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Energy Cost and IAQ Performance of Ventilation Systems and Controls

Executive Summary

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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 heating, ventilating, and air conditioning (HVAC) 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 questions:

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 62-1989 (and subsequently 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

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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.

operational engineers during the most common applications of the indoor air quality and thermal comfort standards as prescribed by ASHRAE.

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.

METHODOLOGY

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 with 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 buildings, ventilation systems and outdoor air control strategies.

Buildings and Climate

One large office building, an education building, and an auditorium formed the basis for most of this study. Summary characteristics of these buildings are presented in Exhibit 1. In addition, however, thirteen variations of the office building were used to examine how these variations impacted the energy costs of increasing outdoor air flow rates from 5 to 20 cfm per occupant, while slight modifications to the education building were made to examine the combined application of energy efficiency and indoor air quality controls.

Each building was modeled with (1) a dual duct constant volume (CV) system with temperature reset; and (2) a single duct variable volume (VAV) system with reheat. The CV system was included to represent many existing CV systems rather than current applications. Outdoor air controls include a fixed outdoor air fraction (FOAF), and a constant outdoor air (COA) flow. The FOAF strategy maintains a constant outdoor air fraction (percent outdoor air) irrespective of the supply air volume. For VAV systems, the FOAF could potentially be approximated in field applications by an outdoor damper in a fixed position (Cohen 1994; Janu 1995; and Solberg 1990), but specific field applications are not addressed in this study. The FOAF strategy was modeled so that the design outdoor air flow rate is met at the design cooling load, and diminishes in proportion to the supply flow during part-load.   The COA strategy maintains a constant volume of outdoor air irrespective of the supply air volume. In a CV system, the FOAF and the COA strategies are equivalent, and are referred to in this report as CV (FOAF). In a VAV system, the COA strategy might be represented in field applications by a modulating outdoor air damper which opens wider as the supply air volume is decreased in response to reduced thermal demands. 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.

In this study two types of air-side economizer strategies were also modeled: one controlled by outdoor air temperature ($ECON_T$) and one controlled by outdoor air enthalpy ($ECON_E$). The economizer is designed to override the minimum outdoor air flow called for by the prevailing strategy (FOAF or the COA) by bringing in additional quantities of outdoor air to provide "free cooling" when the outdoor air temperature (or enthalpy) is lower than the return air temperature (or enthalpy). In addition, the temperature economizer is prevented from operating when outdoor air temperatures exceed 65°F in order to avoid potential humidity problems. While the enthalpy economizer was modeled for comparison purposes, the temperature economizer was the primary economizer control used in various parts of this study.

Climates and Utility Rates

Each building was modeled using TMY formatted weather data for three different cities, each representing distinctly different climate regions: Minneapolis, MN (cold climate regions), Washington, DC (temperate climate regions), and Miami, FL (hot and humid climate regions). Five different utility rate structure were modeled to determine the extent to which energy cost impacts from various parametric changes were dependent on utility rate structures. The base utility rate structure represents the average of prices taken from utilities in 17 major cities around the country in 1994. The price of electricity was modeled at $0.044 per kilowatt-hour, and $7.89 per kilowatt. Gas for space heating and DHW service was modeled at $0.49 per therm. The absolute, rather than the percentage increase in the cost of raising outdoor air flow rates, and the absolute dollar savings of energy efficiency projects, for example, would be greater (less) with higher (lower) utility rates. The sensitivity of the results in this study to alternative utility rate structures (varying relationships between gas and electric rates) was also tested.

LIMITATIONS

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

- The analysis is ultimately constrained by the extent to which the model used accurately reflects real world performance.

- While a large number of building and ventilation parameters were used, they are limited in comparison to the many and varied building and ventilation characteristics in the nation's building stock. While the parameters were chosen to capture important variations, they are not necessarily representative.

- The model assumes that all equipment functions as it was intended to function. Faulty design, improper installation, and malfunctioning equipment due to poor maintenance, which are not uncommon in buildings, were not modeled.

ISSUES ADDRESSED IN THE PROJECT

Seven reports, covering the following questions describe the issues addressed in this project:

<u>Project Report #1</u>: Project Objectives and Methodology

- What is the purpose of the project?

- What modeling tool was used and what modifications were made to meet the needs of this project?

- What buildings, HVAC systems, outdoor air control strategies, and utility rate structures were used, and how were they combined in simulations which constitute the database for this project?

<u>Project Report #2</u>: Assessment of CV and VAV Ventilation Systems and Outdoor Air Control Strategies for Large Office Buildings-- Outdoor Air Flow Rates and Energy Use

- Are there significant differences in outdoor air flow and energy cost among different HVAC systems and outdoor air control strategies?

- What HVAC system/outdoor air control strategy combinations offer the best and the worst results?

- What are the trade-offs and compatibilities between energy cost and outdoor air performance among the combinations studied?

<u>Project Report #3</u>: Assessment of CV and VAV Ventilation Systems and Outdoor Air Control Strategies for Large Office Buildings-- Zonal Distribution of Outdoor Air and Thermal Comfort Control

- How well do HVAC systems and outdoor air control strategies deliver design quantities of outdoor air to individual zones?

- Can shortfalls in particular zones be easily corrected and at what energy cost?

Project Report #4: Energy Impacts of Increasing Outdoor Air Flow Rates from 5 to 20 cfm per Occupant in Large Office Buildings

- What are the energy costs of raising outdoor air flow rates from 5 to 20 cfm per occupant for office buildings?

- How does the cost impact vary among different ventilation systems, outdoor air control strategies, and climates?

Project Report #5: Peak Load Impacts of Increasing Outdoor Air Flow Rates from 5 to 20 cfm per Occupant in Large Office Buildings

- Do HVAC system capacity problems result when outdoor air flow rates are raised in existing buildings (designed for 5 cfm of outdoor air per occupant) to conform with ASHRAE 62-1999?

- How significant are such problems and when are they most likely to occur?

- What implications do peak load impacts have on desires to downsize equipment in order to reduce first costs and save energy?

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 Education Buildings, Auditoriums, and Other Very High Occupant Density Buildings

- What operational difficulties are presented by the requirement for large quantities of outdoor air for schools, auditoriums and other buildings with high occupant densities and how can these difficulties best be solved?

- What are the energy costs of increasing outdoor air flow from 5 to 15 cfm per occupant as per ASHRAE Standard 62-1999 for schools, auditoriums, and other buildings with high occupant densities, and how much can these costs be mitigated?

Project Report #7: The Impact of Energy Efficiency Strategies on Energy Use, Thermal Comfort, and Outdoor Air Flow Rates in Commercial Buildings

- What energy efficiency measures are compatible and what measures are incompatible with indoor environmental quality?

• What are the energy savings and penalties associated with measures to protect the indoor environments during energy efficiency projects?

• What protections and enhancements to indoor environmental quality can reasonably be employed in energy management and retrofit projects without sacrificing energy efficiency?

KEY RESULTS

* **VAV Systems Save Energy:** The variable air volume systems provided $0.10 - $0.20 energy savings per square foot over constant volume systems modeled for a savings of !0% to 21% of HVAC energy cost. Since the modeled CV system is much more energy efficient than other CV systems, the results tend to underestimate the advantages of conversion from CV to VAV systems. See Report #2.

*** VAV with Fixed Outdoor Air Fractions Caused Outdoor Air Flow Problems**: VAV systems may require a different outdoor air control strategy at the air handler to maintain adequate outside air for indoor air quality than the constant volume predecessor. Because the fixed outdoor damper strategy of the CV system, which is commonly used in the VAV systems, was modeled to provide a fixed outdoor air fraction, the outdoor air delivery rate at the air handler was cut to about one half to two thirds the design level during most of the year. See Report #2.

*** Core Zones Received Significantly Less Air than Perimeter Zones and space temperatures tended to be higher:** Both the CV and VAV systems provided an unequal distribution of supply air and outdoor air to zones. The south zone received the highest and the core zone received the least outdoor air. The core zone received only about two thirds of the building average outdoor air flow and had higher space temperatures. The impact of zonal differences on indoor air quality in the under-ventilated zones will depend on the degree of air mixing between zones. See Report #3.

*** Core Zones in VAV Systems with a Fixed Outdoor Air Fraction Received Very Little Outdoor Air**: The VAV system with fixed outdoor air fraction diminished the outdoor air delivery to the core zone to only about one third of the design level. With a design level of 20 cfm of outdoor air pe occupant, the core zone received only 6-8 cfm per occupant, and only 2-3 cfm per occupant with a design level of 5 cfm per occupant. Along with higher temperatures in the core zone, this shortfall could contribute to higher indoor air quality complaint rates in the core relative to the perimeter zones in some circumstances. See Report #3.

* **VAV with Constant Outdoor Air Control Displayed Improved Indoor Air Performance without any Meaningful Energy Penalty.** The VAV system with an outdoor air control strategy that maintains the design outdoor air flow at the air handler all year round had slightly lower energy cost in the cold climate, and slightly more energy cost in the hot and humid climate.

It is therefore comparable in energy cost, but preferred for indoor air quality. See Reports #2 and #3.

* **Economizers on VAV Systems May Be Advantageous for Both Indoor Air Quality and Energy in Cold and Temperate Climates.** By increasing the outdoor air flow when the outside air temperature (or enthalpy) is less than the return air temperature(or enthalpy), economizers can reduce cooling energy costs. For office buildings, economizers may operate to provide free cooling even at winter temperatures (e.g. at zero degrees Fahrenheit), provided that coils are sufficiently protected from freezing. For the office building, energy savings of about $0.05 per square foot were experienced by the VAV system economizer over the non-economizer VAV system in cold and temperate climates. The economizer on the CV system modeled was much less advantageous due to increases in heating energy costs for this particular system, and was actually more expensive under some utility rate structures. However, this would not likely be the case for dual fan, dual duct CV systems with separate economizers for the hot and cold coils. The need to control relative humidity and the potential introduction of outdoor contaminants are potential disadvantages of economizer systems. See Reports #2 and #3.

* **VAV with Constant Outdoor Air Control and an Economizer Offers Significant Advantages, while VAV with Fixed Outdoor Air Fraction and No Economizer offers offers Significant Disadvantages:** Of all the ventilation systems and controls studied, the VAV system with constant outdoor air flow, which in cold and temperate climates is combined with an economizer and proper freeze control and humidity control, provided good overall performance considering outdoor air flow, thermal comfort and energy efficiency. The VAV system with a fixed outdoor air fraction and no economizer provided poor overall performance because it failed to deliver adequate outdoor air and displayed no energy benefit. See Reports #2 and #3.

* **Raising Outdoor Air to Meet ASHRAE Standard 62-1999 in Most of the Office Buildings Resulted in Very Modest Increases in Energy Costs**. The main factor affecting the energy cost of raising outdoor air flow was occupant density, such that buildings with higher occupant density experienced higher energy cost increases. But for office buildings with 7 persons per thousand square feet, with moderate chiller and boiler efficiencies, and operating in daytime mode for 12 hours per work day, raising outdoor air flow from 5-20 cfm (2 - 9 L/s) per occupant raised HVAC energy costs by 2% - 10% depending upon system and climate variations. Considering the total energy bill, this increase amounted to approximately 1% - 4%. This is generally less than is commonly perceived and suggests that the issue needs a more careful examination by practitioners. The cooling cost increases in the summer months were counterbalanced by cooling cost savings during cooler weather. Cost increases were higher for economizer systems than systems without economizers because much of the cost savings from higher outdoor air flow rates during cooler weather was already captured by the economizer system. For buildings with occupant densities of 3 persons per thousand square feet, energy costs increases were less. By contrast, office buildings modeled with 15 persons per 1000

square feet experienced up to 21% increase in HVAC energy (or up to 8% increase in the total energy bill). See report #4.

*** VAV Systems in Education, Auditoriums, and Other Buildings with Very High Occupant Densities May Require Special Adjustments for Meeting the High Outdoor Air Flow Rates of ASHRAE 62-1999.** In the education and auditorium buildings, the higher per occupant outdoor air requirements sometimes exceeded the total supply air needed to control thermal comfort. Even with the constant outdoor air damper control on the VAV system, the VAV box minimum settings had to be raised to what appear to be uncommonly high levels (e.g. 50% - 100% of peak flow), in order to maintain 15 cfm per occupant during part load. See Report #6.

* **Controlling Humidity Can be a Problem for Education Buildings, Auditoriums or Other Buildings with Very High Occupant Densities where HVAC Systems Must Deliver High Outdoor Air Flows to Meet ASHRAE Standard 62-1999.** Relative humidity frequently exceeded 60% and occasionally exceeded 70% in all climates in the education buildings and the auditoriums even though the cooling coils were adequately sized to handle peak loads and the indoor temperatures were well controlled. Problems occurred at part load during mild weather when the outdoor relative humidity was high. The increased dominance of the outdoor air at 15 cfm per occupant meant that the heating and cooling system had to deal with wide ranges in the sensible to latent heat ratio, so that humidity as well as temperature had to be part of the control regime. Controlling humidity may be a subject of special concern in buildings with very high occupant densities which meet the outdoor air flow requirements of ASHRAE Standard 62-1999. See Report #6.

*** The Outdoor Air Requirements of ASHRAE Standard 62-1999 for Education Buildings, Auditoriums and Other Buildings with Very High Occupant Densities Can Create a Significant Energy Burden**. When outdoor air ventilation rates were raised from 5 to 15 cfm per occupant in the education building and the auditorium, and when all adjustments were made to insure adequate outdoor air flow rates at part load, and relative humidity was controlled to 60% or below, HVAC energy costs rose by $0.13 -$0.27 per square foot (15%-32%) in the education building, and by $0.36 -$0.88 per square foot (26% - 67%) in the auditorium. This was judged to be a significant energy burden. See report #6.

*** Peak Loads, and therefore Equipment Capacity Requirements, may be Significantly Impacted when Outdoor Air Ventilation Rates are Raised.** Raising the rate from 5 to 20 cfm per occupant in office buildings often raised peak coil requirements by 15% - 25%, and created preheat requirements where none had previously existed. Raising the outdoor air flow rate from 5 to 15 cfm increased the peak loads by 25%-35% in the education building, and by 35% - 40% in the auditorium. This could provide real limits to downsizing strategies which are often part of an energy efficiency strategy, and calls for specific steps to reduce peak loads without sacrificing outdoor air requirements. It also suggests indoor air consultants advise clients of existing buildings to raise outdoor air flow rates in order to reduce indoor air quality complaints, should first consider the potential need to either increase capacity or reduce peak loads. Buildings without sufficient capacity may find themselves unable to maintain thermal comfort in the face of these higher outdoor ventilation rates, or in the worst scenario, may experience coil damage. See Report #5.

*** Energy Recovery Technologies May Potentially Reduce or Eliminate the Humidity Control, Energy Cost and Sizing Problems Associated with ASHRAE Standard 62-1999 in Education Buildings, Auditoriums, and Other Buildings with Very High Occupant Density.** While DOE-2 has limited capabilities to adequately model energy recovery technologies, some literature suggests that both latent and sensible energy recovery systems may significantly reduce or eliminate the associated problems of controlling thermal comfort, reducing energy costs, and downsizing equipment needs while meeting the outdoor air requirements of ASHRAE Standard 62-1999 in high occupant density buildings. Cost issues would include the capital cost of the energy recovery equipment, capital cost savings from downsizing, and the annual energy savings from the energy recovery system. Corroborating research could be of great value. See Report #7.

*** Protecting or Improving Indoor Environmental Quality During Energy Efficiency Projects Need Not Hamper Energy Reduction Goals.** Many energy efficiency measures with the potential to degrade indoor environmental quality appear to require only minor adjustments to protect the indoor environment. When energy efficiency measures (including lighting upgrades), which were adjusted to either enhance or not degrade indoor environmental quality, were combined with measures to meet the outdoor air requirements of ASHRAE Standard 62-1999, total energy costs were cut by 42% - 43% for the office building, and 22% - 37% for the school. Not included were savings from reduced lighting during unoccupied hours that could provide 12% - 22% savings, or improved equipment operations that could provide 5% - 15% savings. Operational measures that could degrade IAQ such as widening the daytime temperature deadband, relaxing the nighttime temperature setback, and reducing HVAC operating hours were not included. Cumulatively, these three measures that are not compatible with IEQ would have reduced total energy costs by only 3%-5% in the office building, and 7%-10% in the education building. Therefore, there appears to be demonstrable compatibility between indoor environmental goals and energy efficiency goals, when energy saving measures and retrofits are applied wisely. See Report #7.

DISCUSSION

Relative Performance of Alternative HVAC Systems and Outdoor Air Control Strategies

Exhibit 2 presents the average outdoor air flow rate by outdoor air temperature for the CV and VAV systems. The design outdoor air flow for each system was set at 20 cfm per occupant. The CV(FOAF) and the VAV(COA) configurations provided 20 cfm of outdoor air per occupant at all times and in all climates. However, the VAV(FOAF) system never provided 20 cfm of outdoor air per person -- except on the design day -- because as the supply air flow rate is throttled back from design conditions, the outdoor air flow into the building is reduced proportionally, to between one third to two thirds the design flow rate most of the time.

Exhibit 3 presents the proportion of occupied hours that each HVAC system in the base office building experiences an outdoor air flow within designated ranges. The outdoor air performance of the VAV(FOAF) systems varied with climate location. The system's outdoor air performance was best in the hot Miami climate, and worst in the cold Minnesota climate. This is because a larger portion of the year is spent at low cooling load conditions in Minneapolis relative to Washington D.C. and Miami. An economizer significantly improved the outdoor air performance of the VAV(FOAF) system for Minneapolis and Washington D.C., but only when the economizer was operational. As expected, the economizer made little difference in the outdoor air performance in the Miami climate.

Variations in outside air distribution due to variations in the thermal loads on the base office building[4] with VAV(COA) in Washington, D.C. are shown in Exhibit 4. For the VAV(COA) system, the outdoor air flow at the air handler is consistently at the design level of 20 cfm (9.2 L/s) per occupant, but there is wide divergence in the outdoor air flow rate to the zones, with the divergence depending on the outdoor temperature. At all temperatures, the core zone is being consistently under ventilated relative to the building design flow rate and receives the least outdoor air during hot weather, when a large portion of the supply air flows to the south zone because of its high cooling load. The zonal pattern would be similar for the VAV(FOAF) system, except that the outdoor air flow rate for each zone is lower, corresponding to the reduced air flow into the building described above.

Since the ventilation disparity between zones is seasonal, the extent to which each zone is over ventilated or under ventilated over the course of the year depends in part on the proportion of occupied hours the building is experiencing various outdoor air temperatures. Exhibit 5 presents the proportion of occupied hours that each zone experiences various outdoor air ventilation rates for different ventilation systems. This table shows that for a design outdoor air flow rate of 20 cfm (9 L/s) per occupant, the core zone of the CV(FOAF) system consistently receives 11-15 cfm (5 - 7 L/s) per occupant, while the core zone for the VAV(COA) system receives this amount about half the time. However, the core zone for the VAV(FOAF) system received only 6 - 10 cfm (2 - 4 L/s) of outdoor air per occupant all year round. While not shown here, patterns for other climates are similar. Also, adjusting VAV box settings (not shown here) did not resolve this potential problem.

Operational modifications (not shown here) to improve the performance of the VAV(FOAF) system were also modeled. Raising the outdoor air setting at design to 30 cfm (14 L/s) per occupant in Miami was sufficient to achieve at least 15 cfm (7 L/s) per occupant year round, but raised HVAC energy costs by $.03 per square foot. For Minneapolis and Washington D.C., raising the design setting to 45 cfm (21 L/s) per occupant was necessary to achieve 15 cfm per occupant year round, raising HVAC energy costs by $ .05 - $.06 per square foot respectively. A seasonal reset strategy was also modeled with similar results. However, making operational adjustments such as these runs the risk of exceeding capacity during extreme weather conditions and may not be advisable.

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[4] The occupant density is the same for each zone.

Exhibit 6 shows the energy costs for the CV system and the VAV systems with and without economizers. Comparisons of the energy costs of the CV(FOAF) and the VAV(COA) demonstrates the energy advantage of the VAV over the CV system. Both systems provide 20 cfm (9 L/s) under all operating conditions, but the energy cost for the VAV system was $0.10 - $0.20 per square foot less than the CV system. Much of this is due to the reduction in fan energy costs.

It is also useful to compare the VAV(FOAF) with the VAV(COA). The VAV(FOAF) system consistently delivered less than 20 cfm (9 L/s) of outdoor air, but offered no energy advantage over the VAV(COA) system which delivered a constant 20 cfm (9 L/s) per occupant. That is, the diminished outdoor air flow of the VAV(FOAF) system did not reduce energy costs over the VAV (COA) system. In fact, for the cold and temperate climates of Minneapolis and Washington D.C., energy costs of the VAV(FOAF) system were marginally greater than the VAV(COA) system, and only marginally less than the VAV(COA) system in Miami. This result is consistent with the fact that additional outside air during cooler weather provides some degree of free cooling, which is the concept underlying the economizer outdoor air control strategy. The added cooling benefit of the additional outdoor air in the VAV(COA) system tends to offset the added cooling burden during the hot summer season. However, when economizers were added to both systems, both systems experienced free cooling. The VAV(FOAF)Econ saved about $.02 per square foot over the VAV(COA)Econ.

Economizers reduced HVAC energy costs 6% - 10% on VAV systems compared to only 1% to 2% for the CV system modeled. The economizer for the CV system provided significant savings in cooling energy for the core zone, but this is partially counterbalanced by a heating penalty for the perimeter zones. While the economizer brings in sufficient outdoor air to reduce the mixed air temperature to 55^{O} F in both systems, the supply air quantity of the CV system is considerably higher than that of the VAV system, and this resulted in a substantial heating penalty for the CV system economizer. Since gas is used for space heating, the advantage of the CV economizer was sensitive to the price of gas relative to electricity. In fact, while not shown here, for pricing structures involving high gas and low electricity prices , the CV economizer raised rather than lowered energy costs. However, this may be unique to the CV system modeled, and would not be expected to apply to a CV system with dual fan, dual ducts and a separate economizer for the hot and cold coils. As expected, economizers have a meaningful impact on energy costs only in cold and temperate climates. Because the Miami climate offers little opportunity for economizer operation, energy savings of the economizer in Miami were minimal.

Impacts of Increased Outdoor Air Flows on Annual HVAC Energy Costs

It is commonly held that raising outdoor air flow rates to accommodate indoor air quality needs will dramatically increase energy use because this increased outdoor air must be conditioned. However, this conventional wisdom ignores the dynamics of energy use of different systems during different seasons. By way of explanation, Exhibit 7 shows the variations on the coil loads at different seasons for selected systems. Exhibit 7 suggests that the annual average change in energy use resulting from increasing the outdoor air flow rate in an office building from 5 - 20 cfm per occupant depends on the relative impact of increases in energy use and decreases in energy use during different seasons. Significant reductions in cooling energy can occur in mild to cold temperatures which offset increases during warm weather, but heating penalties may also occur in some CV systems during cold weather periods. The actual impact depends on the nature of the energy impact during each season, the utility rate structure, and the amount of time the system is operating within each seasonal range. Increases in CV systems modeled tended to be higher than VAV systems because of the heating penalty in winter, while economizer systems tended to result in higher energy cost increases because much of the cooling cost savings in the mild to cold weather is already accounted for in the economizer system.

Exhibit 8a presents the HVAC energy cost changes when outdoor air flow rates were raised from 5 cfm (2 L/s) per occupant to 20 cfm (9 L/s) per occupant for the office building, and to 15 cfm (7 L/s) per occupant for the education and assembly buildings. All buildings have VAV(COA) systems with economizers. For the office building shown, the outdoor air increase resulted in only a 6% - 10% increase in HVAC energy cost, (or approximately 2% - 4% increase of total energy cost). While not shown here, altering the HVAC system and controls and using alternative energy pricing structures did not significantly change this range for the base building; however, lowering the occupant density reduced the energy impact, while raising the occupant density raised the energy impact considerably (see Project Report #4).

Raising outdoor air flow rates resulted in a considerably higher HVAC energy cost increase in the school, amounting to 15% - 31% (5% - 14% total energy cost), while the auditorium experienced an HVAC energy cost increase of 26% - 67% (9% - 25% total energy cost), for the HVAC system shown. The range of increase for other systems was very similar. This large increase was due to many factors. Because of the high occupant densities in these buildings, the required per occupant outdoor air flows may exceed supply air flow during periods of the year when thermal loads are low. In these cases, supply air flows must be increased to maintain minimum outdoor air flows in the building, increasing annual fan energy costs. This was done by adjusting the VAV box minimum settings. In addition, the large volumes of outdoor air subjected the cooling system to wide ranges in the sensible to latent heat ratio, making it difficult for the system to keep indoor air relative humidity below 60% when controlling only for temperature. Particularly on mild but humid days, indoor relative humidity frequently rose above 60% and occasionally rose above 70%. As a result, cooling coil temperatures had to be lowered when needed to insure that indoor relative humidity did not exceed 65%.

Surprisingly, the total increase in HVAC energy cost from raising the outdoor air flow rate in the education building and auditorium was least in Miami. While heating energy costs did not increase in the office buildings with a VAV system in any climate, heating cost penalties in the education and assembly buildings were substantial, often accounting for more than half of the increase in total HVAC energy cost in the cold and temperate climates. However, in the hot and humid climate of Miami, heating energy and fan energy penalties were very low. As a result, the total energy cost increase in Miami was less than it was in either Minneapolis or Washington, D.C. This result may be a function of the limitations of DOE-2 and modeling parameters established for this project and should therefore be interpreted cautiously.

Exhibit 8b shows annual HVAC energy cost increases when outdoor air flow was raised from 5 to 20 cfm (2 - 9 L/s) per person for various office building configurations. It suggests that occupant density is the single most important factor affecting the energy penalty from raising outdoor air flow rates. The occupant density of the base building was 7 persons per 1000 square feet. Dropping occupant density to 3 persons per 1000 square feet reduced the energy penalty to about a third of that in the base building, while raising the occupant density to15 persons per 1000 square feet approximately doubled the energy penalty. Other building variations that were modeled included changes in building shell efficiency, changes in boiler and chiller efficiency, increased exhaust, changes in building shape, and increases in HVAC operating hours. None of these variations showed consistent and significant effects on the energy penalty.

Impacts of Increased Outdoor Air Flows on HVAC System Capacity

Research on the impact of increased outdoor air flows on HVAC system capacity is important because ASHRAE Standard 62-1999 and the IAQ litigation environment may have the effect of forcing building operators to increase outdoor air flow rates in buildings in response to occupant complaints. When these situations occur, the existing cooling and heating systems (designed for 5 cfm of outdoor air per occupant) may not have the capacity to handle the increased load caused by the increased outdoor air flows.

Exhibit 9 presents DOE-2.1E predicted peak load impacts for the 3 types of buildings for VAV(COA) and the CV (FOAF) systems with economizers. While only economizer systems are presented, there were no meaningful differences in the results between systems with and without economizers Peak cooling load increases tended to be higher for the CV system than the VAV system, and also higher in the education and assembly buildings when compared to the office building. Increases in peak cooling loads ranged from 15% - 21% in the office building, from 20% - 33% in the education building, and from 26% to 45% in the assembly building. Increases tended to be higher in warmer climates[5]. Since increases in peak cooling loads caused by the increase in outdoor air occured during the day, capacity limitations on the cooling coil would most likely bring about thermal discomfort of occupants from midday to late afternoon.

Absolute increases in peak heating loads are modest (below 500 kBTU/hr) for all buildings in all climates, but percentage increases can be substantial due to relatively small initial peak loads. Peak preheat coil load increases can be higher (0 - 1100 kBTU/hr) and often occurred in situations where no preheat was required at the lower outdoor air flow rate. The

[5] Peak cooling load increases show the same climatic pattern in Eto (1988).

increase in both the peak heating and peak preheat coil load caused by the increase in outdoor air occurred consistently at the first hour of occupancy when the outdoor air damper was first opened. This suggests that heating and preheat coil capacity limitations may therefore prevent the system from maintaining thermal comfort in the morning, and, with high outdoor air flow rates, potentially throughout the day. In the worst scenario, inadequate preheat capacity could result in coil damage if the outdoor dampers are not closed. But closing the outdoor dampers would add indoor air quality problems to the thermal comfort problems.

The Energy Consequences of Protecting Indoor Environmental Quality in Energy Efficiency Projects

The indoor environmental factors that most influence occupant health and welfare are the thermal conditions, the lighting, and the concentrations of indoor pollutants. Thermal control and lighting are familiar subjects in energy management. Accordingly, energy professionals are in a strong position to affect these two important aspects of indoor environmental quality (IEQ) while they are often less knowledgeable about indoor pollutant concentrations. Energy activities that are compatible with IEQ, either because they are likely to enhance or have little effect on IEQ if properly instituted, are identified in Exhibit 10. In general, the compatibility with IEQ is dependent on the cautions and adjustments which are outlined in this exhibit. In this modeling project, unless otherwise stated, the cautions and limitations described in this exhibit were either directly or implicitly incorporated into the modeling runs when energy efficiency measures were modeled.

Much of the perceived conflict between IEQ and energy efficiency results from just two elements of an energy strategy– the tendency to minimize outdoor air ventilation rates and the willingness to relax controls on temperature and relative humidity to save energy. Energy reduction activities that are generally recognized as having a significant potential for degrading the indoor environment and causing problems for the building owner (client) and the occupants are identified in Exhibit 11.

A staged energy retrofit on an office building and education building was modeled to quantify the energy gains and losses from energy activities which protect or enhance indoor environmental quality and which avoid measures that compromise it. The office building had a VAV system with fixed outdoor air damper and an economizer, while the education building had a VAV system, constant outdoor air flow control and an economizer. The parameters of these buildings and the energy measures taken are presented in Exhibit 12. The staged retrofit included operational (tune-up) measures in Stage 1, load reduction measures in Stage 2, air distribution system upgrades in Stage 3, central plant upgrades in Stage 4, and selected IEQ upgrades in Stage 5. For analytic convenience, most of the operational measures normally included in Stage 1 were modeled and analyzed separately and not included in Stage 1.

Exhibits 13-14 present the energy cost results from the staged energy activities for the office building (Exhibit 13) and the education building (Exhibit 14). Exhibit 15 presents the percent savings (from the base and from the previous stage) of the total energy cost for both buildings[6].

Stage 1 included only a simple seasonal supply air temperature reset strategy which increased the supply air temperature from 55^0 F to 65^0F from January 1 to March 31 in each climate. Therefore, it does not reflect an optimal control logic for the fans and chiller. As a result, the energy savings for Stage 1 (-2% - 1%) are not substantial and not uniformly positive, and do not reflect values that would normally be achieved with a more sophisticated control strategy (See discussion of other operational measures below).

A further reduction beyond Stage 1 of 28% - 33% was achieved in this building through a lighting retrofit and increased efficiency of office equipment in Stage 2. The Stage 3 upgrades relied solely on variable speed drives which reduced the energy costs an additional 5% -10%. Finally, in Stage 4, central plant efficiency upgrades (including down-sizing the equipment because of reduced loads[7]) added another 13% -15% to the total energy savings, bringing the combined savings to 44% - 45% for the office building. The results for the education building were similar but less dramatic, resulting in a total energy savings of 31% - 40%. While many of these activities implemented in Stages 1 through 4 above could adversely impact IEQ, all the necessary adjustments identified in Exhibit 10 were made or are implicit in the model's algorithms to insure that IEQ would not be degraded.

The base buildings provided only 5 cfm of outdoor air per occupant (i.e. does not meet the current ASHRAE ventilation requirements for indoor air quality (ASHRAE Standard 62-1999)). To meet the requirements of ASHRAE Standard 62-1999, a set of IEQ controls were instituted as part of Stage 5. The first control was to raise the outdoor air setting from 5 cfm per occupant to 20 cfm per occupant in the office building, and 15 cfm per occupant in the education building. The second control was to provide a constant outdoor air control damper to the office building to insure 20 cfm of outdoor air per occupant at all times. In the education building, VAV boxes were adjusted to insure 15 cfm per occupant at all times, and relative humidity was controlled so as not to exceed 60%.

When compared to the previous stage, meeting these indoor environmental requirements raised total energy costs 3% -4% for the office building and 5% -14% for the education building. When compared to from the base building, the IAQ requirements amounted to a sacrifice of 2%-3% of annual energy savings for the office building, and 3%-9% for the education building. Accordingly, the staged energy retrofits which include provisions to protect indoor environmental quality and which provide additional outdoor air to meet ASHRAE Standard 62-1999 achieved total energy savings of 42% - 43% for the office building, and 22% - 37% for the education building. While the modeling capability in DOE-2.1E does not allow adequate representation of energy recovery systems, some literature suggests that the energy

[6]Total energy costs are defined here to include only energy from HVAC, lighting, and office equipment.

[7]The equipment was downsized, but not below that necessary to accommodate increased outdoor air flow in Stage 5 of 20 cfm/occ for the office building, and 15 cfm per occupant for the education building. as per ASHRAE Standard 62-1999.

burden of providing additional outdoor air may be substantially reduced or eliminated through energy recovery technology (Rengarajan, el al. 1996; Shirey and Rengarajan, 1996). This issue, including the capital cost of the energy recovery equipment, the capital saving due to downsizing, and the potential energy saving is worthy of further research.

Many energy measures with significant potential to adversely impact IEQ occur in Stage 1, and involve either relaxing temperature (and humidity) controls and/or reducing HVAC operating hours. Exhibit 16 summarizes the results of these modeling runs. Widening the day time temperature dead band from 71 - 77^0 F to 68-80^0 F reduced energy costs by 2% -3% in the office building, and by 7% - 8% in the education building. Relaxing the night time temperature setback from +/- 10^{o}F to +/- 15^{o}F reduced energy costs from 0% - 1% in the office and from 1% - 2% in the education building. Reducing the HVAC operating time by two hours (including a reduction of startup time from 2 hours to 1 hour), reduced the energy costs by 0% - 1% for the office building and by 2% - 4% in the education building. All of these operational measures are attractive because they are inexpensive to implement. However, the savings are small relative to other operational measures or retrofit measures, and cumulatively amount to savings of only 3%-5% for the office building and to 7% - 10% in the education building.

In contrast, other operational measures for Stage 1 that do not degrade IEQ can provide significant savings. For example, simply commissioning the building to insure that controls and equipment are functioning properly (not modeled) have been shown to typically reduce total energy costs by 5% - 15%, and also tend to improve IEQ (Gregerson, 1997). Reducing lighting and office equipment usage during unoccupied hours can also result in significant savings. The base office building was modeled with lighting during unoccupied hours operated at 20% of daytime use and office equipment operated at 30% of daytime use. Exhibit 17 compares the modeling results for this case (20%/30%) with both greater usage during unoccupied hours (40% /50%) in Stage 1, and reduced usage (10%/15%) after Stage 4 modifications.

As indicated in Exhibit 17, had the usage of the lighting/office equipment during unoccupied hours been at 40%/50% of day time levels and then reduced to the original levels of 20%/30% that was modeled in the office building, 12% savings would have been possible in Stage 1 from this activity. This result is consistent with field data which showed that energy savings of 15% on average are associated with operational controls (mostly lighting) during unoccupied hours (Herzog, et al.1992). In addition, an aggressive program to reduce nighttime use of lights and office equipment after the building is made energy efficient and IEQ compatible could provide additional reductions of equal magnitude.

In sum, the energy savings from operational controls that could degrade IEQ amounted to only 3% - 10% of total energy costs. Considering the energy savings of 31% - 45% associated with IEQ-compatible upgrades through Stage 4, plus the potential for additional savings of 12% or more from reduced use of lights, and savings of 5% - 15% from improved equipment performance, the energy savings of 3% - 10% from controls that are incompatible with IEQ are very small in comparison. It appears to make little sense to pursue energy reduction activities that compromise IEQ and run the risk of potential liability of IEQ-related illnesses and complaints, when the energy saving potential for compatible measures is so much greater in comparison.

SUMMARY

This study contains DOE-2.1E modeling data and analysis which shed light on several important issues related to the performance of ventilation systems in terms of energy use, thermal comfort, and outdoor air flow. Three fundamental questions were examined.

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 results suggest that VAV systems with fixed outdoor air fraction (VAV(FOAF)) do not provide adequate outdoor air to the building even when design settings are consistent with ASHRAE 62-1999. CV and VAV (COA) do not pose such a problem. However, all systems provide less than average outdoor air to the core zones and more than average outdoor air to the perimeter zones. For the VAV (FOAF) system the core zone was particularly vulnerable to being starved for outdoor air even with design settings meeting ASHRAE Standard 62-1999.

The cost of increasing outdoor air flow to meet ASHRAE standards can be modest for office buildings, except that the energy penalty can rise substantially with higher occupant densities. For schools and auditoriums, with high occupant densities, the energy penalty can be substantial. Controlling humidity could also be a problem with higher outdoor air flow rates in schools and auditoriums. It was noted, however, that energy recovery ventilation may have the potential to significantly improve humidity control and reduce the energy penalty.

Finally, the study suggests that protecting indoor environmental quality in energy efficiency projects need not hamper the achievement of energy reduction goals, provided that the projects are instituted wisely. Avoiding measures that could degrade IEQ involved energy sacrifices that were small compared to the potential for energy savings from measures that are compatible with IEQ. Some guidelines for insuring that energy efficiency measures do not degrade IEQ were also presented.

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Exhibit 1: Characteristics of the Base Buildings Modeled in this Study

	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	0.6
infiltration rate (ach)	0.25**	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

**0.5 when HVAC is not operating

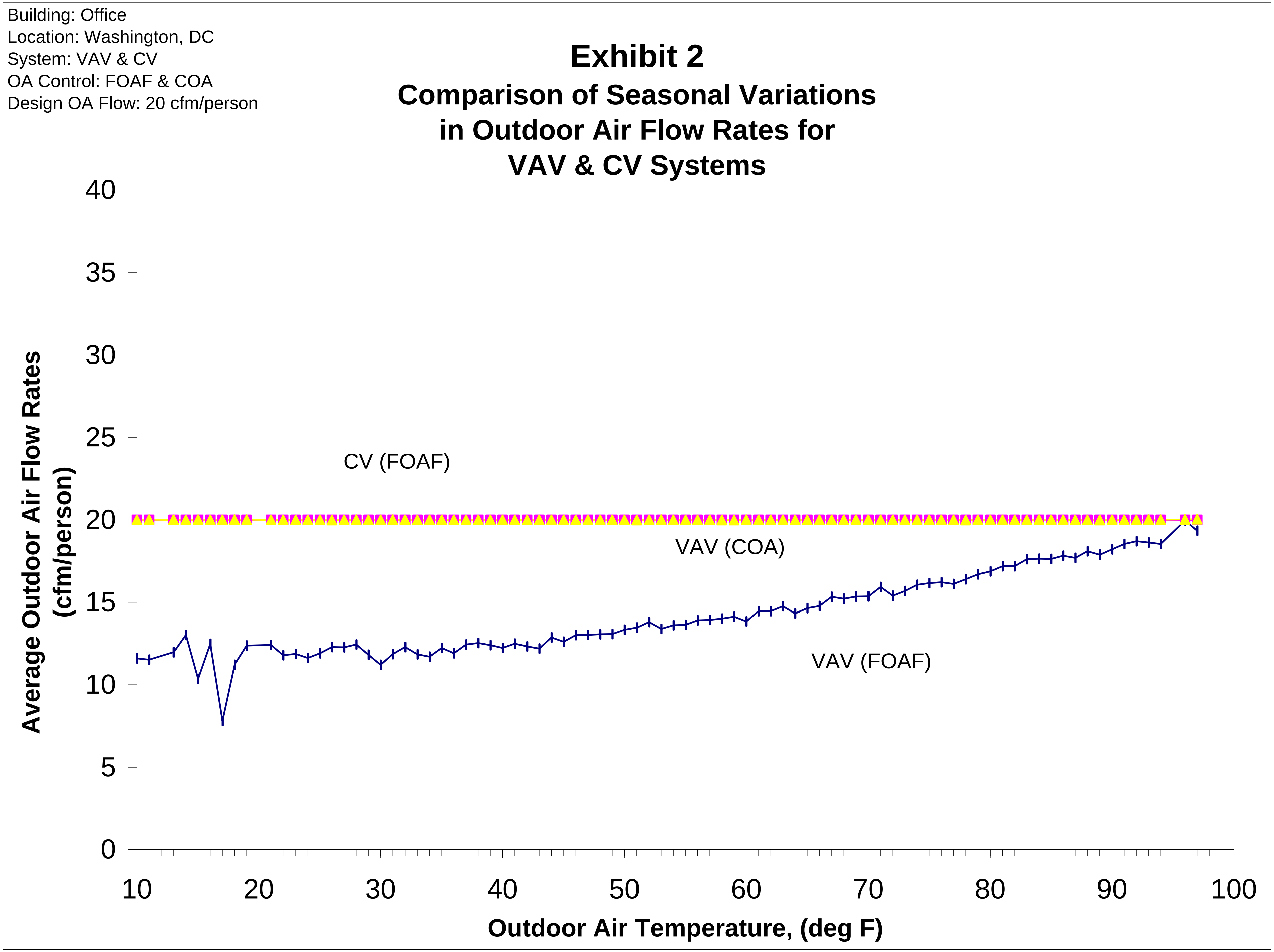
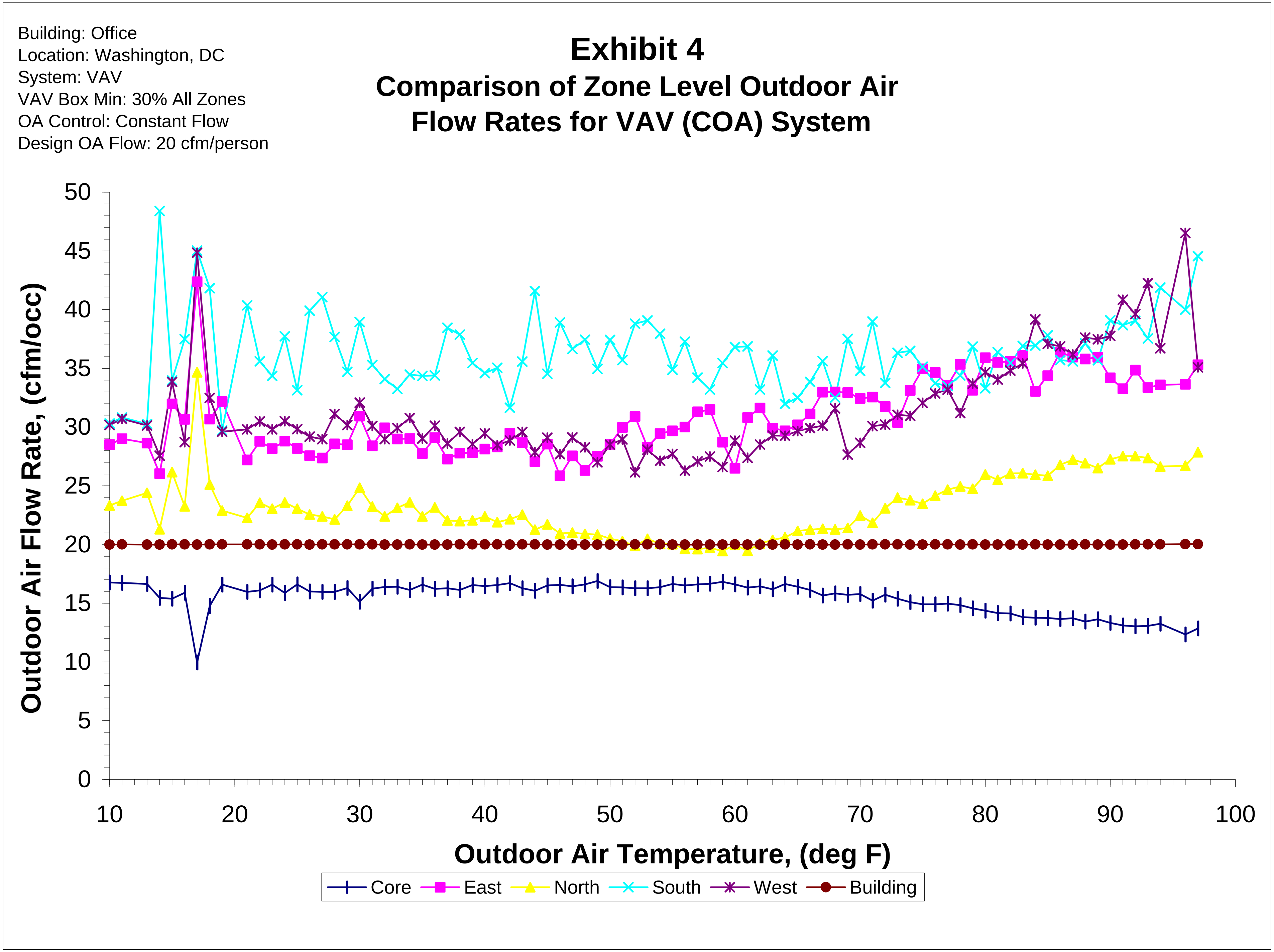

Building: Office
Location: Washington, DC
System: VAV & CV
OA Control: FOAF & COA
Design OA Flow: 20 cfm/person
Exhibit 2
Comparison of Seasonal Variations
in Outdoor Air Flow Rates for
VAV & CV Systems
CV (FOAF)
VAV (COA)
VAV (FOAF)
Average Outdoor Air Flow Rates
(cfm/person)
0
5
10
15
20
25
30
35
40
10
20
30
40
50
60
70
80
90
100
Outdoor Air Temperature, (deg F)

Exhibit 3: Comparison of Outdoor Air Flows {design = 20 cfm (9 L/s) per person} for a Large Office Building with Alternative HVAC Systems

(% of Occupied Hours)

HVAC System Type and Climate Location	Outdoor Air Flow Rates Achieved (cfm per person)				
	<= 5	6-10	11-15	16-19	>= 20
CV(FOAF)					
Minneapolis, MN	0.0%	0.0%	0.0%	0.0%	100.0%
Washington, DC	0.0%	0.0%	0.0%	0.0%	100.0%
Miami, FL	0.0%	0.0%	0.0%	0.0%	100.0%
VAV(COA)					
Minneapolis, MN	0.0%	0.0%	0.0%	0.0%	100.0%
Washington, DC	0.0%	0.0%	0.0%	0.0%	100.0%
Miami, FL	0.0%	0.0%	0.0%	0.0%	100.0%
VAV(FOAF)					
Minneapolis, MN	0.0%	42.0%	56.3%	1.7%	0.0%
Washington, DC	0.0%	16.6%	78.1%	5.3%	0.0%
Miami, FL	0.0%	0.0%	42.5%	57.5%	0.0%
VAV(FOAF) Econ					
Minneapolis, MN	0.0%	0.0%	32.5%	0.0%	67.4%
Washington, DC	0.0%	0.1%	48.6%	0.5%	50.8%
Miami, FL	0.0%	0.0%	62.3%	31.9%	5.8%



Building: Office
Location: Washington, DC
System: VAV
VAV Box Min: 30% All Zones
OA Control: Constant Flow
Design OA Flow: 20 cfm/person
Exhibit 4
Comparison of Zone Level Outdoor Air Flow Rates for VAV (COA) System
Outdoor Air Flow Rate, (cfm/occ)
Outdoor Air Temperature, (deg F)
0
5
10
15
20
25
30
35
40
45
50
10
20
30
40
50
60
70
80
90
100
Core
East
North
South
West
Building

Exhibit 5: Comparison of Zone Level Outdoor Air Flow Rates {design = 20 cfm (9 L/s) per person} for Three Types of HVAC Systems in Office Buildings in Washington, DC.

(% of Occupied Hours)

System Type and Zone	without Economizer: OA Flow Rate Achieved (cfm/person)					with Economizer: OA Flow Rate Achieved (cfm/person)				
	<6	6-10	11-15	16-19	>19	<6	6-10	11-15	16-19	>19
CV(FOAF)										
Core			100.0					49.9	0.6	49.5
East					100.0					100.0
North					100.0					100.0
West					100.0					100.0
South					100.0					100.0
VAV(COA)										
Core		0.2	51.5	48.3	0.0			39.0	10.4	50.6
East					100.0					100.0
North				14.4	85.6				4.1	95.9
West					100.0					100.0
South					100.0					100.0
VAV(FOAF)										
Core	0.7	99.3	0.0	0.0	0.0		49.3	0.1	0.1	50.6
East			46.4	15.3	38.3			6.3	13.5	80.2
North			69.4	24.4	6.2		3.5	19.4	23.8	53.2
West			54.2	15.0	30.8			9.7	14.5	75.7
South			35.2	10.7	54.1			6.1	8.8	85.1

Exhibit 6: Comparison of Annual Energy Costs for the Base Office Building with Alternative HVAC Systems and in Different Climates

HVAC System Type and Climate Location	Annual HVAC Energy Use Summary				
	Fan ($/SF)	Cooling ($/SF)	Heating ($/SF)	Total ($/SF)	Total (KBtu/sf)
CV(FOAF)					
Minneapolis, MN	0.32	0.52	0.04	0.88	47.4
Washington, DC	0.29	0.56	0.01	0.86	41.1
Miami, FL	0.30	0.72	0.00	1.02	50.6
CV(FOAF) Econ					
Minneapolis, MN	0.32	0.45	0.10	0.87	53.7
Washington, DC	0.29	0.50	0.06	0.85	46.4
Miami, FL	0.30	0.71	0.00	1.01	50.7
VAV(COA)					
Minneapolis, MN	0.19	0.49	0.10	0.78	49.7
Washington, DC	0.17	0.52	0.05	0.74	38.9
Miami, FL	0.18	0.65	0.00	0.83	38.9
VAV(COA) Econ					
Minneapolis, MN	0.19	0.43	0.11	0.73	45.9
Washington, DC	0.17	0.47	0.05	0.69	35.8
Miami, FL	0.18	0.64	0.00	0.83	38.5
VAV(FOAF)					
Minneapolis, MN	0.19	0.49	0.10	0.79	50.6
Washington, DC	0.17	0.52	0.05	0.74	39.5
Miami, FL	0.18	0.62	0.00	0.81	38.3
VAV(FOAF) Econ					
Minneapolis, MN	0.19	0.42	0.11	0.71	45.7
Washington, DC	0.17	0.46	0.05	0.68	35.5
Miami, FL	0.18	0.61	0.00	0.80	37.8

Exhibit 2
Change in Coil Loads with Increased Outdoor Air Flow Rate for Building A with CV (FOAF) in Washington, DC

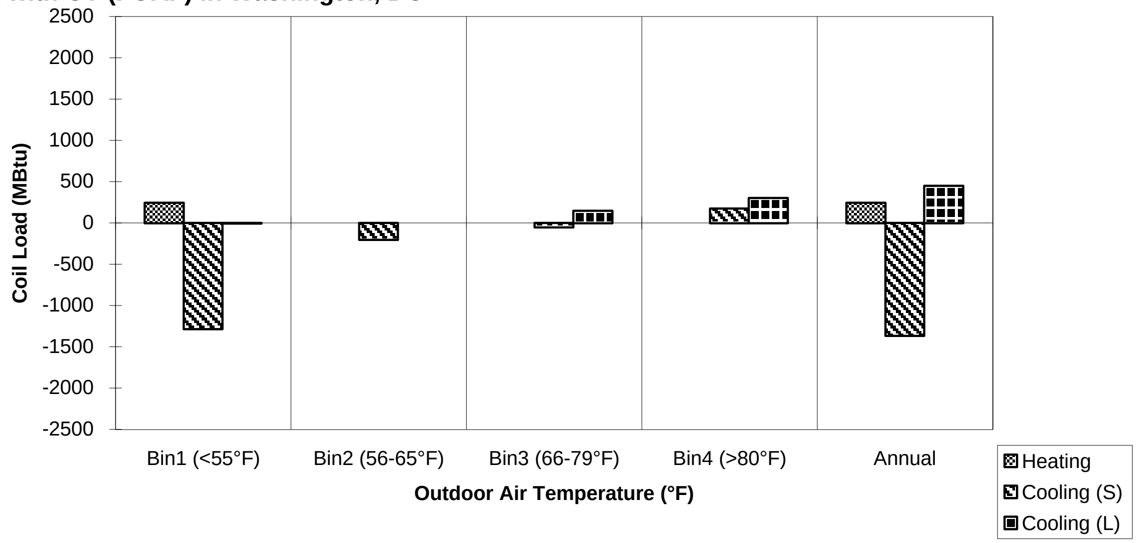


Exhibit 3
Change in Coil Loads with Increased Outdoor Air Flow Rate for Building A with CV (FOAF) $Econ_T$ in Washington, DC

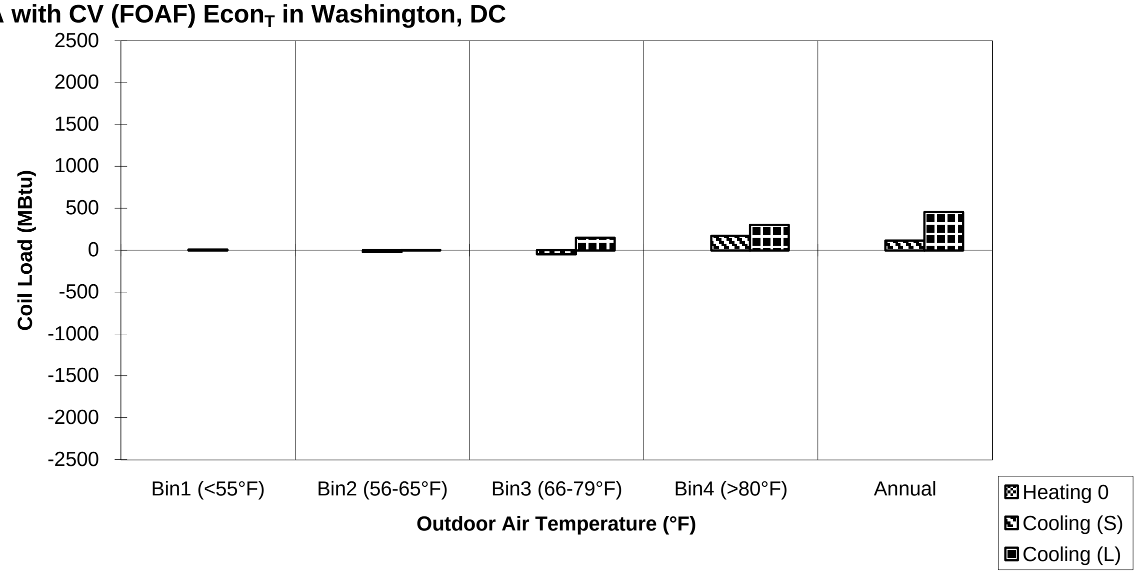


Exhibit 7a
Change in Coil Loads with Increased Outdoor Air Flow Rate for Building A with VAV (COA) in Minneapolis, MN

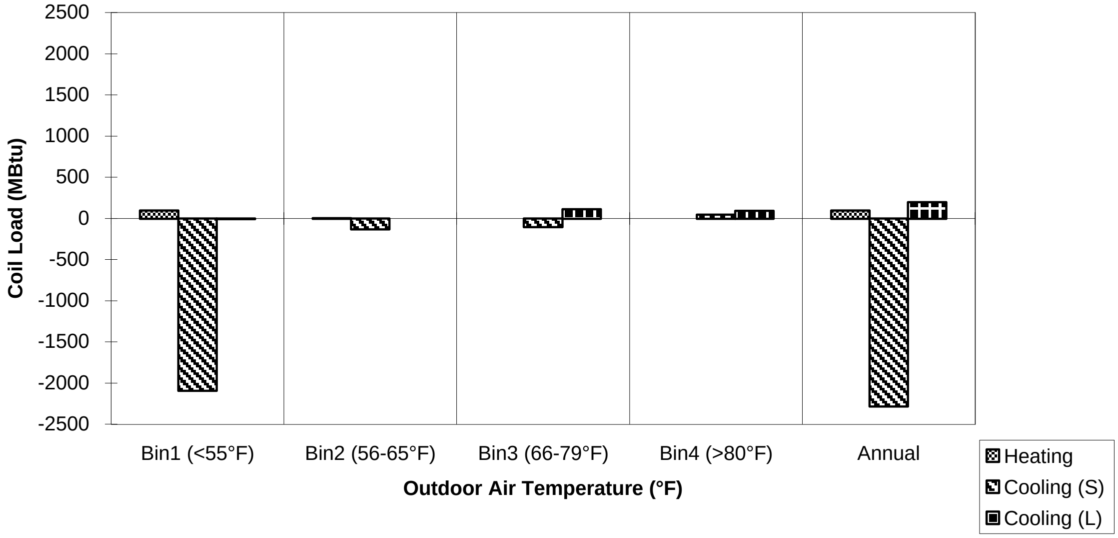


Exhibit 7b
Change in Coil Loads with Increased Outdoor Air Flow Rate for Building A with VAV (COA) $Econ_T$ in Minneapolis, MN

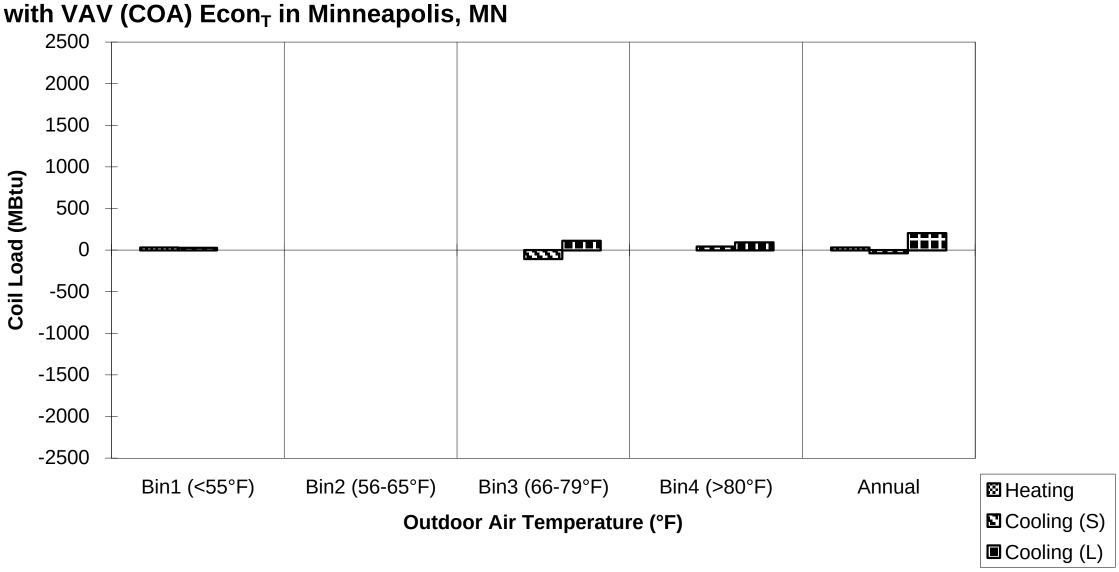


Exhibit 9: Impacts of Increased Outdoor Air Flows on Peak HVAC Coil Loads for the CV(FOAF) and VAV(COA) Systems with Economizers

Climate	Office Building			Education Building			Assembly Building		
	5 cfm	Increase	Percent Increase	5 cfm	Increase	Percent Increase	5 cfm	Increase	Percent Increase
End Use	kBTU/Hr	kBTU/hr	(%)	kBTU/hr	kBTU/hr	(%)	kBTU/hr	kBTU/hr	(%)
CV(FOAF) Econ$_T$									
Minneapolis, MN									
Cooling	9288	1421	15.0%	2068	679	33.0%	1193	527	44.0%
Heating	6707	623	9.0%	2609	588	23.0%	1476	41	3.0%
Preheat	0	0	0.0%	0	0	0.0%	0	44	Increase
Washington, DC									
Cooling	9017	1756	19.0%	2071	562	27.0%	1157	522	45.0%
Heating	3936	35	1.0%	1884	234	12.0%	1153	74	6.0%
Preheat	0	0	0.0%	0	77	Increase	0	38	Increase
Miami, FL									
Cooling	9258	1876	20.0%	2409	503	21.0%	1394	611	44.0%
Heating	3446	0	0.0%	17	560	3366.0%	63	478	758.0%
Preheat	0	0	0.0%	0	0	0.0%	0	0	0.0%
VAV(COA) Econ$_T$									
Minneapolis, MN									
Cooling	8688	1336	15%	1841	370	20%	958	271	28%
Heating	6148	444	7%	2819	1	0%	1134	106	9%
Preheat	0.00	897	Increase	246	1282	521%	330	919	279%
Washington, DC									
Cooling	8517	1659	19%	1951	488	25%	1067	275	26%
Heating	4638	None	None	1935	93	5%	822	177	22%
Preheat	0.00	None	None	90	750	832%	224	553	247%
Miami, FL									
Cooling	8670	1862	21%	2213	529	24%	1229	349	28%
Heating	1949	None	None	397	271	68%	99	291	293%
Preheat	0.00	None	None	0.00	200	Increase	0.00	151	Increase

Exhibit 10: Energy Measures that are Compatible with IEQ

Measure	Comment
Improve building shell	- May reduce infiltration. May need to increase mechanically supplied outdoor air to ensure applicable ventilation standards are met.
Reduce internal loads (e.g. lights, office equipment)	- Reduced loads will reduce supply air requirements in VAV systems. May need to increase outdoor air to meet applicable ventilation standards. - Lighting must be sufficient for general lighting and task lighting needs
Fan/motor/drives	- Negligible impact on IEQ
Chiller/ boiler	- Negligible impact on IEQ
Energy recovery	- May reduce energy burden of outdoor air, especially in extreme climates and/or when high outdoor air volumes are required (e.g. schools, auditoria).
Air-side economizer	- Uses outdoor air to provide free cooling. Potentially improves IEQ when economizer is operating by helping to ensure that the outdoor air ventilation rate meets IEQ requirements. - On/off set points should be calibrated to both the temperature and moisture conditions of outdoor air to avoid indoor humidity problems. May need to disengage economizer during an outdoor air pollution episode.
Night pre-cooling	- Cool outdoor air at night may be used to pre-cool the building while simultaneously exhausting accumulated pollutants. However, to prevent microbiological growth, controls should stop pre-cooling operations if dew point of outdoor air is high enough to cause condensation on equipment.
Preventive Maintenance (PM) of HVAC	- PM will improve IEQ and reduce energy use by removing contaminant sources (e.g. clean coils/drain pans), and insuring proper calibration and efficient operation of mechanical components (e.g. fans, motors, thermostats, controls)..
CO_2 controlled ventilation	- CO_2 controlled ventilation varies the outdoor air supply in response to CO_2 which is used as an indicator of occupancy. May reduce energy use for general meeting rooms, studios, theaters, educational facilities etc. where occupancy is highly variable, and irregular. A typical system will increase outdoor air when CO_2 levels rise to 600-800 ppm to ensure that maximum levels do not exceed 1,000 ppm. The system should incorporate a minimum outside air setting to dilute building related contaminants during low occupancy periods.
Reducing demand (KW) charges	- Night pre-cooling and sequential startup of equipment to eliminate demand spikes are examples of strategies that are compatible with IEQ. Caution is advised if load shedding strategies involve changing the space temperature set points or reducing outdoor air ventilation during occupancy.
Supply air temperature reset	- Supply air temperature may sometimes be increased to reduce chiller energy use. However, fan energy will increase. Higher supply air temperatures in a VAV system will increase supply air flow and vice versa.

Exhibit 10 (continued)

Equipment down-sizing	- Prudent avoidance of over-sizing equipment reduces first costs and energy costs. However, capacity must be sufficient for thermal and outdoor air requirements during peak loads in both summer and winter. Latent load should not be ignored when sizing equipment in any climate. Inadequate humidity control has resulted in thermal discomfort and mold contamination so great as to render some buildings uninhabitable. - Energy recovery systems may enable chillers and boilers to be further downsized by reducing the thermal loads from outdoor air ventilation.

Exhibit 11: Energy Measures that May Degrade IEQ

Measure	Comment
Reducing outdoor air ventilation	- Applicable ventilation standards usually specify a minimum continuous outdoor air flow rate per occupant, and/or per square foot, during occupied hours. They are designed to ensure that pollutants in the occupied space are sufficiently diluted with outdoor air. Reducing outdoor air flow below applicable standards can degrade IEQ and has low energy saving potential relative to other energy saving options.
Variable Air Volume (VAV) Systems with fixed percentage outdoor air	- VAV systems can yield significant energy savings over Constant Volume (CV) systems in many applications. However, many VAV systems provide a fixed percentage of outdoor air (e.g. fixed outdoor air dampers) so that during part load conditions when the supply air is reduced, the outdoor air may also be reduced to levels below applicable standards. - VAV systems should employ controls which maintain a continuous outdoor air flow consistent with applicable standards. Hardware is now available from vendors and involves no significant energy penalty.
Reducing HVAC operating hours	Delayed start-up or premature shutdown of the HVAC can evoke IEQ problems and occupant complaints. - An insufficient lead time prior to occupancy can result in thermal discomfort and pollutant-related health problems for several hours as the HVAC system must overcome the loads from both the night-time setbacks and from current occupancy. This is a particular problem when equipment is downsized. Shutting equipment down prior to occupants leaving may sometimes be acceptable provided that fans are kept operating to ensure adequate ventilation. However, the energy saved may not be worth the risk .
Relaxation of thermal control	Some energy managers may be tempted to allow space temperatures or humidity to go beyond the comfort range established by applicable standards. Occupant health, comfort and productivity are compromised. The lack of overt occupant complaints is NOT an indication of occupant satisfaction.

Exhibit 12: Modeling Parameters for the Office and Education Building

Building Parameter	Office Building		Education Building	
	Base	**Modification**	**Base***	**Modification**
Stage 1: Operational/Tune-up Measures				
Day Temp. Set Points	71^o- 77^o F	*(68^o - 80^o F)*	71^o - 77^o F	*(68^o - 80^o F)*
Night Set Back	+/- 10^o F	*(+/- 15^oF)*	+/- 10^o F	*(+/- 15^oF)*
Day HVAC Hours	8am - 6pm	*(9am - 5pm)*	7am -10pm	*(8am - 9pm)*
Seasonal Reset	No	Yes	No	Yes
Entries in parentheses were modeled separately–not part of the retrofit project				
Stage 2: Load Reduction Measures				
Lighting	2.5 W/f2	30% reduction	3.0 W/f^2 rms 2.0 W/f^2 corr	30% reduction
Office Equipment	1.0 W/f^2	30% reduction	0.25 W/f^2	30% reduction
Stage 3: Air distribution System Upgrades				
VSD	no	yes	no	yes
Stage 4: Central Plant Upgrades				
Chiller COP	3.0	5.5	3.0	5.5
Boiler Efficiency	70%	85%	70%	85%
Stage 5: IEQ Ventilation Modifications Required to meet ASHRAE 62-1999				
Outdoor Air Setting	5 cfm/occ	20 cfm/occ	5 cfm/occ	15 cfm/occ
Outdoor Air Control	fixed damper	constant flow	constant flow	const. flow-VAV box adjustment
Humidity Control	not needed	not needed	not needed	60% RH

*For the base education building used for the energy retrofit: infiltration rate = 0.5ach; window U value = 0.99 (Btui/hr ft^2 ^{0}F); and window shading coeff. = 0.90.

Exhibit 13: Energy Cost for Office Building with Energy and IEQ Modifications

Building Parameter	Washington D.C. ($/sf)						Minneapolis ($/sf)			Miami ($/sf)		
	Fan	Cool	Heat	Total HVAC	Light & Off. Equip	**Total**	Total HVAC	Light & Off. Equip	**Total**	Total HVAC	Light & Off. Equip	**Total**
Base Bldg	0.17	0.42	0.05	0.64	0.94	**1.58**	0.68	0.94	**1.62**	0.74	0.94	**1.68**
Stage 1 Seas. Reset	0.18	0.41	0.04	0.63	0.94	**1.57**	0.66	0.94	**1.60**	0.78	0.94	**1.72**
Stage 2 Ltng/Off Equip	0.15	0.30	0.08	0.52	0.57	**1.08**	0.58	0.57	**1.16**	0.57	0.57	**1.15**
Stage 3 VSD	0.09	0.28	0.06	0.43	0.57	**1.00**	0.47	0.57	**1.04**	0.52	0.57	**1.09**
Stage 4 Chiller/Boiler	0.09	0.16	0.05	0.30	0.57	**0.87**	0.33	0.57	**0.90**	0.35	0.57	**0.93**
Stage 5												
OA Setting	0.09	0.18	0.06	0.32	0.57	**0.89**	0.36	0.57	**0.93**	0.38	0.57	**0.95**
OA Control	0.09	0.19	0.06	0.33	0.57	**0.90**	0.37	0.57	**0.94**	0.40	0.57	**0.9**

Exhibit 14: Energy Cost for the Education Building with Energy and IEQ Modifications

Building Parameter	Washington D.C. ($/f²)						Minneapolis ($/f²)			Miami ($/f²)		
	Fan	Cool	Heat	Total HVAC	Light & Off Equip	**Total**	Total HVAC	Light & Off Equip	**Total**	Total HVAC	Light & Off. Equip	**Total**
Base Bldg	0.21	0.62	0.28	1.11	0.97	**2.08**	1.42	0.97	**2.40**	1.22	0.97	**2.19**
Stage 1 Seasonal Reset	0.21	0.61	0.25	1.07	0.97	**2.04**	1.38	0.97	**2.36**	1.23	0.97	**2.21**
Stage 2 Lights/off equip	0.19	0.53	0.33	1.04	0.67	**1.71**	1.42	0.67	**2.10**	1.08	0.67	**1.76**
Stage 3 VSD	0.11	0.50	0.33	0.94	0.67	**1.62**	1.30	0.67	**1.97**	0.98	0.67	**1.65**
Stage 4 Chiller/boiler	0.11	0.29	0.28	0.67	0.67	**1.35**	0.98	0.67	**1.65**	0.64	0.67	**1.31**
Stage 5* OA Setting	0.12	0.35	0.35	0.82	0.67	**1.49**	1.19	0.67	**1.68**	0.73	0.67	**1.40**
OA & RH control	0.13	0.36	0.38	0.87	0.67	**1.54**	1.20	0.67	**1.87**	0.71	0.67	**1.38**

** Only the education building required RH control*

Exhibit 15: Percent Savings in Total Energy Cost from Energy and IEQ Modifications
(Top figure in each cell is for office building; bottom figure is for education building)

	Washington			**Minneapolis**			**Miami**		
	$/f²	**From Base**	**From Prev. Stage**	**$/f²**	**From Base**	**From Prev. Stage**	**$/f²**	**From Base**	**From Prev. Stage**
Base Bldg	1.58 2.08			1.62 2.40			1.68 2.19		
Stage 1 Seasonal Reset	1.57 2.04	01% 02%	01% 02%	1.60 2.36	01% 2%	01% 2%	1.74 2.21	-02% -01%	-02% -01%
Stage 2 Lights/Off Equip	1.08 1.71	32% 18%	31% 16%	1.16 2.10	28% 13%	28% 11%	1.15 1.76	32% 20%	33% 20%
Stage 3 VSD	1.00 1.62	37% 22%	07% 05%	1.04 1.97	36% 18%	10% 6%	1.09 1.65	35% 25%	5% 6%
Stage 4 Chiller/boiler	0.87 1.35	45% 35%	13% 17%	0.90 1.65	44% 31%	13% 16%	0.93 1.31	45% 40%	15% 21%
Stage 5* OA setting with OA & RH control	0.90 1.54	43% 26%	-03% -14%	0.94 1.87	42% 22%	-04% -13%	0.97 1.38	42% 37%	-04% -05%

** Only the education building required RH control*

Exhibit 16: Energy Costs of Operational Measures that May Have Adverse Effects on IEQ

Building Parameter	Washington D.C. $/sf							Minneapolis $/sf			Miami $/sf		
	Fan	Cool	Heat	Total HVAC	Light & Off Equip	Total	% Save	Total HVAC	Total	% Save	Total HVAC	Total	% Save
Base Off. Bldg	**0.17**	**0.42**	**0.05**	**0.64**	**0.94**	**1.58**		**0.68**	**1.62**		**0.74**	**1.68**	
Day Temp. Set	0.17	0.40	0.04	0.61	0.94	1.56	**01%**	0.64)	1.58	**03%**	0.71	1.65	**02%**
Night Set Back	0.16	0.41	0.04	0.62	0.94	1.56	**01%**	0.66	1.60	**01%**	0.72	1.66	**01%**
Day HVAC Hrs.	0.17	0.42	0.04	0.63	0.94	1.57	**01%**	0.66	1.60	**01%**	0.75	1.69	**00%**
Base Edu. Bldg	**0.21**	**0.62**	**0.28**	**1.11**	**0.97**	**2.08**		**1.42**	**2.40**		**1.22**	**2.19**	
Day Temp. Set	0.18	0.55	0.22	0.95	0.97	1.93	**07%**	1.25	2.23	**07%**	1.06	2.03	**01%**
Night Set Back	0.21	0.62	0.27	1.10	0.97	2.07	**00%**	1.40	2.38	**01%**	1.22	2.19	**00%**
Day HVAC Hrs	0.20	0.61	0.25	1.06	0.97	2.02	**03%**	1.34	2.31	**04%**	1.18	2.15	**02%**

Exhibit 17: Savings from Reduced Lights and Office Equipment when Unoccupied

Operational Control	Office Building in Washington D.C.				
% of daytime use during unoccupied hours	Energy Cost ($/f ^{2})			Saving	
	HVAC	Light/off equip	Total	$/f^2	%
Stage 1					
40% lights/50% office equipment (base case)	0.71	1.08	1.79		
20% lights/30% office equipment	0.64	0.94	1.58	0.21	12%
Stage 4 (retrofitted building)					
20% lights/30% office equipment	0.33	0.57	0.90		
15%lights/20% office equipment	0.29	0.40	0.70	0.20	22%