Energy Cost and IAQ Performance of Ventilation Systems and Controls

Project Report # 1 Project Objective and Methodology

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INTRODUCTION

Purpose and Scope of this Report

In it's 1989 Report to Congress on Indoor Air Quality, the United States Environmental Protection Agency provided a preliminary assessment of the nature and magnitude of indoor air quality problems in the United States, the economic costs associated with indoor air pollution, and the types of controls and policies which can be used to improve the air quality in the nation's building stock. In that report, EPA estimated that the economic losses to the nation due to indoor air pollution was in the "tens of billions" of dollars per year, and suggested that because of the relative magnitude of operating costs, labor costs, and rental revenue in most buildings, it is possible that modest investments toward improved indoor air quality would generate substantial returns. Since that time, EPA has attempted to further define the costs and benefits to the building industry of instituting indoor air quality controls.

This project - *Energy Cost and IAQ Performance of Ventilation Systems and Controls* - is part of that effort. Adequate ventilation is a critical component of design and management practices needed for good indoor air quality. Yet, the energy required to run the ventilation system constitutes about half of a building's energy cost. Since energy efficiency can reduce operating costs and because the burning of fossil fuels is a major source of greenhouse gases, energy efficiency has become an important concern to the building industry and the promotion of efficient energy utilization has become a matter of public policy. It is important, therefore, to examine the relationship between energy use and indoor air quality performance of ventilation systems.

This project represents a substantial modeling effort whose purpose is to assess the compatibilities and trade-offs between energy, indoor air quality, and thermal comfort objectives in the design and operation of HVAC systems in commercial buildings, and to shed light on potential strategies which can simultaneously achieve superior performance on each objective.

This project seeks to examine three related fundamental issues:

1. How well can commonly used HVAC systems and controls be relied upon to satisfy generally accepted indoor air quality standards for HVAC systems when they are operated according to design specifications?

2. What is the energy cost associated with meeting ASHRAE indoor air quality performance standards for HVAC systems?

3. How much energy reduction would have to be sacrificed in order to maintain minimum acceptable indoor air quality performance of HVAC systems in the course of energy efficiency projects?

The outdoor air flow rates contained in ANSI/ASHRAE Standard 62-1999[1], *Ventilation for Acceptable Indoor Air Quality*, along with the temperature and humidity requirements of ANSI/ASHRAE Standard 55-1992, *Thermal Environmental Conditions for Human Occupancy* were used as the indoor air quality design and operational criteria for the HVAC system settings in this study. The outdoor air flow rates used were 20 cfm per occupant for office spaces, and 15 cfm per occupant for educational buildings and auditoriums as per ANSI/ASHRAE Standard 62-1999[2]. The flow rates were established for design occupancy conditions and were not assumed to vary as occupancy changed during the day. Space temperature set points were designed to maintain space temperatures between 70^0 F - 79^0 F and relative humidity levels not to exceed 60%, consistent with ANSI/ASHRAE Standard 55-1992[3]. With these design and operational settings, the actual outdoor air flows, space temperatures, and space relative humidity were then compared with these criteria to assess the indoor air quality performance of the system. When the design or operational set points were not maintained, operational changes were selectively undertaken to insure that the criteria were met so that the associated changes in energy cost could be examined.

While indoor air quality can arguably be controlled by different combinations of source control, ventilation control, and/or air cleaning technologies, no attempt was made in this project to study the potential for maintaining acceptable indoor air quality at reduced ventilation rates through the application of source control and air cleaning methods. In addition, while the impact of polluted outdoor air on the indoor environment is noted in discussions of outdoor air flow rates, no attempt was made to assess the implications of treating the outdoor air prior to entry into the building. In general, this project attempted to examine issues facing HVAC design and operational engineers during the most common applications of the indoor air quality and thermal comfort standards as prescribed by ASHRAE.

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[1] This project was initiated while ASHRAE Standard 1989 was in effect. However, since the outdoor air flow rates for both the 1989 and 1999 versions are the same, all references to ASHRAE Standard 62 in this report are stated as ASHRAE Standard 62-1999.

[2] The outdoor air flow rates specified in ASHRAE 62-1999 are designed to dilute indoor generated contaminants to acceptable levels where no significant indoor sources of pollution are present, and where the outdoor air quality meets applicable pollution standards. Thus, where significant indoor sources of pollution are present, these would have to be controlled. In addition, unacceptable concentrations of contaminants in the outdoor air would have to be removed prior to its entering occupied spaces. These issues were not specifically addressed in this modeling project.

[3] ASHRAE Standard 55-1992 describes several factors which affect thermal comfort, including air temperature, radiant temperature, humidity, air speed, temperature cycling and uniformity of temperature, when establishing criteria for thermal comfort. The modeling in this project addresses only the air temperature and relative humidity factors.

In addition, since outdoor air flow rates of 5 cfm per occupant were allowed by ASHRAE Standard 62-1981, energy costs for both 5 cfm per occupant (which were commonly used prior to 1989) as well as the above referenced 15 and 20 cfm per occupant, were estimated in order to determine the cost implications of raising the outdoor air flow rates from the previously allowed to the current ASHRAE outdoor air requirements.

This is a modeling study, subject to all the limitations and inadequacies inherent in using models to reflect real world conditions that are complex and considerably more varied than can be fully represented in a single study. Nevertheless, it is hoped that this project will make a useful contribution to understanding the relationships studied, so that together with other information, including field research results, professionals and practitioners who design and operate ventilation systems will be better able to save energy without sacrificing thermal comfort or outdoor air flow performance.

Seven reports, covering the following topics, describe the findings of this project:

- Project Report #1: Project objective and detailed description of the modeling methodology and database development
- Project Report #2: Assessment of energy and outdoor air flow rates in CV and VAV ventilation systems for large office buildings:
- Project Report #3: Assessment of the distribution of outdoor air and the control of thermal comfort in CV and VAV systems for large office buildings
- Project Report #4: Energy impacts of increasing outdoor air flow rates from 5 to 20 cfm per occupant in large office buildings
- Project Report #5: Peak load impacts of increasing outdoor air flow rates from 5 to 20 cfm per occupant in large office buildings
- Project Report #6: Potential problems in IAQ and energy performance of HVAC systems when outdoor air flow rates are increased from 5 to 15 cfm per occupant in auditoriums, education, and other buildings with very high occupant density
- Project Report #7: The energy cost of protecting indoor environmental quality during energy efficiency projects for office and education buildings

GENERAL METHODOLOGY AND LIMITATIONS

The process of investigating indoor air quality (IAQ) and energy use can be time-consuming and expensive. In order to streamline the process, this study employed a building simulation computer modeling procedure. The computer modeling approach enabled the investigation of multiple variations of building configurations and climate variations at a scale which would not otherwise be possible at considerably less cost than field study investigations.

The methodology used in this project has been to refine and adapt the DOE-2.1E building energy analysis computer program for the specific needs of this study, and to generate a detailed database on the energy use, indoor climate, and outdoor air flow rates of various ventilation systems and control strategies. Constant volume (CV) and variable air volume (VAV) systems in different buildings in three climate regions using four alternative outdoor air control strategies provided the basis for parametric variations in the database. The database generated by these simulations provide a rich body of information. Comparisons in energy performance, outdoor air performance, and thermal comfort performance allows the analyst to quantify the compatibilities and tradeoffs between performance objectives.

Any analysis, however thorough, is inevitably constrained by the state of the art and resource available. Several fundamental limitations to the analysis in this project must be recognized.

- The analysis is ultimately constrained by the inability of the model to reflect real world conditions. Problems with the model and how they were resolved is discussed below.

- While a large number of building parameters were used to capture the relevant variations in the building stock and their ventilation systems, as a whole, they can not be considered representative because of the exceptionally large variety building and ventilation system features which are currently available.

- Knowledge and understanding of the inventory and performance of building equipment in the current building stock is limited, so that the ability to model representative variations in actual equipment performance is also limited.

- The modeling assumed that all equipment functioned as it was intended to function. Poor design, poor operations, and malfunctioning equipment, which are not uncommon in existing buildings, could not be directly modeled.

- For comparison purposes, the same buildings were modeled for each climate and does not reflect climatic differences in building construction.

DOE-2 BUILDING SIMULATION MODEL

The building energy simulation model DOE-2.1E was used in this study to assess energy and indoor air quality performance of ventilation systems in large buildings. The initial development of the DOE-2 computer model was funded by the U.S. Department of Energy in the late 1970's. Improvements to the original model and the development of additional features have continued throughout the last fifteen years at a total cost of over 10 million dollars. The latest version of DOE-2 (i.e., version 2.1E), released in late 1993, was used in this project. DOE-2 is widely accepted as one of the most fully featured and sophisticated building energy analysis computer models in the world. The purpose of the DOE-2 computer program is to provide a means of investigating in detail the behavior of the many individual building components which affect energy use. More importantly, DOE-2 provides a means of assessing the combined or aggregate effect of these energy systems.

MODIFICATIONS MADE TO DOE-2.1E

The data quality checking procedures used in this project revealed many inherent weaknesses of the model in its application to the purposes of this project. As a result, modifications to the model were made. Problems which required specific modifications to DOE-2.1E were encountered in several areas: infiltration, HVAC equipment sizing, outdoor air controls, control strategies, exhaust systems, and heat recovery systems. Each of these is discussed below.

Infiltration

General Problem

Infiltration can account for a significant portion of energy use in buildings. Many forces create air movement in buildings-- mechanical building pressurization, horizontal wind effects, inter-zone air movement, and vertical stack effects (i.e., buoyancy). There have been numerous studies performed on residential infiltration, but relatively few studies on large commercial structures. Therefore, in commercial buildings, these factors are not well understood.

Due to the difficulty in modeling infiltration, sophisticated infiltration models are not available in the DOE-2.1E program. DOE-2.1E has three different approaches available for modeling infiltration (i.e., air-change, crack, and residential). The most appropriate infiltration model for commercial buildings is the air change method. In this method, the user must provide an appropriate value for the infiltration rate for each zone. DOE-2.1E assumes that this value is for a ten mile per hour wind speed. The infiltration rate is linearly adjusted each hour by DOE-2.1E for wind speeds different from ten miles per hour. Although the wind speed adjustment is reasonable, it leaves the resolution of other variables, most notably wind direction and HVAC system operation, unaccounted for.

Since the thermal loads in a perimeter zone are strongly affected by infiltration, a thermal zone oriented towards the dominant wind direction will be affected more than a thermal zone with another orientation. The modeling of infiltration will affect the outdoor air flow rates in each zone predicted by DOE-2.1E because supply air is dependent on thermal loads, and because outdoor air is a fractional component of the supply air provided to meet the thermal loads.

Resolution

The distribution of outdoor air to individual zones was the subject of considerable analysis for the office building. Accordingly, the DOE-2.1E infiltration algorithm was modified for application to the office building. No modification was undertaken for the education or the assembly building.

After discussing this problem with several DOE-2 modelers, and the developers of the DOE-2 program, the infiltration algorithm in DOE-2.1E was modified to allocate infiltration air to the perimeter zones in accordance with wind direction. The wind pressure exerted on the exterior shell was assumed to force air into that thermal zone, through the interior of the building, and out the other side. Thus, only the windward sides of the building was assumed to experience infiltration. The floor average infiltration must enter the building only through these windward zones. Since each perimeter zone in the office building comprises approximately one twelfth of the total floor area, the air change rate in this zone must be twelve times higher than the floor average value.

In the modification, infiltration is treated differentially depending on the wind direction relative to the surface of the building. If the wind direction at a given hour is perpendicular to a surface of the building, then infiltration is only allowed in the zone with that orientation. If instead, the wind hits the building at an angle, then the infiltration is equally distributed to the two windward zones with each receiving half the infiltration air. When the wind speed is less than 5 mph, it is assumed that the wind direction did not influence the distribution of the infiltration air, so that all perimeter zones received an equal portion of the infiltration air.

Another modification involves a change in the infiltration rate when the HVAC system is operating. When operating, the building is positively pressurized relative to the outside so that some reduction is warranted relative to the non-operating mode. In this modification, the infiltration rate doubles when the HVAC system is not operating (e.g. during the night).

These changes were incorporated into the DOE-2.1E program using a function[4] in the input file for each thermal zone. A simple version of these functions is provided in Appendix A. The revised infiltration model was designed to provide the same floor average infiltration rate as the original DOE-2.1E model. However, the infiltration is distributed to each of the thermal zones on the floor more realistically. Two example scenarios are provided in Exhibits 1 and 2 - a case with a North wind, and a case with a North-East wind, respectively.

In general, for the office building, the nominal (HVAC system not operating, wind speed is at 10 mph) <u>building</u> average infiltration rate was set at 0.5 ACH. This was reduced to 0.25 ACH during operating hours. Depending on specific wind speed and direction conditions, the infiltration rate into each zone was varied hourly. For the North wind example in Exhibit 1, when the wind speed is less than 5 mph, all the <u>perimeter</u> zones are modeled with an equal infiltration rate. This nominal rate is linearly adjusted based on the wind speed in that hour and cut in half during daytime operating hours. For wind speeds greater than 5 mph, only the North zone experiences infiltration - at a nominal rate of 6.0 ACH (3 ACH when HVAC is operating). In the example in Exhibit 2, where the wind direction is coming from the Northeast and is greater than 5 mph, the North and East zones each receive a nominal rate of 3.0 ACH (1.5 ACH when HVAC is operating).

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4 In the DOE-2.1E program, the user is able to write new computer code which will override the original DOE-2.1E code. This capability is called a "function" in DOE-2.1E. The function capability enables the DOE-2.1E user to develop a custom version of DOE-2.1E without having to change the source code and re-compiling the program.

HVAC Equipment Sizing

General Problem

The DOE-2.1E computer program was developed primarily as a design tool for new buildings and includes the capability to model many new and innovative equipment and systems. Generally, DOE-2.1E's built-in defaults tend toward new higher efficiency equipment and design strategies, and assume that the systems are intended to be sized and operated correctly. As a result, DOE-2.1E does not reliably model undersized or inefficient equipment such as may be found in some older buildings.

One of the purposes of this project was to address the feasibility of raising the outdoor air flow in existing buildings. In particular, one objective was to evaluate ventilation system performance when outdoor air flow levels were raised from 5 to 20 cfm per occupant in buildings which were designed for 5 cfm per occupant. In doing this, two problems related to HVAC equipment size were identified in the DOE-2.1E runs

- The energy consumed by under-sized systems is underestimated, and
- The auto-sizing algorithm often provides inadequate supply air to core zones that are served by VAV systems.

Estimating the Energy Used by Undersized Systems: The first modeling difficulty was related to DOE-2.1E's inability to model undersized central plant equipment properly. The problem was realized when after a certain run was completed, the results showed that the energy use by the undersized system was less than the energy used by the properly sized system. In analyzing the results carefully, it was determined that the system did not fully satisfy the thermal loads in all zones all of the time and that the undersized system was operating more efficiently than the properly sized system during each hour when it was not able to meet the thermal load.

This problem is related to DOE-2.1E's algorithms for modeling of equipment part-loading. DOE-2.1E calculates the energy use of each piece of equipment in a building based on its peak capacity and efficiency, as well as the degree of part-loading and the change in efficiency with part-loading. However, the algorithms for some of the equipment do not recognize that the loading of a piece of equipment cannot exceed its capacity. For example, if a chiller is overloaded because it is undersized, DOE-2.1E will model the chiller as increasingly more efficient as it is increasingly overloaded. This presents bizarre results in which the chiller uses less energy when it is overloaded than when it is properly sized.

Inadequate supply air to core zones: The DOE-2 program automatically calculates the amount of supply air that is required to satisfy the peak thermal load in each zone of a building. For VAV systems, this design flow rate is used to size the VAV boxes so that they provide sufficient supply air to maintain the thermostat set point (desired space temperature) at all times. A major problem encountered in the DOE-2.1E system sizing algorithms was that the design supply air flow rates for the core zone were inadequate, causing the core zone to overheat on peak or near peak cooling days. It remains unclear why this shortfall occurs.

Resolution

Because of the problems encountered with undersized systems, the auto sizing algorithm in DOE-2.1E was not used, even for runs at 5 cfm per occupant. Rather, a concerted effort was made to ensure that the HVAC equipment was adequately sized for all runs. Further, attempts to examine ventilation system behavior when existing buildings designed for 5 cfm of outdoor air per occupant were run at 20 cfm per occupant were abandoned. Instead, peak loads were examined to determine capacity restraints. For consistency in this project, all HVAC systems in all buildings were sized large enough to handle both 5 and 20 cfm of outdoor air per occupant.

To resolve the problems with overheating of the core zone, DOE-2.1E's default design supply air flows to the core zones were increased sufficiently to ensure that desired space temperatures were maintained during all occupied hours of the year in the core zones.

Outdoor Air Controls

General Problem

Two DOE-2.1E modeling issues related to outdoor air flow control were identified :

- DOE 2 does presumes that a constant outdoor air flow is always maintained for VAV systems at all load conditions, and
- The default settings for VAV box minimum settings, and for night time operation create problems.

Outdoor air control for VAV systems: If the outdoor air damper in a VAV system is maintained in a fixed position, the flow of outdoor air through the damper can, under some circumstances, to decrease as the fan slows down at low-load conditions.

DOE-2.1E does not provide the ability to model this type of outdoor air flow. Instead, in DOE-2.1E, it is always assumed that the outdoor air flow rate into a building is constant (at the required outdoor air flow rate) when the HVAC system is on, regardless of the part-loading of the supply air fan.

Default settings: In order to simplify the data input process, DOE-2.1E has numerous built-in defaults. Most of DOE-2.1E's defaults are very helpful. However, a general problem with defaults is that it is often unclear what values are being assumed by the program as the defaults. In this study, two of these defaults were identified as inappropriate for typical buildings. These defaults are for the following two parameters:

- VAV box minimum setting

- night operation of the outdoor air dampers

The minimum flow setting for the VAV boxes assumed in DOE-2.1E is the minimum outdoor air requirement for a zone relative to its peak supply air flow. This value is often as low as one or two percent of peak supply air flow. Accurate control at such a low setting is infeasible. Further, most buildings are designed with minimum flow settings of 20 to 50 percent. This problem was discovered when some runs were presenting exceptionally low supply air and outdoor air flows in perimeter zones during no load conditions, and were also presenting suspiciously low fan energy use during these conditions.

Another questionable default in DOE-2.1E is the treatment of the outdoor air damper at night when the HVAC system cycles on. Typically, most building operators only allow the outdoor air damper to open during the daytime when the building is occupied. It is unlikely that very many building operators would allow the outdoor air dampers to open on a cold winter night when the heat comes on to maintain the building at its night setback temperature. However, DOE-2.1E maintains an open outdoor air damper at night whenever the HVAC system cycles on. The problem was discovered when detailed hourly night time data was reviewed. This default can be easily over-ridden by the user, but it is not obvious to the user that this is DOE-2.1E's default outdoor air damper operating strategy.

Resolution

Another DOE-2.1E function was created in order to model an outdoor air control strategy in which the outdoor air is always a constant to the supply air (see description of ventilation system and outdoor air control strategies below). In this modified VAV system outdoor air strategy, the outdoor air flow rate varies continually as the supply air flow varies. The original VAV system with constant outdoor air flow (COA) is called VAV/COA in this project. The new VAV system with a fixed outdoor air fraction is called VAV/FOAF. A simple version of the DOE-2.1E function that modifies the outdoor air control for the VAV/FOAF system is provided in Appendix B of this report.

After consulting with many design professionals, the minimum VAV box setting for all runs was set to 30 percent. Opinions were varied as to the proper setting (between 20 and 50 percent), and there was no clearly defined method for calculating the minimum flow setting. An analysis of the effect of various minimum flow settings for VAV boxes is presented in Report #2 of this project.

Control Strategies

General Problem

There are numerous possible control strategies which can be implemented for any energy-consuming equipment in a building. The strategies used in a building are dependent on the sophistication of the controls and the building operator's understanding of their effects. Some buildings are run under a very simple automated control logic. In other buildings, the control logic is so complex that a centralized computer controller is used.

Most controls strategies have three operating modes per day: (1) occupied or day mode, (2) unoccupied or night mode, and (3) morning startup mode. Weekday and week end control strategies are usually different. Further, each of these strategies may be varied in each of the four seasons.

Demand and energy use can be significantly affected by control strategies. For example, a building which is operated using a single control mode throughout each 24 hour day will use almost twice as much energy as a building which is "shut-down" at night. This significant variation in energy use has nothing to do with the efficiency of the equipment.

DOE-2.1E offers great flexibility in modeling control strategies, but with little guidance. Thus, the user's ability to model complex controls strategies is limited by his understanding of DOE-2.1E controls modeling algorithms. Also, some common control strategies (e.g., return air reset for supply air) cannot be modeled using DOE-2.1E.

Resolution

A series of parametric studies were performed to assess the effects of various common control strategies on energy use and outdoor air flow rates. These studies showed that simulation input files must be carefully prepared, reviewed, and tested. To identify "common" operating and control strategies, many building operators, controls experts, and simulation experts were contacted. From these discussions, some important lessons were learned including:

- ! The more energy efficient the building is, the more complex the controls strategies are likely to be.
- ! DOE-2.1E can model most controls strategies, but it may take twenty or more inputs to model a building's controls effectively.

For this study, the most commonly used control strategies were identified for each type of equipment modeled. Detailed input models were carefully tested and refined prior to generation of the results of this study.

Exhaust Systems

General Problem

DOE-2.1E is very limited in its ability to model operational strategies for exhaust systems. Exhaust systems usually require the use of make-up air systems to provide outdoor air to replace the exhausted air. In the extremes of winter and summer, it is energy efficient to use exhaust only when needed.

In the DOE-2.1E computer program, an exhaust rate can be specified. If exhaust is specified, then it operates during every hour that the HVAC system operates, including night-cycling. If it is not specified, then DOE-2.1E models the building without exhaust. Thus, exhaust cannot be controlled to operate only in the daytime and off at night, as is usually the case.

Further, whenever the HVAC system is on, the exhaust system must be on, and consequently the outdoor air damper open to provide make-up air for the exhaust system. There is no way provided to override this operating strategy.

Resolution

For this project, a DOE-2.1E "function" was written and inserted into the DOE-2.1E input file to shut-off the exhaust and the outdoor air during hours when the buildings was unoccupied. A separate function was required for each type of outdoor air damper control strategy. A simple version of these functions is provided in Appendix C.

Heat Recovery Systems

General Problem

In buildings with large occupant densities and/or special use spaces (i.e., labs, operating rooms), large amounts of outdoor ventilation air must be brought into the building - even during the peak heat of summer and the extreme cold of winter. Also, in some climates, the outdoor air can be very humid. This outdoor ventilation air can cause a large burden on the central heating or cooling coils. Heat recovery between the outdoor air inlet duct and the relief air exhaust duct can significantly reduce this energy penalty.

An important objective of this project was to simulate various energy conservation strategies, quantify the energy savings, and examine their outdoor air flow and thermal comfort consequences. Attempts were therefore made to model heat recovery from the relief air in the winter, and heat removal from the outdoor air stream in the summer. DOE-2.1E cannot model both of these types of heat recovery.

Resolution

In this project, latent and sensible heat recovery were separately modeled. The data were then split by heating and cooling season. The heating season data with heat recovery was then combined with the cooling season data with latent recovery to approximate a single simulation in which both systems were employed.

Results of this exercise were not satisfactory and were ultimately abandoned. Additional work is needed to develop a strategy to add these capabilities to the DOE-2.1E program. Heat recovery technologies are expected to be a very effective means of addressing the demand and energy burdens caused by outdoor ventilation air.

Summary of Problems Identified and How They Were Resolved

The problems and resolutions are summarized below.

Type of Problem	DOE-2.1E Limitation	Resolution
Infiltration	Cannot model wind direction dependency	See Function in Appendix A
Equipment Sizing	Does not model undersized equipment properly Does not size supply air for core zone properly for VAV systems	Checked to ensure that equipment is not under-sized Oversized supply air flow to core zone by 20 to 30%
Outdoor Air Flow	Cannot model fixed position outdoor air dampers Does not set VAV box minimum flow settings appropriately	See Function in Appendix B Set MIN-CFM-RATIO = 0.30
Controls Strategies	Most control strategies are very complex to model, and some common strategies cannot be modeled.	Used most common operating strategy for each type of equipment modeled
Exhaust System	Cannot schedule On/Off	See Function in Appendix C
Energy Recovery	Cannot model both latent (cooling season) and sensible (heating season) heat recovery in the same simulation	Energy Recovery was not modeled

DESCRIPTION OF BUILDINGS MODELED

An office building, education building, and an assembly building were modeled in this project. There were 14 office building configurations (Buildings A-N) that were modeled. The base office building (Building A) is representative of a typical large office building in the U. S. The assumptions about its characteristics were based on data obtained from the U.S. DOE’s CVECS database. To examine the effects of high occupant density on indoor air and energy parameters, one education building and one assembly building were also modeled. The general characteristics of the base office building (Building A), the education building and the assembly building are outlined in Exhibit 3 . Building characteristics showing the variations in building parameters for Office Buildings B-N in comparison to Office Building A are presented in Exhibit 4. Operating schedules for the office, education, and assembly buildings are presented in Exhibit 5. Occasionally, modifications to these building parameters were modeled to address a specific issue. These modifications are described in the individual reports where these issues are addressed.

DESCRIPTION OF VENTILATION SYSTEMS AND OUTDOOR AIR CONTROL STRATEGIES MODELED

HVAC System Components

Ventilation systems have two main functions: (1) pollutant dilution, and (2) thermal control. Because both of these functions require delivery of air to the occupied spaces of the building, they are typically combined into a single air distribution system.

Central HVAC systems for large commercial buildings consist of three interrelated components: air handlers, air distribution systems, and a central heating and cooling plant. An air handler is designed to mix fresh outdoor air with recirculated air, in some proportion, and condition the resulting mixed air stream to temperature and sometimes humidity levels to satisfy the thermal comfort requirements of the occupied spaces.

Once the mixed air is conditioned, it is distributed to occupied spaces through ducts and plenums which comprise the air distribution system. The central heating and cooling plant supplies all necessary heating and cooling energy to the air handlers.

In this project, all HVAC systems are equipped with a central gas- fired boiler for heating and DHW service, and a central chiller and cooling tower. The HVAC equipment is always sized to meet the design loads in each climate.

The air handler for each system serves four perimeter zones representing each compass direction and a core zone. The dimensions and relative area of the perimeter and core zones for each building type is presented in Exhibit 6.

Air Flow Control Strategies

Two types of air handling systems -a constant volume (CV) and a variable volume (VAV) system were modeled. The CV system is a dual duct system with temperature reset capability, and was included to represent many older systems currently in use today. The VAV system is a single duct system with reheat coils at each zone. The configuration for the CV and VAV systems are presented in Exhibits 7 and 8.

<u>Constant Volume (CV) Systems</u>

When a CV system is designed, the volume of supply air is established to satisfy the design cooling requirement (maximum cooling load) of the system. Once established, the supply air volume remains constant. The daily and seasonally varying cooling and heating loads in each zone are satisfied by varying the temperature of the supply air which is delivered to the zone.

A dual duct CV system, which is the most common, is modeled in this project (see Exhibit 7). In this system, two main ducts are used to distribute a constant volume of supply air to each zone. One duct is maintained at a relatively hot temperature (e.g. 110°F) to provide heating as needed, and the other is maintained at a relatively cool temperature (e.g. 55°F) to provide cooling as needed. The hot and cold air streams are proportionally mixed at each zone to a temperature that will meet the heating or cooling load of that zone. In addition, the CV system modeled has a temperature reset capability to improve its energy efficiency. The temperatures of the hot and cold air streams can be reset if the total building heating or cooling loads are small. For example, in the Spring and Fall the hot air duct temperature may be reduced from 110°F to 78°F if no heating is called for in any zone.

Variable Air Volume (VAV) Systems

In the single duct VAV system (see Exhibit 8), the temperature of the supply air at the air handler is held constant, while the volume of supply air is varied in response to daily and seasonal variations in cooling and heating loads. The supply air delivered to a zone is thermostatically controlled by a VAV box serving that zone, and the volume of air delivered is dependent on the cooling and heating requirements of the zone. Heating is provided at each zone on an as-needed basis by a reheat coil in the VAV box. During off-peak conditions, most zones are operated at reduced supply air flow lowering the system supply air flow. The air handler must constantly adjust the total supply air flow provided in order to meet the combined needs of the individual zones.

Outdoor Air Control Strategies

Three general types of outdoor air control strategies are examined in this project. (1) fixed outdoor air fraction (FOAF), (2) constant outdoor airflow (COA), and (3) air-side economizers (ECON). These three outdoor air strategies are briefly introduced below. The actual quantity of outdoor air introduced into a building depends on both the outdoor air control strategy and the type of air handler (CV or VAV).

> Fixed Outdoor Air Fraction(FOAF): This strategy maintains a constant outdoor air fraction. The strategy might, for example, employ a fixed outdoor air damper which is set to introduce a pre-determined quantity of outdoor air to meet the outdoor air requirements of the building at the design cooling load[5]. Once the damper setting is

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[5] A fixed outdoor air damper is usually locked into a position or "setting" where it is sometimes expected to provide a given fraction of outdoor air. This setting (percent outdoor air) is often mistakenly used as a substitute for outdoor air per occupant. That is, as a rule of thumb, 5% outdoor air is often interpreted to correspond to 5 cfm per occupant, while 20% is used for 20 cfm per occupant etc. The accuracy of this rule of thumb depends on the total supply air flow, the occupant density, and the airflow per occupant which is desired. In general, this rule of thumb can be grossly inaccurate.

fixed, it is assumed to maintain a constant outdoor air fraction even if supply air quantities change[6].

In CV systems, where supply air quantities are constant, this strategy maintains a constant outdoor air flow rate and is equivalent to the COA strategy (see below). When VAV systems were introduced as being more energy efficient, they often employed this same strategy. However, in a VAV system, the total supply air flow in a VAV system varies in response to varying thermal loads. It is expected that, in many applications, a fixed position outdoor air damper will approximately maintain a constant outdoor air fraction, where the outdoor air flow rate into the building would increase or decrease approximately in proportion to the supply air flow (Cohen 1994; Janu 1995; and Solberg 1990).
In this project, the VAV(FOAF) system achieves the outdoor air requirements of the occupied spaces during design (or peak) cooling load conditions. This is common practice among operating engineers. At off-peak conditions, as the total supply air is reduced, the outdoor air flow is reduced proportionally. Therefore, except at the rare circumstance when the building is at or close to design cooling load, the occupied spaces will always receive less than the intended quantities of outdoor air.

Constant Outdoor Air Flow (COA): Constant outdoor air flow strategies maintain a constant quantity of outdoor air to the air handler under all operating conditions. In CV systems, this is accomplished with a fixed damper and is equivalent to the FOAF strategy. In VAV systems, the strategy will employ some mechanism, such as a modulating outdoor air damper which alters the outdoor air fraction as the supply air quantities are varied in response to changing thermal loads. Specific control mechanics which would achieve a VAV (COA) have been addressed by other authors, (Haines 1986, Levenhagen 1992, Solberg 1990) but are not addressed in this modeling project. The outdoor air fraction is at its maximum when the supply air flow is at its minimum and visa versa. However, the outdoor air quantity at the air handler remains the same under all operating conditions.

Air-Side Economizers (ECON): Air-side economizers are a commonly used energy efficiency measure in CV and VAV systems. Economizers may be based on

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6 The damper mechanisms described here are used as examples only. It is recognized that the damper position is not always a reliable indicator of the outdoor air fraction, and it is recognized that different building and HVAC configurations can greatly complicate the determination of the actual quantity of outdoor air that would enter a building with a fixed outdoor damper position. The purpose of this project is to assess implications of any design which approximately achieves the outcome described by each strategy. The specific controls and circumstances that would necessarily create such an outcome is not the subject of this modeling study.

temperature or enthalpy (i.e. total heat content of the air considering both temperature and humidity level). The three basic systems (CV-FOAF, VAV-FOAF, and VAV-COA) can accommodate either temperature or enthalpy economizers. The basic system determines the mode of operation when the economizer is in the off position.

Temperature economizers will use cold outdoor air to meet the cooling needs of occupied spaces whenever possible. This "free cooling" effect would otherwise be provided by the mechanical cooling system. Thus, economizers can significantly reduce the cooling energy requirements while increasing the outdoor air flow rates of an HVAC system. Because large buildings require cooling year round, it is usually cost-effective to use economizers whenever the outdoor air is colder than the recirculated air stream in the air-handler. However, to avoid indoor humidity problems when outdoor humidity levels are high, temperature economizers are often switched off when the outdoor temperature exceeds some predetermined set point. In this project, this set-point is 65°F. At outdoor air temperatures greater than 65°F, the temperature economizer is in the ?off ?position and the outdoor air damper position is reset to the prevailing outdoor airflow setting[7]. At temperatures less than 65°F, the temperature economizer causes the outdoor air damper to open sufficiently to provide a mixed air temperature of approximately 55°F. Thus, in mild Spring and Fall weather, the economizer causes the building to operate with as much as 100 percent outdoor air. As the outdoor air temperature becomes increasingly colder, the outdoor air damper position is increasingly reduced by the economizer until the outdoor air flow position of the basic strategy (FOAF or COA) is reached. At the outdoor air temperature when this occurs and at all colder temperatures, the basic outdoor air flow strategy is maintained (i.e. the economizer is off).

Enthalpy economizers function similarly to temperature economizers, except that the enthalpy of the outdoor air and the return air (rather than the temperature alone) is compared to determine the outdoor air fraction that will minimize energy use. However, because the control parameter is enthalpy, no automatic shut-off set-point is used. Whereas a temperature controlled economizer may allow the economizer to operate on a mild day even when the humidity level is high, an enthalpy-controlled economizer would not. Contrastingly, on a warm dry day, the enthalpy-controlled economizer may allow the economizer to operate as long as the return air is

[7] The set-point is climate dependent. Thus, one might use a higher temperature set-point for a very dry climate.

warmer than the outdoor air and the outdoor air humidity is acceptably low, while the temperature-controlled economizer would automatically shut off.

Since the supply air volume in CV systems is constant, the CV(FOAF) and the CV(COA) are equivalent and are referred to as CV(FOAF) only. There are thus three basic systems -CF(FOAF), VAV(FOAF), and VAV(COA). Each system may be modeled without an economizer, or with either a temperature or enthalpy economizer. Thus, each CV system may be modeled with three, and each VAV system may be modeled with six optional outdoor air control strategies. There are therefore a total of nine possible HVAC system types available for modeling in each building .

CLIMATE REGIONS

Three climate regions are used to represent the range of weather conditions in the United States. Minneapolis weather data are used to represent cold (heating dominated) climates, Washington D.C. weather data are used to represent temperate (mixed cooling/heating)climates, and Miami weather data are used to represent hot/humid (cooling dominated) climates. Most analysis is conducted in each of these three climates.

DATABASES FOR BUILDING CONFIGURATIONS

DOE-2.1E output data are assembled for each operating hour during the year. The data are then post processed by the SAS statistical program. Post processed data include the supply air flow, outdoor air flow, outdoor and indoor temperature and relative humidity, as well as energy use and loads for the heating, cooling, and distribution system.

Indoor air quality is represented by the supply and outdoor airflows, as well as the indoor temperature and humidity data. The airflow and temperature data are provided at the zone level. Since indoor humidity cannot be derived from DOE-2.1E at the zone level, relative humidity in the return air stream is used to represent the average humidity conditions in the occupied spaces of the building. The energy data in each database is converted to energy costs using alternative energy price assumptions.

The analysis and findings are based upon the results from a large set of DOE-2.1E simulations comprising data for the 14 office buildings and for the education and assembly buildings.

The two databases are described below.

Office Buildings Database

This database is derived from over 600 DOE-2.1E simulations which are defined as follows:

Fourteen Office Building Variations

This project employed a base building (Building A) with thirteen variations of building and equipment parameters (Buildings B-N).

Three Climate Regions

All 14 buildings were modeled in each of three cities representing three climate regions: Mineapolis (cold), Washington, D.C. (temperate), and Miami (hot, humid).

Six HVAC System/Outdoor Air Control Combinations

Each building in each climate is modeled with both the CV and the VAV system.

Supply air is made up of some portion which is outdoor air and some portion which is recirculated air. This proportional split is dependent on both the type of system (CV or VAV) and the outdoor air control strategy. Of the nine possible combinations described above, six are used throughout the office buildings analysis: CV(FOAF), CV(FOAF)(ECONt), VAV(FOAF), VAV(FOAF)(ECONt), VAV(COA), and VAV(COA)(ECONt).

Two Outdoor Air Flow Settings

Each outdoor air control strategy is modeled at a setting which delivers 5 cfm of outdoor air per occupant and 20 cfm of outdoor air per occupant at the design cooling load.

Database Composition

Fourteen buildings in three climates, each with six HVAC system/outdoor air control combinations, modeled at both 5 and 20 cfm of outdoor air per occupant yielded 504 simulations. In addition, 31 special simulations were provided to further examine the VAV(FOAF) and the VAV(FOAF)(ECONt) system on the base building (Building A) in all three climates. These included outdoor air design settings of 30 and 45 cfm per occupant, seasonal outdoor air reset capability, and alterations in the VAV box settings. An additional 66 simulations were performed to examine the impact of energy saving measures, including enthalpy economizers, on energy reduction and IAQ issues.

Education and Assembly Building Database

The education and assembly building database consists of output from over 100 DOE-2.1E simulations which are described below.

One prototype education building, and one prototype auditorium building were modeled in each of the three designated climates. Four ventilation system configurations - CV(FOAF) and a VAV(COA) system, each configured with and without a temperature economizer - were modeled..

Each building and ventilation system configuration was modeled at a design setting of 5 and 15 cfm of outdoor air per occupant. Additional simulations were therefore provided in which the VAV box minimum settings for each zone were adjusted in various ways to insure a minimum of 15 cfm per occupant, and in which humidity was controlled not to exceed 60%. A variety of energy conservation measures were also modeled for the education building.

UTILITY PRICE ASSUMPTIONS

Energy use is translated into energy cost using assumed utility rate structures. The utility rate structures used in this study are not meant to represent specific utilities . Rather they were established to represent a reasonable range of rate structures and to provide an opportunity to test the sensitivity of energy cost to relative prices of gas and electricity and electricity demand charges. The Base rate structure which used throughout this study was derived from commercial rates obtained from a survey conducted in 1994 of major electric and natural gas utilities. The average utility rates were based on average effective usage and demand rates for the 17 major cities. Four optional rate structures were designed to systematically vary the relative price of gas and electricity in order to provide a reasonable range of prices for sensitivity analysis.

Exhibit 9 shows the rate structures modeled in this study. Exhibit 10 shows the 17 utility rate structures from the survey, along with the minima, maxima, averages, and standard deviations of the distributions. The Average Electric Rates and Average Gas Rates in Exhibit 9 were determined using a weighted average of building fuel-specific energy charges applied to Office "A" in this study, including energy taxes. Average Electric Demand was determined using a weighted average building electric demand charge for Office "A", based on average monthly peak demand. The high and low rates modeled in the optional rate structures were derived by adding and subtracting one standard deviation of the rate distribution for the 17 utility rates.

In addition, Exhibit 10 shows Ratchet Clauses for the 17 utilities surveyed. A ratchet clause determines the minimum demand charge per kW demand. It is the higher of the monthly peak or a percentage of the highest monthly peak over the previous 11 months. Since only four of the utilities surveyed had ratchet clauses, and none of these were 100%, the rate structures used in this study do not use a ratchet clause.

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Exhibit 1: Infiltration Modeling Scheme for Scenario with North Wind

Zone Affected	Infiltration Rate (ACH)*		
	Daytime Operation (HVAC System On)		Night Operation (HVAC System Off)
	Wind Speed < 5 mph	Wind Speed > 5 mph	
Core	0	0	0
North	0.75	3.0	6.0
East	0.75	0	0
South	0.75	0	0
West	0.75	0	0

* The nominal infiltration rate--the rate for a 10 mph wind with HVAC system off--is 0.5ACH. The actual hourly rate is linearly adjusted based on the actual wind speed at that hour (not shown above). In addition, the actual hourly rate is cut in half during daytime operations (shown above) and distributed to zones according to wind direction.

Exhibit 2: Infiltration Modeling Scheme for Scenario with North-East Wind

Zone Affected	Infiltration Rate (ACH)*		
	Daytime Operation (HVAC System On)		Night Operation (HVAC System Off)
	Wind Speed < 5 mph	Wind Speed > 5 mph	
Core	0	0	0
North	0.75	1.5	3.0
East	0.75	1.5	3.0
South	0.75	0	0
West	0.75	0	0

* The nominal infiltration rate--the rate for a 10 mph wind with HVAC system off-- is 0.5ACH. The actual hourly rate is linearly adjusted based on the actual wind speed at that hour (not shown above). In addition, the actual hourly rate is cut in half during daytime operations (shown above) and distributed to zones according to wind direction.

Exhibit 3: General Characteristics of Base Buildings

	Office	Education	Assembly
Building Characteristics			
shape	square	L-shaped	square
zones/floor	5	6	5
floor area (ft^2)	338,668	50,600	19,600
number of floors	12	2	1
floor height (ft)	12	15	30
wall construction	steel-reinforced concrete, curtain wall	concrete block	concrete block
net window area (%)	42%	34%	7%
window U-value (Btu/hr ft^2 ^0F)	0.75	0.59	0.59
window shading coefficient	0.8	0.6	0.6
wall R-value (hr ft^2 ^0F/Btu)	R-7	R-8	R-8
roof R-value (hr ft^2 ^0F/Btu)	R-8	R-12	R-12
perimeter/core ratio*	0.5	1	0.6
nominal infiltration rate (ach)	0.5	0.25	0.25
Occupancy			
number of occupants	2,130	1,518	588
occupant density (occup/1000ft^2)	7	30	60
HVAC			
air distribution system	central (CVor VAV)	central (CVor VAV)	central (CVor VAV)
heating and DHW	central gas boiler - 70% efficiency	central gas boiler - 80% efficiency	central gas boiler - 80% efficiency
cooling	chiller - 3 COP w/cooling tower	chiller - 4 COP w/cooling tower	chiller - 4 COP w/cooling tower

* ratio of perimeter to core floor area, where perimeter space is up to 15 ft. from the exterior walls

Exhibit 4: Energy Related Charactierisics of Office Buildings and HVAC System Parameters

Building Configuration	Window R-Value	Window Shading Coeffic.	Roof Insulation	Infiltration Rate	Chiller COP	Boiler Effic. (%)	Occup. Density (Occup/ 1000 SF)	P/C Ratio	Exhaust Flow Rate (cfm)	Daily Operating Hours (hrs/day)
A. Base Case	2.0	0.8	10	0.5	3.5	70	7	0.5	750	12
B. High Effic. Shell	3.0	0.6	20	0.75	3.5	70	7	0.5	750	12
C. Low Effic. Shell	1.0	1.0	5	0.25	3.5	70	7	0.5	750	12
D. High Effic. HVAC System	2.0	0.8	10	0.5	4.5	80	7	0.5	750	12
E. Low Effic. HVAC System	2.0	0.8	10	0.5	2.5	60	7	0.5	750	12
F. High P/C Ratio	2.0	0.8	10	0.5	3.5	70	7	0.8	750	12
G. Low P/C Ratio	2.0	0.8	10	0.5	3.5	70	7	0.3	750	12
H. High Exhaust Rate	2.0	0.8	10	0.5	3.5	70	7	0.5	1500	12
I. High Occup. Density	2.0	0.8	10	0.5	3.5	70	15	0.5	750	12
J. Medium Occup. Density	2.0	0.8	10	0.5	3.5	70	10	0.5	750	12
K. Low Occup. Density	2.0	0.8	10	0.5	3.5	70	5	0.5	750	12
L. Very Low Occup. Density	2.0	0.8	10	0.5	3.5	70	3	0.5	750	12
M. Extended Oper. Hours	2.0	0.8	10	0.5	3.5	70	7	0.5	750	18
N. 24 Hour Operation	2.0	0.8	10	0.5	3.5	70	7	0.5	750	24

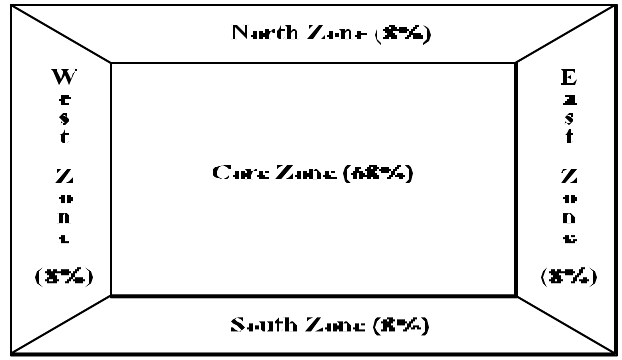
Exhibit 5 Occupant & Operating Schedules for Office, Education, and Assembly Buildings

	Office Building					Education Building						Assembly			
	Occupancy		HVAC			Occupancy			HVAC			Occupancy		HVAC	
Hour	Mon-Fri	WE/Hol	Mon	Tue-Fri	WE/Hol	Mon-Fri	Sat.	Sun/Hol	Mon-Fri	Sat	Sun/Hol	Mon-Fri	WE/Hol	Mon-Fri	WE/Hol
1-5	0%	0%	night	night	night	0%	0%	0%	night	night	night	0%	0%	night	night
6	0%	0%	St Up	night	night	0%	0%	0%	St Up	St Up	night	0%	0%	night	night
7	0%	0%	St Up	St Up	night	0%	0%	0%	St Up	St Up	night	0%	0%	night	night
8	25%	0%	day	day	night	10%	0%	0%	day	day	night	0%	0%	night	night
9	75%	0%	day	day	night	100%	10%	0%	day	day	night	10%	10%	St Up	St Up
10-12	95%	0%	day	day	night	100%	10%	0%	day	day	night	10%	10%	day	day
13	75%	0%	day	day	night	100%	10%	0%	day	day	night	50%	75%	day	day
14-15	95%	0%	day	day	night	100%	0%	0%	day	day	night	50%	75%	day	day
16	95%	0%	day	day	night	50%	0%	0%	day	day	night	50%	75%	day	day
17	75%	0%	day	day	night	50%	0%	0%	day	day	night	50%	75%	day	day
18	50%	0%	day	day	night	50%	0%	0%	day	day	night	50%	75%	day	day
19	25%	0%	night	night	night	15%	0%	0%	day	night	night	100%	100%	day	day
20-21	10%	0%	night	night	night	20%	0%	0%	day	night	night	100%	100%	day	day
22	10%	0%	night	night	night	10%	0%	0%	day	night	night	100%	100%	day	day
23-24	0%	0%	night	night	night	0%	0%	0%	night	night	night	50%	75%	day	day

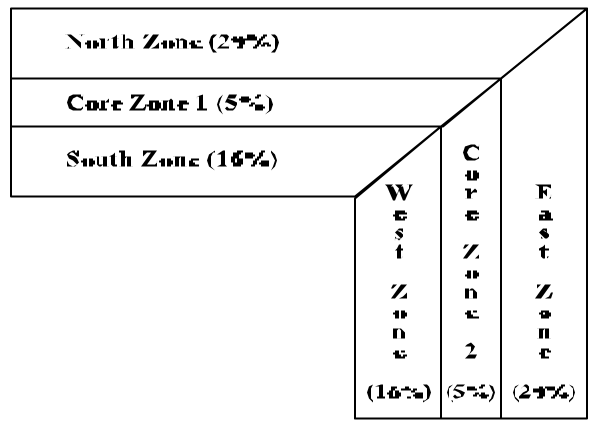
St Up = startup; day = full operation; night = operating with night temperature set back of 10^{o}F.

Exhibit 6: Zone Size and Distributions for Office, Education, and Assembly Buildings

12 Story Office Building with Percent Area:



2 Story Education Building with Percent Area:



1 Story Assembly Building with Percent Area:

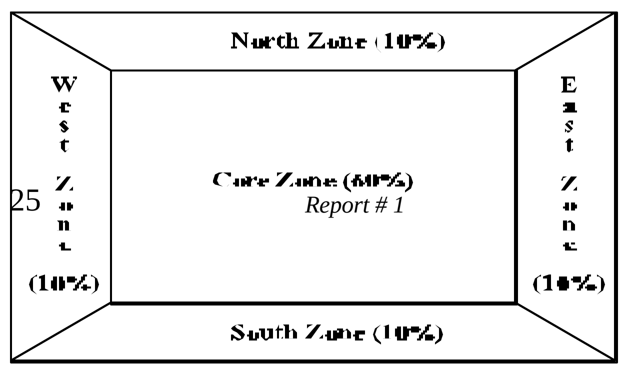


Exhibit 7: Constant Volume HVAC System Configuration

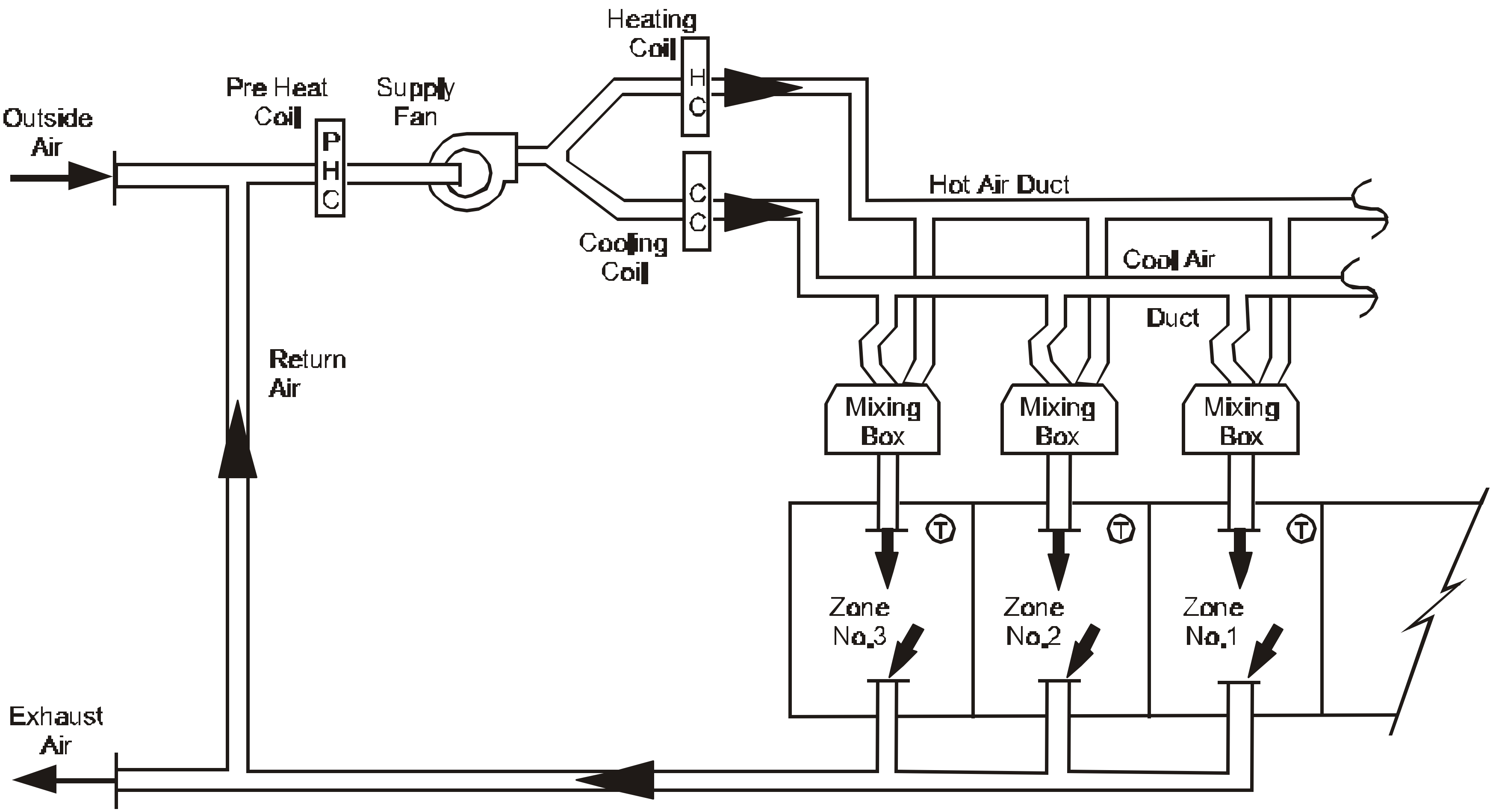


Exhibit 8: Variable Air Volume HVAC System Configuration

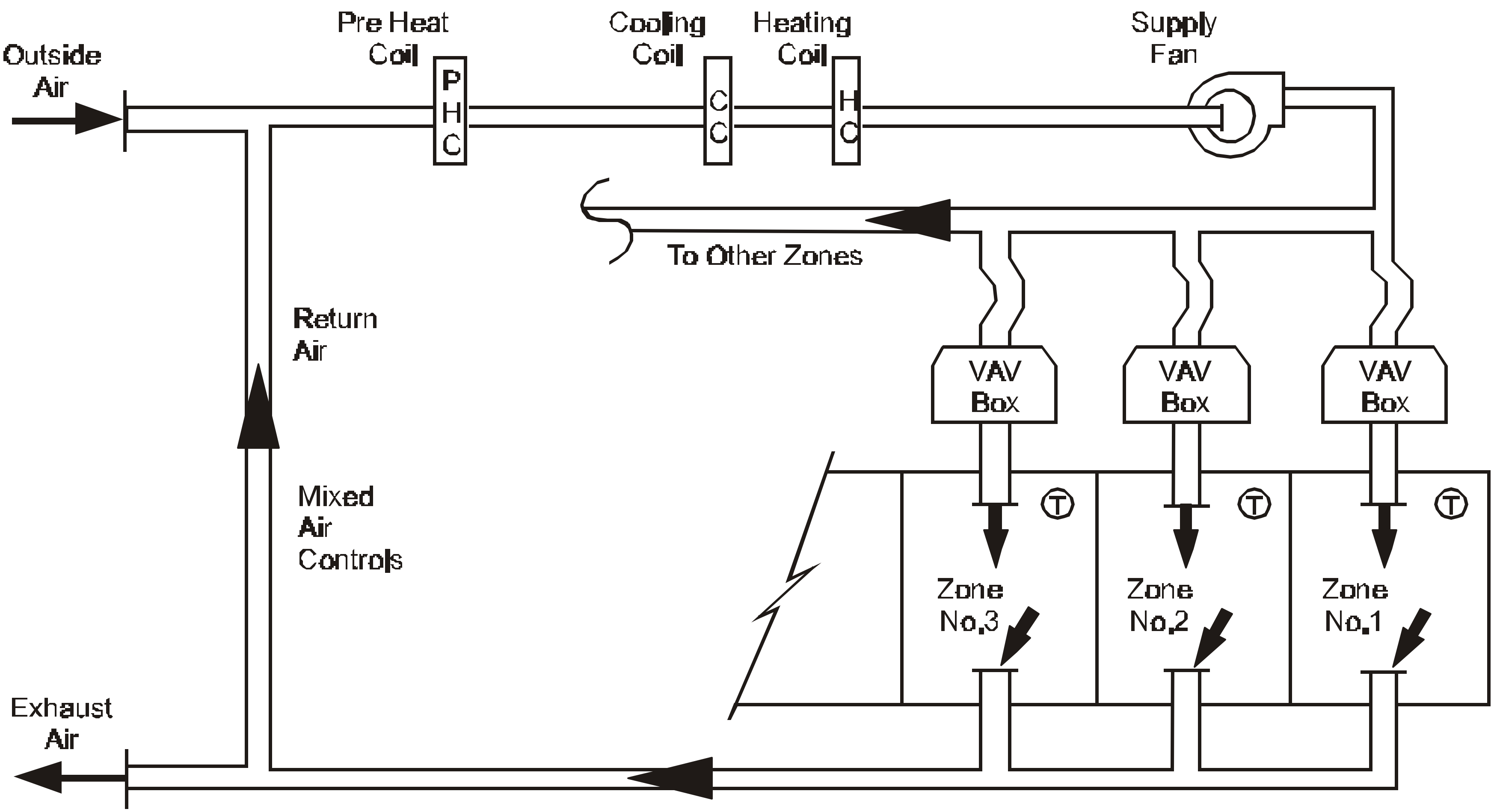


Exhibit 9: Utility Rate Structures Modeled

	Rate Class			Rate Structure			
Rate Structures	Gas Rate	Electric Rate	Electric Demand	Gas Rate	Electric Rate	Electric Demand	Ratchet Clause
Base	Average	Average	Average	$0.490	$0.044	$7.890	No
Option 1	Low	High	Average	$0.330	$0.063	$7.890	No
Option 2	High	Low	Average	$0.650	$0.025	$7.890	No
Option 3	Average	Average	High	$0.490	$0.044	$11.710	No
Option 4	Average	Average	Low	$0.490	$0.044	$4.070	No

Exhibit 10: Actual Utility Rate Structure

City	Average Gas Rate ($/therm)	Average Electric Rate	Average Electric Demand	Ratchet Clause	Office A Bill ($/sf/yr)	Effective Average Rate
Anchorage	$0.318	$0.045	$9.640	80%	$1.16	$0.073
Phoenix	$0.562	$0.075	$6.029	No	$1.46	$0.092
Los Angeles	$0.597	$0.073	$7.000	50%	$1.48	$0.093
San Francisco	$0.637	$0.088	$2.550	No	$1.52	$0.096
Denver	$0.337	$0.027	$8.480	75%	$0.82	$0.051
Washington, DC	$0.826	$0.053	$3.650	No	$1.01	$0.064
Miami	$0.627	$0.039	$8.100	No	$1.00	$0.063
Chicago	$0.158	$0.050	$12.205	No	$1.36	$0.085
Boston	$0.273	$0.030	$16.720	No	$1.25	$0.079
Minneapolis	$0.490	$0.033	$5.738	No	$0.79	$0.050
Omaha	$0.694	$0.044	$5.775	No	$0.97	$0.061
Cleveland	$0.522	$0.043	$11.950	No	$1.24	$0.078
Memphis	$0.454	$0.035	$13.220	30%	$1.16	$0.073
Dallas	$0.514	$0.009	$6.990	No	$0.46	$0.029
San Antonio	$0.469	$0.032	$6.000	No	$0.79	$0.049
Salt Lake City	$0.388	$0.032	$8.450	No	$0.90	$0.056
Seattle	$0.534	$0.035	$1.635	No	$0.63	$0.039
Minimum	$0.158	$0.009	$1.635	None	$0.458	$0.029
Maximum	$0.826	$0.088	$16.720	80%	$1.518	$0.096
Average	$0.494	$0.044	$7.890	n/a	$1.058	$0.067

Standard Deviation	$0.160	$0.019	$3.824	n/a	$0.299	$0.019

Appendix A
Infiltration Modifications

Function Used to Modify the Infiltration Model for the South Zone

```
FUNCTION NAME     = INF5  ..
ASSIGN
        INITIL           = IPRDFL
        IZNAME           = IZNM
        IDTYPE = ISCDAY
        IMONTH           = IMO
        IDAY             = IDAY
        IHOUR   = IHR
        WS               = WNDSPD       $ wind speed
        WD               = IWNDDR       $ wind direction
        ZCFMINF          = ZCFMWSC      $ infiltration rate
        CFMINF           = 0  ..

CALCULATE  ..

        ZCFMINF = 0.25*3*19508*1.15/60/10

        IF ((IHOUR.GE.7).AND.(IHOUR.LE.18).AND.(IDTYPE.GE.2)
*         .AND.(IDTYPE.LE.6).AND.(WS.LE.5))  GOTO 15

        CFMINF = 0

        IF((WD.GE.5).AND.(WD.LE.11)) CFMINF = ZCFMINF*2
        IF(WD.EQ.8) CFMINF = ZCFMINF*4

        ZCFMINF = CFMINF

15      CONTINUE

         END
```

END-FUNCTION ..

Appendix B
Outdoor Air Flow Control Modifications

Function Used to Modify Outdoor Air Control for the VAV/FIX System

```
FUNCTION NAME = economan  ..

ASSIGN
        IHR          = IHR
        IDAY         = IDAY
        IMO          = IMO
        IDTYPE = ISCDAY
        INILZE  = INILZE  ..
        PO      = PO          $ Outdoor air fraction
        TR      = TR          $ Return Air Temp.
        TAPPXX       = TAPPXX
        POMXXX       = MIN-OUTSIDE-AIR
        DBT     = DBT  $ Outdoor air temperature
        ECONOLT      = ECONO-LIMIT-T
        ECONOLL      = ECONO-LOW-LIMIT
        MAXOA        = MAX-OA-FRACTION  ..

CALCULATE  ..
        PO = POMXXX
        IF( ABS( DBT-TR ) .gt. 0.1 )
        PO = AMAX( POMXXX, (TAPPXX-TR)/(DBT-TR) )
        PO = AMIN( PO, MAXOA )
        IF( (ECONOLL .ne. 0) .and. (DBT .lt. ECONOLL) ) PO = POMXXX
        IF( (ECONOLT .ne. 0) .and. (DBT .ge. ECONOLT) ) PO = POMXXX

$ Schedule outdoor air off at night and weekends

        IF( (IDTYPE .lt. 2) .or. (IDTYPE .gt. 6) ) PO = 0
        IF( (IHR .lt. 9) .or. (IHR .gt. 18) ) PO = 0

        END

END-FUNCTION  ..
```

Appendix C
Exhaust Modifications

Function Used for Scheduling Exhaust

```
FUNCTION    NAME   = AIRZONE  ..

        ASSIGN IMONTH  = IMO
               IDAY    = IDAY
               IHOUR   = IHR
               IDTYPE  = ISCDAY
               IZTYPE  = NZ
               INILZE  = INILZE  ..

        ASSIGN ZTEMP   = TNOW
               ZCFM    = CFMZ
               ZCFMH   = FH
               ZCFMC   = FC
               CFMINF  = CFMINF
               QL      = QL
               OAT     = DBT
               WS      = WNDSPD
               WD      = IWNDDR
               OATW    = WBT
               OAH     = ENTHAL
               ZQH     = ZQH
               NVARAA  = 0
               ZP1     = ZP1
               EXH     = EXHAUST-CFM
               CFMZEX  = EXCFM  ..

        CALCULATE  ..

               EXH=0

               IF ( INILZE .LT. 4 ) RETURN
               IF ( IDTYPE .LT. 2 ) RETURN
               IF ( IDTYPE .GT. 6 ) RETURN
               IF ( IHOUR  .LT. 6 ) RETURN
               IF ( IHOUR  .GT. 18) RETURN
```

```
IF ((IZTYPE .EQ. 1).AND.(IHOUR.GT.8)) EXH=750
IF ((IZTYPE .EQ. 6).AND.(IHOUR.GT.8)) EXH=750

IF ( IZTYPE .LT. 6 ) RETURN

NVARAA=ZP1+7
IF (ZCFM.EQ.0) ZCFM=ACCESS(NVARAA)

END

END-FUNCTION  .
```