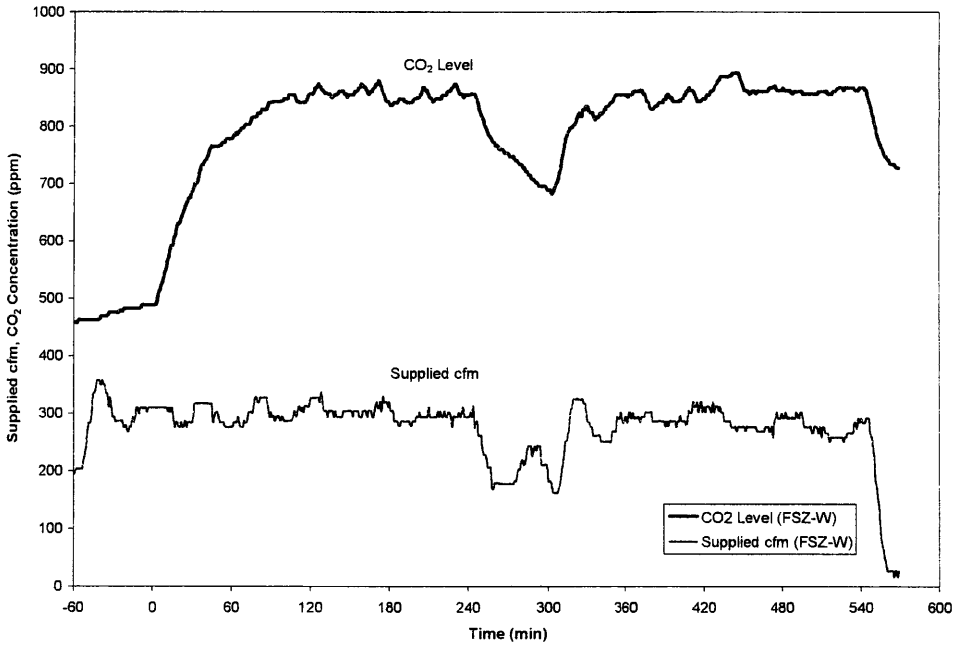


Figure 4.7a: Two-zone Control of CO_2 and Temperature in the FSZ-W (ON/OFF Control)

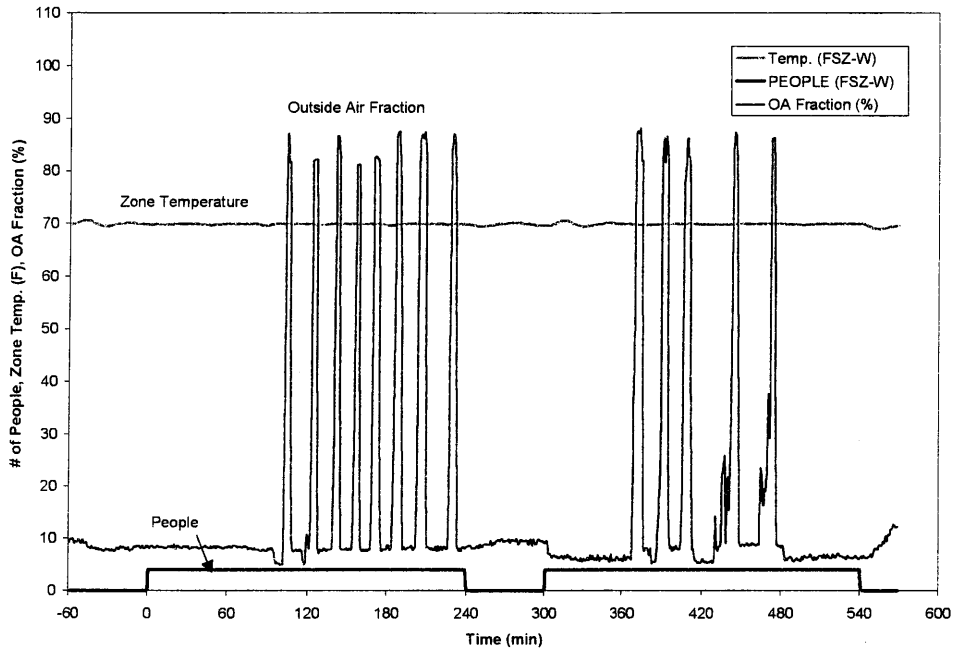


Figure 4.7b: Two-zone Control of CO_2 and Temperature Setpoints in the FSZ-W (ON/OFF Control)

4.2.1.3 PID Control Strategy

Figures 4.8a, 4.8b, 4.9a, and 4.9b show the results of a PID controller for both zones using K_p = 5.0, K_i = 0.11, and K_d =0.09 in Equation 4.2. These constants were the best possible constants found by trial and error for this experiment. For the FSZ-East zone, the CO_2 concentration is within the range of $\Delta C = \pm 50$ ppm, with an average CO_2 concentration of 900 ppm. The CO_2 concentration profile in the FSZ-West zone, as predicted, has the same behavior as that of the base case and the On/Off control strategy. However, the average CO_2 concentration in the West zone is different. With the PID control strategy, the average concentration in the West zone is 782 ppm, while with the On/Off control strategy, the concentration is 838 ppm. This resulted in further reduction of total outside air consumption in favor of the On/Off control strategy.

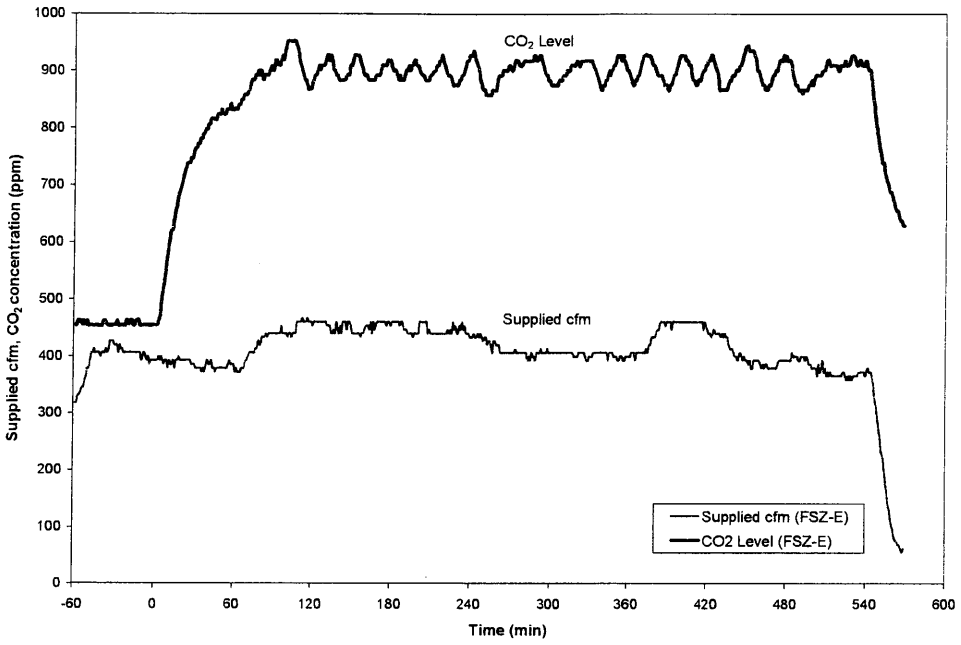


Figure 4.8a: Two-zone Controls of CO_2 and Temperature in the FSZ-East (PID Control)

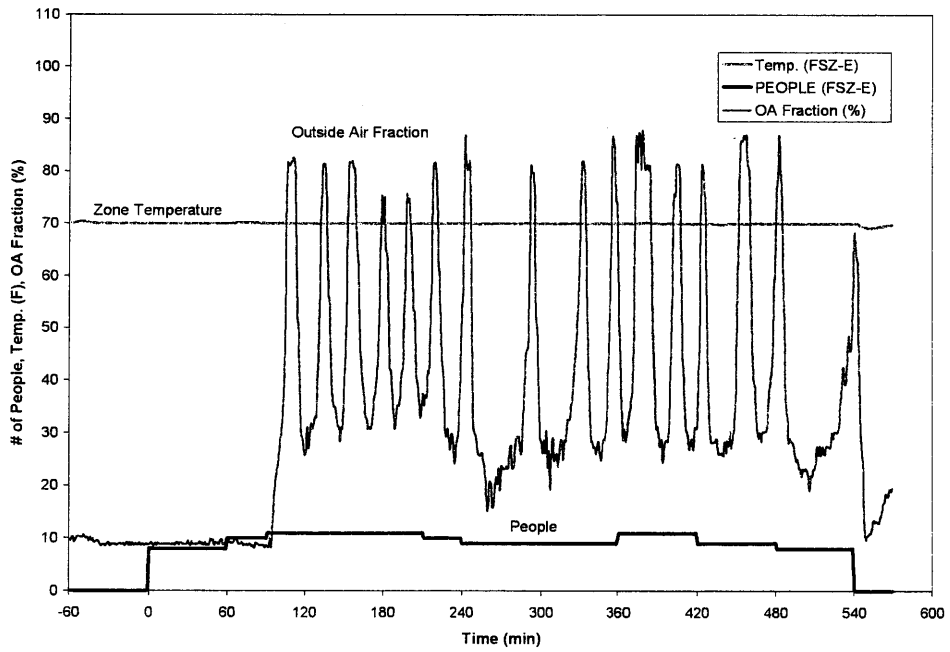


Figure 4.8b: Two-zone Controls of CO_2 and Temperature in the FSZ-East (PID Control)

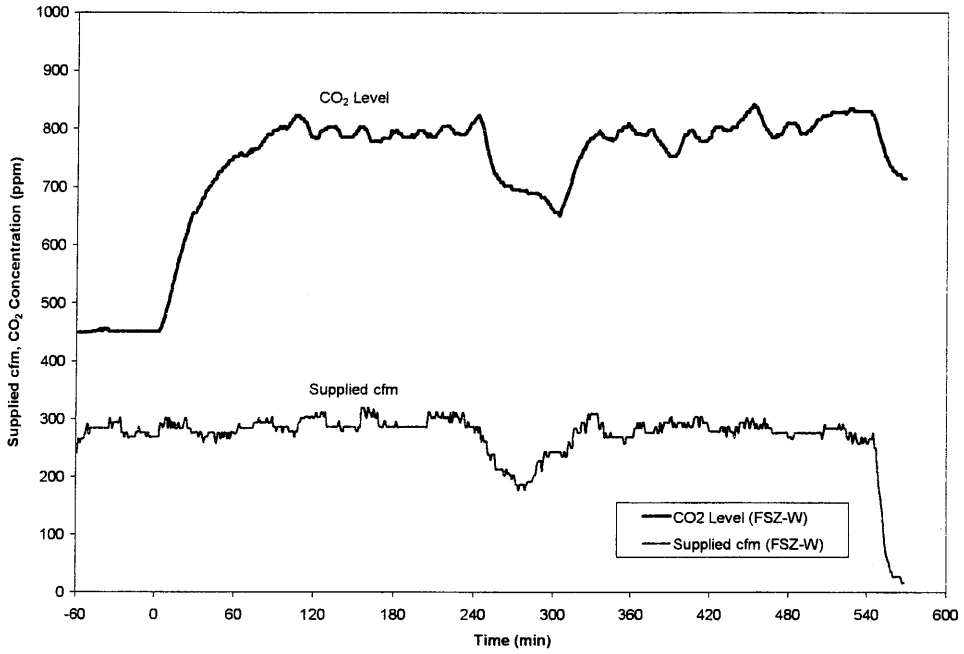


Figure 4.9a: Two-zone Controls of CO_2 and Temperature in the FSZ-West (PID Control)

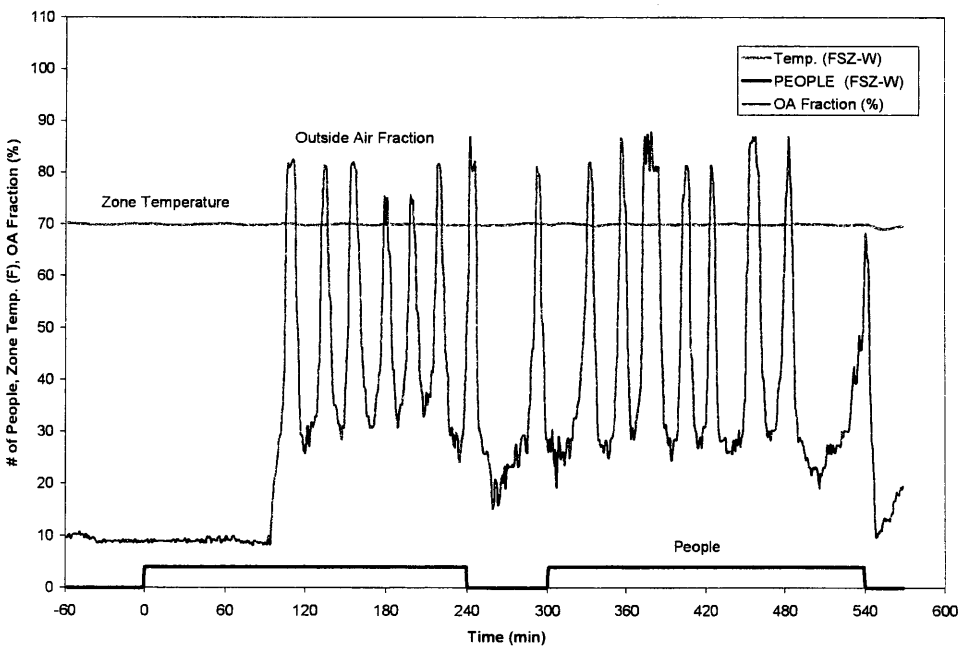


Figure 4.9b: Two-zone Controls of CO_2 and Temperature in the FSZ-West (PID Control)

4.2.2 One-Zone Control of CO_2 and Temperature

The experiments for a one-zone control of CO_2 concentration and temperature are performed to control both indoor temperature and CO_2 level in one zone only. The CO_2 set-point is 900 ppm and the indoor temperature set-point is 70 °F.

4.2.2.1 Base-Case Experiment

In the base case experiment, the indoor CO_2 level is monitored when 52 % outside air (20 cfm/person and 12 people) is supplied to the East critical zone (refer to Figures 4.10a and 4.10b). The average concentration in the zone during the test is 832 ppm implying that there is a change of outdoor air reduction and thus energy savings when the CO_2 level is raised to 900-ppm setpoint.

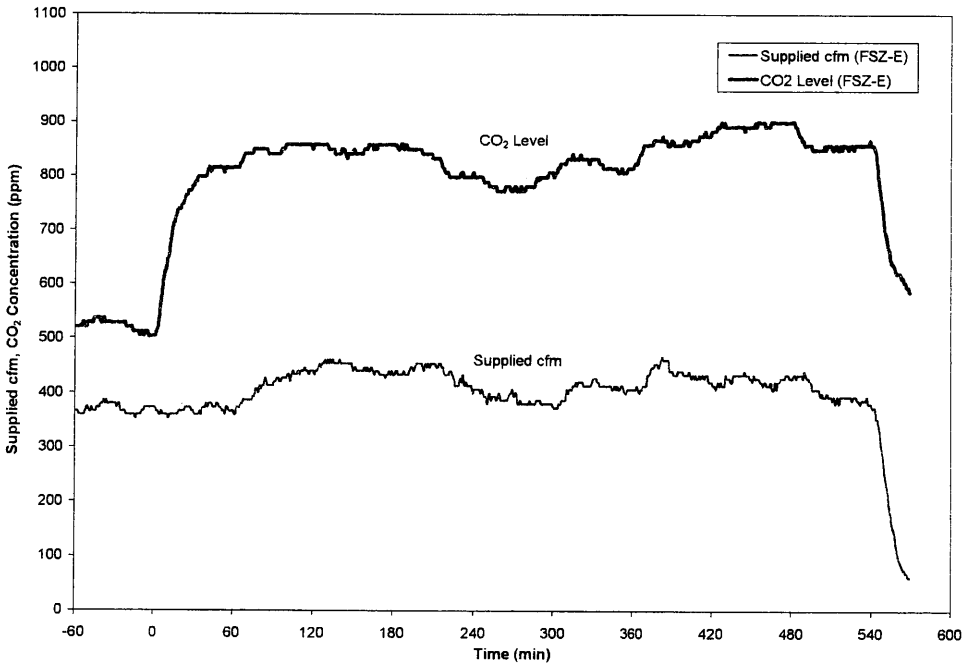


Figure 4.10a: One-zone Control of CO_2 and Temperature in the FSZ-East (Base-Case, OA Fraction =52%)

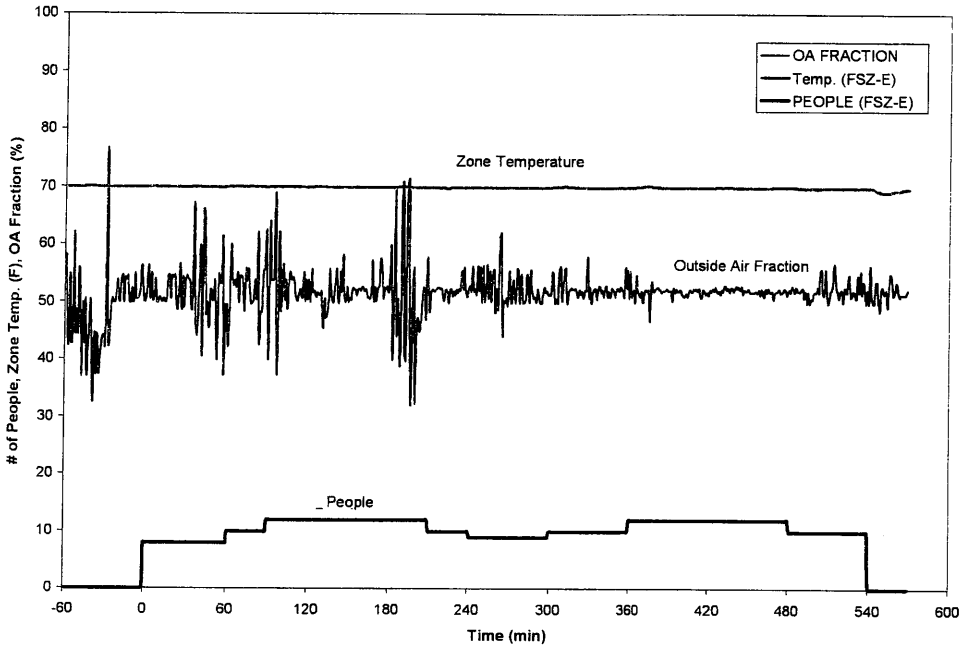


Figure 4.10b: One-zone Control of CO_2 and Temperature in the FSZ-East (Base Case, OA Fraction =52%)

4.2.2.2 ON/OFF Controls Strategy

For the ON/OFF strategy, the damper position is controlled using the following actions:

- CO_2 set-point = 900 ppm
- Outside damper 100% open at CO_2 Level > 910ppm
- Outside damper 100% close at CO_2 Level < 890 ppm

The same virtual channel described for the two-zone experiments is used to control the damper position. Figures 4.11a and 4.11b show the results of the experiment performed using the ON/OFF strategy for the FSZ-E zone. Figure

4.11a shows that the CO_2 concentration starts with the low value close to the outdoor CO_2 level at the beginning of the test, and increases gradually when people start to occupy the zone. When the CO_2 concentration reaches the setpoint level, the outside air damper opens fully to lower the CO_2 concentration back to the set-point. The outside damper fully closes when the concentration starts to go below the set-point. During the duration of the test, the temperature is maintained well within the 70 °F set-point. The average CO_2 level in the East zone is 880 ppm.

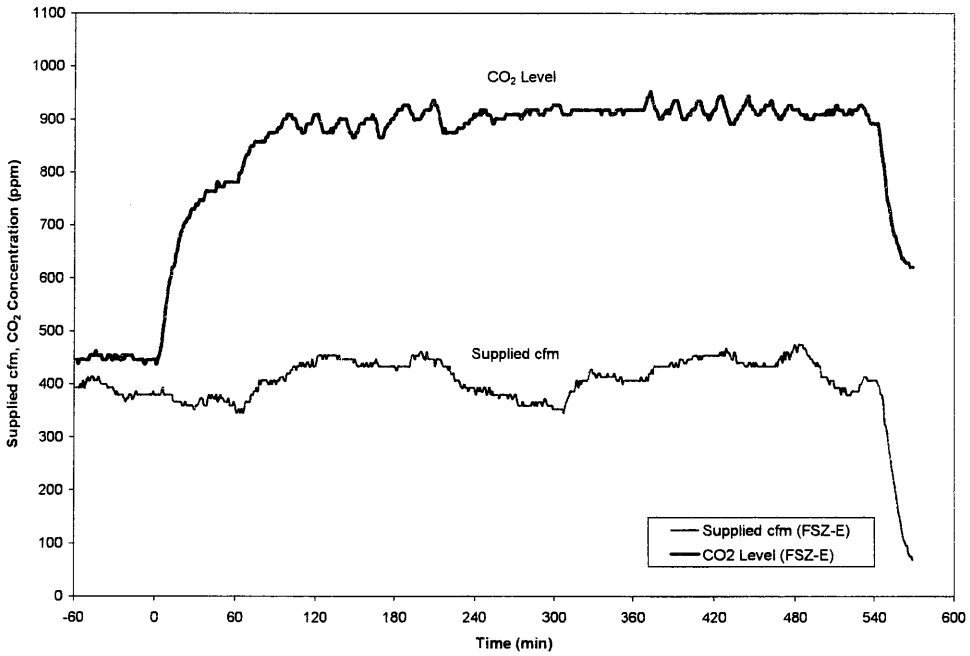


Figure 4.11a: One-zone Control of CO_2 and Temperature in the FSZ-East (ON/OFF Control)

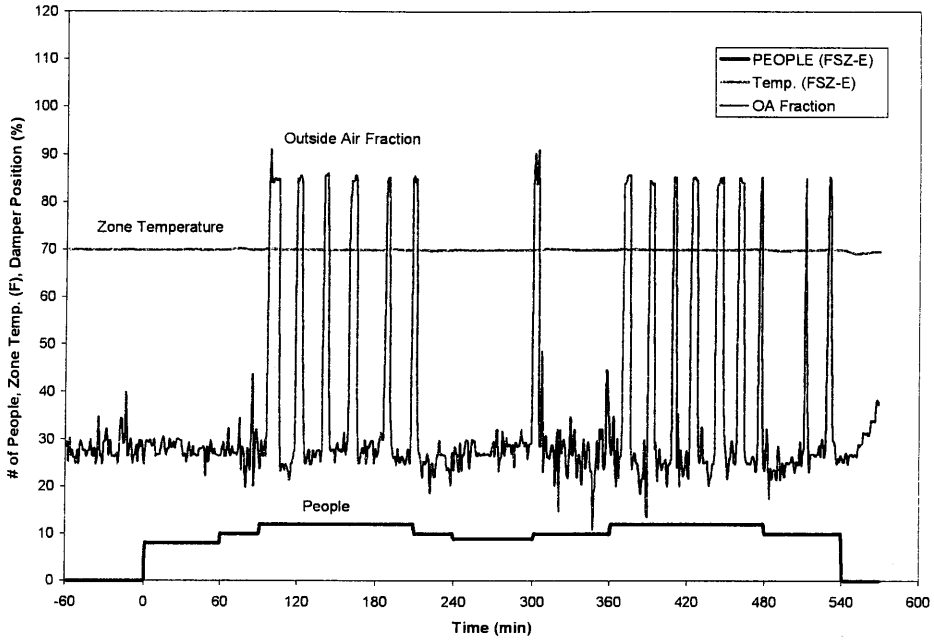


Figure 4.11b: One-zone Control of CO_2 and Temperature in the FSZ-East (ON/OFF Control)

4.2.2.3 PID Control Strategy

The PID control strategy applied to one zone maintains successfully the CO_2 concentration within the acceptable range with an average value of 901 ppm. When compared to the previous On/Off experiments, where the average value is 880 ppm, it becomes very clear that the PID controller must consume less amount of outside air. Therefore, the PID controller in this case worked better. The results of the PID control strategy are shown in Figures 4.12a and 4.12b. The amount of total supply cfm to the zone is maintained at the same level for all one-zone tests and the temperature kept at the desired set-point.

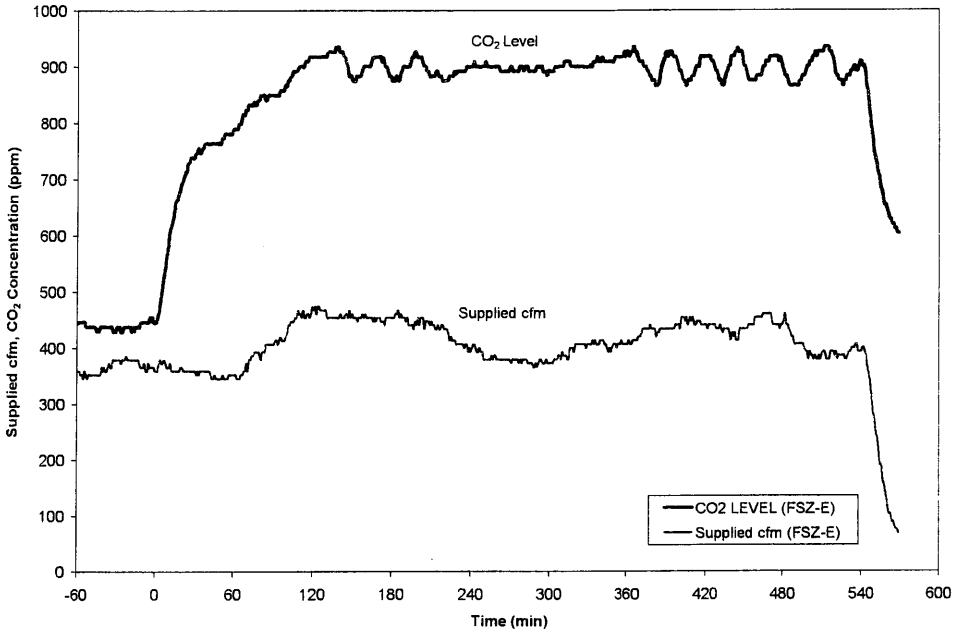


Figure 4.12a: One-zone Control of CO_2 and Temperature in the FSZ-East (PID Control)

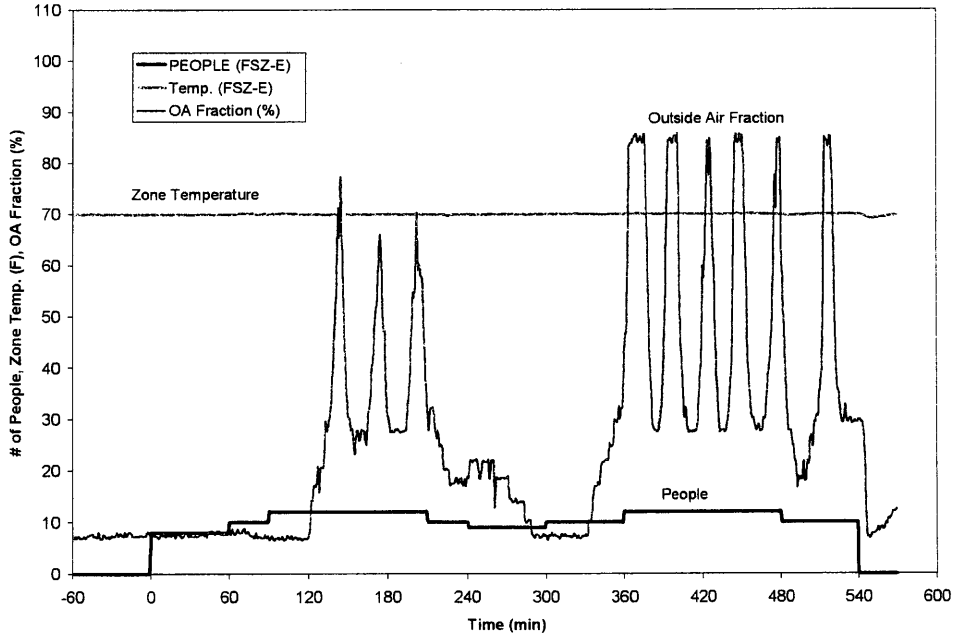


Figure 4.12b: One-zone Control of CO_2 and Temperature in the FSZ-East (PID Control)

4.3 Comparative Analysis of the Performance of Control Strategies

Table 4.2 lists a summary of the total amount of outside air ventilation required for each control strategy in the one-zone experiments when both indoor air temperature and CO_2 level are controlled. The PID control strategy required the lowest amount of outside air with only 64,874 ft^3 (percent reduction of about 44% compared to the base case). More savings can be expected if the true PID constants are found. The ON/OFF control strategy is also better than the base case, but with only a reduction of 32.4% of total outside airflow. The following points can be noted from the results:

1. When the outside opens fully for the On/Off control strategy, the outside air percent is only 87% even though the damper is 100% open. This means there is still 13% of recirculated air leakage.
2. The fluctuations in the outside air fraction in the two-zone base case experiments at 52% are caused by the return fan as it tries to maintain the differential pressure in the zone at the 0.05 in WG setpoint. That is, as the pressure in the zone increases, the fan pulls more air and pushes it to the air-handling unit, which in turn increases the recirculated air. The mixed air damper, then reacts and tries to adjust itself by closing the recirculated air damper and so on.
3. All CO_2 concentration values in the control strategies fall within the limits of ± 50 ppm.
4. The temperature in the zone for all tests was maintained at the 70 °F setpoint.

Table 4.2: Comparisons of Control Strategies Parameters of the One-zone Experiments

Parameter	Control Strategy			Percent Difference (%)	
	Base Case	PID	On/Off	PID	On/Off
Average Supplied Air (cfm)	414±10	414±10	414±10	0.0	0.0
Supply Duct Static Pressure (in WG)	0.8±0.008	0.8±0.008	0.8±0.008	0.0	0.0
Initial Zone CO_2 Level (ppm)	520±27	445±27	446±27	14.42±6.8	14.23±6.8
Zone Differential Pressure (in WG)	0.05 ±0.0005	0.05 ±0.0005	0.05 ±0.0005	0.0	0.0
Zone Average Temperature (F)	70.0±0.1	70.0±0.1	70.0±0.1	0.0	0.0
Initial Zone Supplied cfm	366±10	360±10	376±10	1.64±3.8	-2.73±3.9
Average CO_2 Concentration During the Test (ppm)	832±27	901±27	880±27	n/a	n/a
Average Outside Air Consumption (cfm)	215±10	120±10	145±10	44.2 ± 5.4 (reduction)	32.6 ± 5.6 (reduction)
Maximum CO_2 Level (ppm)	858±27	936±27	952±27	n/a	n/a
Minimum CO_2 Level (ppm)	841±27	866±27	866±27	n/a	n/a

For the two-zone experiments, the results are shown in Table 4.3. In particular, the On/Off control strategy required the lowest amount of outside air (63.2% reduction), followed by the PID and then the base case control strategy. The average CO_2 level in the critical zone (East) during the test is 899 ppm for the PID controller and 897 ppm for the On/Off.

Comparing the control strategies for the maximum concentration reached, the base case is the best. The On/Off and the PID control strategies worked most of the time within the upper limit (950 ppm). Some general notes can be listed for the experiments as follows:

1. All tests start at 0 minute time and finish at 540 minutes time, that is each test last for one full business day of 9 hours (from 8 a.m. to 7 p.m.).
2. The critical zone is the FSZ-E zone, while the FSZ-W zone is the non-critical for all tests.
3. The percent differences in the tables are differences compared with the base case where applicable.
4. For all tests, the cooling load consists of people load and lighting load. The maximum cooling load is set so that it does not exceed the maximum load on the VAV box (550 cfm).

Table 4.3: Comparisons between Control Strategies Parameters for the Two-zone Tests

Parameter	Zone	Control Strategy			Percent Difference	
		Base Case	PID	ON/OFF	PID	ON/OFF
Average total Supplied Air (cfm)		755±14	725±14	730±14	3.9±2.6	3.3±2.6
Average Supplied Air to Zone (cfm)	East	481±10	447±10	448±10	7.0±2.8	6.9±2.8
	West	294±10	278±10	282±10	5.4±4.7	4.1±4.7
Supply Duct Static Pressure (in WG)		0.8 ±0.008	0.8 ±0.008	0.8 ±0.008	0.0	0.0
Zone Differential Pressure (in WG)	East	0.05 ±0.0005	0.05 ±0.0005	0.05 ±0.0005	0.0	0.0
Initial Zone CO_2 Level (ppm)	East	420±27	452±27	463±27	-7.5±10.5	-10.2±10.7
	West	454±42	457±42	474±42	-0.5±13.1	-4.4±13.4
Zone Temperature During the Test	East	70.0±0.1	70.0±0.1	70.0±0.1	0.0	0.0
	West	70.0±0.1	70.0±0.1	70.0±0.1	0.0	0.0
Initial Zone Supplied cfm	East	462±10	422±10	457±10	8.5±2.9	1.0±3.0
	West	294±10	278±10	291±10	5.0±4.7	1.0±4.8
Average CO_2 During the Test (ppm)	East	819±27	899±27	897±27	n/a	n/a
	West	732±42	782±42	838±42	n/a	n/a
Maximum CO_2 Value during the Test (ppm)	East	892±27	953±27	935±27	n/a	n/a
	West	789±42	842±42	894±42	n/a	n/a
Minimum CO_2 Value during the Test (ppm)	East	798±27	858±27	858±27	n/a	n/a
	West	619±42	651±42	683±42	n/a	n/a
Average Outdoor Air Consumption (ft^3)		349+10	275+10	128+10	21.2±3.6 (reduction)	63.3±3.1 (reduction)

4.4 Uncertainty Analysis

The uncertainty in the readings of CO_2 sensors shown in both Tables 4.2 and 4.3 are calculated according to the following equation (Holman, 1994):

$$w = \pm t_{\alpha,n-2} \frac{\sigma}{\sqrt{n}} \qquad \text{Eq. 4.5}$$

where:

w = uncertainty in the sensor reading, ppm

n = number of data points for each sensor in the calibration process (n=9)

$t_{\alpha,n-2}$ = value of the t distribution with n-2 degrees of freedom at a confidence level of α= 95, t_{95}= 2.365

σ= standard error, ppm

The standard error is calculated according to the following equation (Holman, 1994):

$$\sigma = \sqrt{\frac{\sum_{i}^{n} \left(Y_{measured_i} - Y_{actual_i}\right)^2}{n-2}} \qquad \text{Eq. 4.6}$$

Table 4.4 shows the standard error and the uncertainty in the CO_2 sensors. The uncertainty values are calculated based on t_{95}= 2.365. Table 4.5 shows the uncertainties in the instrumentation used in the laboratory as provided by the manufacturers.

Table 4.4: Uncertainty in CO_2 Sensors Readings

Sensor	Standard Error (ppm)	Uncertainty (ppm)
H1	62	49
H2	27	21
H3	34	27
E4	101	80
J6	103	81

Table 4.5: Equipment and Instrumentation Accuracy

Measurement	Manufacturer	Method	Accuracy
Temperature (°F)	MAMAC Systems	Calibrated Resistance Temperature Detectors (RTDs)	±0.1
Pressure (in WG)	MAMAC Systems	Pressure Transducers	±1.0%
Airflows (ft/min)	Ebtron, Inc.	Heated thermistor sensors	±10 for <500 ft/min

Since sensors H1 and H2 are placed in the FSZ-E zone and sensors H3 and E4 are placed in the FSZ-W zone, the average CO_2 concentration is calculated as follows:

$$C_{avg} = \frac{C_{S1} + C_{S2}}{2} \qquad \text{Eq. 4.7}$$

where:

C_{avg} = Average concentration, ppm

C_{S1}, C_{S2} = Concentration measured by sensor 1 and sensor 2, ppm

Hence, the uncertainty in the average sensor value is calculated as follows:

$$w^2_{C_{avg}} = \left(\frac{\partial C_{avg}}{\partial C_{S1}} w_{C_{S1}}\right)^2 + \left(\frac{\partial C_{avg}}{\partial C_{S2}} w_{C_{S2}}\right)^2 \quad \text{Eq. 4.8}$$

where:

w_{Cavg} = the uncertainty in the average CO_2 concentration, ppm

w_{Cs1} = the uncertainty in CO_2 concentration of sensor 1, ppm

w_{Cs2} = the uncertainty in CO_2 Concentration of sensor 2, ppm

The solution of this equation is:

$$w_{C_{avg}} = \sqrt{\frac{\left(w_{C_{S1}}\right)^2 + \left(w_{C_{S2}}\right)^2}{4}} \quad \text{Eq. 4.9}$$

Hence, the combined uncertainty in sensors H1 and H2 is 27 ppm and the combined uncertainty in sensors H3 and E4 is 42 ppm.

Likewise, the uncertainty in the outside air fraction is calculated in the same manner as above, where the equation for calculating the outside air fraction is:

$$F = \frac{cfm_{OA,BC} - cfm_{OA,Control}}{cfm_{OA,BC}} \quad \text{Eq. 4.10}$$

where:

F= outside air fraction

$cfm_{OA,BC}$ = outside air cfm for the base case control strategy

$cfm_{OA,Control}$ = outside air cfm for the control strategy

And the uncertainty in the outside air fraction is:

$$w_F = \sqrt{\left(\frac{1}{cfm_{OA,BC}}\right)^2 \times (w_{control})^2 + \left(\frac{cfm_{OA,control}}{cfm^2_{OA,BC}}\right)^2 \times (w_{BC})^2} \qquad \text{Eq. 4.11}$$

where:

w_F = uncertainty in the outside air fraction

w_{BC} = uncertainty in cfm for the base case control strategy

$w_{control}$ = uncertainty in cfm for the control strategy

4.5 Conclusions

In this chapter, the results of experiments are presented to evaluate the effect of various CO_2-based demand-controlled ventilation strategies on HVAC system energy consumption under controlled and repeatable test conditions. Three control strategies were evaluated: (i) the base case control strategy that controls the outside damper to provide a fixed outside air fraction equivalent to 20 cfm/person; (ii) the ON/OFF control strategy that opens and closes the damper as a function of the CO_2 level deviation from the set-point; and (iii) a PID controller that controls the damper position according to the error involved in the control process. Another control strategy was also evaluated with the damper controlled proportionally to the zone CO_2 concentration level.

The results of the one-zone control of CO_2 experiments showed that a PID controller maintained acceptable indoor CO_2 level with the minimum outdoor air

requirements, followed by the proportional control, and lastly the ON/OFF control. A reduction in outside air of about 33 % of outside air can be obtained when applying the PID controller instead of the On/Off control strategy.

The one-zone controls to maintain indoor CO_2 concentration and temperature performed in the third set of experiments indicated that the PID control strategy provides the minimum outdoor air requirements followed by the On/Off controller with a percent reduction of about 44.0% and 32.4%, respectively. The two-zone experiments, however, showed that the On/Off provided the lowest requirement of outside air. The percent reduction for the PID and the On/Off control strategies as compared to the base case is 21.2% and 63.2%, respectively.

CHAPTER FIVE

Modeling of Multiple Zone Pollutant Transport

This chapter describes a computer program that was developed to model the transport of indoor pollutants through the HVAC system of a multiple-zone building. The model described is based on the module developed by Knoespel (1991). Besides the new features added to the program, the model was modified to be more flexible, especially with regard to the simulation time step so it can provide results for any time-step (as short as one minute).

5.1 Model Description

A pollutant-mass-balance model was used to develop a computer program that can be used to simulate the transport of contaminants within a multiple-zone building conditioned by a HVAC system. The model is based on a nodal analysis applied to various building elements. In particular, the model assumes that the nodes have a capacitance for pollutant and uses a pollutant mass balance for each node to solve for the pollutant concentrations. Therefore, the equations of the model can be solved as an initial-value problem resulting in pollutant concentration values as a function of time for any period during which the pollutant source and inter-zone flows may vary. The model developed in this chapter is a transient model with time-dependent variables. Some of the important features of the model are summarized below:

1. Infiltration and inter-zone flows are input variables to the program. Thus, they can be calculated outside the program or estimated from experimental tests.
2. Ventilation requirements in the zones are calculated by the VAV model. The VAV model is discussed in the next chapter.
3. The model is capable of varying the pollutant source level within each zone during the simulation by defining adequately the number of people present.
4. The circulation time of the air traveling through the ventilation system is taken into account by sizing the ducts from the air-handling unit to the zones.
5. Different occupancy schedules during the weekday and the weekend can be simulated.
6. Removal efficiency of the pollutant (due for instance to air filters installed in the ductwork) can be an input variable in the model.
7. Calculation of pollutant concentrations can be calculated in three different locations, namely: in the zones, the return duct, and the supply duct.

5.2 Model Equation System

Figure 4.1 shows a schematic of the transport model of the pollutant within the ventilation system and the zones. In general, for a typical zone, the transport of a pollutant occurs through three possible paths as indicated in Figure 4.1:

1. Circulation flow in and out of the zone.
2. Air infiltration in and out of the zone.
3. Inter-zone flow in and out of the zone.

The flow of the pollutant can be determined by multiplying the airflow rates by the concentration of pollutant within the air streams. The zones may also have pollutant generation sources .

The pollutant transport model is based on a mass balance of pollutant applied to each zone. The relationship between pollutant-mass flow in, pollutant-mass flow out, and the change in mass is provided in Equation 5.1 (Knoespel, 1991).

$$\frac{dm_p}{dt} = \dot{m}_{p,\,in} - \dot{m}_{p,\,out} \qquad \text{Eq. 5.1}$$

where:

$\dot{m}_{p,in}$ = pollutant mass flow in the zone
$\dot{m}_{p,out}$ = pollutant mass flow out of the zone

For this model, the density of the pollutant is assumed to be 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 5.1 are divided by the pollutant density, a volume balance equation is obtained as indicated in Equation 5.2. Again, the volumes are partial volumes occupied by the pollutant.

$$\frac{dV_p}{dt} = \dot{V}_{p,\,in} - \dot{V}_{p,\,out} \qquad \text{Eq. 5.2}$$

where:

$\dot{V}$ = partial volume of the pollutant in the zone

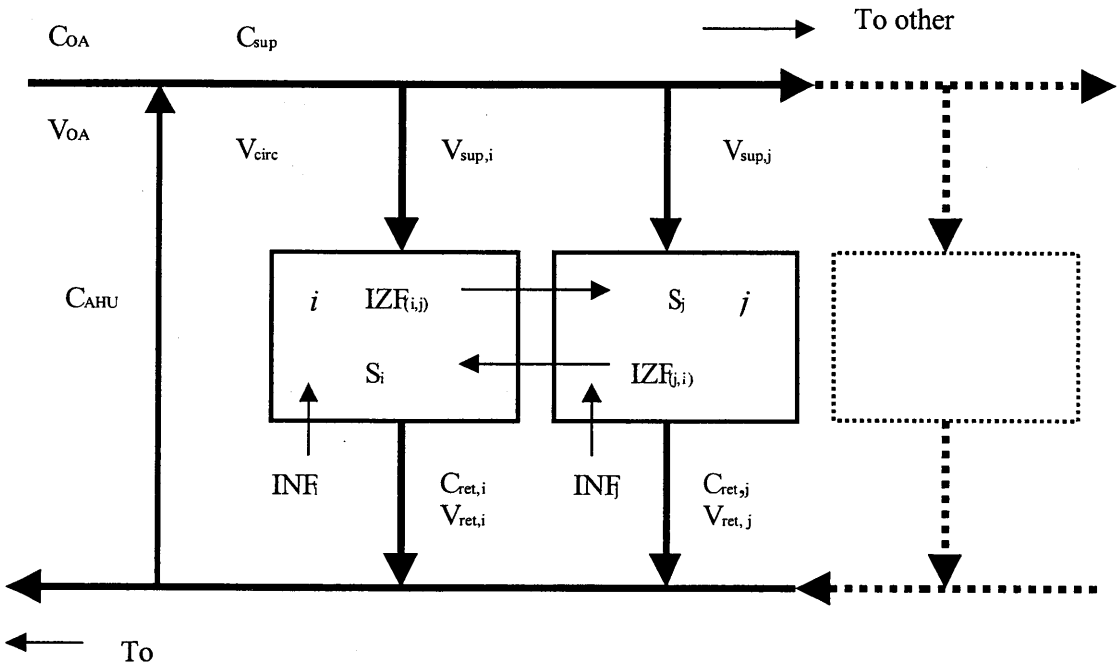


Figure 5.1: Model Used for the Pollutant Transport

The pollutant volume flow rate is equal to the air-volume flow rate times the volume fraction (or volume concentration) of the pollutant as indicated in Equations 5.3 and 5.4. The subscript *a* refers to air.

$$\frac{dV_p}{dt} = \dot{V}_{a,in}\left(\frac{V_p}{V_a}\right)_{in} - \dot{V}_{a,out}\left(\frac{V_p}{V_a}\right)_{out} \qquad \text{Eq. 5.3}$$

and,

$$\frac{dV_p}{dt} = \dot{V}_{a,in}C_{in} - \dot{V}_{a,out}C_{out} \qquad \text{Eq. 5.4}$$

If both sides of Equation 5.4 are now divided by the air volume in the zone (essentially equal to the nominal zone volume for dilute pollutant concentrations), the expression provided in Equation 5.5 is obtained. 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.

$$\frac{d\left(\frac{V_p}{V_a}\right)}{dt} = \frac{\dot{V}_{a,in}}{V_a}C_{in} - \frac{\dot{V}_{a,out}}{V_a}C_{out} \qquad \text{Eq. 5.5}$$

The left-hand side of equation 5.5 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 5.6.

$$\frac{dC}{dt} = \frac{\dot{V}_{a,in}}{V_a}C_{in} - \frac{\dot{V}_{a,out}}{V_a}C_{out} \qquad \text{Eq. 5.6}$$

If equation 5.6 is now expanded to include all of the possible airflow paths into or out of zone *i* as shown in Figure 5.1, four in-flow terms and two out-flow terms are obtained as illustrated in Equation 5.7. 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.

$$\frac{dC_i}{dt} = \frac{S_i}{V_{a,i}} + \frac{V_{\text{inf},i}}{V_{a,i}} C_{OA} + \frac{\dot{V}_{\text{sup},i}}{V_{a,i}} C_{\text{sup}} - \frac{\dot{V}_{ret,i}}{V_{a,i}} \varepsilon_{c,i} C_i + \frac{\sum_j (\dot{V}_{izf(j,i)} C_j)}{V_{a,i}} - \frac{\left(\sum_j \dot{V}_{izf(i,j)}\right) C_i}{V_{a,i}} \qquad \text{Eq. 5.7}$$

The description of the terms in equation 5.7 is as follows:

1. The first term on the right-hand side of equation 5.7 is the pollutant source located in zone *i*. The source S_i has units of volume per unit time.
2. The second term is the air infiltration volume flow rate into the zone, multiplied by the outside air pollutant concentration (typically constant).
3. The third and the fourth terms are for the air circulation flow through the zone. The supply airflow is multiplied by a variable supply air concentration to determine the pollutant flow into the zone. The return airflow 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 concentration divided by the zone average concentration and it accounts for the fact that the pollutant may not be fully mixed in the zone air. The value of the ventilation

effectiveness used in the model is determined as calculated experimentally in chapter four.

4. The last two terms are for the inter-zone flows of pollutants. The next to the last term is the flow into zone *i* from all other zones *j*. The terms $V_{izf(j,i)}$ represent the volume airflow from zone *j* into zone *i*. These terms are multiplied by the average concentration in zone *j* and summed over all zones to determine the total inter-zone pollutant flow into zone *i*. The last term, the inter-zone pollutant flow out of zone *i*, is obtained using a similar procedure, except that the air flows are multiplied by the pollutant concentration in zone *i*.

The air volume flow rate returning to the ventilation system from zone *i* is calculated by applying a mass balance of all the other air flows for the zone as indicated in equation 5.8.

$$\dot{V}_{ret,i} = \dot{V}_{\sup,i} + \dot{V}_{\inf,i} + \dot{V}_{izf(i,j)} - \dot{V}_{izf(j,i)} \qquad \text{Eq. 5.8}$$

The supply air concentration is calculated by a two-step procedure. 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 the zones as shown in equation 5.9. In this equation, t refers to the present time while DT is the return air time delay and is equal to the zone return air path volume divided by the return air volume flow rate for the zone.

$$C_{ahu} = \frac{\sum_i \left(\dot{V}_{ret,i} \varepsilon_{c,i} C_{i,t-DT} \right)}{\sum_i \dot{V}_{ret,i}} \qquad \text{Eq. 5.9}$$

The second step in calculating the supply air pollutant concentration C_{sup} accounts for some of the circulation air that may be exhausted to the outside and replaced with an equal volume of outdoor air. The concentrations are again flow-normalized as shown in equation 5.10.

$$C_{\text{sup}} = \frac{\left(\sum_i \dot{V}_{\text{sup},i} - \sum_i \dot{V}_{OA,i} \right) C_{ahu} + \left(\sum_i \dot{V}_{OA,i} \right) C_{OA}}{\sum_i \dot{V}_{\text{sup},i}} \qquad \text{Eq. 5.10}$$

5.3 Program Implementation

The pollutant transport model represented by equations 5.7 through 5.10 was implemented as a FORTRAN 77 simulation program. The differential equation represented by Equation 5.7, is solved by the iterative-modified Euler method for solving differential equations (TRNSYS, 1994). The concentrations for each zone at time $t+\Delta t$ are first estimated using the simple Euler method using the zone concentrations at time t as shown in equation 5.11. The superscript on concentration represents the iteration step number. The function F represents the

expression provided in the right-hand side of equation 5.7, and Δt is the program time step.

$$C^{1}_{t+\Delta t} = C_t + F(C_t)\Delta t \qquad \text{Eq. 5.11}$$

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 illustrated equation 5.12.

$$C^{n+1}_{t+\Delta t} = C_t + F\left(C^{n}_{t+\frac{\Delta t}{2}}\right)\Delta t \qquad \text{Eq. 5.12}$$

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 $t+\Delta t$, as indicated in equation 5.13.

$$C^{n}_{t+\frac{\Delta t}{2}} = \frac{C_t + C^{n}_{t+\Delta t}}{2} \qquad \text{Eq. 5.13}$$

The iterations continue until two successive estimates of concentrations for each zone differ by less than a specified tolerance. For this program, the tolerance for convergence was set to 1.0E-8.

The circulation of outside air flows are calculated externally of the program from the VAV model and provided as input for each simulation time step. The building description necessary for the program calculations such as zone volume,

zone removal effectiveness, infiltration and inter-zone flows are read from an input data file at the beginning of the simulation. The variable pollutant source in the program is accomplished by reading 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.

The program output variables include the average and return duct concentration for each zone, and the concentrations within the air-handling unit, including the supply air and the return duct. Also available is the volume flow of outdoor, air divided by the source-level multiplier for each zone. This parameter is useful because if the unit source strength is equivalent to the pollution generated by one person, then the level multipliers are equivalent to the number of people present for each zone during each simulation time step. The outside air fraction at each time step, and the total outside air consumption, are also output of the program.

5.4 Additional Features of the Program

The program includes some other features that were not mentioned in the above description. Figure 5.1 shows the location of three filters in the ventilation system that can be used for other pollutants other than CO_2. These filters are modeled as ideal, nonsaturating filters that change the pollutant concentration in the air stream by a filter efficiency factor as indicated in equation 5.14. The filter capability was not used in the program because the pollutant modeled is CO_2.

$$C_{after} = C_{before}\varepsilon_{filter} \qquad \text{Eq. 5.14}$$

Although the inter-zone flows used in the program are input as constants, at times when there is no ventilation system airflow (that is the system is shut off), the inter-zone flows are set to zero.

The zone pollutant removal effectiveness, ε_c, is also a constant in the model. 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 condition by setting ε_c for a zone equal to 1.0 when there is no pollutant source in the zone. The value of removal effectiveness in the West zone is equal to 0.86 as measured in experiments (refer to chapter 3). The value of the removal effectiveness in the East zone is assumed to be 0.82.

The occupancy schedule in the model can be changed every 30 minutes, allowing flexible variations in the amount of CO_2 released in the zones.

The model calculates the CO_2 concentrations in three different locations, namely: the concentration in the zones, the mixed return air concentration at the air-handling unit (AHU), or the supply air concentration. Control of outside air control can be performed based on any one of these concentrations (i.e. based on where the sensor is located).

5.5 The Control Model

A control program has been integrated with the contaminant transport model. This control model adopts three control strategies as explained in the following sections.

5.5.1 Base-Case Control Strategy

The damper position in the base-case control strategy is always fixed at a predefined percentage. The base-case constant outside airflow modeling is based on the ASHRAE 62-1989 Standard, providing 20 cfm/person (10 L/s/person) for office buildings. For a multiple-zone system, equation 6.1 in the Standard is used. This equation determines the new outdoor air fraction that should be used as shown below.

$$Y = \frac{X}{(1 + X - Z)} \qquad \text{Eq. 5.15}$$

where,

$$Y = \frac{V_{ot}}{V_{st}} = \text{corrected fraction of outdoor air in system supply}$$

$$X = \frac{V_{on}}{V_{st}} = \text{uncorrected fraction of outdoor air in system supply}$$

$$Z = \frac{V_{oc}}{V_{sc}} = \text{fraction of outdoor air in critical space}$$

V_{ot} = corrected total outdoor airflow rate (cfm)

V_{st} = total supply flow rates, i, e., the sum of all supply for all branches of the system (cfm)

V_{on} = sum of outdoor airflow rates for all branches on system (cfm)

V_{oc} = outdoor airflow rate required in critical spaces (cfm)

V_{sc} = supply flowrate in critical space (cfm)

5.5.2 The On/Off Control Strategy

The On/Off control strategy uses the following operating modes:

- When the high concentration limit set-point is exceeded, the outside damper is fully open, providing 100% outside air.
- When the concentration drops below the low concentration limit set-point, the outside damper is fully closed or supplying minimum outside air.

The maximum and minimum limit set-points can be any values above and below the CO_2 concentration set-point. According to this control strategy, a maximum value of damper position can be set so that the damper is not 100% open, even when there is a need for a large amounts of outside air. That is, when the damper is fully open for outside air, the percentage of outside air can be less than 100%, allowing some of the return air to mix with the entering outside air.

This feature is implemented because the experiments performed in the laboratory cannot provide 100% outside air, even when the damper is fully open. The maximum value can be any value less than 100%.

The On/Off control strategy has another feature to prevent the outside damper from oscillating too frequently between on and off. That is, the damper cannot close or open unless a preset time has already passed, regardless of the outside air demand. The time is determined by the user and can be any value between 3 to 10 minutes. This value has an effect on the total outside air consumption. Clearly, as the preset time period increases, the outside air consumption increases.

5.5.3 The PID Control Strategy

The PID control strategy varies the outside air with the CO_2 concentration according to equation 4.2 presented in chapter four. The control program allows the user to input and change the gain constants (K_p, K_i, and K_d) to conform with the experimental values. However, the program accepts any set of values.

The control program can account for outside air leakage that can be input in the program as a percentage of total supply air. This value is assumed to be constant during the simulation period.

5.6 Model Validation

The developed program has been tested against the experimental results performed and discussed in chapter three. Two experiments were selected. The first experiment is the base-case control of both CO_2 and temperature set-points,

and the other experiment is the PID control of CO_2 and temperature. Both experiments are two-zone tests.

The results of the validation are shown in Figures 5.2 and Figure 5.3. As indicated in Figure 5.2 (base case validation), the curves of the experimental, and the model CO_2 levels, almost coincide for both zones. Figure 5.3 shows the validation of the model for the PID control strategy. The oscillations of the CO_2 in the critical (East) zone seem to follow that of the experiments. However, in the noncritical zone, the oscillations seem to be less significant because it does not have any effect on the final energy consumption.

In addition to finding the maximum error between the experimental and the model data, the chi-square test of goodness of fit was used. The chi-square is defined as follows (Holman, 1994):

$$\chi^2 = \sum_{i=1}^{n} \frac{[(X)_i - (Y)_i]^2}{(Y)_i} \qquad \text{Eq. 5.16}$$

where:

n = number of CO_2 concentration values in one test, n =540

X = model data

Y = experimental data

The R^2 is also calculated as shown in the Table 5.1. The R-squared value can be interpreted as the proportion of the variance in the model data attributable to the variance in the experimental data. R^2 is calculated according to the following equation:

$$r = \frac{n\left(\sum_{i=1}^{n} X_i Y_i - \left(\sum_{i=1}^{n} X_i\right)\left(\sum_{i=1}^{n} Y_i\right)\right)}{\sqrt{\left[n\sum_{i=1}^{n} X_i^2 - \left(\sum_{i=1}^{n} X_i\right)^2\right]\left[n\sum_{i=1}^{n} Y_i^2 - \left(\sum_{i=1}^{n} Y_i\right)^2\right]}} \qquad \text{Eq. 5.17}$$

The average concentration is also calculated for the experimental data and the model data and compared to each other for each zone. The outdoor air consumption is also compared and the percent error is calculated. The summary of the comparison is shown in Table 5.1 below. It is clear that the model predictions of the average CO_2 concentration agree well with the experimental data for both cases and both zones. However, the time variation of CO_2 predicted by the model is not as accurate for the West zone as it is for the East zone, especially for the PID control case. The lack of accuracy of the model predictions in the West zone may be attributed to the value of the ventilation effectiveness assumed for the West zone (as opposed to the measured value used for the ventilation effectiveness in the East zone). However, the model predictions are generally accurate, especially when the accuracy of the measurements are taken into account (refer to chapter three for detailed discussion on the accuracy of CO_2 sensors).

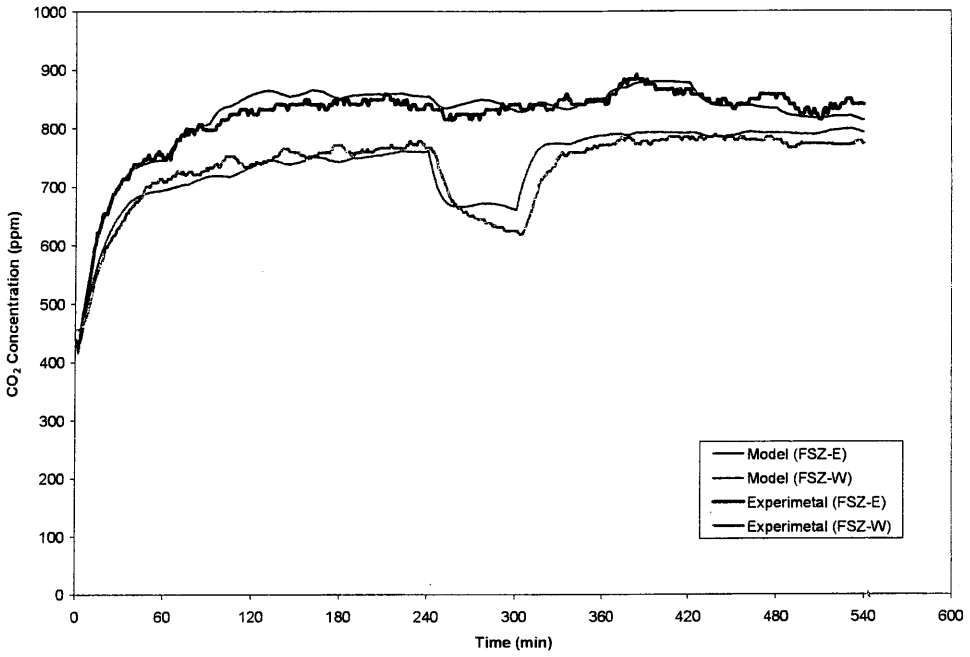


Figure 5.2: Model Validation: Two–zone Control of CO_2 and Temperature Setpoints (Base-Case Strategy)

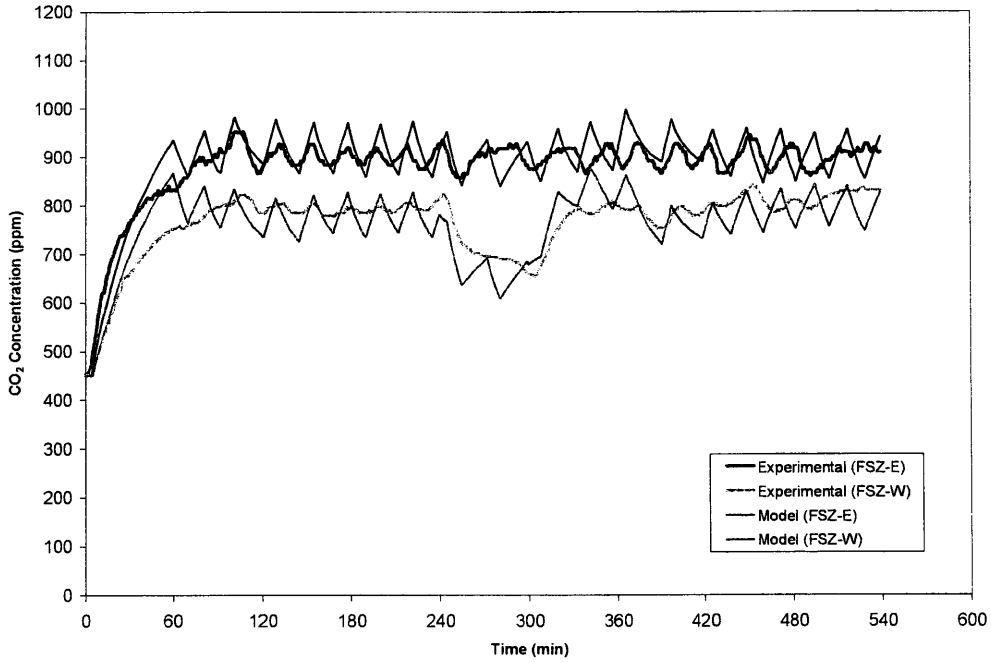


Figure 5.3: Model Validation: Two-zone Control of CO_2 and Temperature Setpoints (PID Control Strategy)

Table 5.1: Summary of Comparison Results

Analysis and Test Performed	Model			
	Base-case Model		Control Model (PID)	
	East Zone	West Zone	East Zone	West Zone
Maximum % Error in Concentration	4.47	13.91	9.13	15.15
R^2 (-1 < C < 1)	0.95	0.82	0.83	0.72
Chi-squared Test Fit at α=0.005	fit	fit	fit	fit
Average Zone Concentration in Model (ppm)	823.2	738.7	888.7	759.4
Average Zone Concentration in Experiment (ppm)	820±27	733±42	880±27	765±42
Percent Error in Concentration	-0.42	-0.81	-1.3	0.75
Outdoor Air Consumption in Model (ft^3)	116917	71418	91513	55644
Outdoor Air Consumption in Experiment (ft^3)	116917	71418	91691	56819
Percent Error in OA	0.0	0.0	0.19	2.1

For the control model, the following modifications are made:

1. Average outside CO_2 concentration is 380 ppm as measured in experiments.
2. Outside air leakage is 10% (refer to experimental graph to reveal this value).
3. The delay time is 6 minutes, the time of the CO_2 sensor to sense the actual CO_2 concentration in the zone.
4. The maximum outside air percentage entering through the outside damper is 87%. That is, even when the damper is fully open for outside air, only

87% is outside air and the remaining 13% is recirculated (refer to experimental graphs to reveal this value).

5. The same gain constants used in the PID equation for the experiments are used in the model as follows: K_p =3.0, K_d = 0.0, and K_i =0.5. These constants may not be the best (ideal) constants (refer to the simulation results in chapter six).
6. Some other input values in the model are as follows:
   - CO_2 generation rate per person = $1.77*10^{-4}$ ft^3/sec ($5.0*10^{-6}$ m^3/sec)
   - Zone Volume (FSZ-E)= 4,000 ft^3 (20 ft × 20 ft × 10 ft) (113.25 m^3)
   - Zone Volume (FSZ-W)= 3,040 ft^3 (20 ft × 19 ft × 8 ft) (86.1 m^3)
   - Zone pollutant removal effectiveness (FSZ-E) = 0.82
   - Zone pollutant removal effectiveness (FSZ-W) = 0.86 (as measured)
   - Zone HVAC return air path volume (FSZ-E) = 464 ft^3
   - Zone HVAC return air path volume (FSZ-W) = 384 ft^3
   - Zone infiltration = 0 (positive pressure in both zones)
   - Inter-zone flow from FSZ-E to FSZ-W (only) = 50 cfm (0.024 m^3/sec)

5.7 Conclusions

This chapter discussed the development of a contaminant transport model that calculates the contaminant concentration in the zone. A contaminant control program is also developed. In this control program, three control strategies were modeled, namely: the base case supplying a fixed rate of outside air at all times,

the On/Off control strategy, and the PID control strategy. Both programs were validated using the experimental work performed in chapter four. Appropriate features and modifications of the two models were discussed.

CHAPTER SIX

Integrated IAQ/HVAC Model

This chapter presents the integration module of the indoor air quality models (including the contaminant transport model and the contaminant control model) developed in the last chapter with a variable air volume (VAV) model and an equipment HVAC model to ultimately calculate the energy consumption for each CO_2-based demand-controlled ventilation strategy. The VAV model is based on the secondary HVAC toolkit (Brandemuehl et al, 1993).

6.1 Air-handling Unit (AHU) System Model

A variable air volume (VAV) system is a single path system that controls the indoor temperature within a zone by varying the amount of air supply. The damper of the VAV terminal unit adjusts the airflow rate supplied to the zone depending on the thermal load. The discharge air temperature is kept constant and equal to the supply air temperature from the air-handling unit plus any temperature increase due to heat gains (for instance, from the motor of the supply fan).

As indicated in Figure 6.1, the VAV system consists of a central air-handling unit and a VAV terminal unit with a reheat coil located at each zone. The AHU unit itself includes a supply fan, a cooling coil, a preheat coil, and an outside air economizer. The AHU supplies air at a controlled fixed temperature while in the

VAV unit terminal, the damper position is adjusted to vary the airflow to meet the cooling load for each zone. In particular, when the zone cooling load decreases, the damper closes until it reaches its minimum position where it supplies the minimum airflow. The reheat coil will be activated to meet the load if the cooling load continues to decrease. A variable speed drive can be used to module the supply fan to vary the air volume flow rate according to the thermal load of the zones .

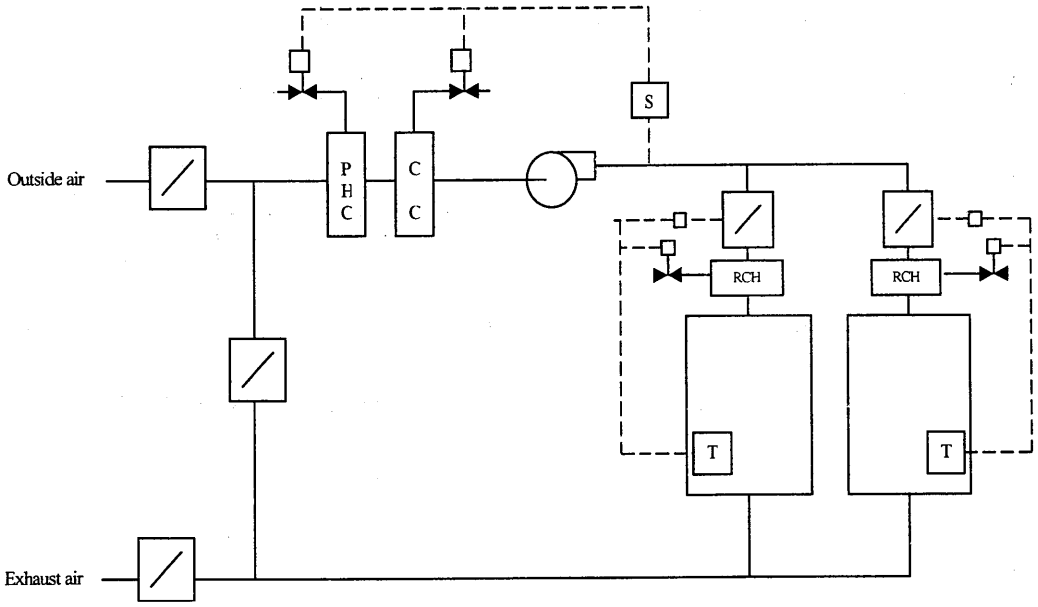


Figure 6.1: Schematic Diagram of the Variable Air Volume (VAV) System

The outside air damper in the AHU can have a fixed-position or a variable-position depending on the control strategy being implemented. For the base case used in the experimental work, the damper position is fixed to satisfy ASHRAE ventilation requirements (ASHRAE Standard 62-1989) . For any other control strategy such as the On/Off or the PID, the position of the damper is varied to maintain the CO_2 setpoint. In addition, the outdoor air damper position is

controlled depending on the ambient air temperature when the economize cycle is used.

The input variables for the AHU simulation model includes the supply air temperature set-point, entering water temperature for the cooling coil, preheat coil and the reheat, the outside airflow rate values obtained from the CO_2 control model for the control strategy, and the economizer temperature set-point. Moreover, the input variables specific to each zone are the sensible and latent loads, the minimum zone supply airflow rate, and the zone heating supply flow rate.

The VAV model calls a number of modules selected from the secondary HVAC toolkit as shown in Figure 6.2. In particular, the VAV model calls three subroutines:

I. COILINV, which is the general routine for coils that calls either the cooling coil (CCDET) or the preheat/ reheat coils (HCDET) whenever any one is needed.

II. FANSIM, which is the routine that models the supply fan and calculates the energy consumption.

III. ECON, which is the routine that calculates the mixed air temperature and enthalpy and determines whether an economizer is used or not.

A brief description of each subroutine is provided in the sections that follow. The mathematical description of each subroutine is presented in the secondary HVAC toolkit (Brandemuehl, 1993).

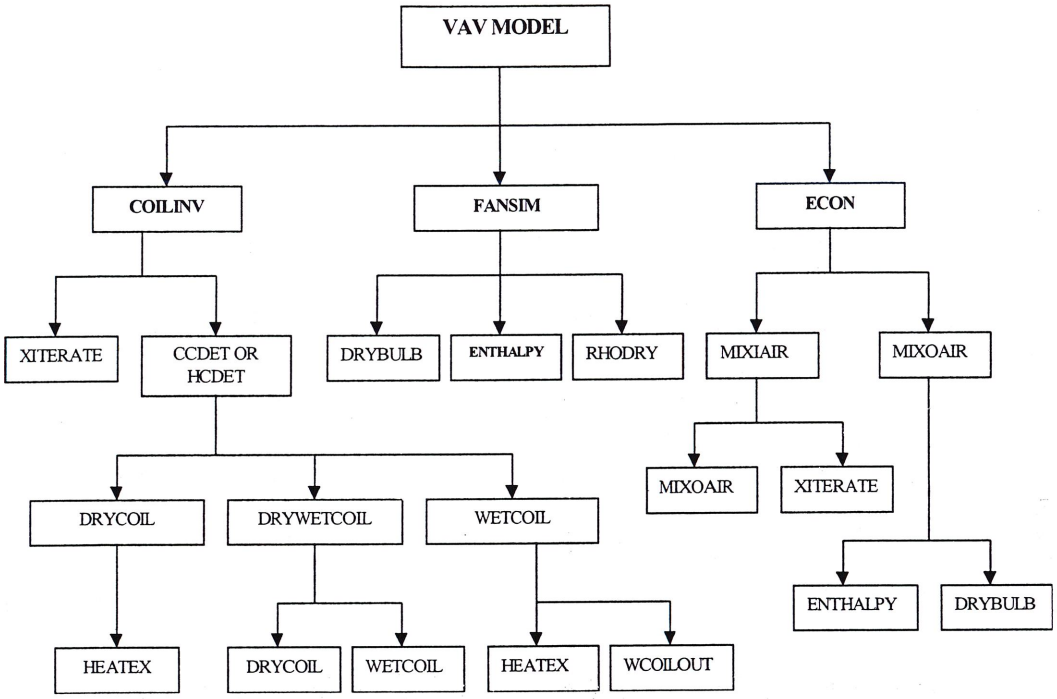


Figure 6.2: Modules used for the VAV simulation model

6.1.1 Cooling and Heating Coils Models (CCDET and HCDET)

The model for the cooling and heating coils calculates the performance of a finned tube, multi-row heat exchanger with a counter-flow cross-flow configuration. Three conditions are possible:

(a) All wet: when the external fin surface is completely wet,

(b) Partially wet and partially dry: when the external fin surface is partly wet and partly dry, and

(c) All dry: when the external fin surface is completely dry.

Calculation of the performance of the cooling and heating coils require two different levels of modeling and approximations. First, the local heat transfer between the air and liquid must be modeled using basic principles of conduction, convection, and radiation heat transfer. Second, the local heat and mass transfer

throughout the coils is modeled to determine the leaving air and liquid conditions. A general model is used to integrate local heat and mass transfer effects using the effectiveness-NTU model. The effectiveness –NTU is defined as the ratio of the actual heat transfer rate to the maximum heat transfer rate for the given entering fluid conditions and flow rates.

The model used in this work assumes a cross-counter-flow configuration with the heat transfer governed by the counter-flow heat exchanger relationships described in the HEATEX subroutine. Indeed, The HEATEX subroutine calculates the leaving conditions of a simple heat exchanger using the effectiveness-NTU method of analysis using counter-flow configurations.

6.1.2 Economizer Control (ECON)

The economizer cycle control is modeled using the subroutine ECON. The economizer cycle uses cooler outside air that is mixed with recirculated return air to reduce the need for mechanical cooling. The economizer algorithm assumes that the mixed airflow rate is known based on an energy analysis that determines the supply airflow rate from zone thermal loads. The amount of the outside air to be mixed with return air is determined using the following control objectives:

1. Maintain a specified mixed air temperature setpoint.
2. Maintain minimum ventilation even at very low outdoor temperature.
3. Reduce outside air to the minimum when it is not beneficial for cooling.

There are four control modes for determining outdoor airflow rate. The first three modes compare the values of outdoor, return and mixed setpoint temperatures, while the fourth mode is based on an estimation of the availability of, and the need for, free cooling. In general, the control objective is to maintain the desired mixed air temperature setpoint, subject to a set of constraints, as follows:

- Regime 1: At very low outdoor temperatures, the mixed air temperature setpoint can only be achieved with very low outdoor airflow rates. If the airflow rate calculated to satisfy the mixed air temperature setpoint is less than the minimum required for ventilation from the control model, the outdoor airflow rate is set to that minimum. The resultant mixed air temperature is less than the setpoint, and thus heating is required by the preheat coil.
- Regime 2: When the mixed air temperature setpoint is between the outdoor air temperature and the return air temperature, the two streams can be mixed to achieve the desired mixed air temperature while satisfying the requirements for minimum ventilation air determined by the control model.
- Regime 3: The mixed air temperature setpoint can not be maintained when both the return and the outdoor air temperature are above the mixed air temperature setpoint. However, if the outdoor temperature is below the return air temperature, it is often desirable to operate with 100% outdoor air to reduce cooling requirements.

When the outdoor air conditions become warmer, it is better to use return air than outdoor air. Under these conditions, the outdoor airflow is reduced to the minimum set by the control model. To determine when it is more advantageous to use return air than outdoor air temperatures or enthalpies of various airstreams can be used. The simplest is to compare the outdoor temperature with a given setpoint. The ventilation airflow rate is reduced to a minimum set by the control model if the temperature is above the setpoint. A better approach recognizes that the energy consumption by the cooling coil is a function of the air enthalpy, rather than temperature. The economizer controller reduces the ventilation rate to the minimum value if the outdoor enthalpy is greater than the return air enthalpy. In this work, the first technique is used.

The mixed air temperature in the ECON subroutine is calculated by two other subroutines called the MIXOAIR and MIXIAIR. These subroutines calculate outlet and inlet states for a mixing process based on conservation equations for moist air. In particular, subroutine MIXOAIR calculates the leaving temperature, humidity ratio and mass flow rate of two mixed airstreams. Meanwhile, subroutine MIXIAIR calculates the mass flow rate of two entering air streams of a mixing box with a known leaving mass flow rate and temperature of all the streams. The outlet temperature and humidity ratios are calculated from basic energy and mass conservation principles.

6.1.3 Supply Fan Model (FANSIM)

The algorithm for the supply fan calculates the fan electric power and leaving air temperature and humidity for a given airflow rate and entering air conditions. The algorithm is suitable for energy calculations when the airflow rate is specified and when the pressure rise across the fan is maintained constant. The assumption of constant fan pressure rise is important to ensure fixed supply air duct static pressure as set in the experimental work (0.8 in WG). This control strategy is common in several existing HVAC systems.

For the purpose of the energy calculations, the fan is characterized by the electric power and volumetric rate at full-load design conditions. Fan electric power consumption at part-load is determined by an empirical relationship as a function of the ratio of operating airflow rate to design airflow rate.

6.1.4 Psychometrics

The psychrometrics subroutines called by the VAV model consists of twelve functions that calculate the individual basic psychrometric moist air properties. These properties include humidity ratio, relative humidity, enthalpy, wet bulb temperature, dew-point temperature, dry and moist air densities, saturated water vapor pressure, saturated water vapor temperature, saturated moist air enthalpy, and saturated moist air temperature.

6.2 HAVC Equipment Models

The HAVC equipment consists of a chiller, a boiler, and a supply fan. The model for each piece of equipment is briefly described below:

6.2.1 Chiller Model

For this work, a hermetic reciprocating chiller is used to provide chilled water to the cooling coil. The electrical power input (P_{el}) for the hermetic reciprocating chiller can be described by the following equation (Krarti, 1996):

$$P_{el} = EIR_{design} * CCAP_{nom}(a + b * PLR(t) + c * PLR^2(t)) \qquad \text{Eq. 6.1}$$

where:

EIR_{design} = chiller efficiency, kW/ton
$CCAP_{nom}$ = nominal chiller capacity, tons
PLR(t) = Part - load ratio at time t
a, b, and c = the chiller performance coefficients (Krarti, 1996)
(a = 0.088065; b = 1.137742; c = -0.225806)

The part load ratio can be estimated as follows:

$$PLR(t) = \frac{Q_{ch}(t)}{CCAP(t)} \qquad \text{Eq. 6.2}$$

and

$$CCAP(t) = CCAP_{nom}(1 + \delta[T_{ref} - T_{DB}(t)]) \quad \text{Eq. 6.3}$$

where:

$\delta = 0.005$

$T_{ref} = 85\ ^{o}F\ (29.4^{o}C)$

$T_{DB}(t)$ = Dry bulb temperature from weather file at time t

6.2.2 Boiler Model

The boiler heat rate input is given by (Kreider and Rabl, 1994):

$$\dot{Q}_{boiler} = \frac{BN}{\eta}\left[a + b*(PLR(t)) + c*(PLR(t))^2\right] \quad \text{Eq. 6.4}$$

where:

BN = Boiler nominal size, kW

η = boiler full-load efficiency

a, b, c = boiler performance coefficients (Kreider, 1994)

(a = 0.0, b =1.368, and c = -0.368)

PLR(t) = boiler part-load ratio and is defined as follows:

$$PLR(t) = \frac{\dot{Q}_{h,tot}}{BN} \quad \text{Eq. 6.5}$$

where:

$Q_{h,tot}$ = total boiler heat rate from preheat and reheat heating coils, kW

6.2.3 Supply Fan Model

The fan electric power requirement is given by (Brandemuehl et al, 1993):

$$\dot{W}_{fan} = FMP * \left(a + b * (PLR(t)) + c * (PLR(t))^2 + d(PLR(t))^3\right) \qquad \text{Eq. 6.6}$$

where:

FMP = fan motor power, kW

a, b, c, and d = fan performance coefficients (Brandemuehl, 1993)

(a = 0.00153, b= 0.005208, c = 1.1086, d= -0.11635563)

PLR(t) = fan part-load ratio

where:

$$\text{PLR(t)} = \frac{\text{total airflow (t), m}^3/\text{s}}{\text{fan capacity, m}^3/\text{s}} \qquad \text{Eq. 6.7}$$

6.3 Integration of IAQ/HVAC Models

The simulation module that calculates the energy consumption of the CO_2-based demand-controlled ventilation strategies is shown graphically in Figure 6.3. A brief description of the calculation procedure follows:

1. DOE-2 program is used to determine hourly sensible and latent loads. Inputs in the DOE-2 program include building characteristics and location, occupancy schedule, zones setpoint temperatures, and weather

files. From the hourly results, the sensible and latent loads are determined to every minute (or any time step smaller than one hour) by linear interpolation.

2. The VAV model reads the sensible and latent loads calculated from the DOE-2 program and determines zone total supply air (Q_{sup}). At the same time, the CO_2 control program determines the fraction of outside air (Q_{OA}) based on the concentration in the zone and the CO_2-based demand-controlled strategy used. The output of the VAV model includes the pre-heat, reheat, and cooling coils heat transfer rates, and the supply fan electric power consumption.
3. The CO_2 transport model used in conjunction with the control program provides the CO_2 concentration when zone outside air (Q_{OA}) changes with time. The variation in the CO_2 concentration is then fed back to the control program depending on the CO_2-based demand-control ventilation strategy.
4. Using the HVAC equipment models, the chiller and boiler energy power consumption are calculated based on the heating and cooling transfer rates data received from the VAV model. Input of the HVAC equipment model include weather data, and both chiller and boiler performance data.

6.4 Calculation of Energy Consumption

In order to calculate the energy consumption of the HVAC equipment for the cooling loads of two full-size zones used in the experiments, the integrated IAQ/HVAC model is used. The integrated program was run for four different

cities. The same sensible and latent loads were used for each city in the VAV model assuming both zones are internal and separated from the outside environment. Since most of the experiments were performed in summer, June 1 was selected as a typical summer day for the daily simulation. The results of the energy consumption along with the energy savings (or penalties) are shown in Table 6.1.

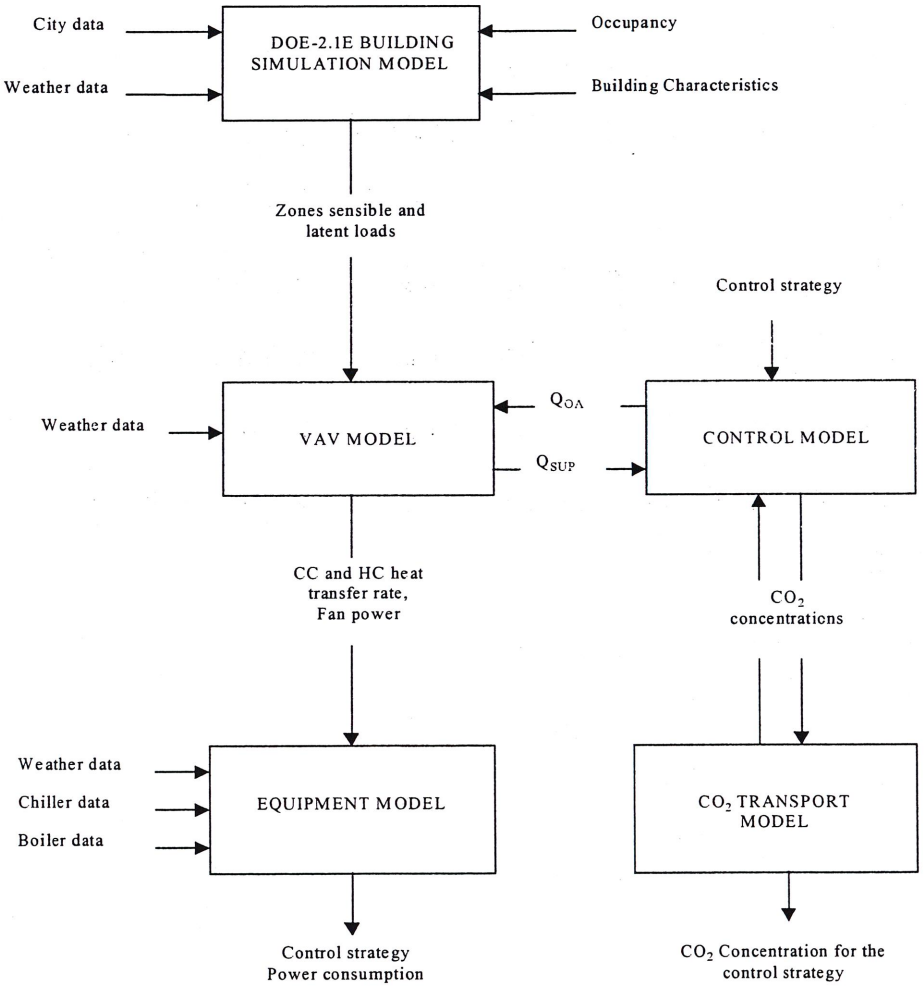


Figure 6.3: Integration of IAQ/HVAC Models

Table 6.1: Equipment Energy Consumption for June 1

City	Equipment	Control Strategy and OA Fraction			Percent Reduction in On/Off*	Percent Reduction in PID*
		Base Case	On/Off	PID		
		45%	**38.6%**	**35.4%**	14.0	21.0
Miami	Chiller (kWh)	999.3	769.1	755.4	23.0	24.4
	Fan (kWh)	4453	4451	4450	0.05	0.07
Phoenix	Chiller (kWh)	629.2	532.3	521.6	15.4	17.1
	Fan (kWh)	4440	4469	4451	-0.6	-0.25
Boulder	Chiller (kWh)	444.3	464.7	472.4	-4.6	-6.3
	Fan (kWh)	4455	4467	4496	-0.27	-0.92
Madison	Chiller (kWh)	552.0	489.6	480.8	11.3	12.9
	Fan (kWh)	4443	4443	4443	0.0	0.0

* (-) sign indicates energy penalty

Table 6.1 shows that the energy consumption of the supply fan is almost constant regardless of the location of the city or the control strategy. This is because the amount of air being pushed from the air-handling unit (AHU) is the same for all cases. The slight changes arise from the temperature or the pressure across the fan. However, the cooling coil total heat transfer rate, and thus, the energy consumption of the chiller, is different. The biggest savings are for Miami, FL with 23.0% and 24.4% for the On/Off and the PID controller, respectively. Madison, MA and Phoenix, AZ also have energy savings. For all three cities mentioned above, the savings of the PID control strategy is bigger than that of the On/Off control strategy. The savings, however, is insignificant, though. On the other hand, there are energy penalties for Boulder, CO. A possible justification for this penalty is that the outside temperature is better and has less demand for conditioning than conditioning the return temperature from the air-handling unit.

The energy savings or penalties for the cities obtained above were compared with those obtained by Brandemuehl and Braun (1999). The results agree in terms of energy savings for Miami and Phoenix, and penalties for Boulder. However, for Madison, the results obtained above showed savings instead of penalties obtained by Brandemuehl and Braun. This can be accounted for the high outside temperature during June 1 selected for the daily run, compared to the annual overall temperature for Madison.

6.5 Sensitivity Analysis

6.5.1 Effect of Outdoor Air Leakage on Energy Consumption

The outdoor air leakage from the outside damper may have significant effect on the energy consumption. To prove this, the value of outside air leakage in the contaminant program was allowed to vary from 0% leakage to 100% leakage, and for each control strategy, the outdoor air consumption was calculated. The results are shown in Figure 6.4.

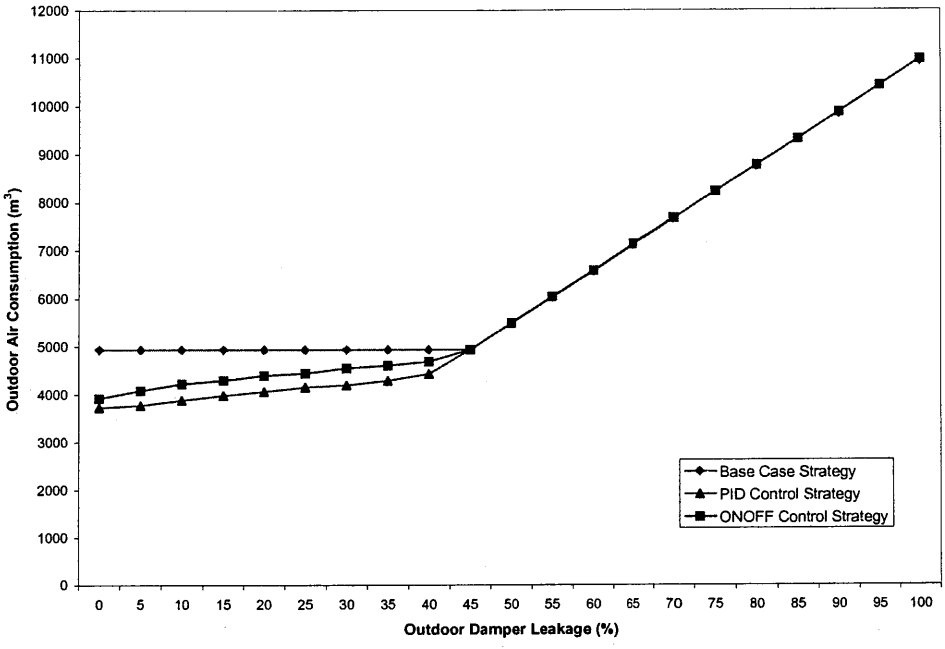


Figure 6.4: Effect of Outdoor Air Leakage on Outdoor Air Consumption

As shown in Figure 6.4, the outdoor air leakage has no effect on outdoor air consumption for the base case control strategy when the value is below the minimum required by ASHRAE (in this situation 45% - refer to experiments in chapter 4). The outdoor air consumption is thus constant. However, for the On/Off and PID control strategies, the consumption increases gradually as the outdoor air leakage increases, until it reaches the minimum required by ASHRAE. After that, the consumption of all three strategies increases by the same rate. It can also be noted that the outdoor air consumption of the PID control strategy is always less than the On/Off control strategy for the same leakage.

6.5.2 Effect of Temperature Economizer on Energy Consumption

To see the effect of temperature economizer on energy consumption, an office building has been modeled using DOE2. The building modeled is a one-story office building that consists of two zones: one external for offices and one internal (meeting room) centered in the middle of the external zone. The area of the building is 20,000 ft^2 (1857 m^2): the perimeter office zone is 17,900 ft^2 (1663 m^2) whereas the area of the internal meeting room is 2100 ft^2 (195 m^2) that is surrounded by the external zone. The height of the building is 10 feet. The total volume of the building is 200,000 ft^3 (5,663 m^3). Windows surround the building from the four directions with a height of 4 ft. The maximum occupancy in the perimeter office space is 60 people (300 ft^2/person) and in the meeting room is 12 people (175 ft^2/person). Description of the building can be seen in the DOE2 input program in Appendix C.

One daily simulation has been run for Boulder, CO., using the three modeled control strategies: one time without temperature economizer, and another time with the temperature economizer. The results are shown in Table 6.2. Without a temperature economizer, Table 6.2 shows energy penalties if either the on/off or the PID control strategies are used instead of the base case. If the temperature economizer is operated, however, savings can be accomplished.

Table 6.2: Effect of Temperature Economizer for Boulder, CO on June 1

Control Strategy	Economizer status	Energy Consumption	
		Cooling Coil (kW)	Chiller (kWh)
Base Case	Without	537.0	670.0
	With	325.7	449.1
	Percent Reduction	39.3	33.0
On/Off	Without	587.3	713.8
	With	325.7	325.7
	Percent Reduction (w/o)	-9.4	-6.5
	Percent Reduction (with)	0.0	0.0
PID	Without	595.7	721.2
	With	325.7	325.7
	Percent Reduction (w/o)	-10.9	-7.6
	Percent Reduction (with)	0.0	0.0

6.5.3 Effect of CO_2 Setpoint on Energy Consumption

The CO_2 setpoint was varied between 800 ppm and 1200 ppm and the energy consumption (no economizer is used) was calculated for two cities, namely: Boulder, CO., and Miami, FL. The results for Boulder, CO., are shown in Table 6.3. Since no economizer is used, there is an energy penalty in reducing the outside air. And since the PID controller consumes less outside air, the penalty is higher compared to the On/Off and the base case. As a result, as the CO_2 setpoint increases, the outdoor air consumption decreases, and the energy consumption increases.

Table 6.3: Effect of CO_2 Setpoint on Energy Consumption for Boulder (June 1)

CO_2 Setpoint	Control Strategy Energy Consumption (kWh)			% Reduction	
	Base Case	On/Off	PID	On/Off	PID
800	444.3	445.4	448.9	-0.2	-1.0
900	444.3	455.8	462.1	-2.6	-4.0
1000	444.3	464.7	472.4	-4.6	-6.3
1100	444.3	472.0	478.2	-6.2	-7.6
1200	444.3	476.5	483.0	-7.3	-8.7

The results for Miami, FL. are shown in Table 6.4. On contrary to Boulder, the energy consumption decreases as the CO_2 setpoint increases and the outdoor air consumption decreases.

Table 6.4: Effect of CO_2 Setpoint on Energy Consumption for Miami (June 1)

CO_2 Setpoint	Control Strategy Energy Consumption (kWh)			% Reduction	
	Base Case	On/Off	PID	On/Off	PID
800	999.3	859.2	856.6	14.0	14.3
900	999.3	808.5	801.8	19.1	19.8
1000	999.3	769.1	755.4	23.0	24.4
1100	999.3	747.6	727.2	25.2	27.2
1200	999.3	727.3	703.2	27.2	29.6

6.5.4 Effect of Proportional Gain Constant (K_P) on Energy Consumption

To investigate the effect of bad tuning of the proportional gain constant (K_P) on energy consumption (refer to Figures 6.5 and 6.6), the value of K_P was varied as shown in Table 6.5. The value of K_P was changed between –100% and +100%, relative to the ideal K_P value ($K_{p,\ ideal}$=10 cfm/ppm). The ideal K_P is selected based on the value of K_P that gives the overall CO_2 concentration equal to the set-point without any oscillations. The energy consumption was then

calculated based on the simulation model. Table 6.5 shows the corresponding change in energy consumption for each percentage change in K_P. Figures 6.7 and 6.8 show the results graphically. As the Figures 6.7 and 6.8 show, increasing the K_P value by 100% (from K_P= 10 to K_P= 20) produces an increase in energy consumption by 2.3%. On the other hand, decreasing the K_P value by 100% (from K_P=10 to K_P=0.01) produces a decrease in energy consumption by 16.9%. The percentage difference in energy consumption between the $K_{P,ideal}$ and $K_{P,experimental}$ (the K_P value used to validate the model) is only 2.5%. It can also be noted that an error by 50% in selecting the ideal (correct) K_P might result in a 2.5% difference in energy consumption. Figure 6.5 shows the effect of K_p on the variation of CO_2 level within the space.

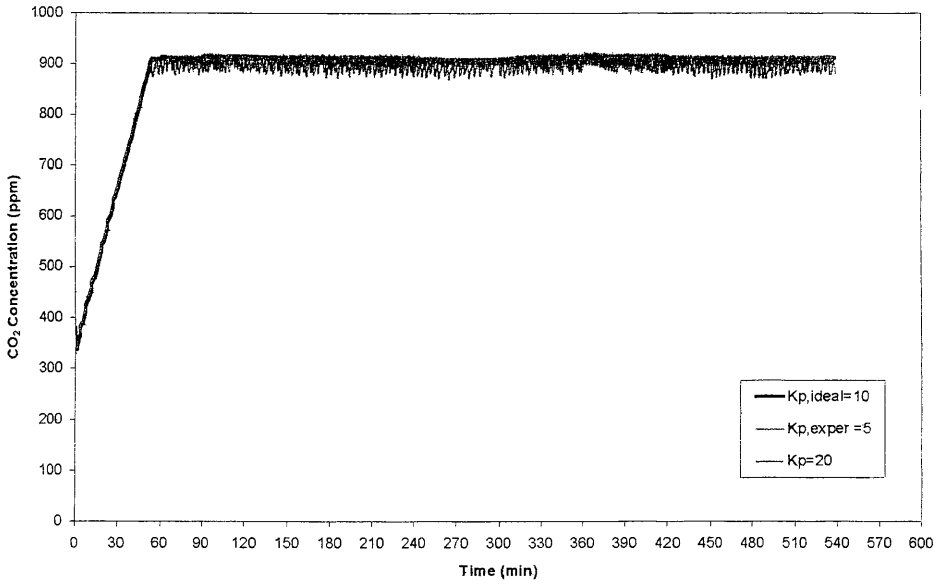


Figure 6.5: Effect of Kp on the Variation of CO_2 Level

Figure 6.6 shows a close-up view of the CO_2 oscillations for a portion of Figure 6.5 between 150 minutes to 240 minutes. Figure 6.5 shows clearly that the oscillations in the CO_2 level found in the experiments were not caused by the Kp value in the PID equation, because the fluctuations in Figure 6.6 have higher frequency and lower amplitude than those obtained in the experiments.

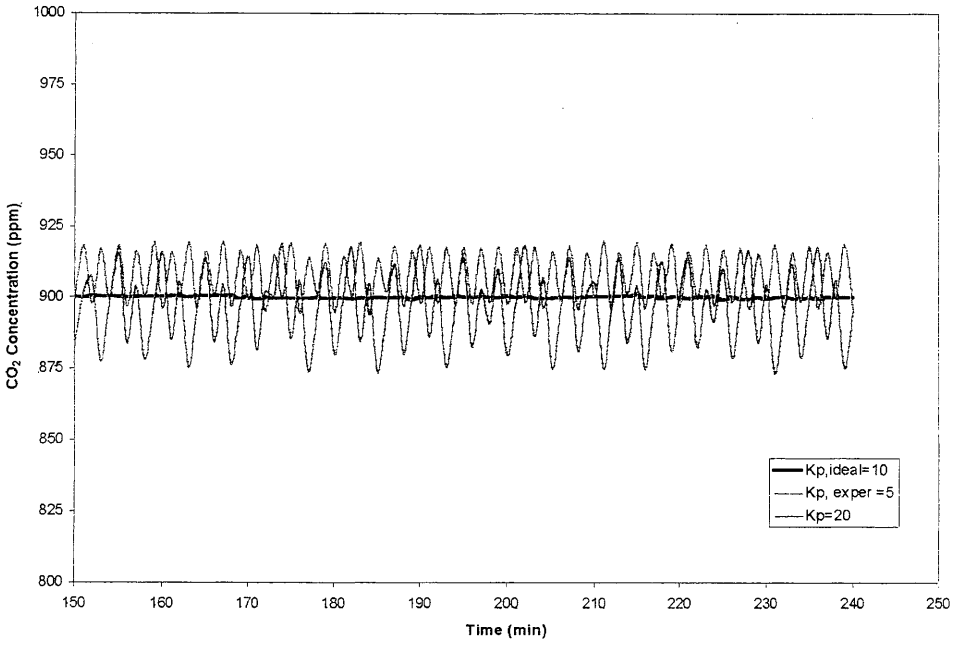


Figure 6.6: Effect of Kp on the Variation of CO_2 level between 150 min and 240 min

Table 6.5: Effect of Proportional Gain Constant (K_P) on Energy Consumption

K_P Value (cfm/ppm)	Percent Change in K_P	Energy Consumption (kWh)	Percent Change in Energy Consumption
0.01	-99.9	141.5	16.9
0.05	-99.5	143.7	15.6
0.5	-95.0	151.7	10.9
1	-90.0	155.6	8.6
5 (model validation)	-50.0	166.2	2.5
8	-20.0	169.0	0.8
9	-10.0	169.7	0.4
10 (ideal)	0.0	170.3	0.0
11	10.0	170.8	-0.3
12	20.0	171.2	-0.5
15	50.0	172.3	-1.2
20	100.0	174.2	-2.3

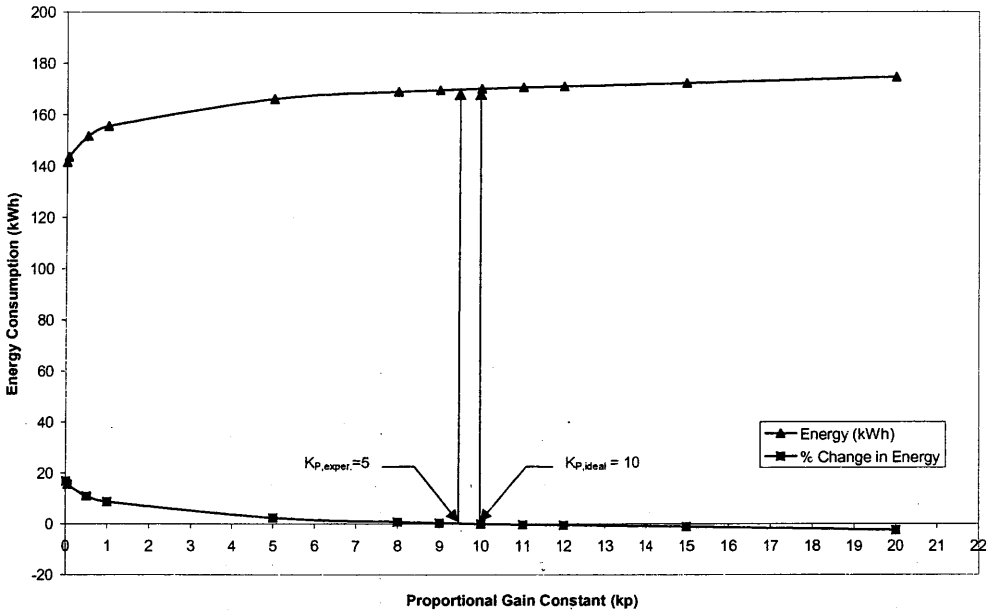


Figure 6.7: Effect of K_P Variation on Energy Consumption

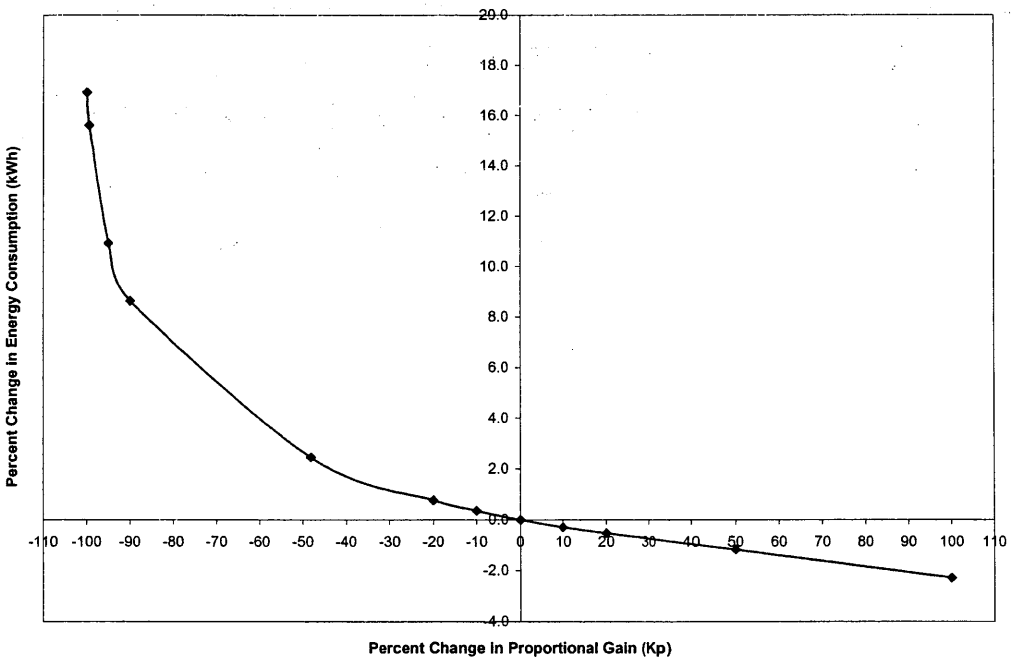


Figure 6.8: Effect of K_P Variation on Energy Consumption

6.5.5 Effect of Sensors Time Delay on Energy Consumption

Table 6.6 shows the effect of sensor's time delay on energy consumption for Miami, FL. The time delay was varied from 0 minute to 20 minutes and the energy consumption was determined.

Table 6.6: Effect of Sensor's Time Constant on Energy Consumption

Time Delay (min)	Energy Consumption (kWh)	Percent Difference (%)
0	999.0	0.0
5	999.2	0.0
10	1008.7	0.9
15	1020.1	2.1
20	1031.3	3.2

The results in Table 6.6 show that energy consumption increases with the time delay of the CO_2 sensor. Figure 6.9 shows the effect of time delay on the variation of CO_2 level within the space. Figure 6.9 reveals that the amplitude of the oscillations in the CO_2 level increases with the time delay of the CO_2 sensor. Moreover, the frequency of the oscillations decreases with increasing time delay of the CO_2 sensor. These results indicate that the probable cause of the oscillations in CO_2 concentration obtained in the experimental tests is the relatively high time delay (or response) of the CO_2 sensors.

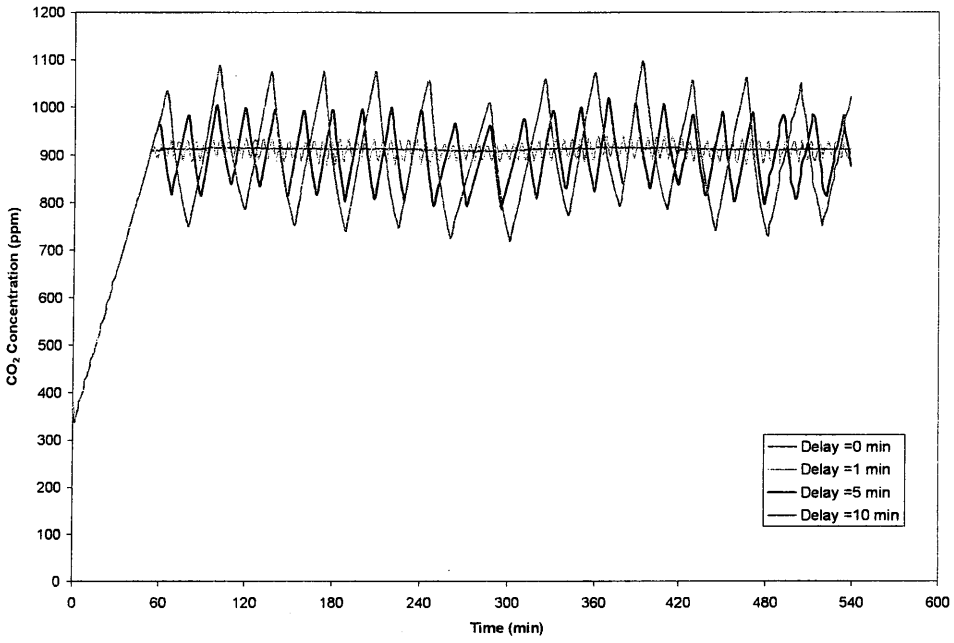


Figure 6.9: Effect of Time Delay on CO_2 Set-point

6.6 Conclusions

This chapter discussed a variable air volume and HVAC models that were integrated with the IAQ contaminant transport and control models. The results is an integrated IAQ/HVAC model that calculates the energy consumption, given the control strategy that determines the method and the quantity outdoor air is introduced to the buildings. The IAQ/HVAC model was run for a number of cities to calculate the energy consumption for each control strategy. Generally, as the outdoor air consumption decreases, the energy consumption decreases. There are cases where the energy consumption increases when the outdoor air introduced to the zone decreases. That is because the outdoor air conditions require less conditioning energy than conditioning the return air from the HVAC

system. The situation with these cases can be corrected by using a temperature economizer. The outdoor air leakage was found to play a major rule in energy consumption. The outdoor air leakage must be kept to minimum in order to avoid wasting energy in conditioning this extra outdoor air. Generally, the energy consumption decreases as the CO_2 setpoint increases and vice versa. The proportional gain constant (K_P) in the PID control equation was found to have little effect on energy consumption if not properly tuned. An error of 50% between the ideal K_P and the chosen K_P can produce a difference of 2.5% in energy consumption. The time delay of the CO_2 sensors was found to be the major cause of the oscillations in the CO_2 level.

CHAPTER SEVEN

Conclusions and Recommendations

7.1 Conclusions

7.1.1 General Conclusions

This research project was intended to measure experimentally the effectiveness of various CO_2-based demand-controlled ventilation strategies in saving energy in HVAC equipment, while at the same time control the indoor contaminants to the allowed limits. The experimental study performed in this research work provided a test-bed to validate the computer simulation model that analyzes the effect of CO_2 controlled ventilation.

CO_2-based demand-controlled ventilation optimizes and resolves the traditional conflict between reducing ventilation to save energy while still maintaining healthy indoor air quality levels. There are three major reasons to consider CO_2-based demand-controlled ventilation:

1. It saves energy by reducing costly over-ventilation when spaces are partially occupied.
2. It always ensures that the target ventilation rate per person (20 cfm/person, in office buildings) is maintained in the space at all times to ensure acceptable indoor air quality.
3. CO_2-based demand controlled ventilation is not particular as to where and how outside air is coming from. If adequate outside air is being

introduced by opening a window, it will reduce the amount of outside air that will be delivered by the mechanical ventilation system.

In CO_2-based demand-controlled ventilation, CO_2 sensors are used. They control ventilation in a space much like a thermostat would control heating or cooling. Just as an economizer saves energy when outside air can be used for cooling, a CO_2 control saves energy when space occupancies are below full capacity.

Before conducting any indoor air quality experiments, a large number of preparations and laboratory setup have been performed that are considered highly time-consuming. Performing such experiments in a laboratory that was not really well suited to conduct IAQ experiments is a rather complicated and frustrating job.

In order for the laboratory to carry out IAQ experiments, several features have been added or modified. In summary, these features include:

1. Installing a total of five CO_2 sensors in the zones and in the supply, return, and outside ducts.
2. Removing the old airflow stations and installing three new stations to accurately measure the airflow in the supply, return, and outside ducts.
3. Installing accurate and reliable CO_2 flow meters to inject the gas in the zones, and simulate CO_2 gas production by people.
4. Installing pressure transducers in each zone and the lab outside environment.

5. Perform the wiring to connect the CO_2 sensors, air flow stations, and the pressure transducers to the laboratory control system.
6. Connecting new and replacing old temperature sensors in the zones.
7. Securing the zones to prevent any unwanted leakage of CO_2, including ceiling, floors, walls below the floors, and covering holes in the walls.
8. Weather-stripping the blades of the outside damper to prevent any outdoor air leakage.

With the modifications, and with the installation of the new laboratory control system, it was possible to conduct the IAQ experiments in the Larson laboratory.

The indoor air quality experiments have been performed in three phases. Phase one verified laboratory setup and included the following preliminary experiments:

1. Measuring the outdoor CO_2 concentration level for a typical day. An average value of 380 ppm has been found.
2. Measuring the CO_2 concentration in the zone with no internal source for CO_2 generation. An average value of 440 ppm has been found.
3. Measuring and monitoring the CO_2 concentration level in the zones to verify that both CO_2 injection flow and sensor readings are accurate.
4. Measuring and monitoring CO_2 concentration level at different ventilation rates to verify ASHRAE requirements. A value of 20 cfm/person proved to be needed to meet ASHRAE requirements.

5. Measuring and monitoring CO_2 concentration level at different zone differential pressures. It was verified that the CO_2 concentration level does not change regardless of the differential pressure value. The zones were set at 0.05 in WG.

Phase two of the experiments involved controlling the CO_2 level only to 1000-ppm setpoint to determine the optimum control strategy that requires the lowest amount of outside air. Three control strategies were investigated, namely: On/Off control strategy, proportional control strategy, and finally the PID controller. By far, the PID controller consumed the lowest amount of outside air with 32.7% reduction than the on/off. The linear proportional also has lower air consumption with 6.3% reduction. The PID controller maintained the CO_2 level within the required range.

Phase three involved controlling both CO_2 concentration and temperature setpoints for one zone and two zones. The CO_2 setpoint was 900 ppm, while the temperature setpoint was 70 ^{o}F. For the one-zone control of CO_2 and temperature setpoints, the PID controller required the lowest amount of outside air followed by the on/off control strategy with reductions of 44% and 32.4%, respectively compared to the base-case control strategy. Both the PID controller and the on/off maintained CO_2 level within ± 50 ppm range.

For the two-zone experiments, the On/Off control strategy consumed the lowest amount of outside air with a reduction of 63.2% than the base-case control strategy. Still the PID controller consumed lower amount of outside air with a percent reduction of 21.2 compared to the base case. The results obtained and the

analysis performed for the experiments were used as test-bed to validate the models developed in this work.

A number of computer models were developed in this research work. These models were validated by the experimental work conducted under controlled conditions of the laboratory. These computer models are:

1. A transient computer program that models the transport of indoor pollutants through a multiple-zone building in the HVAC system. With this model, CO_2 concentrations can be measured at three different locations in the HVAC system, namely: the zone, the supply duct, and the return duct.
2. A control model that controls the CO_2 level in the HVAC system. Three control strategies were modeled, namely: the base-case control, the On/Off control strategy, and the PID controller. Each control strategy has its own features validated from the experiments performed. The minimum outside damper leakage has been modeled at 10%, which is a typical value in buildings.
3. A secondary HVAC equipment model that calculates the performance of a cooling coil, preheat coil, reheat coil, and a supply fan. This model is a modified version of an existing VAV model developed in the ASHRAE toolkit.
4. A primary HVAC equipment model that calculates the energy consumption of a chiller, a boiler, and a supply fan.

All of these models are integrated together to form an IQA/HVAC model that calculates the energy consumption of a CO_2-based demand controlled ventilation strategy.

With this IAQ/HVAC model, the energy consumption was calculated for the experiments performed using a typical day in June for four cities in the United States. Energy savings were found for both the On/Off and the PID control strategies over the base case for three cities: Miami, FL.; Phoenix, AZ.; and Madison, MA. There were energy penalties for Boulder, CO., however, when using the On/Off or the PID over the base-case strategy. Decreasing the amount of outside air, in this case, increases the energy consumption. With the temperature economizer, however, savings can be accomplished.

The outside air leakage from the outside damper proved to play a major role on energy consumption. As the leakage increases, the outdoor air consumption increases, and thus, the energy consumption increases.

Generally, as the CO_2 setpoint increases, the outside air consumption decreases, resulting in less energy required to condition the outside air. Special cases exist, though, where decreasing outside air results in energy penalties.

The tuning of the proportional gain constant (K_P) in the PID equation has an effect on the energy consumption. This effect in not significant, though, in that an error of 50% compared to the ideal K_P can produce only 2.5% difference in energy consumption. At extreme cases, where the error reaches 100% between the ideal and the selected K_P, the difference in energy consumption can be as high as 17%.

7.1.2 Guidelines for Implementing IAQ Demand-Controlled Ventilation

From the experiments performed in this work, some results indicated that it is not uncommon for a CO_2 sensor reading to vary by more than ± 50 ppm. This difference can potentially impact the ability to meet the desired IAQ conditions as well as maximize the benefit/cost of a CO_2-based demand-controlled ventilation strategy. This is particularly true when considering that a typical control range of CO_2 will be between 700 ppm and 1000 ppm. For example, if a sensor incorrectly indicates a CO_2 concentration 100 ppm lower than the actual level, CO_2 levels will be greater than 1000 ppm, and hence, less outdoor air will be supplied to the zones than what is actually needed and the minimum indoor air quality conditions will not be satisfied. On the other hand, if the CO_2 sensor indicates higher concentrations than actual, IAQ conditions will be maintained at the cost of unnecessarily conditioning more excess outdoor air than what is needed. This is because, as the sensor incorrectly indicates a CO_2 concentration above 1,000 ppm, the outside damper will open to provide more ventilation air to lower the concentration. Since the actual CO_2 concentration is below the limiting upper level of 1,000 ppm, the damper is unnecessarily opened.

As a guideline, CO_2 sensors should always be calibrated and maintained over a regular basis. They are energy-saving devices that will soon be incorporated into most new building designs. However, the technology of manufacturing CO_2 sensors is still in its early stages, and thus, a sensor can easily

be damaged and affected by misuse, improper handling, vibrations, pressure change, and temperature change.

As noted from the calibration of CO_2 sensors performed in this study, similar sensors (sensors built by the same manufacturer) tend to behave similarly. It is recommended that a building be furnished by the same type of sensor throughout its zones and ducts. Besides, this action makes the calibration as well as the installation and maintenance, of the sensors easier.

The airflow stations in the HVAC system should be reliable enough to accurately measure supply, return, and outside air flows. This is very important when determining the outside air fraction. Relying on a fixed damper position does not guarantee that the correct amount of outside air is introduced to the building. In the experimental work performed in this research, the outside damper was controlled to supply a specific percentage of outside air at all times. Without accurate air flow stations, this feature can not be accomplished.

The leakage of outside air through the outside damper, which is a common problem in HVAC systems, must be kept to a minimum. Conditioning more outside air than what is needed will always result in more energy consumption.

The PID control strategy appeared to work better than the On/Off control strategy in CO_2-based demand-controlled ventilation. The PID control strategy consumes less amount of outside air, and better maintains the CO_2 concentration within limits. Furthermore, the outside damper adapts more to a PID controller. No significant energy penalties are obtained when the proportional gain constant

(K_P) of the PID controller is not well tuned. The difference in energy consumption between the PID and the On/Off control strategy is insignificant and negligible in most cases.

7.2 Recommendations

For this research project, much time was spent in modifying and adapting the Larson laboratory to be able to carry out experiments such as the IAQ experiments. Much time was also spent on experiment setup and conducting the real experiments required for this research. It can be easily said that the laboratory consumed 90% of total time spent on this research. It is thought that much more time is needed to make several computer simulation runs to measure the effectiveness of the discussed strategies in saving energy. Since the time step used in the integrated IAQ/HVAC model is one minute, each annual run is estimated to take from 45-50 hours, depending on the number of iterations used in the model to calculate the required output values. It is recommended then that future work begins where this project ends.

Yet, more experiments can still be conducted to enhance the models developed. For instance, measurements of the zone removal effectiveness in the zones can be measured and mapped in the zones at different ventilation rates. This will make the contaminant transport model more realistic than measuring one single value.

The variation of the CO_2 in the zone can also be mapped and measured at different points in a zone. This project measured CO_2 concentrations at only two

to three points and the average value is then calculated and considered as representative of the zone CO_2 concentration. Mapping the CO_2 concentration in the zone when modeled can improve the control model.

The contaminant transport model assumes constant inter-zone flow from one zone to another, and constant infiltration throughout the entire testing period. A more realistic model would include a module that considers the effects of the inter-zone flow and the variation in the infiltration rate.

Some major modifications might be implemented to the Larson HVAC laboratory to handle experiments such as the IAQ experiments. The outside conditioning station was not used in this research to simulate a pre-determined temperature or humidity that simulate the outdoor conditions in any city in the United States. Operating the station's fan allows the outside air to leak through the supply duct (reaching 30%), and thus, ruins the experiment. Reconfiguration of the lab arrangement would help improve the results. Separating the return duct from the outside air duct by more distance than what is currently used can also improve the IAQ experiments. And finally, reducing the size of the air-handling unit would control the amount of supplied air and circulated air throughout the system.

It is very important when conducting IAQ experiments that the CO_2 sensors be calibrated at least every three months. The technology of the commercially available sensors is still in its early stages. The sensors are very delicate. If subjected to severe weather conditions or high air velocities, they

might be damaged. Moving the sensors frequently might also affect their accuracy.

Because this research work investigates the effect of CO_2-based demand-controlled ventilation strategies in office buildings only, additional investigations should be carried out for other types of buildings, including schools, libraries, museums, grocery stores, and retail buildings.

The movement behavior and airflow characteristics of the outside damper have not been modeled in the control model developed in this work. In the On/Off control strategy, for example, as soon as the CO_2 concentration goes above the setpoint, the control model fully opens the outside-recirculated damper. A more realistic control model would open or close the damper gradually, depending on the characteristics of the damper, until it reaches 100%.

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Appendix A

Contaminant Transport Model

```
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C MULTIPLE ZONE INDOOR CONTAMINANT TRANSPORT MODEL
C
C CALCULATES THE CONCENTRATION OF A CONTAMINANT IN
C A MULTIPLE-ZONE VENTILATED VOLUME AS A FUNCTION
C OF SIMULATION TIME
C
C
C
C SUBROUTINES CALLED:
C
C    SUBROUTINE SHORTCALC
C    SUBROUTINE CALC
C
C INPUT FILES
C
C    BUILDING DESCRIPTION (FORT.300)
C    OCCUPANCY SCHEDULE (FORT.400)
C
C
C VARIABLE LISTING
C
C I- INPUT VALUE
C C- CALCULATED VALUE
C
C
C     A1       C ZONE SOURCES INDEPENDENT OF ZONE CONCENTRATION
C     A2       C ZONE SOURCES DEPAENDENT ON ZONE CONCENTRATION
C     A3       C ZONE SINKS DEPENDANT ON ROOM RETURN DUCT
C                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 CONTAMINAT CONCENTRATION AT THE
C                AHU
C     COA      I CONTAMINANT CONCENTRATION IN OUTSIDE AIR
C     CRET     C RETURN DUCT CONCENTRATION AT THE ZONE
C     CSUP     C SUPPLY DUCT CONTAMINAT CONCENTRATION
C     DELTAT   I SIMULATION TIME STEP
C     DT       I TIME STEP FOR EACH CALCULATION (S)
C     ERR      C ERROR TOLERANCE FOR CHANGE BETWEEN ITERATION
C                STEPS
C     ERRFLAG C ERROR FLAG SET WHEN ERROR TOLERANCE IS NOT MET
C     F        C FUNCTION FOR RHS OF CONCENTRATION DIFFERENTIAL
C                EQUATION
C     FILTEFF  I FRACTIONAL FILTER EFFECIENCY
C     HOD      C CURRENT HOUR OF THE DAY BASED ON SIMULATION TIME
C     HOW      C CURRENT HOUR OF THE THE WEEK BASED ON SIMULATION
C                TIME
```

```
C              I C TIME STEP INDEX
C     IDELAY   C NUMBER OF TIME STEPS FOR AIR TO FLOW THROUGH
C                RETURN PATH
C     INFO     C 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     NI       C NUMBER OF INPUT
C     NP       C NUMBER OF PARAMETERS
C     NSTEPS   C NUMBER OF TIME STEPS
C     NZONES   I NUMBER OF ZONES
C     OUT      C OUTPUT ARRAY
C     PAR      I CONSTANT INPUT ARRAY
C     QCIZF    C TOTAL INTER-ZONE CONTAMINANT FLOW INTO A ZONE
C                (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     QIZFOUT C INTER-ZONE FLOW LEAVING A ZONE (M3/S, FT3/S)
C     QOA      I OUTSIDE AIR VOLUME FLOW RATE INTO A ZONE
C                (M3/S,FT3/S)
C     QOAPU    C ZONE FLOW RATE OF OUTSIDE AIR PER SOURCE UNIT
C                 (L3/S-U,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
C                (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
C                (M3/S, FT3/S)
C     QCIRC    C TOTAL CIRCULATION AIR VOLUME FLOW RATE (M3/S,
C                FT3/S)
C     RE       I ZONE POLLUTANT REMOVAL EFFECTIVENESS WITH HVAC ON
C     REMEF    C ZONE REMOVAL EFFETIVENESS USED IN CALCULATIONS
C     RETVOL   I VOLUME OF THE ZONE'S RETURN AIR PATH TO THE AHU
C                (M3, FT3)
C     RHOAIR   C CONSTANT AIR DENSITY (KG/M3, LBM/FT3)
C     SOURCE   I CONTAMINANT SOURCE VOLUME FLOW RATE IN A ZONE
C                (M3/S, FT3/S)
C     SMULT    C SOURCE MULTIPLIER FOR THE CURRENT TIME STEP
C     SMULTWD I WEEKDAY SCHEDULED SOURCE MULIPLIER
C     STIMEWD I WEEKDAY SCHEDULE TIME THAT SOURCE MULTIPLIER
C                IS APPLIED
C     SMULTWE I WEEKEND SCHEDULED SOURCE MULIPLIER
C     STIMEWE I WEEKEND SCHEDULE TIME THAT SOURCE MULTIPLIER
C                IS APPLIED
C     SUNIT    I UNIT CONTAMINANT SOURCE VOLUME FLOW RATE
C                (M3/S,FT3/S)
C     TIME     C SIMULATION TIME (HR)
C     TIMEPREVC TIME OF THE PREVIOUS SIMULATION STEP
C     UNITFLAGI FLAG TO INDICATE SI OR ENGLISH UNITS
C                (1=SI, 2=ENGLISH)
C     ZONEVOL C ZONE VOLUME (M3, FT3)
C     NUMWD I NUMBER OF HOURLY DIVISIONS IN THE WEEKDAY SCHEDULE
C     NUMWE I NUMBER OF HOURLY DIVISIONS IN THE WEEKEND SCHEDULE
C
      SUBROUTINE TYPE60 (TIME,XIN,OUT,T,DTDT,PAR,INFO,CSUP)
```

```
      IMPLICIT NONE

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

      REAL TIME, XIN(600), OUT(77), T, DTDT, PAR(10)
      REAL C(-60:60,25), CAHU, COA, FILTEFF(3)
      REAL CSUP, ISTEP
      REAL SOURCE(25), SUNIT(25), SMULT(25), RHOAIR
      REAL STIMEWD(25,19),SMULTWD(25,19)
      REAL STIMEWE(25,19),SMULTWE(25,19)
      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 A1(25), A2(25), A3(25)

      COMMON A1, A2, A3
C
C     SET PARAMETERS, DO CHECKS, SET ALL INITIAL ZONE
C     CONCENTRATIONS EQUAL TO THE OUTSIDE AIR CONCENTRATION, AND
C     READ IN SOURCE AND ZONE INFORMATION ON THE FIRST CALL OF
C     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 UINTS IF THE UNITS FLAG IS SET, OTHERWISE
C 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

      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
      NUMWD=19
      NUMWE=19
      OPEN (LUS,STATUS = 'OLD')
      REWIND (LUS)
      DO 10 J =1, NZONES
        READ (LUS,*) SUNIT(J)
C       READ (LUS,*) (STIMEWD(J,K), K=1,NUMWD)
C       READ (LUS,*) (SMULTWD(J,K), K=1,NUMWD)
C       READ (LUS,*) (STIMEWE(J,K), K=1,NUMWE)
C       READ (LUS,*) (SMULTWE(J,K), K=1,NUMWE)
      DO K =1,NUMWD
        READ (LUS,*) STIMEWD(J,K),SMULTWD(J,K)
      ENDDO
      DO K=1,NUMWE
        READ (LUS,*) STIMEWE(J,K),SMULTWE(J,K)
      ENDDO
10    CONTINUE
      CLOSE (LUS)

      OPEN (LUB,STATUS = 'OLD')
      REWIND (LUB)
C     READ (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,
C RETURN TO PREVENT CHANGING THE CONCENTRATION TWICE DURING A
C 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
C APPROPRIATE FOR THE CURRENT SIMULATION TIME, IF THERE IS NO
C SOURCE IN A ZONE, SET THE ZONE REMOVAL EFFECTIENESS 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,NUMWD)
            DO 22 K=1,NUMWD-1
      IF ((HOD.GT.STIMEWD(J,K)) .AND.
     1       (HOD.LE.STIMEWD(J,K+1))) THEN
      SMULT(J) = SMULTWD(J,K)
      GOTO 24
      ENDIF
22    CONTINUE

        ELSE
            SMULT(J) = SMULTWE(J,NUMWE)
            DO 23 K=1,NUMWE-1
      IF ((HOD.GT.STIMEWE(J,K)) .AND.

     1       (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
C PER HOUR 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
C CALCULATE THE RETURN AIR FLOW BY A FLOW BALANCE, IF THE
C VENTILATION SYSTEM IS OFF, INFILTRATION CONTINUES BUT
C INTER-ZONE FLOW IS ASSUMED 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
C SYSTEM IS OFF, A SHORTENED CALCULATION WITHOUT RECIRCULATION
C IS PRERFORMED
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,
     1      QIZF,REMEFF,NSTEPS,NZONES,ZONEVOL,FILTEFF)
      ENDIF
C
C CALCULATE VALUES FOR OUTPUT, AIR FLOWS ARE CONVERTED BACK TO
C AIR 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 I CHANGED THE VALUES BACK TO M3/S INSTEAD OF L/S/PERSON
C (VALUES ARE NOT PER PERSON ANY MORE)
C

       IF (SMULT(J) .LE. 0.0) THEN
        QOAPU = 0.0
       ELSEIF (UNITFLAG .EQ. 2) THEN
C       QOAPU = QOA(J)* 60.0/ SMULT(J)
        QOAPU = QOA(J)
       ELSE
C       QOAPU = QOA(J)*1000.0/SMULT(J)
          QOAPU = QOA(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))+
```

```
     1       QOATOT*COA*(1-FILTEFF(2)))*(1-FILTEFF(3))/QCIRC
        ENDIF
        DO ISTEP = PAR(1)+1, 3*PAR(1)+2
              OUT(ISTEP)=OUT(ISTEP)*1000000.0
          ENDDO


        RETURN
        END

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
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 SOUBROUTINE 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

          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 SLOP
C 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
C READY FOR THE NEXT 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
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
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 SUBROUTINE CALC CALLS:
C      FUNCTION F
C

      SUBROUTINE CALC (C,COA,DT,IDELAY,QRET,QRETTOT,QCIRC,QOATOT,
     1      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
C CONCENTRATION FOR EACH ROOM AND THEN PROCEDDING TO THE
C 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))+
     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 SLOPE ESTIMATE
C AT STEP I+1/2
C

        ERRFLAG = 0
        DO 40 J=1, NZONES
C       PRINT*,'CSUP=',CSUP,'QCIZF(J)',QCIZF(J)
          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
C READY FOR THE NEXT ENTRY INTO THIS SUBROUTINE
C
C

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

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C FUNCTION F
C
C THIS FUNCTION IS FOR A ZONE EITH 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
        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

Contaminant Control Model

```
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C CONTAMINANT CONTROL MODEL
C CONTROL CO2 USING THREE CONTROL STRATEGIES:
C
C           1. BASE CASE
C           2. ON/OFF
C           3. PID
C
C
C CREATED BY:
C
C     MOHSIN ALALAWI
C
C DATE:
C
C     2/21/1999
C
C
C SUBROUTINES CALLED:
C
C     MULTIPLE ZONE INDOOR CONTAMINANT TRANSPORT
C
C INPUT FILES:
C
C     BUILDING DESCRIPTION FILE LUB (FORT.300)
C     OCCUPANCY SCHEDULE FILE LUS (FORT.400)
C     TOTAL AIR SUPPLY TO EACH ZONE IN KG/S (FORT.888)
C
C OUTPUT FILES:
C
C     OUTSIDE AIR FLOWS TO EACH ZONE IN M3/S (QOAMIN.DAT)
C     OUTSIDE AIR CONSUMPTION FOR EACH ZONE IN M3 (FORT.666)
C     CONTAMINANT CONCENTRATION (A.OUT)
C
C VARIABLE LISTING
C
C I- INPUT VALUE
C C- CALCULATED VALUE
C
C     CMAX    C MAXIMUM CONTAMINANT CONCENTRATION IN THE ZONES
C     COA     I CONTAMINANT CONCENTRATION IN OUTSIDE AIR
C     CON     I UPPER LIMIT FOR CONCENTRATION IN THE ON/OFF
C               STRATEGY
C     COFF    I LOWER LIMIT FOR CONCENTRATION IN THE ON/OFF
C               STRATEGY
C     CSET    I CONCENTRATION SETPOINT IN THE PID CONTROL STRATEGY
C     DELTAT  I SIMULATION TIME STEP
C     DT      I TIME STEP FOR EACH CALCULATION (S)
C     ICONTROL I OUTSIDE AIR CONTROL STRATEGY: (1= BASE CASE,
```

```
C                2=PID, 3=ON/OFF)
C      ID      C DELAY TIME IN UOUT
C      IDELAY  I TIME DELAY OF OUTSIDE AIR TO REACH THE ZONES (MIN)
C      IX      C VALUE OF UOUT IN ON/OFF CONTROL STRATEGY
C      KD      I DERIVATIVE GAIN IN THE PID CONTROL EQUATION
C      KI      I INTEGRAL GAIN IN THE PID CONTROL EQUATION
C      KP      I PROPORTIONAL GAIN IN THE PID CONTROL EQUATION
C      QOAMIN  C OUTSIDE AIR SUPPLY IN THE MINUTE TO ALL
C                ZONES(M3/MIN)
C      QOATOT  C TOTAL OUTSIDE AIR SUPPLY TO THE ZONES (M3)
C      SUPMIN  C SUPPLY AIR TO THE ZONES IN THE MINUTE TO ALL ZONES
C                (M3/MIN)
C      SUPTOT  C TOTAL SUPPLY AIR TO THE ZONES (M3)
C      XLEAK   I FRACTION OF OUTSIDE AIR LEAKAGE IN OUTSIDE/RECIRC
C                DAMPER (VALUES BETWEEN 0 TO 1, WHERE: 0=NO
C                        LEAKAGE,1=FULL LEAKEAGE OF OUTSIDE AIR)
C
C
C
      PROGRAM MAIN
      IMPLICIT NONE
      INTEGER INFO(10), I, J, IDELAY, ID, TIME1,UNITFLAG
      INTEGER TIMEB,TIMEE
      INTEGER IX, ICONTROL, LUB
      REAL TIME, XIN(4), OUT(77), T, DTDT, PAR(10),CSUP
      REAL CMAX, CSET ,KP, KI, KD, CON, COFF, QOAVOL, QOAMIN
      REAL UOUT, SUMOLD, COLD, ERR, UOUT(-10:1440), XLEAK
      REAL SUM1, SUM2, SUPMIN, SUPTOT, ZONEVOL(25),RHOAIR
      INTEGER IFLAGON,INCNT,IONOFF
      OPEN (UNIT=10, FILE='XIN.DAT')
      OPEN (UNIT=19, FILE='QOAMIN.DAT')
C
C     PROGRAM CONTROL PARAMETERS AND CONSTANTS
C
      PAR(1)=2
      PAR(2)= 60.0
      PAR(3)=1./60.
      PAR(4)=0.000380
      PAR(5)=400
      PAR(6)=300
      PAR(7)=0.
      PAR(8)=0.
      PAR(9)=0.
      PAR(10)=1
      INFO(6)=3
      INFO(7)=-1
      XIN(3)=0.
      XIN(4)=0.
      SUMOLD = 0.
C
C     DETRMINING THE VALUE OF PHOAIR- CHANGE THIS WITH EACH CITY
C
      UNITFLAG =NINT(PAR(10))
      IF (UNITFLAG.EQ. 2) THEN
      RHOAIR = 0.07517
      ELSE
      RHOAIR=0.9932
```

```
      ENDIF
C
C     OPEN LUB (FORT.300) FILE TO GET THE VOLUME OF ZONES NEEDED
C     TO CALCULATE THE REQUIRED ACH SUPPLIED TO EACH ZONE
C

      LUB= NINT(PAR(6))
      OPEN (LUB,STATUS = 'OLD')
      REWIND (LUB)
      READ (LUB,*) (ZONEVOL(J), J=1, 2)
      CLOSE (LUB)

C
C     THE SETPOINT AND GAIN TERMS FOR THE PID CONTROL STRATEGY
C
      CSET = 1000.
      KP = 3000.0/1000.
      KD = 0.01/1000.
      KI = 0.5/1000.

C
C     THE UPPER AND LOWER LIMITS OF CONCENTRATIONS FOR THE ON/OFF
C     CONTROL STRATEGY

      CON=1005
      COFF=995
C
C     DAMPER CONTROL PARAMETERS FOR THE ON/OFF CONTROL STRATEGY
C     TO PREVENT THE DAMPER FROM OSCILLATING TOO MUCH BETWEEN ON
C     AND OFF WHERE IONOFF IS THE NUMBER OF MINUTES THE DAMPER
C     SHOULD STAY BEFORE SWITCHING ITS POSITION
C

      IFLAGON=0
      INCNT=0
      IONOFF=8

C
C     AHU SYSTEM CHARECTESTICS: OUTSIDE AIR LEAKEAGE, CONTROL
C      STRATEGY, AND TIME DELAY OF OUTSIDE AIR TO REACH ZONES
C

      XLEAK=0.05
      ICONTROL=1
      IDELAY=4
C
C     FOR BASE CASE, OUTSIDE/RECIRCULATED AIR DAMPER POSITION:
C

      IF(ICONTROL.EQ.1) XLEAK=0.10

C     SYSTEM INITIAL PARAMETERS

      COLD=380.-CSET
      DO ID=-10,0,1
      UOUT(ID)=XLEAK
      ENDDO
```

```
      ID=0
      SUM1=0.
      SUM2=0.
      SUMOLD=0.
      IX=0
      QOAVOL=0.
      SUPTOT=0.


      timeb=6.
      timee=19.

      DO TIME=timeb,timee,1./60.

      ID=ID+1

      READ(10,*)(XIN(I),I=1,2)

C
C     CHANGING THE UNITS OF SUPPLIED AIR FLOW OF EACH ZONE FROM
C     KG/S IN FORT.888 FILE OBTAINED FROM VAV PROGRAM TO ACH
C     VALUES WHICH IS THE CORRECT FORMAT FOR THE CONTROL PROGRAM
C     TO READ IN
C     ****RHOAIR IS OBTAINED FROM ABOVE BASED ON CITY
C     ****ZONEVOL IS OBTAINED FROM (FORT.300)
C     ****XIN ARE OBTAINED FROM FORT.888 (OUTPUT FROM VAV)
C
      READ(888,*)(XIN(I),I=1,2)

      DO I=1,2
         XIN(I)=XIN(I)*3600./(RHOAIR*ZONEVOL(I))
      ENDDO

      CALL TYPE60 (TIME,XIN,OUT,T,DTDT,PAR,INFO,CSUP)

C
C     SELECTING THE MAXIMUM CONCENTRATION OF ALL ZONES FOR
C     CONTROL
C
      CMAX = OUT(5)
      IF(OUT(6).GT.CMAX) CMAX = OUT(6)
C
C     CONTROL STRATEGIES:
C
C     ICONTROL = OUTSIDE AIR CONSUMPTION IS CALCULATED ACCORDING
C                TO
C         1      =    BASE CASE CONTROL STRATEGY
C         2      =    PID CONTROL STRATEGY
C         3      =    ON/OFF CONTROL STRATEGY
C
C
C     ICONTROL =1 OUTSIDE AIR CONSUMPTION IS CALCULATED ACCORDING
C                   TOTHE BASE CASE OUTSIDE AIR FRACTION

      IF(ICONTROL.EQ.1) THEN

         UOUT(ID)=0.11
```

```
C
C      ICONTROL =1 OUTSIDE AIR CONSUMPTION IS CALCULATED ACCORDING
C                  TOTHE PID CONTROL STRATEGY

       ELSEIF (ICONTROL.EQ.2)THEN

          ERR = CMAX-CSET
          SUMOLD =SUMOLD+ERR
          UOUT(ID) = KP*(ERR+(KI*SUMOLD))
                IF(UOUT(ID).GE.0.87) UOUT(ID)=0.87
                IF(UOUT(ID).LT.XLEAK) UOUT(ID)=XLEAK
C
C      ICONTROL =3 OUTSIDE AIR CONSUMPTION IS CALCULATED ACCORDING
C                  ON/OFF CONTROL STRATEGY

       ELSEIF (ICONTROL.EQ.3) THEN
C
       IF(INCNT.LT.IONOFF)THEN
         UOUT(ID)=UOUT(ID-1)
         INCNT=INCNT+1
       ELSE
C        DURATION MORE THAN IONOFF MINUTES (4)
         IF(IFLAGON.EQ.0)THEN
C        FLAG WAS OFF BEFORE
          IF(CMAX.LT.COFF)THEN
            UOUT(ID)=XLEAK
            INCNT=INCNT+1
          ENDIF
          IF(CMAX.GT.CON)THEN
           IFLAGON=1
           INCNT=1
           UOUT(ID)=0.87
          ENDIF
         ELSE

          IF(CMAX.LT.COFF)THEN
            UOUT(ID)=XLEAK
            INCNT=1
           IFLAGON=0
          ENDIF
          IF(CMAX.GT.CON)THEN
           INCNT=INCNT+1
           UOUT(ID)=0.87
          ENDIF
       ENDIF
       ENDIF
       IF(UOUT(ID).LT.XLEAK)UOUT(ID)=XLEAK
       IF(UOUT(ID).GT.0.87)UOUT(ID)=0.87

       ENDIF
       QOAMIN=0.
       SUPMIN=0.

       DO J =1,2

          XIN(2+J)=XIN(J)*UOUT(ID-IDELAY)
C
```

```
C      CALCULATING QOAMIN AND SUPMIN (UNITS ARE M3/S
C      [1/HR * M3 * 1 HR/3600 S])
C
          QOAMIN=QOAMIN+XIN(J)*UOUT(ID-IDELAY)*ZONEVOL(J)/3600.
          SUPMIN=SUPMIN+XIN(J)*ZONEVOL(J)/3600.

       ENDDO
C
C      WRITING THE TOTOL AND OUTSIDE AIR VOLUME CONSUMPTION IN
C      FILE:
C      QOAMIN.DAT (QOAVOL IS TOTAL OUTSIDE AIR VOLUME IN M3)
C      (SUPTOT IS THE TOTAL SUPPLIED AIR IN M3)
C
       QOAVOL=QOAVOL+QOAMIN
       SUPTOT=SUPTOT+SUPMIN
       WRITE(19,*)UOUT(ID-IDELAY),QOAMIN,QOAVOL,SUPMIN,SUPTOT

       CALL TYPE60 (TIME,XIN,OUT,T,DTDT,PAR,INFO,CSUP)

       SUM1=SUM1+XIN(3)*ZONEVOL(1)/60.
       SUM2=SUM2+XIN(4)*ZONEVOL(2)/60.
       TIME1=TIME*60.-timeb*60

       WRITE(*,98)INT(TIME1),(OUT(J),J=1,3*PAR(1)+2)


       WRITE(666,*)INT(TIME1),UOUT(ID-IDELAY),SUM1,SUM2,SUM1+SUM2
       INFO(7)=1
       ENDDO
98     FORMAT(I4,2X,9(F10.2, 2X))
       STOP
       END
```

Appendix C

Variable Air Volume (VAV) Model

```
C*************************************************************
C*    SUBROUTINE:              VAV
C*
C*    LANGUAGE:                FORTRAN 77
C*
C*    PURPOSE:                 To simulate a variable air volume
C*                             multizone system with reheat at
C*                             the zone and an outside air
C*                             temperature controlled economizer
C*                             cycle.
C*************************************************************
C*    INPUT VARIABLES:
C*    TAirAmb       Ambient air temperature
(C)
C*    WAirAmb       Ambient humidity ratio                    (-)
C*    MZHeat        Heating supply air flow rate           (kg/s)
C*    TZSetPt       Setpoint temperature for                  (C)
C*    QZsen         Zone sensible load                        (W)
C*    QZlat         Zone latent load                          (W)
C*    TAirPri       Primary air design temperature            (C)
C*    MZMin         Minimum design air flow to zone        (kg/s)
C*    MAmbMin       Minimum flow rate for ambient a        (kg/s)
C*    EconClose     Ambient air temperature setpoint for minimum
C*                  damper position
(C)
C*    THCLiqEnt     Inlet water temperature for heating coils(C)
C*    TCCLiqEnt     Inlet water temperature for cooling coil (C)
C*
C*    PARAMETERS
C*    PPHCoil(i)    Array containing preheat coil parameters
C*    PRHCoil(i)    Array containing reheat coil parameters
C*    PCCoil(i)     Array containing cooling coil parameters
C*    CoilType      Coil type                                 (-)
C*                  0 = Flat continuous fins
C*                  1 = Circular fins
C*    APri          Primary surface area/face area            (-)
C*    ASec          Secondary surface area/face area          (-)
C*    AInt          Internal surface area/face area           (-)
C*    AMinFlow      Minimum air flow area/face area           (-)
C*    KFin          Fin thermal conductivity             (W/m C)
C*    AFace         Coil face area
(m2)
C*    FinPerM       Number of fins per meter
(1/m)
C*    TubePerRow    Number of tubes per row                   (-)
C*    NumRow        Number of rows                            (-)
C*    OdTube        Outside diameter of tub                   (m)
C*    IdTube        Inside diameter of tu                     (m)
C*    FinThick      Fin thickness                             (m)
C*    TubeSpace     Tube longitudinal spacing                 (m)
C*    FinDia        Fin diameter (coil type = 1) or           (m)
```

```
C*                  distance between rows (coil type = 0)
C*    KTube         Tube thermal conductivity              (W/m C)
C*    FoulFactor    Fouling factor for tubes              (C m2/W)
C*
C*    PSFan(i)      Array containing supply fan parameters
C*    FlowRated     Rated air volumetric flow rate          (m3/s)
C*    BkWRated      Rated shaft power (brake kW)               (W)
C*    EffMot        Motor drive efficiency                     (-)
C*    MotorToAir    Fraction of motor loss to airstream        (-)
C*    ModeCont      Mode for fan control                       (-)
C*                  ModeCont = 1:  Discharge dampers
C*                  ModeCont = 2:  Inlet vanes
C*                  ModeCont = 3:  Variable speed drive
C*
C*    PROPERTIES
C*    Prop(i)       Array containing air and liquid properties
C*    CpAir         Air specific heat                    (J/kg C)
C*    CpLiq         Liquid specific heat                 (J/kg C)
C*    PrAir         Air Prandtl number                         (-)
C*    PrLiq         Liquid Prandtl number                      (-)
C*    DViscAir      Air dynamic viscosity                (kg/m s)
C*    DViscLiq      Liquid dynamic viscosity             (kg/m s)
C*    Patm          Atmospheric pressure                      (Pa)
C*    RhoAir        Density of air                        (kg/m3)
C*    KLiq          Liquid thermal conductivity           (W/m C)
C*    Hfg           Latent heat of vaporization of water
C*                  referenced at 0 C                       (J/kg)
C*    CpVap         Specific heat of water vapor         (J/kg C)
C*    RAir          Gas constant for air                 (J/kg C)
C*********************************************************************
C*     OUTPUT VARIABLES:
C*     TotQPHSen    Preheat coil sensible heat transfer rate  (W)
C*     TotQRHSen    Reheat coil sensible heat transfer rate   (W)
C*     TotQCCSen    Cooling coil sensible heat transfer rate  (W)
C*     TotQCCTot    Cooling coil total heat transfer rate     (W)
C*     TotSFanPow   Supply fan power consumption              (W)
C*     ErrStat      Error status flag                         (-)
C*********************************************************************
C*     MAJOR RESTRICTIONS:
C*
C*    DEVELOPER:              Shauna Gabel
C*                            Michael J. Brandemuehl, PhD, PE
C*                            University of Colorado at Boulder
C*
C*     DATE:                  January 1, 1992
C*
C*     MODIFIED BY:           Mohsin Alalawi
C*                            University of Colorado at Boulder
C*
C*     DATE:
C*
C*                            April 15, 1999
C*    MAJOR MODIFICATIONS:
C*
C*                      1. VARIABLE OUTSIDE AIR FLOW RATE
C*                      2. SENSIBLE LOADS CAN BE NEGATIVE
C*                      3. TIME STEP
```

```
C*
C*
C     INCLUDE FILES:
C                    prop.inc
C                    coildet.inc
C                    fansim.inc
C
C     INPUT FILES:
C                    WEATHER FILE (W2.DAT)
C                    ZONES DATA (VAVINP.DAT)
C                    OUTSIDE AIR CONSUMPTION FROM CONTROL
C                    PROGRAM (QOAMIN.DAT)
C     OUTPUT FILES:
C                    SIMULATION RESULTS (OUTPUT)
C                    EQUIPMEANT WATTAGE (FORT.567) TO ENERGY
C                    TOTAL ZONE SUPPLY AIR FLOW RATE (FORT.888)
C
C     SUBROUTINES CALLED:      ECON
C                              FANSIM
C                              COILINV
C     FUNCTIONS CALLED:        None
C
C     REVISION HISTORY:        None
C
C     REFERENCE:  ASHRAE 1983, Simplified Energy Analysis
C                 Using the Modified Bin Method, Atlanta:
C                 American Society of Heating, Refrigeration,
C                 and Air-Conditioning Engineers, Inc..
C*****************************************************************
C   INTERNAL VARIABLES:
C   mAirPri    Primary air mass flow rate                    (kg/s)
C   mZAir      Zone air mass flow rate                       (kg/s)
C   tZAir      Required zone supply air temperature             (C)
C   wCCAirLvg Cooling coil leaving humidity ratio               (-)
C   oldW      Previous cooling coil leaving air humidity ratio(-)
C   wDum       Humidity ratio dummy variable for reheat coil   (-)
C   wAirMix    Mixed air humidity ratio                        (-)
C   tAirMix    Air temperature leaving economizer              (C)
C   wAirRet    Return air humidity ratio                       (-)
C   tAirRet    Return air temperature                          (C)
C   qZlatTot   Total latent load for all zones                 (W)
C   rANum      Numerator for return air temp. calculation
C   sFanPow    Hourly supply fan power                         (W)
C   tCCAirLvg Air temperature leaving cooling coil             (C)
C   tLiqPHLvg Liquid temperature leaving preheat coil          (C)
C   tLiqRHLvg Liquid temperature leaving reheat coil           (C)
C   tLiqCCLvg Liquid temperature leaving cooling coi           (C)
C   pHFWet     Fraction preheat coil wet                       (-)
C   rHFWet     Fraction reheat coil wet                        (-)
C   cCFWet     Fraction cooling coil wet                       (-)
C   qCCtot     Cooling coil total heat transfer rate           (W)
C   qCCsen     Cooling coil sensible heat transfer rate        (W)
C   qPHsen     Preheat coil sensible heat transfer rate        (W)
C   qRHsen     Reheat coil sensible heat transfer rate         (W)
C   mPHLiq     Preheat coil water flow rate                 (kg/s)
C   mRHLiq     Reheat coil water flow rate                  (kg/s)
C   mCCLiq     Cooling coil water flow rate                 (kg/s)
```

```
C   numHr      Number of hours in simulation                    (hr)
C   zNum       Number of zones                                   (-)
C   iter,n,h   Loop counters                                     (-)
C   itmax      Iteration maximum                                 (-)
C   lUR4,lUR5  Logical read units                                (-)
C   lUW        Logical write unit                                (-)
C   error      Deviation of dependent variable in iteration
C   icvg       Iteration convergence flag
C   F1,F2      Previous values of dependent variable in XITERATE
C   X1,X2      Previous values of independent variable in XITERATE
C***********************************************************************

      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd
      REAL Prop(16), h, XITERATE

      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)
      INTEGER NPFanPmp
      PARAMETER (NPFanPmp=13)

      INTEGER EffMot
      INTEGER MotorLoss
      INTEGER DiamWheel
      INTEGER DPRated
      INTEGER FlowRated
      INTEGER PowRated
      INTEGER PlrContl
      INTEGER Coef0Fflp
      INTEGER Coef1Fflp
      INTEGER Coef2Fflp
      INTEGER Coef3Fflp
      INTEGER Coef0Head
      INTEGER Coef1Head
      INTEGER Coef2Head
      INTEGER Coef3Head
      INTEGER Coef4Head
      INTEGER Coef0Eff
      INTEGER Coef1Eff
      INTEGER Coef2Eff
      INTEGER Coef3Eff
      INTEGER Coef4Eff
```

```
PARAMETER (EffMot    = 1)
PARAMETER (MotorLoss = 2)
PARAMETER (DiamWheel = 3)
PARAMETER (FlowRated = 3)
PARAMETER (PowRated  = 4)
PARAMETER (DPRated   = 5)
PARAMETER (PlrContl  = 5)
PARAMETER (Coef0Fflp = 6)
PARAMETER (Coef1Fflp = 7)
PARAMETER (Coef2Fflp = 8)
PARAMETER (Coef3Fflp = 9)
PARAMETER (Coef0Head = 4)
PARAMETER (Coef1Head = 5)
PARAMETER (Coef2Head = 6)
PARAMETER (Coef3Head = 7)
PARAMETER (Coef4Head = 8)
PARAMETER (Coef0Eff  = 9)
PARAMETER (Coef1Eff  =10)
PARAMETER (Coef2Eff  =11)
PARAMETER (Coef3Eff  =12)
PARAMETER (Coef4Eff  =13)
PARAMETER (NPCDet=27)

INTEGER CoilType
INTEGER AFace
INTEGER ARatPri
INTEGER ARatSec
INTEGER ARatInt
INTEGER ARatFlow
INTEGER FinsPerM
INTEGER TubePerRow
INTEGER NumRow
INTEGER NumCircuit
INTEGER OdTube
INTEGER IdTube
INTEGER FinThick
INTEGER RowSpace
INTEGER FinDia
INTEGER KTube
INTEGER KFin
INTEGER FoulFactor

INTEGER TubeLength
INTEGER DiamFin
INTEGER FinHt
INTEGER DHydCoil
INTEGER ARatTot
INTEGER ASecTot
INTEGER AIntTot
INTEGER ATot
INTEGER CondResist

PARAMETER (CoilType=1)
PARAMETER (AFace=2)
PARAMETER (ARatPri=3)
PARAMETER (ARatSec=4)
```

```
      PARAMETER (ARatInt=5)
      PARAMETER (ARatFlow=6)
      PARAMETER (FinsPerM=7)
      PARAMETER (TubePerRow=8)
      PARAMETER (NumRow=9)
      PARAMETER (NumCircuit=10)
      PARAMETER (OdTube=11)
      PARAMETER (IdTube=12)
      PARAMETER (FinThick=13)
      PARAMETER (RowSpace=14)
      PARAMETER (FinDia=15)
      PARAMETER (KTube=16)
      PARAMETER (KFin=17)
      PARAMETER (FoulFactor=18)
      PARAMETER (TubeLength=19)
      PARAMETER (DiamFin=20)
      PARAMETER (FinHt=21)
      PARAMETER (DHydCoil=22)
      PARAMETER (ARatTot=23)
      PARAMETER (ASecTot=24)
      PARAMETER (AIntTot=25)
      PARAMETER (ATot=26)
      PARAMETER (CondResist=27)

      EXTERNAL HCDET,CCDET
      CHARACTER*12 weatherfile,zonefile,outfile
      DIMENSION PCCoil(NPCDet),PRHCoil(NPCDet)
      DIMENSION PPHCoil(NPCDet),PSFan(NPFanPmp)
      real QZlat(3)
      REAL MZAir(3),MZMin(3),MZHeat(3),QZSen(3),TZSetPt(3),ratio
      REAL mPHLiq,mRHLiq,mCCLiq,mAirPri,MAmbMin,mAirRet
      REAL MAirAmb,mDum
      INTEGER zNum,ErrStat
      DATA itmax/20/

      ErrStat = 0

C1*** Initialize the array containing air and water properties
      DATA Prop(Patm) /101325./
      DATA Prop(CpAir) /1006./
      DATA Prop(CpWat) /4186./
      DATA Prop(CpVap) /1805./
      DATA Prop(CpLiq) /4186./
      DATA Prop(DViscAir) /0.0000182/
      DATA Prop(DViscLiq) /0.00144/
      DATA Prop(KAir) /0.0260/
      DATA Prop(KLiq) /0.604/
      DATA Prop(RhoLiq) /998./
      DATA Prop(Hfg) /2501000./
      DATA Prop(RAir) /287.055/
      DATA Prop(TKelMult) /1./
      DATA Prop(TAbsAdd) /273.15/
      DATA Prop(PaMult) /1./
      DATA Prop(PAbsAdd) /0./

C *** Initialize cooling coil parameters
      PCCoil(CoilType) = 1.0
```

```
      PCCoil(ARatPri) = 5.218
      PCCoil(ARatSec) = 74.66
      PCCoil(ARatInt) = 5.06
      PCCoil(ARatFlow) = 0.53
      PCCoil(KFin) = 228.3
      PCCoil(AFace) = 2.235
      PCCoil(FinsPerM) = 288.714
      PCCoil(TubePerRow) = 22.0
      PCCoil(NumRow) = 2.0
      PCCoil(NumCircuit) = 22.
      PCCoil(OdTube) = .016357
      PCCoil(IdTube) = .015341
      PCCoil(FinThick) = .0001905
      PCCoil(RowSpace) = .0254
      PCCoil(FinDia) = .0381
      PCCoil(KTube) = 385.7
      PCCoil(FoulFactor) = 0.00003

C *** Initialize reheat coil parameters
      PRHCoil(CoilType) = 1.0
      PRHCoil(ARatPri) = 5.218
      PRHCoil(ARatSec) = 74.66
      PRHCoil(ARatInt) = 5.06
      PRHCoil(ARatFlow) = 0.53
      PRHCoil(KFin) = 228.3
      PRHCoil(AFace) = 0.8
      PRHCoil(FinsPerM) = 288.714
      PRHCoil(TubePerRow) = 10.0
      PRHCoil(NumRow) = 2.0
      PRHCoil(NumCircuit) = 5.
      PRHCoil(OdTube) = .016357
      PRHCoil(IdTube) = .015341
      PRHCoil(FinThick) = .0001905
      PRHCoil(RowSpace) = .0254
      PRHCoil(FinDia) = .0381
      PRHCoil(KTube) = 385.7
      PRHCoil(FoulFactor) = 0.00003

C *** Initialize preheat coil parameters
      PPHCoil(CoilType) = 1.0
      PPHCoil(ARatPri) = 5.218
      PPHCoil(ARatSec) = 74.66
      PPHCoil(ARatInt) = 5.06
      PPHCoil(ARatFlow) = 0.53
      PPHCoil(KFin) = 228.3
      PPHCoil(AFace) = 2.235
      PPHCoil(FinsPerM) = 288.714
      PPHCoil(TubePerRow) = 22.0
      PPHCoil(NumRow) = 4.0
      PPHCoil(NumCircuit) = 11.
      PPHCoil(OdTube) = .016357
      PPHCoil(IdTube) = .015341
      PPHCoil(FinThick) = .0001905
      PPHCoil(RowSpace) = .0254
      PPHCoil(FinDia) = .0381
      PPHCoil(KTube) = 385.7
      PPHCoil(FoulFactor) = 0.00003
```

```
C     air density at sea level
C     RhoAir=1.2

C *** Initialize fan parameters
      PSFan(FlowRated) = 5.6
      PSFan(PowRated) = 17500
      PSFan(EffMot) = 0.8
      PSFan(MotorLoss) = 1
      PSFan(PlrContl) = 3

C *** Initialize system parameters

C     PRIMARY SUPPLY AIR TEMPERATURE
      TAirPri =  12.8
C
C     MAmbMin IS NOT ANY MORE CONSTANT. IT VARIES WITH
C     THE OUTSIDE/RECIRCDAMPER POSITION
C     MAmbMin =  1.2 (MODIFICATION FROM THE ORIGINAL PROGRAM)
C
C     INLET WATER TEMPERATURE FOR HEATING COILS
      THCLiqEnt =  82.
C
C     INLET COOLING TEMPERATURE FOR COOLING COIL
      TCCLiqEnt =  4.4
C
C     ECONOMIZER STATUS
      EconMode = 1.0
C
C     ECONCLOSE IS THE AMBIENT TEMPERATURE AT WHICH
C     THE DAMPER WILL BE SETAT MINIMUM POSITION

      EconClose =  22.0


      TotSFanPow = 0.0
      TotQPHsen = 0.0
      TotQRHsen = 0.0
      QtotCCTot = 0.0
      QsenCCTot = 0.0

      LUR4=7
      LUR5=8
      LUW=10

      PRINT *, 'Enter name of weather data file'
      READ(*,1001) weatherfile
      OPEN(UNIT=19,FILE='QOAMIN.DAT')
      OPEN(UNIT=LUR4,FILE=weatherFile)
      PRINT *, 'Enter the name of the zone input file'
      READ(*,1001) zonefile
      OPEN(UNIT=LUR5,FILE=zonefile)
      WRITE(*,*) 'Enter name of output file:'
      READ(*,1001) outFile
      OPEN(UNIT=LUW,FILE=outFile)
      WRITE(LUW,*) 'VARIABLE AIR VOLUME SYSTEM SIMULATION'
      READ*,RHOAIR
C *** Begin hour loop
```

```
      READ(LUR5,*) numHr, zNum
      DO 600 h = 1./60.,numHr,1/60.
        READ(LUR4,*) TAirAmb,WAirAmb
C     READING QOAMIN FROM A SEPARATE FILE CALLED qoamin.dat
c     the value of qoa is volumetric change m3/s
      READ(19,*)ratio
c     ratio=0.45
c     to change it to kg/s
      MAmbMin=MAmbMin*RhoAir

        qZlatTot = 0.0
        mAirPri = 0.0
        rANum = 0.0
        WRITE(LUW,1011) h,TAirAmb,WAirAmb

C *** Begin zone loop
      READ(LUR5,*) (QZsen(m),m=1,zNum),(QZlat(m),m=1,zNum),
     1       (TZSetPt(m),m=1,zNum),(MZMin(m),m=1,zNum),
     1       (MZHeat(m),m=1,zNum)
        DO 200 n = 1,zNum
          READ(LUR5,*)
QZsen(n),QZlat,TZSetPt(n),MZMin(n),MZHeat(n)

C *** Calculate the required air flow to satisfy zone load in
C *** kg/s

          mZAir(n) = QZsen(n)/(Prop(CpAir)*(TZSetpt(n)-TAirPri))
      if(QZsen(n).lt.0.)then
      mZAir(n)=mZmin(n)
      endif

C *** Sum zone primary air mass flow rates and latent loads

          mAirPri = mAirPri+MZAir(n)
          qZlatTot = qZlatTot+QZlat(n)

C *** Numerator for return air calculation

          rANum = TZSetPt(n)*MZAir(n)+rANum

WRITE(LUW,1012)n,MZAir(n),QZsen(n),QZlat(n),TZSetPt(n),MZMin(n)

 200    CONTINUE
      write(888,*)(mzair(n),n=1,znum)
C *** End of zone loop

C *** Return air temperature calculation, considering that
C *** each zone may have a different setpoint and assuming that
C *** the return air will be the same as the setpoint temperature
C *** for the room

        tAirRet = rANum/mAirPri

C *** Iterate on the cooling coil leaving air humidity ratio
C *** until convergence
```

```
C *** Initial guess at cooling coil leaving air humidity
C *** ratio, relating to air leaving conditions of 55 DB and 90%
C *** RH

        wCCAirLvg = 0.0077
DO 300 iter= 1,itmax

          oldW = wCCAirLvg

C *** Return air humidity ratio calculation

          wAirRet = wCCAirLvg+qZlatTot/(Prop(Hfg)*mAirPri)

C *** Calculate the supply fan power consumption and
C *** the leaving air temperature
      wCCAirLvg1=wCCAirLvg
          mDum = -mAirPri
          CALL FANSIM (Prop,PSFan,mDum,TAirPri,wCCAirLvg,
     &                  tCCAirLvg,wCCAirLvg1,sFanPow,Errstat)

C *** Calculate the mixed air temperature and humidity
C *** ratio from the economizer

    CALL ECON (Prop,tAirRet,wAirRet,TAirAmb,WAirAmb,mAirPri,
     &         MAmbMin,tCCAirLvg,TAirAmb,EconClose,EconMode,
     &         TAirMix,wAirMix,mAirRet,mAirAmb,ErrStat,ratio)

       IF (tAirMix .LT. tCCAirLvg) THEN

C *** Preheat is necessary.  Calculate the coil load
C *** required to heat mixed air to desired leaving
C *** air temperature.

     CALL COILINV (HCDET,Prop,PPHCoil,THCLiqEnt,mAirPri,
     &             tAirMix,wAirMix,tCCAirLvg,wCCAirLvg,mPHLiq,
     &             tLiqPHLvg,qPHtot,qPHsen,pHFwet,Errstat)

            qCCtot = 0.0
            qCCsen = 0.0
            mCCLiq = 0.0

          ELSE IF (tAirMix .GT. tCCAirLvg) THEN

C *** Cooling is necessary.  Calculate the cooling coil
C *** load, chilled water flow rate, and leaving air humidity
C *** ratio for the desired leaving air temperature

     CALL COILINV (CCDET,Prop,PCCoil,tCCLiqEnt,mAirPri,tAirMix,
     &             wAirMix,tCCAirLvg,wCCAirLvg,mCCLiq,tLiqCCLvg,
     &             qCCtot,qCCsen,cCFWet,Errstat)

            qPHsen = 0.0
            qPHtot = 0.0
            mPHLiq = 0.0

          ELSE
```

```
C ***  Zone cooling loads can be met with economizer alone.

            qCCtot = 0.0
            qCCsen = 0.0
            mCCLiq = 0.0
            qPHsen = 0.0
            qPHtot = 0.0
            mPHLiq = 0.0
            wCCAirLvg = wAirMix

          ENDIF

C1*** Compare newly calculated humidity ratio with
C *** old value and determine new estimate

      error = oldW-wCCAirLvg
      wCCAirLvg = XITERATE(wCCAirLvg,error,X1,F1,X2,F2,iter,icvg)

C1*** If converged, leave loop
          IF (icvg .EQ. 1) GO TO 400

300     CONTINUE
C1*** If not converged after itmax iterations,
C1*** return error code

        WRITE(*,1010) itmax
1010    FORMAT(/1X,'*** ERROR IN SUBROUTINE CAV ***'/
     &          1X,'The humidity ratio has not converged
after,'I2,
     &          'iterations'/)
        ErrStat = 1

C *** End of humidity ratio loop

400   CONTINUE

C *** Begin zone loop for reheat coil load calculations

      DO 500 n = 1,zNum

C *** If the minimum supply air flow does not satisfy
C *** the zone load then reheat supply air at zone and
C *** sum reheat coil load

      If(Qzsen(n).lt.0.)then
         mZAir(n)=MZHeat(n)
         tZAir=TZSetPt(n)-QZsen(n)/(Prop(CpAir)*MZheat(n))

          CALL COILINV (HCDET,Prop,PRHCoil,THCLiqEnt,MZHeat1,
     &                  TAirPri,wCCAirLvg,tZAir,wCCAirLvg,mRHLiq,
     &                  tLiqRHLvg,qRHtot,qRHsen,rHFwet,Errstat)
      else

        IF (mZAir(n) .LT. MZMin(n)) THEN
          mZAir(n) = MZHeat(n)
          tZAir = TZSetPt(n)-QZsen(n)/(Prop(CpAir)*MZheat(n))
```

```
            MZHEAT1=MZHEAT(N)

            CALL COILINV (HCDET,Prop,PRHCoil,THCLiqEnt,MZHeat1,
     &                    TAirPri,wCCAirLvg,tZAir,wCCAirLvg,mRHLiq,
     &                    tLiqRHLvg,qRHtot,qRHsen,rHFwet,Errstat)
          ELSE
            mRHLiq=0.0
            qRHsen=0.0
          ENDIF
          ENDIF

          qZRHsen = qZRHsen+qRHsen

500     CONTINUE

C *** Write hourly results

        WRITE(LUW,1015) qCCtot,qCCsen,qZRHsen,qPHsen,sFanPow
        qhtot = abs(qZRHsen)+abs(qPHsen)
        write(567,*)h,qcctot,qhtot,sfanpow,TAirAmb

C *** Sum hourly totals

            TotSFanPow = sFanPow+TotSFanPow
            TotQPHsen = qPHsen+TotQPHsen
            TotQRHsen = qZRHsen+TotQRHsen
            QtotCCTot = qCCtot+QtotCCTot
            QsenCCTot = qCCsen+QsenCCTot

c       print*,TAirAmb,tAirMix,ratio
600     CONTINUE

C *** End of hour loop

        CLOSE(UNIT=7)
        CLOSE(UNIT=8)

C *** Write output to a file

        WRITE(LUW,'(/1X,''SIMULATION SUMMARY'')')
        WRITE(LUW,1002) QtotCCTot/60.
        WRITE(LUW,1003) QsenCCTot/60.
        WRITE(LUW,1004) TotQRHsen/60.
        WRITE(LUW,1005) TotQPHsen/60.
        WRITE(LUW,1006) TotSFanPow

 1001 FORMAT(A)

 1002 FORMAT(1X,'Cooling Coil Total Heat Transfer Rate'T47,F13.2,
     &          ' W')
 1003 FORMAT(1X,'Cooling Coil Sensible Heat Transfer
Rate'T47,F13.2,
     &          ' W')
 1004 FORMAT(1X,'Reheat Coil Total Heat Transfer Rate'T47,F13.2,
     &          ' W')
 1005 FORMAT(1X,'Preheat Coil Total Heat Transfer Rate'T47,F13.2,
     &          ' W')
```

```
 1006 FORMAT(1X,'Total Supply Fan Power'T47,F13.2,' W')
 1011  FORMAT(/1X,'TIME = ',f7.3,' hr'/,
     &   1X,'Outdoor Air Temperature and
Humidity:'3X,F10.2,F10.4/
     &     1X,'ZONE INFORMATION'/ 1X,10X,'   AirFlow  SenLoad',
     &     '   LatLoad   SetPoint   MinFlow')
 1012  FORMAT(1X,'  Zone ',I2,':',F10.3,2F10.0,2F10.3)
 1015  FORMAT(1X,'AIR HANDLER LOADS (W)'/
     &       1X,'   TotCoil   SenCoil   Reheat   Preheat
FanPower'/
     &       1X,4F10.0,f10.4)

      CLOSE(UNIT=10)

      END
```

Appendix D

Equipment Model

```
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
CC
C
C  EQUIPMENT MODEL
C
C  CALCULATES ENERGY CONSUMPTION OF HVAC EQUIPMENT
C
C  CREATED BY:
C                  MOHSIN ALALAWI
C
C  DATE:
C                  MAY 25, 1999
C
C  SUBROUTINE CALLED:
C
C     NONE
C
C
C  INPUT FILE:
C
C       COOLING AND HEATING COILS HEAT TRANSFER RATES, AND
C        SUPPLY FAN POWER CONSUMPTION (FORT.567- FROM VAV.F)
C
C  OUTPUT FILE:
C
C       ENERGY CONSUMPTION OF CHILLER, BOILER AND SUPPLY FAN
C       (A.OUT)
C
C  VARIABLE LISTING
C
C
C  I-INPUT VALUE
C  C-CALULATED VALUE
C
C     AB     I  BOILER (A) COEFFICIENT IN THE PART-LOAD
C                CHARACTERISTICS
C     AC     I  CHILLER (A) COEFFICIENT IN THE PART-LOAD
C                CHARECTERISTICS
C     BB     I  BOILER (B) COEFFICIENT IN THE PART-LOAD
C                CHARECTERISTICS
C     BBN    I  BOILER NOMINAL (BOILER SIZE) CAPACITY (W)
C     BC     I  CHILLER (B) COEFFICIENT IN THE PART-LOAD
C                CHARECTERISTICS
C     CB     I  BOILER (C) COEFFICIENT IN THE PART-LOAD
C                CHARECTERISTICS
C     CC     I  CHILLER (C) COEFFICIENT IN THE PART-LOAD
C                CHARECTERISTICS
C     CCAP   C  CHILLER LOAD (W.MIN)
```

```
C      CCAPN I  NOMINAL CHILLER CAPACITY (TONS)
C      DLETA I  DELTA OF CHILLER (SLOPE IN THE CHILLER CAPACITY
C               EQUATION)
C      EFFB  I  BOILER FULL-LOAD EFFICIENCY
C      EGYB  C  ENERGY CONSUMPTION OF BOILER (KWH)
C      EGYC  C  ENERGY CONSUMPTION OF CHILLER (KWH)
C      EIR   I  ENERGY INPUT RATIO OF CHILLER
C      PEL   C  ELECTRICAL POWER OF CHILLER, BOILER, OR SUPPLY FAN
C      PLR   C  PART LOAD RATIO OF BOILER AT THE MINUTE
C      PLRH  C  PART LOAD RATIO OF CHILLER AT THE MINUTE
C      SFANTOT C  TOTAL SUPPLY FAN POWER CONSUMPTION (W)
C      TREF  I  REFERENCE TEMPERATURE IN CHILLER MODEL
C
C
C      HERMETIC RECIPROCATING LIQUID CHILLER MODEL DATA
C

       DELTA=0.005
       TREF=29.44
       CCAPN=50.
       EIR=0.9
       AC=0.088065
       BC=1.137742
       CC=-0.225806
C
C      CONVERT CHILLER NOMINAL CAPACITY FROM TONS TO WATTS
C
       CCAPN=CCAPN*3.517*1000

C
C      BOILDER MODEL DATA
C
       AB=0.
       BB=1.368
       CB=-0.368
       EFFB=0.8
       BBN=129000.
C
C      TOTAL ENERGY CONSUMPTION FOR CHILLER, BOILER, AND FAN
C
       EGYCTOT=0.
       EGYBTOT=0.
       EGYFTOT=0.
C
C      QHTOT = QRH + QPH
C
C      START CALCULATION OF ENERGY CONSUMPTION OF HVAC EQUIPMENT
C
C

10     READ(567,*,END=999)H,QCC,QHTOT,SFANPOW,TAIR

       EGYC=0.
       EGYB=0.
       EGYF=0.

       IF (QCC.GT.QHTOT)THEN
```

```
C
C     CHILLER IS OPERATED
C
      CCAP=CCAPN*(1.+DELTA*(TREF-TAIR))
      PLRH=QCC/CCAP
      PEL=EIR*CCAPN*(AC+BC*PLRH+CC*PLRH*PLRH)

C     WATT.MIN IS CONVERTED TO KWH BY DIVIDING BY (60*1000)

      EGYC=PEL/60000.
      EGYCTOT=EGYCTOT+EGYC

      ELSE
C
C     BOILER IS OPERATED
C
      PLR=QHTOT/BBN
      PEL=BBN/EFFB*(AB+BB*PLR-CB*PLR*PLR)

C     WATT.MIN IS CONVERTED TO KWH BY DIVIDING BY (60*1000)

      EGYB=PEL/60000.
      EGYBTOT=EGYBTOT+EGYB

      ENDIF

C
C     CALCULATION OF ENERGY CONSUMPTION OF SUPPLY FAN
C

C     WATT.HR IS CONVERTED TO KWH BY DIVIDING BY (1000)

      EGYF=SFANPOW/1000.
      EGYFTOT=EGYFTOT+EGYF


      PRINT*,H,EGYC,EGYB,EGYF



      GOTO 10
999   PRINT*,'TOTAL ENERGY CONSUMPTION FOR CHILLER (KWH) =',
EGYCTOT
      PRINT*,'TOTAL ENERGY CONSUMPTION FOR BOILER  (KWH) =',
EGYBTOT
      PRINT*,'TOTAL ENERGY CONSUMPTION FOR FAN     (KWH) =',
EGYFTOT
      EGYTOT=EGYCTOT+EGYBTOT+EGYFTOT
      PRINT*,'TOTAL ENERGY CONSUMPTION             (KWH) =',
EGYTOT

      STOP
      END
```

Appendix E

DOE2e Building Input File

```
$INPUT LOADS .
$=============================================================
$ LOADS, LOADS, LOADS, LOADS

$=============================================================

$=============================================================
$ MATRIAL, LAYERS, CONSTRUCTION  MATERIAL, LAYERS, CONSTRUCTION
$=============================================================
$FUNDAMENTAL BUILDING MATERIALS

EXT-WALL1=LAYERS          MATERIAL=(CC26,IN45,CB46,GP04) I-F-R=0.68
..
$PRECAST CONCRETE PARAPET
$8"CONC 2" SPARYED ON INSULATION, 5 5/8 CMU USED 6"
$ L. WEIGHT HOLLOW CONCRETE BLOCK

FLOOR-WALL1=LAYERS          MATERIAL=(IN45,CC26) I-F-R=0.68    ..
$PRECAST CONCRETE PARAPET
$8"CONC 2" SPARYED ON INSULATION, 5 5/8 CMU USED 6"
$ L. WEIGHT HOLLOW CONCRETE BLOCK

ROOF-1=LAYERS    MATERIAL=(RG02,IN46,CC24)   ..
$1"ROOF GRAVEL,2"RIGID INSULATION (USED 3"), 4"CONCRETE

$CONSTRUCTION_WALLS/ROOFS

WALL1=CONSTRUCTION        LAYERS=EXT-WALL1  ABS=0.65    RO=3   ..

FLOOR1=CONSTRUCTION       LAYERS=FLOOR-WALL1   ..

RF-1= CONSTRUCTION        LAYERS=ROOF-1  ABS=0.29  RO=1   ..

GLASS1=GLASS-TYPE         G-T-C=2201
$INSULATED,TINTED (DBL TINTED BRONZE IG)
     FRAME-CONDUCTANCE=3.037   ..  $THERMALLY UNBROKEN ALUMINUM

INTER1=CONSTRUCTION  U=2.5   ..


$=============================================================
$                         SCHEDULE
$=============================================================


P-SCH=SCHEDULE THRU DEC 31
             (WD) (1,7) (0) (8) (0.3) (9,12) (1) (13) (0.5) (14,17) (1)
                  (18) (0.3) (19,24) (0)
             (WEH) (1,7) (0) (8) (0.1) (9,12) (0.5) (13) (0.1) (14,17) (0.5)
                   (18) (0.1) (19,24) (0)  ..
```

```
L-SCH=SCHEDULE THRU DEC 31
             (WD)(1,5)(0)(6,18)(1)(19,24)(0)
             (WEH)(1,5)(0)(6,18)(.6)(19,24)(0) ..
E-SCH=SCHEDULE THRU DEC 31
             (WD)(1,5)(0)(6,18)(1)(19,24)(0)
             (WEH)(1,5)(0)(6,18)(0.6)(19,24)(0) ..



$===================================================================
$                  SPACE CONDITIONS

$===================================================================

EXT-CON=S-C
      P-SCH=P-SCH
      NUMBER-OF-PEOPLE=60
      PEOPLE-HG-LAT=200
      PEOPLE-HG-SENS=250
      L-SCH=L-SCH
      LIGHTING-TYPE=REC-FLUOR-NV
      LIGHTING-KW=5
      E-SCH=E-SCH
      EQUIPMENT-KW=50
      ..

INT-CON=S-C
      P-SCH=P-SCH
      NUMBER-OF-PEOPLE=12
      PEOPLE-HG-LAT=200
      PEOPLE-HG-SENS=250
      L-SCH=L-SCH
      LIGHTING-TYPE=REC-FLUOR-NV
      LIGHTING-KW=5
      E-SCH=E-SCH
      EQUIPMENT-KW=25
      ..

$===================================================================
$                         SPACES
$===================================================================

EXT-ZONE=SPACE
        AREA=17900
        VOLUME=179000       $HEIGHT=10 ASSUMED
        X=0
        Y=0
        Z=0
        AZIMUTH=0
        S-C=EXT-CON
        ..

EXT-S-WALL  =EXTERIOR-WALL
            H=10 W=200    $SOUTH
            CONS=WALL1
            X=0
            Y=0
```

```
              Z=0
              AZ=180
              TILT=90  ..

S-WINDOW      =WINDOW
              GLASS-TYPE=GLASS1
              H=4 W=180
              X=10
              Y=4 ..

EXT-N-WALL    =EXTERIOR-WALL
              H=10 W=200     $NORTH
              CONS=WALL1
              X=200
              Y=100
              Z=0
              AZ=0
              TILT=90  ..

N-WINDOW      =WINDOW
              GLASS-TYPE=GLASS1
              H=4 W=180
              X=10
              Y=4 ..

EXT-E-WALL    =EXTERIOR-WALL
              H=10 W=100     $EAST
              CONS=WALL1
              X=200
              Y=0
              Z=0
              AZ=90
              TILT=90  ..

E-WINDOW      =WINDOW
              GLASS-TYPE=GLASS1
              H=4 W=80
              X=10
              Y=4 ..

EXT-W-WALL    =EXTERIOR-WALL
              H=10 W=100     $WEST
              CONS=WALL1
              X=0
              Y=100
              Z=0
              AZ=270
              TILT=90  ..

W-WINDOW      =WINDOW
              GLASS-TYPE=GLASS1
              H=4 W=80
              X=10
              Y=4 ..


ROOF-EXT1     =ROOF
```

```
            H=100 W=65
            CONS=RF-1
            X=0
            Y=0
            Z=10
            AZ=180
            TILT=0   ..

ROOF-EXT2   =ROOF
            H=100 W=65
            CONS=RF-1
            X=135
            Y=0
            Z=10
            AZ=180
            TILT=0   ..

ROOF-EXT3   =ROOF
            H=35 W=70
            CONS=RF-1
            X=65
            Y=0
            Z=10
            AZ=180
            TILT=0   ..

ROOF-EXT4   =ROOF
            H=35 W=70
            CONS=RF-1
            X=65
            Y=65
            Z=10
            AZ=180
            TILT=0   ..

FLOOR-EXT1  =UNDERGROUND-WALL
            AREA=17900
            CONSTRUCTION=FLOOR1   ..

INT-ZONE=SPACE
        AREA=2100
        VOLUME=21000        $HEIGHT=10 ASSUMED
        X=65
        Y=35
        Z=0
        AZIMUTH=0
        S-C=INT-CON
        ..

INT-W-WALL  =INTERIOR-WALL
            H=10
            W=30
            X=0
            Y=30
            Z=0
            AZ=270
            TILT=90
```

```
              CONSTRUCTION=INTER1
              NEXT-TO = EXT-ZONE   ..

INT-N-WALL    =INTERIOR-WALL
              H=10
              W=70
              X=70
              Y=30
              Z=0
              AZ=0
              TILT=90
              CONSTRUCTION=INTER1
              NEXT-TO = EXT-ZONE   ..

INT-S-WALL    =INTERIOR-WALL
              H=10
              W=70
              X=0
              Y=0
              Z=0
              AZ=180
              TILT=90
              CONSTRUCTION=INTER1
              NEXT-TO = EXT-ZONE   ..

INT-E-WALL    =INTERIOR-WALL
              H=10
              W=30
              X=70
              Y=0
              Z=0
              AZ=90
              TILT=90
              CONSTRUCTION=INTER1
              NEXT-TO = EXT-ZONE   ..

ROOF-INT1     =ROOF
              H=30 W=70
              CONS=RF-1
              X=0
              Y=0
              Z=10
              AZ=180
              TILT=0   ..

FLOOR-INT1    =UNDERGROUND-WALL
              AREA=2100
              CONSTRUCTION=FLOOR1   ..



LOADS-REPORT
      VERIFICATION = (LV-A)
      SUMMARY = (ALL-SUMMARY)
      REPORT-FREQUENCY = HOURLY
      HOURLY-DATA-SAVE = FORMATTED   ..
```

```
ALL-ON = SCHEDULE
      THRU DEC 31 (ALL)(1,24)(1) ..

EXT-Z-LRB = REPORT-BLOCK
       VARIABLE-TYPE = EXT-ZONE
       VARIABLE-LIST =(42,43)    ..

INT-Z-LRB = REPORT-BLOCK
       VARIABLE-TYPE = INT-ZONE
       VARIABLE-LIST =(42,43)    ..

LOAD-DAT = HOURLY-REPORT
      REPORT-SCHEDULE = ALL-ON
      REPORT-BLOCK = (EXT-Z-LRB,INT-Z-LRB)    ..


END  ..
COMPUTE LOADS  ..
SAVE-FILES  ..

##INCLUDE C:\DOE21E\INPUT\WACH-SYS.INP


$===============================================================
$ SYSTEMS
$===============================================================

INPUT SYSTEMS ..

TITLE LINE-1 *MOHSIN*
      LINE-2 *AL-ALAWI*  ..
$===============================================================
$               DEFAULT
$===============================================================


$===============================================================
$                         SCHEDULE
$===============================================================

HEATING-SET=SCHEDULE THRU DEC 31
(ALL)(1,5)(71)(6,19)(71)(20,24)(71)  ..


COOLING-SET=SCHEDULE THRU DEC 31
(ALL)(1,5)(71)(6,19)(71)(20,24)(71)  ..

FAN1=SCHEDULE THRU DEC 31 (ALL)(1,24)(1)  ..

HEAT1=SCHEDULE THRU DEC 31 (ALL) (1,24)(1) ..

CHILL1=SCHEDULE THRU DEC 31 (ALL) (1,24)(1)  ..

$===============================================================
$                         ZONES
$===============================================================
```

```
EXT-ZONE=ZONE
        H-T-SCH=HEATING-SET
        C-T-SCH=COOLING-SET
        THERMOSTAT-TYPE=REVERSE-ACTION
        $OA-CFM/PER=20
        ..

INT-ZONE=ZONE
        H-T-SCH=HEATING-SET
        C-T-SCH=COOLING-SET
        THERMOSTAT-TYPE=REVERSE-ACTION
        $OA-CFM/PER=20
        ..

$================SYSTEMS====================$


VAV1   =SYSTEM

             S-TYPE=VAVS
             MAX-S-T=85
             MIN-S-T=55
             MAX-OA-FRACTION=0.45
             MIN-OUTSIDE-AIR=0.45
             MIN-CFM-RATIO=0.05
             MIN-FAN-RATIO=0.1
             OA-CONTROL=FIXED
             FAN-CONTROL=SPEED
             REHEAT-DELTA-T=30

             ZONE-NAMES=(EXT-ZONE,INT-ZONE)
             H-SCH=HEAT1
             C-SCH=CHILL1
             COOL-CONTROL=CONSTANT
             COOL-SET-T=55
             F-SCH=FAN1

             HEAT-SOURCE=ELECTRIC
             ZONE-HEAT-SOURCE=ELECTRIC
             PREHEAT-SOURCE=ELECTRIC
             BASEBOARD-SOURCE=ELECTRIC
             ..


PA1=PLANT-ASSIGNMENT
      S-N = (VAV1) ..

SYSTEMS-REPORT
      VERIFICATION = (SV-A)
      SUMMARY = (ALL-SUMMARY)
      REPORT-FREQUENCY = HOURLY
      HOURLY-DATA-SAVE = FORMATTED  ..

ALL-ON = SCHEDULE
      THRU DEC 31 (ALL)(1,24)(1) ..
```

```
EXT-Z-SRB = REPORT-BLOCK
       VARIABLE-TYPE = EXT-ZONE
       VARIABLE-LIST =(1,2,14,32)    ..

INT-Z-SRB = REPORT-BLOCK
       VARIABLE-TYPE = INT-ZONE
       VARIABLE-LIST =(1,2,14,32)    ..

LOAD-DAT = HOURLY-REPORT
      REPORT-SCHEDULE = ALL-ON
      REPORT-BLOCK = (EXT-Z-SRB,INT-Z-SRB)    ..


END  ..
COMPUTE SYSTEMS  ..
SAVE-FILES  ..
STOP  ..
```