ASSESSMENT OF POTENTIAL CARBON DIOXIDE-BASED DEMAND CONTROL VENTILATION SYSTEM PERFORMANCE IN SINGLE ZONE SYSTEMS

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Abstract

Heating, ventilation, and air conditioning (HVAC) use accounts for 43% of commercial energy consumption, with close to 5% used for ventilation purposes. Federal government agencies face both energy consumption reduction mandates and reduced funding. Carbon dioxide (CO_2) based demand control ventilation (DCV) is a technology that allows for reduced energy consumption by allowing facility designers to introduce outside air based on facility occupancy, per American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) standards. This research aims to create a generalized methodology assessing energy and cost reductions from CO2–based DCV and then apply it to a specific facility at multiple locations.

This research creates a generalized methodology for future researchers to follow based on the present body of knowledge. The model application then applies this methodology to one of the Department of Energy's (DOE) commercial benchmark facility models. The selected DOE model is a small office building with single zone HVAC air systems, assessing DCV impact on energy consumption and costs for 52 United States locations. Although the model application is not life cycle cost effective for the building modeled, it successfully identifies which areas experience the greatest cost and energy savings from DCV.

To my fiancé, who supported me across continents and time zones.

To my parents, who raised me to be the man I am today.

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First and foremost, I would like to my advisor Colonel Paul Cotellesso for his mentorship and guidance. I would also like to thank my committee members Captain James Wyatt, Dr. Al Thal, and Mr. Jeep Wedding. Their input and recommendations substantially improved the quality of my research. Thanks to Mr. Juan Lopez, who first introduced me to this topic, reigniting my interest in energy modeling.

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Joseph G. Pickenpaugh

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I. Introduction

Since the oil crisis of 1973, the United States government has placed an increased emphasis on energy conservation and independence (EIA, 2000). This event served as a warning of growing energy demand with no regard to supply. As a result, the federal government has developed several federal mandates and goals to limit the nation's energy consumption.

Executive Order (EO) 13423, enacted in 2007, requires all federal agencies to reduce energy intensity three percent each year starting in October 2008 to a total of 30 percent in October 2015 (EPA, 2012a). The 2007 Energy Independence and Security Act adopted EO 13423's energy goals, effectively making these energy reduction requirements law (EPA, 2012a). The President updated the requirements of EO 13423 with EO 13514, signed in 2009. This order requires individual government agencies to plan and track energy reduction goals and progress (EPA, 2011). The Department of Defense's (DoD) revised goal is to reduce facility energy intensity 30% by 2020, using 2003 as a baseline once again (DoD, 2012b). Additionally, this order sets a requirement that beginning in Fiscal Year (FY) 2020, all newly designed federal buildings must be able to "achieve 'net zero energy' by FY 2030" (EPA, 2011). To be "net zero" means that a building produces as much energy as it consumes over the course of a year (DOE, 2009).

In addition to meeting these federally mandated energy reduction goals, the government is also decreasing the budgets of its different agencies. The DoD is required to reduce its FY 2013 budget by $31.8 billion, specifically reducing the operations and maintenance budget by $11.2 billion (DoD, 2012a).

To meet these goals, the government must alter how it functions. Implementing new technologies and practices will be paramount for the United States government to continue to operate and be a responsible environmental steward in the future. In considering new technologies, the government should first consider technologies presenting the greatest potential energy and cost reduction, and also a wide area of implementation.

Background

Though the rest of this section frames itself around the commercial realm, the federal government (specifically the United States Air Force) is fairly representative of the commercial realm. Air Force bases have commercial facilities and industrial facilities, depending on the mission set of an installation. This similarity means that these facilities are analogous to typical U. S. building stock.

The largest element of energy consumption in the U. S. is facility operation. In commercial facilities, almost 43 percent of total energy consumption comes from heating, ventilation, and air conditioning (HVAC) requirements (DOE, 2012b). Therefore, HVAC operations are one of the first areas that should be considered for implementing new technologies for facility operation.

One of the goals for both U. S. and Air Force facilities is for HVAC systems to provide adequate outdoor air to ensure occupants avoid exposure to too many indoor air quality (IAQ) contaminants. Overexposure to IAQ contaminants can have different kinds of detrimental effects on occupants. Sick Building Syndrome (SBS) occurs when occupants are not receiving the proper amount of outdoor air (EPA, 2012b). Facility occupants with SBS suffer from symptoms like "irritation of the eyes, nose and throat; headache; stuffy nose; mental fatigue; lethargy, and skin irritation" (EPA, 2012b). SBS can lead to decreased performance at work, poor attendance, and negative attitude at work (EPA, 2012b).

Though outside air ventilation is necessary for facility occupants' health, it is an energy intensive endeavor. In 2010, the energy required to ventilate outside air was 6.1% of the overall consumed in energy in the commercial sector (DOE, 2012a). Research by the Department of Energy (DOE) calculated a 5.2% percent difference between typical commercial facilities and those facilities modeled with no minimum mechanical ventilation (Benne et al., 2009). Regardless of how calculated, the treatment and ventilation of outside air is a significant factor in energy consumption.

When following good engineering practice, the first step in properly providing outdoor air for facility occupants is to find the required outdoor air for maximum occupancy. Though not always considered, building designers have the option of employing outdoor air control systems that incorporate changes in occupancy. The American Society of Heating, Refrigeration, and Air Condition Engineers (ASHRAE) serves as the arbiter of all HVAC issues in the United States. All sectors have adopted the standards created by this organization as guidance for HVAC system design and operation. ASHRAE 62.1, the accepted ventilation standard, refers to an adjustable ventilation system as “dynamic reset” (ASHRAE, 2010b). A dynamic reset system is one which “resets outdoor air intake flow and/or space or ventilation zone airflow as operating conditions change.” One of the most common types of dynamic reset, based on changes in facility occupancy, is demand control ventilation (DCV) (ASHRAE, 2010b). ASHRAE Standard 62-1989, *Ventilation for Acceptable Indoor Air Quality*, allowed the implementation of DCV systems for the first time in 1989 (Di Giacomo, 2006).

There are several types of DCV systems. Examples include: “population counters, carbon dioxide (CO_2) sensors, timers, occupancy schedules, or occupancy sensors” (ASHRAE, 2010b). Each DCV system estimates how much outside air occupants need based on occupancy at a given time. Population counters and occupancy sensors typically employ either a contact-based sensor or motion-based sensor to monitor how many people are in a facility (Liu et al., 2012). Occupancy schedules and timers use a time-based estimate to predict how many people are in a facility at a given point in the day. CO_2-based DCV systems measure CO_2 concentration in zone air. Each of these measurement or estimates controls how much outside air an HVAC system introduces into a control zone.

Problem

Typically, commercial facilities that do not use DCV technology use a designed outdoor airflow rate based on maximum occupancy to determine outside air flow ventilation requirements. This binary form of outdoor air ventilation control provides excess outdoor air to the facility because the air flow rate is for maximum facility occupancy. HVAC equipment requires additional energy to humidify or dehumidify and heat or cool outdoor air as required based on the season and climate. Therefore, DCV systems have the potential to reduce a facility's overall energy consumption by introducing the actual amount of outdoor air needed for occupants, not the maximum potential outdoor air requirement.

Research Questions

This research's overall objective is to develop a systematic, customizable approach to compare DCV system energy efficiency or cost effectiveness to traditional facility HVAC operation. This goal leads to two overarching primary research questions:

- Can a generic model be developed to predict the performance and operation of a DCV system compared to a baseline facility?
- If so, can this model simulate the operation of a generic building at multiple geographic locations and determine which locations lend themselves to DCV system use?

Several other dimensions need to be considered to answer these primary research questions. These factors lead to the following secondary research questions:

- How do environmental factors, such as climate and facility type, influence the decision to incorporate a DCV system?
- Based on model results, how much energy and money can be saved by using a DCV system?
- Which geographic locations present the best opportunity for DCV implementation?

Methodology

To answer these research questions, a framework for research must be created. The first secondary research question can be answered by investigating the present body of knowledge of HVAC system performance, specifically DCV system performance. The second secondary research question can be answered by comparing an energy model and a cost model for a baseline facility without a DCV system to the same facility with the DCV system. Applying the energy and cost model to multiple locations across the United States will address the third secondary research question. Though analyzed locations are Air Force installations, the overall methodology is applicable to any location. Executing research in this fashion will answer the overall research questions. Figure 1 presents a visual, holistic representation of the research approach.

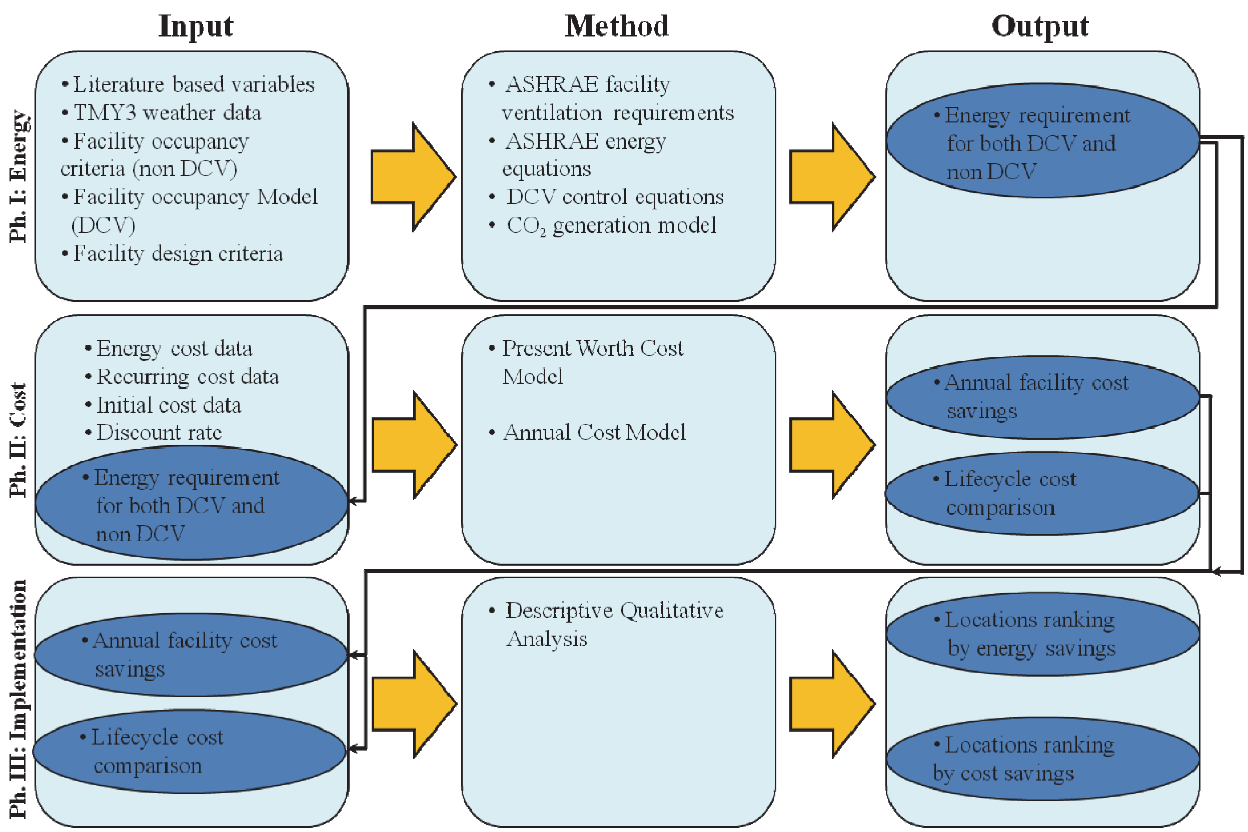

Figure 1. Overall Research Methodology

First, this energy model will compare the energy requirements to treat the ventilation air for a small office building for both the baseline small office building and for the small office building with DCV controls. This comparison between two systems can be thought of as an "energy differential," as shown in Figure 2. The energy differential is the savings in energy that can be expected by implementing a DCV system.

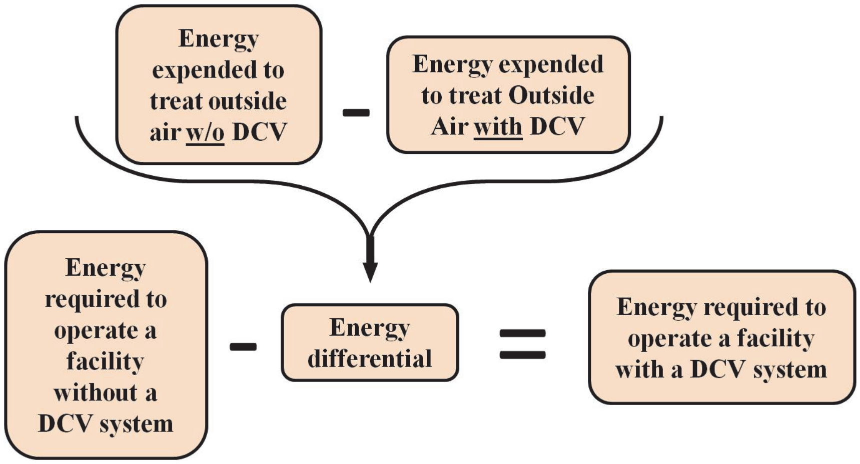

Figure 2. Energy Differential Comparison

Using the energy differential, an economic model will first evaluate the annual energy cost savings of the DCV system for each selected location. The model will then compare these savings over a 25 year period to the life cycle costs of the DCV system for each location (DOC, 2012). Finally, tabulated and rank-ordered energy and cost results will show which locations experience the greatest potential energy and cost savings.

The energy modeling and cost modeling components of this research can be generalized for all CO_2 monitoring systems. A specific type of atmospheric sensor (CO_2) will be evaluated for the life cycle cost analysis portion of the cost model. The OptiNet multipoint monitoring system is the selected sensor to be modeled. These models can be easily modified to simulate other CO_2 monitoring systems based on initial and recurring system costs.

Assumptions

This analysis requires several assumptions to be considered. Facility size and construction type will need to be assumed for the energy model. First, one of the DOE's

"Commercial Reference Building" facility models will determine facility design parameters. Facility heating and cooling of the facility using a DCV system will require further assumptions.

Second, required ventilation rates will be calculated for both the DCV and baseline cases. Assumed occupancy rates, schedules, and DCV performance will determine DCV ventilation flow rates. These calculated flow rates must ensure compliance with criteria presented in ASHRAE Standard 62.1 for the selected building type (2010). Chapter III will provide further discussion of the energy model inputs.

Third, the cost model assumes the following for each location: utility rates, utility escalation rates, a discount factor for recurring costs, and natural gas effectiveness. Information from both the National Institute of Standards and Technology (NIST) Handbook Supplement 135 and the Energy Information Administration guide these assumptions (DOC, 2011; DOC, 2012; Fuller and Petersen, 1996).

Organization

The following chapters will examine the research supporting an approach to determine the energy and economic effectiveness of a DCV system. Chapter II will explain the operation of HVAC systems and the applicability of DCV to these systems, exploring the current body of research regarding DCV systems, specifically CO_2 sensors. Chapter III will explain a methodology that can evaluate a CO_2-based control system and an application of it. Following the simulation Chapter III explains, Chapter IV will provide an analysis of the results and discuss data outputs. Finally, Chapter V will discuss the ramifications for the work presented in Chapter IV and provide recommendations regarding further implementation and research.

II. Literature Review

This chapter reviews the existing literature relating to demand control ventilation (DCV) systems and their application. First, this chapter will present different heating, ventilation, and air conditioning (HVAC) system goals, HVAC types, and includes an assessment of which HVAC systems are feasible for DCV system implementation. Second, the chapter will discuss previous research assessing DCV systems at multiple locations. Third, the chapter will compare previous research efforts modeling DCV systems at single locations. In discussing different models and systems, readers may see unit abbreviations that may be unfamiliar. Readers can refer to Appendix A for any acronyms or unit abbreviations that need clarification.

Factors to Consider in HVAC Systems

HVAC system goals should be considered when determining the best HVAC system type. Table 1 provides a list of these system goals. DCV system usage affects several of these goals. Specifically, goal four and goal six directly relate to the use of a DCV system. There are three criteria that drive the potential effectiveness of DCV systems: highly variable facility occupancy, a facility type and location with a steady heating or cooling requirement, and few non-building occupant sources of indoor air quality contaminants (Emmerich and Persily, 2001).

Table 1. HVAC System Goals (Doty and Turner, 2009)

1	Provide sensible heating to each of several spaces in the building to offset heat loss from the building envelope and to maintain thermal comfort at some desired space temperature
2	Provide humidification at the system level, or at the space level if required to maintain space relative humidity set-points
3	Provide sensible and latent cooling to each of several spaces in the building to offset heat gain from the building envelope and internal gains and maintain thermal comfort at some desired space temperature and humidity
4	Provide ventilation for each space to maintain good ventilation effectiveness for human comfort and to meet mandated ventilation needs for process, dilution, infection control, or other requirement
5	Provide pressurization control for the building to the outside elements and in some cases pressurization control of some spaces with respect to each other for safety, process, or infection control reasons
6	Provide outdoor air for the building for dilution of odors, to make up for building exhaust and to provide desired indoor air quality
7	Provide filtration of air to maintain good indoor air quality and/or to meet specific process and infection control requirements
8	Provide regulation and automated control of system components to maintain desired space temperatures as environmental and operating conditions change

HVAC Air Handling Systems

The way an HVAC system supplies and removes air plays a large part in whether or not it should be considered for DCV usage. The two primary types of air handling systems are either single zone (SZ) or multiple zone (MZ) systems (ASHRAE, 2010b). SZ and MZ systems can further be classified by the method used to treat air and whether return air is recirculated. Because of the lower energy requirement associated with recirculating zone air, most air handling systems recirculate air—those that do not recirculate return air do so because of unique outside air criteria. Laboratories or hospitals are examples of facilities that have unique outside air requirements.

Single Zone Systems

In a SZ system, the air handling system only conditions one zone. The typical design process determines the required zone air flow rate based on maximum occupancy, zone floor area, and the way the air handler distributes outside air to the zone (ASHRAE, 2010b). This outdoor air flow rate is typically the set-point a facility uses during HVAC operation. This set-point means that the air handler introduces the same amount of outside air regardless of actual occupancy. Typically, SZ systems use a constant air volume (CAV) air handling system. A CAV system supplies a constant volumetric flow rate of air; heating and cooling coils control air temperature (ASHRAE, 2008).

Multiple Zone Systems

In MZ systems, standard design calculates each zone's required outside air flow the same way as for a single zone system. For a MZ system that recirculates air, the standard design determines the overall system ventilation efficiency by comparing the smallest required zone outdoor air flow rate to the overall air handler flow rate. Calculating required system outside airflow incorporates system ventilation efficiency with the sum of all zone requirements. In an MZ system that does not recirculate air, the total outside air requirement is the sum of individual zone requirements. The calculated MZ outdoor air flow rate is typically the set-point used during facility operation. Using this flow rate means that the air handler introduces the same amount of outside air to each zone regardless of actual occupancy. (ASHRAE, 2010b)

Application of DCV Systems

DCV systems cannot presently be applied to every HVAC system type. This exclusion occurs because a DCV system cannot ensure it will supply an adequate amount of outdoor air in all cases. There are several cases of successful CO_2-based DCV implementation for SZ systems, especially for SZ CAV systems (Liu et al., 2012). There are no known cases of using CO_2-based DCV in MZ systems; no documents provide guidance for MZ DCV implementation (Liu et al., 2012). The control strategies implemented in SZ systems cannot be applied to MZ systems (Lau, 2012). These strategies cannot be used because MZ systems "receive a mixture of first-pass outdoor air and recirculated air from all zones" and the outdoor air design equations for MZ systems cannot be applied to any non-maximum loading (Lau, 2012). In order for a DCV system to be used in a MZ system, the MZ system's air handling system would need to be able to ensure that each zone receives enough outside air flow in addition to meeting additional HVAC requirements. For this reason, CO_2-based DCV system case studies focus on SZ systems.

To control the outdoor airflow to the zone, control algorithms must be employed that tie outdoor airflow to conditions within the zone. In the current body of knowledge, the only known comparison of established CO_2-based DCV controls found is the work by Schell et al. (1998). According to Schell et al. (1998), there are three primary types of CO_2-based DCV control algorithms—set-point control, proportional control, and exponential control. Each of these control schemes will require less energy than the typical system design because DCV controls do not operate at maximum load conditions at all times (Schell et al., 1998).

Schell et al. (1998) discusses the operation of a set-point control scheme in DCV systems. The set-point control method is the simplest of the three control methods. In the set-point control method, intake flow increases whenever the CO_2 sensors in the zone achieve a certain threshold. The damper closes to its initial starting point once the CO_2 concentration in the zone reaches another activation level. This strategy is not suitable for most design conditions; however, one of the few instances that it may be useful is if a zone reaches peak design occupancy quickly. For most other occupancy densities and types, the lag time for proper zone dilution is unacceptable.

Schell et al. (1998) also discusses the proportional control method, which operates under a different logic type. Instead of allowing a set amount of outdoor airflow only once CO_2 concentration meets a certain threshold, the DCV system allows a varying amount of outdoor air in response to how high the zone CO_2 concentration is above a lower threshold. This control type works well for a broad array of occupancy densities and schedules.

The final control method Schell et al. (1998) discusses is the exponential control method, which is similar to the proportional control method. It is a broad category of control that also weighs the rate at which CO_2 concentration changes. This control type lends itself to minimal occupancy densities or large densities with considerable air volumes.

Demand Control Ventilation Models for Multiple Locations

By allowing HVAC designers the latitude to modulate outdoor air flow based on DCV control schemes, new technologies can be developed to more effectively predict or monitor the required amount of outdoor air. To ensure that these technologies are embraced and employed, models that show potential energy savings and financial benefits need to be developed. The following discussion presents two case studies of models that use two different approaches to evaluate DCV effectiveness for several climate zones. This section presents only two case studies because there are no other known case studies found for DCV implementation at multiple locations; neither case study found was for CO_2-based DCV.

DCV Economic Comparison Case Study

The California Energy Commission (CEC) funded a 2009 DCV assessment that models a medium-sized office building based on the Department of Energy (DOE) commercial reference building. This office building uses an MZ variable air volume (VAV) air handling system. This assessment also included the prescriptive requirements set by the CEC's energy efficiency standard for nonresidential buildings. Hong and Fisk (2009) model the office building in five different climates of California for five different cases, three with DCV and two without DCV.

Hong and Fisk (2009) use three different design occupancies for the DCV cases. They then calculate the required outdoor air flow rate as the greater between either 0.76 $L/s/m^2$ or 8.3 L/s/person multiplied by design occupancy and the occupant schedule percentage. Following their calculations, they establish the three design occupancies by halving, keeping the original value, and doubling the design occupancy used in the DOE commercial reference building data. The occupancy schedule used is from the CEC's efficiency standards. Hong and Fisk (2009) calculate outdoor air flow rate per person using an assumed CO_2 generation rate per person and a maximum outdoor air differential of 600 ppm CO_2.

For the cases without DCV, Hong and Fisk (2009) use two different ventilation rates (13.2 L/s/person or 38.2 L/s/person). Hong and Fisk (2009) choose these two flow rates because they are representative of U. S. buildings—even though they are both above the minimum required flow rate. These two flow rates are based on a previously conducted survey of 100 office buildings in the United States.

Hong and Fisk (2009) compare each of the DCV cases to each of the non-DCV cases for all five California locations to assess energy consumption. They compute energy consumption differences using the DOE energy modeling program EnergyPlus. A cost comparison between the operating costs of the two systems and system costs of a DCV system uses these energy consumption rates. (Hong and Fisk, 2009)

Overall, DCV systems are not cost effective unless comparing the higher assumed flow rate (38.2 L/s/person) to DCV systems or in cases where the climate is more severe. The authors concede that there is a large uncertainty in the base case ventilation rates, which makes it difficult to generalize results. (Hong and Fisk, 2009)

Hong and Fisk's (2009) research effort presents several engaging points in modeling DCV system performance. First, using an assumed occupancy schedule and occupancy density allows a designer to estimate facility population at different times of facility operation. Second, the researchers use EnergyPlus to model their energy requirements. Third, the researchers model a DCV system in a medium-sized office building with a MZ VAV air handling system. Based on the work of Liu et al. (2012) and Lau (2012), modeling a VAV system with DCV presents a limitation in Hong and Fisk's research (2012). The research by Liu et al. (2012) and Lau (2012) both show that SZ systems can successfully use DCV; they also show that DCV use in MZ systems is not yet adequately developed. Fourth, Hong and Fisk's (2009) research shows the application of a DOE commercial reference building model across several different climate zones.

DCV Monte Carlo Simulation

In the previous case study, Hong and Fisk (2009) use DOE energy modeling software and a DOE commercial reference building to compare DCV usage at five different locations. The second case study compares six different ventilation strategies in six different cities, comparing concentrations of eight different groups of contaminants and indoor air processes in a 2,000 iteration per season Monte Carlo simulation. CO_2 concentration was one of the eight types of contaminants modeled for multiple ventilation strategies, one of which was strictly a DCV system. The zone modeled in the case study is a 92.5 m^2 floor area with a 2.4 m ceiling height (Rackes and Waring, 2013).

Rackes and Waring (2013) make several assumptions to simulate facility and environmental factors. The model does not use economizer cycling, a technology that allows the use of untreated outside air to meet HVAC needs. Rackes and Waring (2013) model maximum infiltration as a lognormal distribution, model building height probabilistically, and model infiltration fraction using a beta distribution.

Rackes and Waring (2013) model DCV outdoor air flow rate based on an area factor added to an "actual zone population" based factor. They model actual zone population using a binomial distribution. The model uses a lognormal distribution to simulate maximum design population. The baseline case uses the design population to model outside air flow, which is the typical outdoor air flow rate area factor added to a maximum design population-based factor. Figure 3 shows the resulting CO_2

concentration for the summer Philadelphia office iterations. The figure shows similar zone CO_2 concentrations for both types of systems.

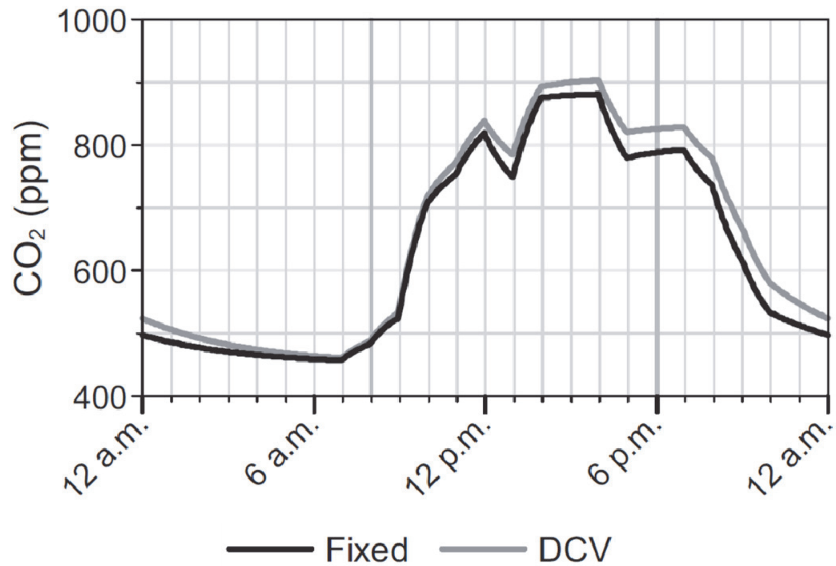

Figure 3. Summer Philadelphia Office Space Indoor CO_2 Concentration (Rackes and Waring, 2013)

Table 2 shows overall model results for CO_2 concentration in the office space for all simulations. The table shows slight deviation in CO_2 concentration between the standard design and the DCV design. These results are for all locations and seasons modeled.

Table 2. Resulting CO_2 Concentration for DCV and Typical Design (Rackes and Waring, 2013)

Summary of daytime mean and peak values under fixed ventilation and DCV, as well as the percent change, which is defined as DCV concentration minus the baseline concentration, divided by the baseline concentration × 100.

City	Fixed VRP			DCV			% change		
	10th	50th	90th	10th	50th	90th	10th	50th	90th
Mean CO_2 (ppm)									
Ave.	605	752	963	611	772	1024	1	3	7
Peak CO_2 (ppm)									
Ave.	685	880	1157	688	892	1194	0	1	4

Though Rackes and Waring's (2013) model uses Monte Carlo simulations to simulate various office parameters, the overall DCV strategy is uncomplicated. This simplicity stems from the assumption that the DCV system will directly follow the ASHRAE-published occupancy parameters (Rackes and Waring, 2013; ASHRAE, 2010b). A shortcoming of Rackes and Waring's research is their model's need for expert opinion to determine model input distribution type and distribution parameters.

DCV Energy Assessment

The final CO2-based DCV case study examines six different facility types, shown in Table 3, for six different locations and employing several different control strategies (Persily et Al., 2004). Persily et al. (2004) model these six facility types as being a large single zone in the airflow and dispersal modeling software CONTAMW. The model uses a constant air infiltration rate of 0.1 ach for each space—even during HVAC system use.

Table 3. Space Characteristics (Persily et al., 2004)

Space Type	**Floor Area**	**Ceiling Height**	**Design Occupancy**	**Occupancy Density**	**Operating Times**
	(m^2)	**(m)**	**(people)**	**(people/m^2)**	
Office	1000	3.0	70	7.0	0600-1900
Conference Room	100	3.0	50	50.0	0600-1800
Lecture Hall	100	6.0	150	150.0	0800-2100
Classroom	100	3.0	35	35.0	0600-1800
Portable Classroom	89	2.6	20	22.5	0700-1700
Fast Food Restaurant	125	5.4	70	56.0	0600-2400

Persily et al. (2004) use a corresponding occupancy schedule for each zone type. Each schedule is dependent upon the facility type. Also, each zone has a schedule for weekdays and weekends. Persily et al. (2004) use these schedules in conjunction with the occupancy density and a calculated contaminant generation rate. Contaminant generation rate will in turn determine how much outside air DCV control schemes will allow.

Persily et al. (2004) modeled eight different outside air control strategies, four of which are CO_2-based DCV. Each of the DCV control schemes follow the proportional control scheme introduced by Schell et al. (1998) with different bounds. Table 4 summarizes the bounds of each DCV control scheme. Figure 4 shows airflow results for a weekday at the office building modeled. Each of the DCV control schemes follows a similar trend; each is well below the prescriptive requirements of ASHRAE 62-2001. Figure 5 shows the corresponding CO_2 concentrations for a weekday at the modeled office building. The CO_2 concentrations for all modeled control schemes is well within typical design parameters.

Table 4. Various Control Schemes

Name	**Basis**	**Minimum Flow Rate**	**Maximum Flow Rate**
C-ZeroMin	ASHRAE 62-2001	0	Max. Occupancy
C-25%Min	ASHRAE 62-2001	25% of maximum	Max. Occupancy
C-62nAreaMin	ASHRAE 62-2001 Addendum	Per Person Requirement from 62-2001 addendum	Max. Occupancy
C-T24	California Title 24	0.76 L/(s*m^2)	7.1 L/(s*person)

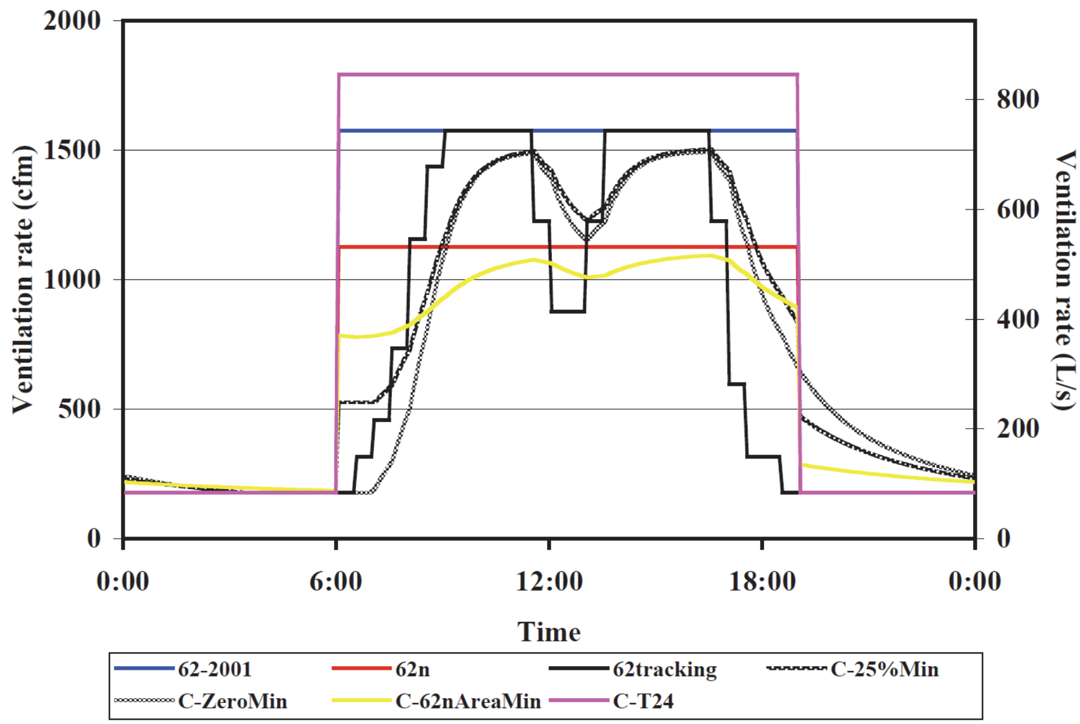

Figure 4. Weekday Office Outside Air Flow Rates (Persily et al., 2004)

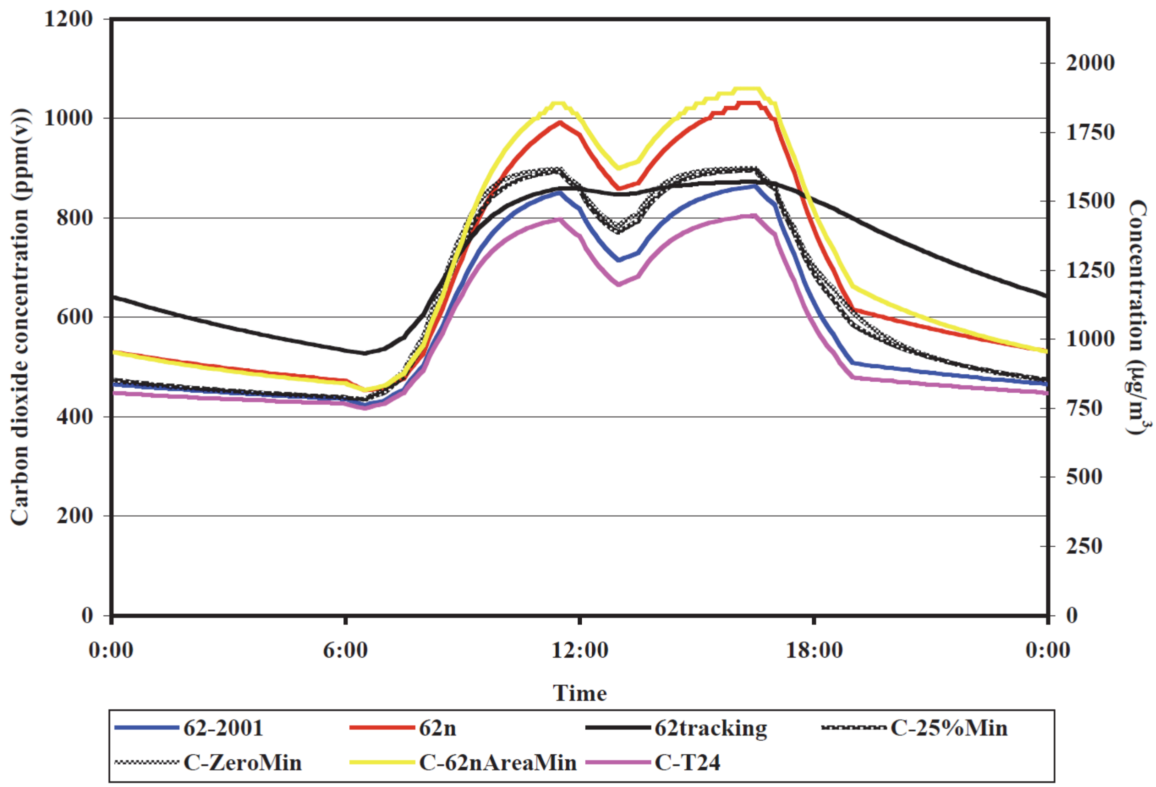

Figure 5. Weekday Office Outside Air Flow Rates (Persily et al., 2004)

The model calculates the required flow rate for each control scheme over the course of a week. Persily et al. (2004) extrapolate the results over a year, and use climatic data for six different location and heating, cooling, and latent load equations to find energy requirements between DCV systems. Specifically for the calculation of latent loads, Persily et al. (2004) assumed that each facility's HVAC system was capable of maintaining 60% relative humidity in the zone. This assumption means that every time outside air was above 60% relative humidity, the HVAC system removed enough moisture to bring it to indoor set-point. Table 5 shows the office building's annual energy requirement from treating outside air for all modeled locations.

Table 5. Annual Office Energy Load (Persily et al., 2004)

	Annual Energy Load due to Ventilation (MJ/m^2)					
Control Strategy	Bakersfield	Los Angeles	Sacramento	San Francisco	Miami	Minneapolis
62/2001	30	6	18	1	117	63
62tracking	24	5	15	1	85	34
C-ZeroMin	26	5	17	1	87	35
C-25%Min	27	5	17	1	93	37
62n	20	4	12	1	79	18
C-62nAreaMin	19	4	12	1	71	13
C-Title24	35	7	21	1	135	94

The research effort by Persily et al. (2004) provides insight into how different control schemes affect CO_2 concentrations and energy requirements. This research also provides several strengths. First, it uses an established control scheme introduced by Schell et al. (1998). Secondly, it uses documented energy relationships to determine how outside air flow drives energy consumption. A limitation of this research is that it uses annual heating and cooling degree day climate information. Using annual degree day information does not incorporate fluctuating temperature's impact throughout daily DCV operation.

Carbon Dioxide Sensor System Modeling Case Studies

One of the most commonly used methods for implementing DCV systems is to use some form of monitoring to track data for a building. Typically these systems will measure an indoor air contaminant—usually CO_2. The typical goal for ensuring HVAC systems supply enough outdoor air to the zone is 700 parts per million (ppm) CO_2 above ambient outdoor conditions, with a typical ambient outdoor condition ranging between 300 ppm to 500 ppm CO_2 (ASHRAE, 2010b). This set-point stems from the idea that CO_2 concentration can be correlated to the presence of bioeffluents in a facility (ASHRAE, 2010b). The only American regulation for CO_2 concentration is to ensure that a work zone does not exposure workers to more than 5,000 ppm CO_2 during an eight hour time weighted average (OSHA, 2006).

For the purposes of this research, the indoor air quality monitoring system selected will only measure CO_2 and only model single zone air distribution systems. Though research modeling MZ DCV exists, work by Lau (2012) and Liu et al. (2012) present that MZ VAV needs more development. The following discussion presents the only known recent case studies for SZ CO_2–based DCV modeling.

Carbon Dioxide Sensor Modeling Case Study 1

The first case study examines the ventilation requirements of a 400 m^3 space in a multiuse facility in South Korea using two types of DCV systems—one CO_2-based and the other uses a radio frequency identification (RFID) device to detect zone occupancy

(Jeong et al., 2010). A dedicated outdoor air system supplies ventilation air to the zone. The assumed maximum occupant density is 40 people per 100 m^2. The model does not include infiltration because Jeong et al. (2010) assume the zone is an interior space. They also assume that the ambient outdoor CO_2 concentration is 300 ppm. The model assumes occupants' CO_2 generation rate is 0.0003 m^3/min. The model also assumes a daily occupancy as shown in Figure 6.

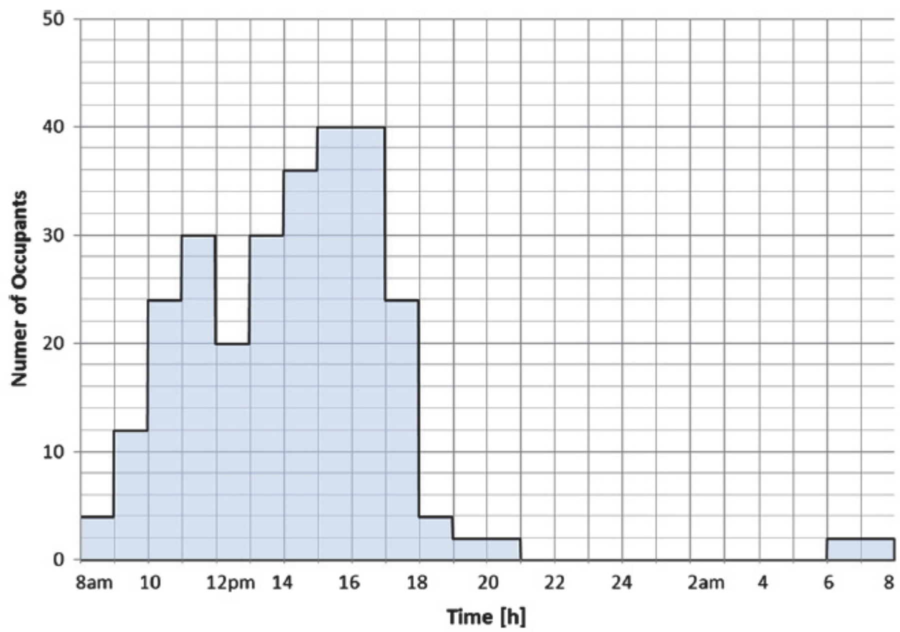

Figure 6. Modeled Zone Occupancy (Jeong et al., 2010)

Using ASHRAE-defined required flow rates, Jeong et al. (2010) adds a zone area-based factor to a zone population-based factor for each model time step to find the supply air flow rate (ASHRAE, 2010b). Figure 7 shows the required ventilation air flow rate for a typical day for both CO_2-based DCV and the RFID-based DCV. The figure shows that CO_2-based DCV flow rates lag behind the RFID-based DCV flow rates.

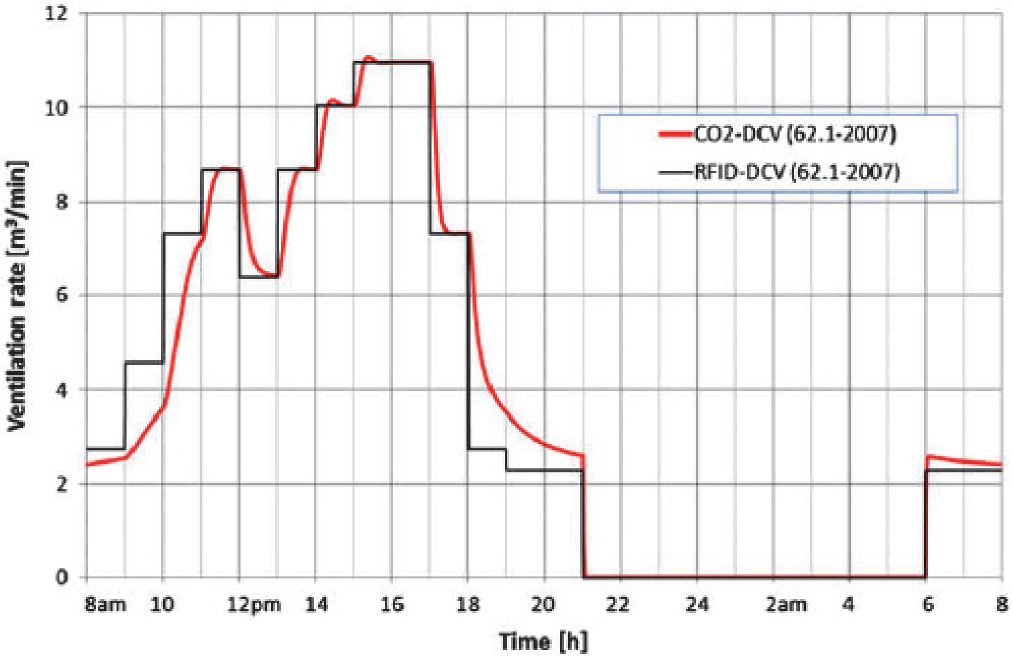

Figure 7. Calculated Outdoor air Ventilation Rate (Jeong et al., 2010)

Finally, Jeong et al. (2010) solve for CO_2 concentration in the zone for each time step using the original CO_2 mass balance equation. Figure 8 presents the corresponding CO_2 concentrations for a typical building day for both CO_2-based DCV and RFID DCV systems. This figure shows that the lagging flow rate between CO_2-based DCV and RFID-based DCV can lead to either system having a higher CO_2 concentration depending on time of day.

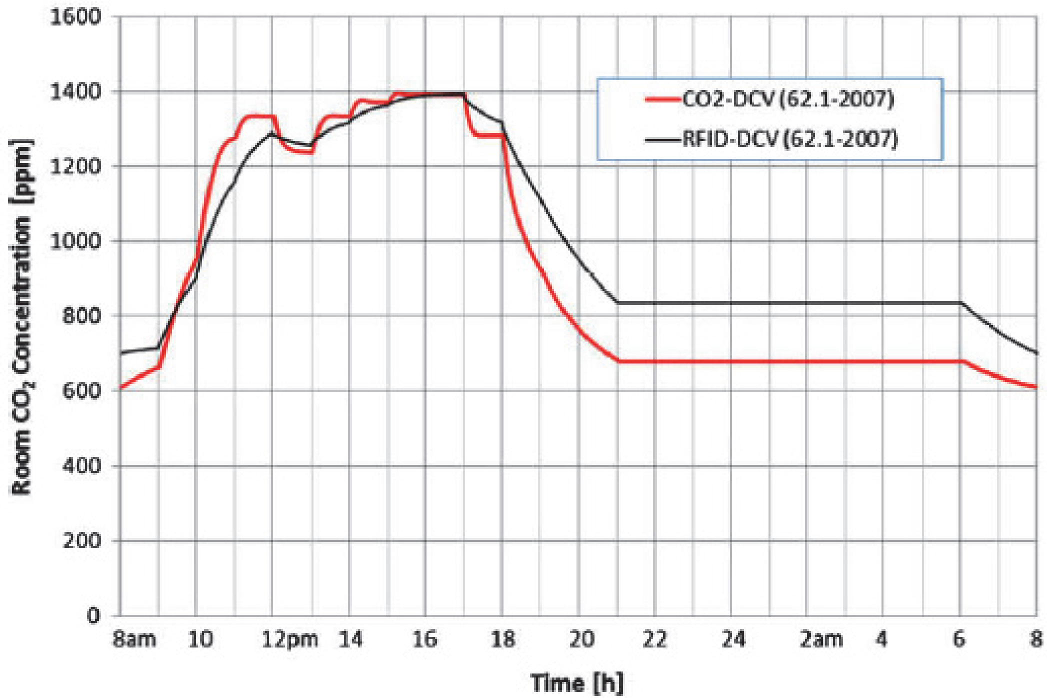

Figure 8. Calculated Zone CO_2 Concentration (Jeong et al., 2010)

Jeong et al. (2010) present a different methodology for determining required outside air flow. They start by calculating the CO_2 concentration based on a CO_2 mass balance equation. They then use a CO_2 concentration as a means of estimating zone population and use ASHRAE per person factors to determine the required outside air flow rate (Jeong et al., 2010). A limitation of this model is that it does not ventilate the room to a typical design room CO_2 concentration nor does it use a known control scheme (ASHRAE, 2010b).

Carbon Dioxide Sensor Modeling Case Study 2

While the first case study examines a single multiuse zone, the second case study examines the ventilation requirements for an elementary school gymnasium in Indiana. The gymnasium uses a SZ CAV system to provide outdoor air to the zone occupants. Ng et al. (2011) model the facility using real world data taken in the facility for 42 days; including CO_2 concentration, temperature, and relative humidity. Using a CO_2 concentration mass balance, Ng et al. (2011) create a measured occupancy profile. Figure 9 displays the results for August 17th of the study. The figure shows the occupancy schedule based on the CO_2 concentration and CO_2 mass balance agree fairly well with the actual, counted occupancy profile.

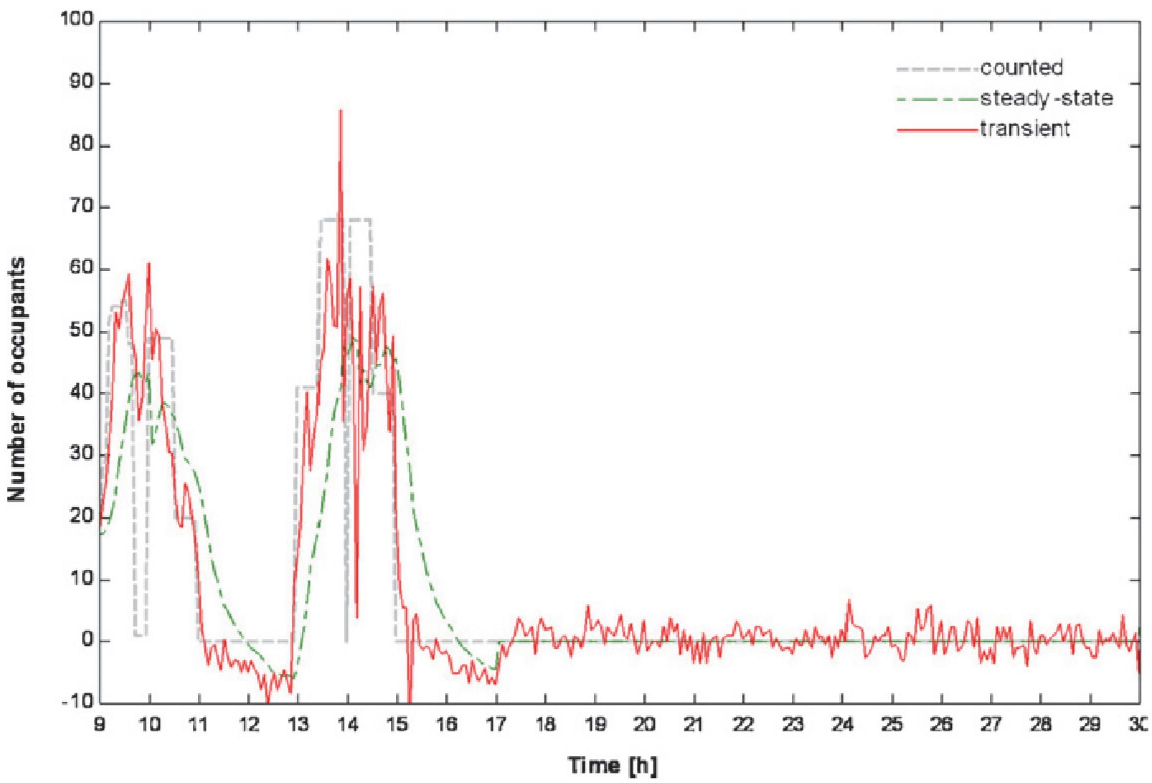

Figure 9. Calculated Occupancy Profile (Ng et al., 2011)

The study compares four different strategies for controlling outdoor air flow. These strategies include two different styles of occupancy detection, CO_2-based proportional control and a fixed five percent airflow. Proportional controls allow the outdoor air flow to fluctuate between a preset minimum to a design maximum based on an upper and lower limit of operation. Ng et al. (2011) calculate the maximum outdoor air flow rate based on an assumed maximum occupancy of 200 people. Figure 10 presents the predicted ventilation rates for eight hours on August 16th of the simulation. The proportional control scheme provides a higher ventilation rate than the fixed five percent operation, but typically less than the other two types of control schemes.

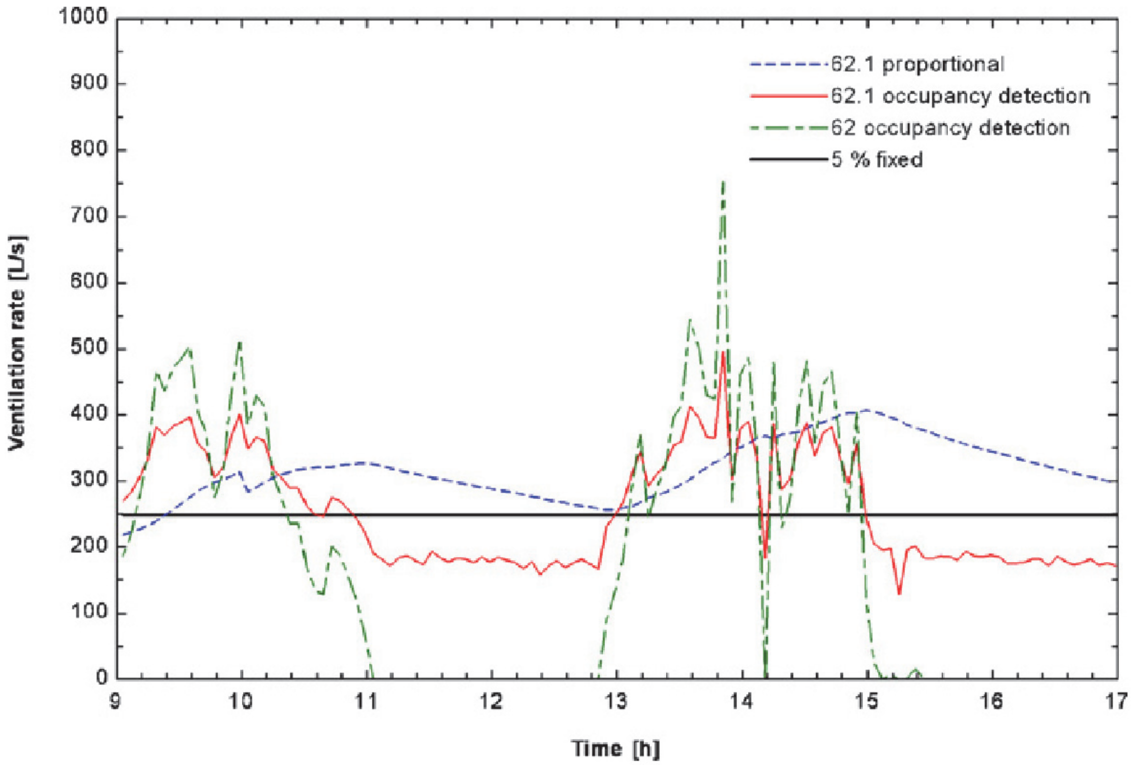

Figure 10. Resulting Air Flows in Different DCV Implementation Techniques (Ng et al., 2011)

Additionally, Figure 11 shows the resulting CO_2 concentration for the same eight hour window on August 17th of the simulation. This figure shows that all modeled control schemes keep the CO2 concentration in the gymnasium well below 1000 ppm. Of the four modeled control schemes, the proportional control scheme keeps the CO_2 concentration lower than the other three for almost all times of the day.

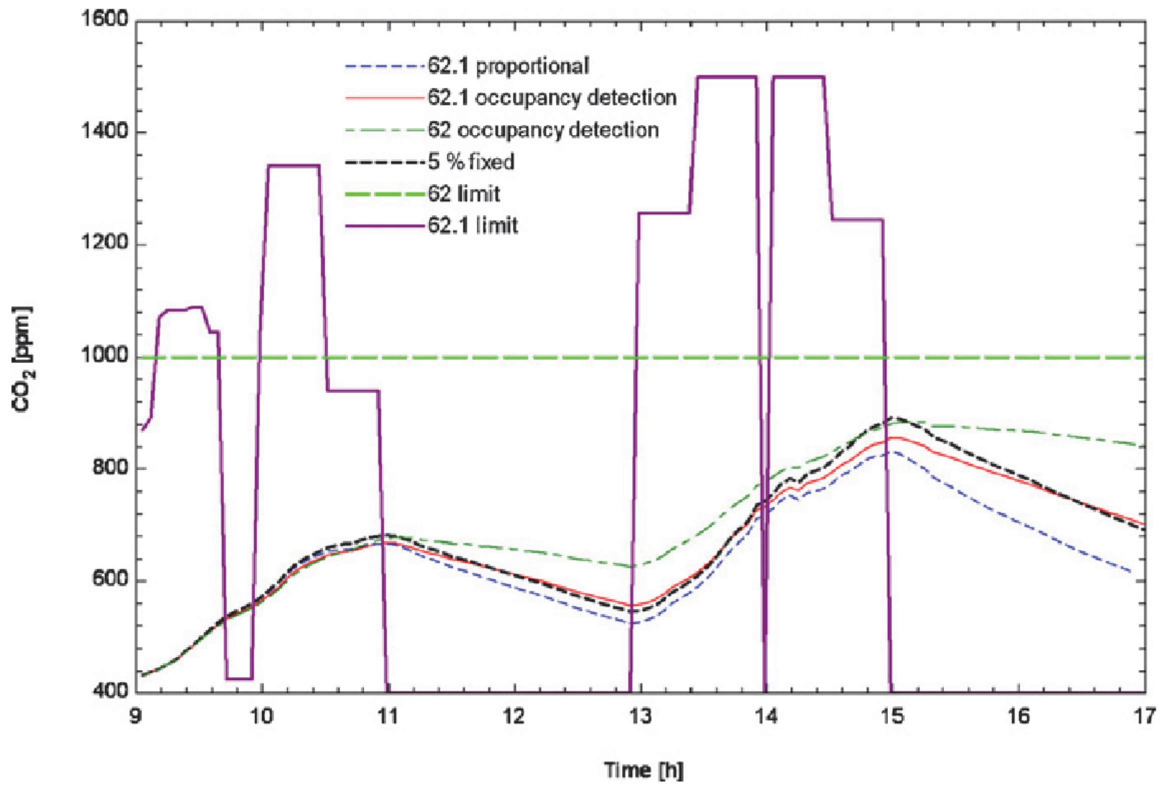

Figure 11. Resulting CO_2 Concentration in Different DCV Implementation Techniques (Ng et al., 2011)

Ng et al. (2011) use a CO_2 mass balance to predict CO_2 concentration. It additionally uses a variety of control schemes to determine outdoor air flow rates, specifically for CO_2-based sensing (Ng et al., 2011). Ng et al. use real world measured data to predict a CO_2-based DCV system using an established control scheme. Model results prove to be compliant with the requirements of ASHRAE 62.1. The shortcoming of this model is that it uses only 42 days of data.

Conclusion

This chapter shows readers that SZ air handling systems are currently the best candidates for CO_2-based DCV systems. This chapter also exposes readers to several ways to predict facility occupancy, estimate required outside air flow using a DCV system, and predict how the DCV system will control outside air flow. Each of these case studies had strengths. Some case studies can easily be reproduced. One case study used real world data and used established control practices. Additionally, each examined case study had limitations. Some case studies did not meet standard design criteria. Some case studies applied DCV technology to systems that have not been shown to be able to support DCV technology. Some case studies are not easy to recreate.

The next chapter will present a methodology that is easily reproducible, follows typical design criteria, and follows an accepted control scheme. It will integrate a select DCV model, occupancy schedule, and a cost model with initial and recurring costs of a select CO_2 monitoring system. It will also justify each selection regarding model creation.

III. Methodology

This chapter develops and justifies the selected methodology for the research effort. The research framework divides this effort into three phases, each directly related to answering the research questions posed in Chapter I. Phase I of research is to use an energy model to compare the energy effectiveness of a demand control ventilation (DCV) system to a baseline facility that does not use DCV controls. Phase II will incorporate energy prices and escalation factors with the energy consumption data found in Phase I. Phase II will then determine the cost savings of implementing carbon dioxide (CO_2) concentration-based DCV controls and the feasibility of a multipoint monitoring system (a specific CO_2 sensor type). Phase III is to compile the results of Phase I and Phase II for all locations chosen for analysis. The remainder of this chapter further details the three areas of research and explains the rationale for each area of research execution. Readers will be presented with a generalized model, followed by a specific application of the model to a selected commercial facility. Figure 12 presents the overall energy and cost model. The numbered components of the model shown in Figure 12 are further explained in the following same-numbered subsections for the generalized model and model application.

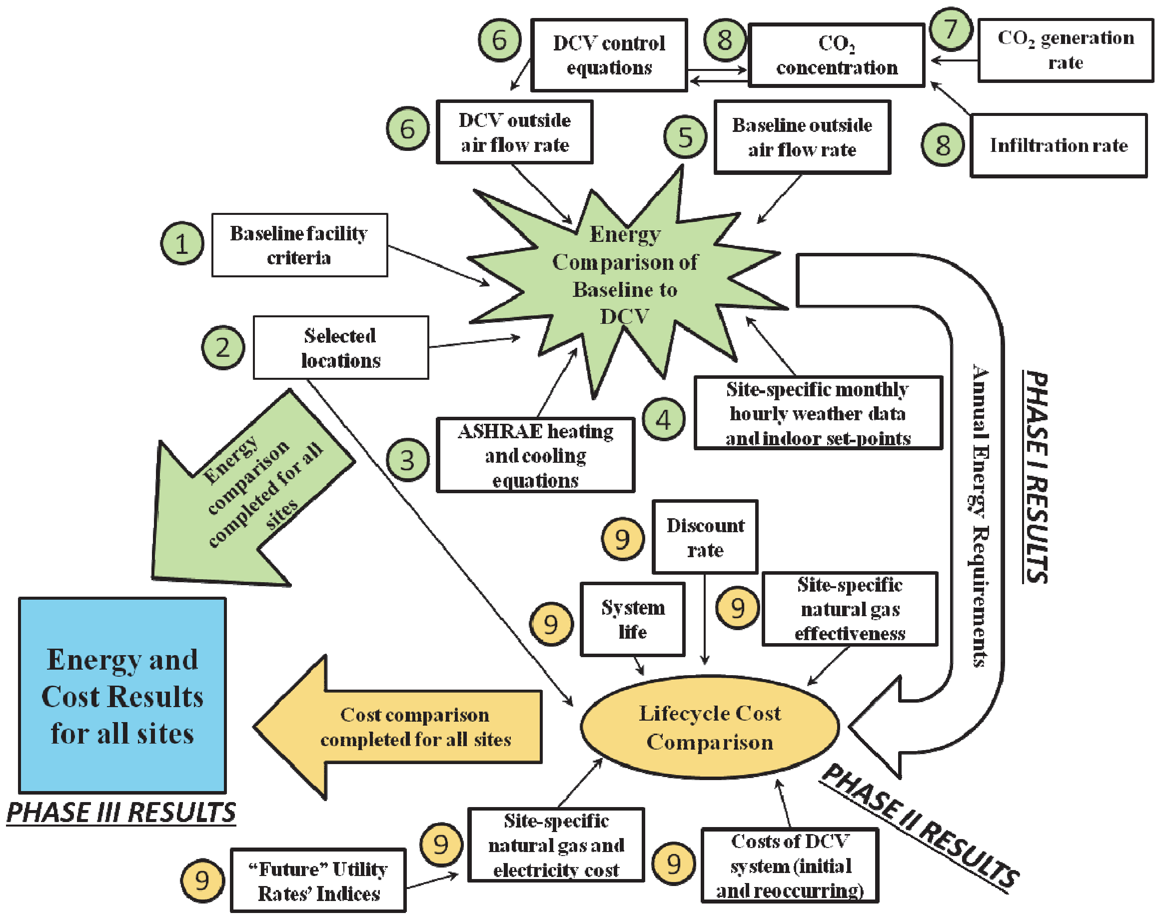

Figure 12. Energy and Cost Model

1. Generalized Model: Baseline Facility Energy Consumption

Metered data is highly desirable when determining how much energy a facility requires in meeting heating, cooling, and ventilation needs. It gives decisions makers an energy consumption value to compare DCV energy savings. In cases where either raw data is unavailable or the facility does not physically exist, energy modeling serves to estimate energy consumption and compare energy consumption differentials between system types. Table 6 provides a list of factors that affect HVAC energy consumption.

Table 6. Factors that Affect Total Energy Consumed by HVAC Systems (Doty and Turner, 2009)

Climate
Type and efficiency of building envelope
Amount of internal heat gain requiring cooling
Amount of fresh air which must be introduced to the spaces in the building to meet code, good practice, or exhaust requirements
Amount of minimum air changes required for good indoor air quality and ventilation effectiveness
Requirement for simultaneous heating and cooling
Requirement for humidification
Space temperature and humidity requirements for heating and cooling
Types of HVAC systems selected to serve the building loads
Hours of operation of the systems
Actual occupied hours of the building spaces
Mechanical equipment efficiencies
Distribution energy requirements
System thermal losses
Equipment condition, including cleanliness of heat transfer elements, duct leaks, etc.

1. Model Application: Baseline Facility Energy Consumption

The application of this generalized model examines the DOE commercial reference facilities, using many of their design parameters. The DOE commercial reference facilities are representative of over 60 percent of the commercial building floor space in the United States (Deru et al., 2011). To represent various types of structures, the DOE developed 16 different models (DOE, 2012d). Table 7 lists a summary of the representative facility types and their corresponding HVAC system.

Table 7. DOE Commercial Reference Facility Types (DOE, 2012d)

Building Type	Heating	Cooling	Air Distribution
Small Office	Furnace	PACU (packaged air-conditioning unit)	SZ CAV (single-zone constant air volume)
Medium Office	Furnace	PACU	MZ VAV (multizone variable air volume)
Large Office	Boiler	Chiller (2) – water cooled	MZ VAV
Primary School	Boiler	PACU	CAV
Secondary School	Boiler	Chiller – air cooled	MZ VAV
Stand-Alone Retail	Furnace	PACU	SZ CAV
Strip Mall	Furnace	PACU	SZ CAV
Supermarket	Furnace	PACU	CAV
Quick Service Restaurant	Furnace	PACU	SZ CAV
Full Service Restaurant	Furnace	PACU	SZ CAV
Small Hotel	ISH (individual space heater), furnace	IRAC (individual room air conditioner), PACU	SZ CAV
Large Hotel	Boiler	Chiller (2) – air cooled	FCU (fan coil unit) and VAV*
Hospital	Boiler	Chiller – water cooled	CAV and VAV**
Outpatient Healthcare	Furnace	PACU	CAV and VAV**
Warehouse	ISH, furnace	PACU	SZ CAV
Midrise Apartment	Furnace	PACU-SS (split system)	SZ CAV

* Hotels may be characterized with two system types serving different areas. Both multizone systems (VAV or CAV) may serve public spaces (lobby/conference rooms), whereas single zone fan coil systems may be common for living areas.

** Hospitals may use CAV systems in some operating and critical care type areas with variable air flow used for pressurization, but classic VAV multizone systems in areas such as offices. CBECS buildings reporting VAV are significantly less common in pre-1980 construction (67% versus 95% in post-1980 hospitals).

The DOE models these reference facilities for 16 different climate regions, using the most populous city in each climate zone as the reference city (Deru et al., 2011). Researchers estimate that slightly less than 80% of the total U. S. population occupies five of these climate zones (Deru et al., 2011). Table 8 lists a summary of the representative city and its corresponding International Energy Conservation Code (IECC) climate region and DOE Building America climate region (DOE, 2012d; Baechler et al., 2010). Figure 13 shows how each county in the United States aligns with its corresponding IECC climate region.

Table 8. DOE Commercial Reference Facility Climate Zones and Representative U. S. City (DOE 2012d)

Location	**IECC**	**Building America**
Miami, FL	1A	Hot-Humid
Houston, TX	2A	Hot-Humid
Phoenix, AZ	2B	Hot-Dry
Atlanta, GA	3A	Mixed-Humid
Los Angeles, CA	3B-CA	Hot-Dry
Las Vegas, NV	3B	Hot-Dry
San Francisco, CA	3C	Marine
Baltimore, MD	4A	Mixed-Humid
Albuquerque, NM	4B	Mixed-Dry
Seattle, WA	4C	Marine
Chicago, IL	5A	Cold
Boulder, CO	5B	Cold
Minneapolis, MN	6A	Cold
Helena, MT	6B	Cold
Duluth, MN	7	Very Cold
Fairbanks, AL	8	Subarctic

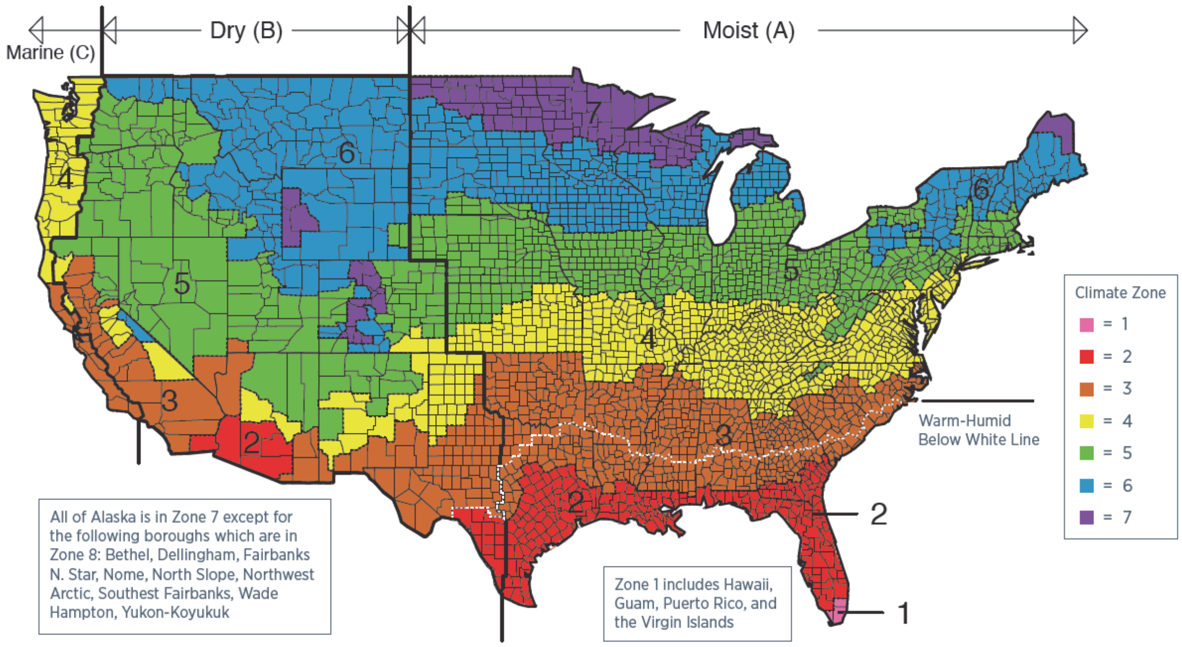

Figure 13. Nationwide IECC Classification. (Baechler et al., 2010)

The DOE designs each commercial reference facility to meet the minimum facility energy design requirements for each different climate region in ASHRAE 90.1-2004, the energy standard for all non-residential commercial facilities (DOE, 2012d). These facility models' energy intensity is within 12 percent of 4,820 measured commercial buildings (Griffith et al., 2008).

The small office building is chosen as a basis for modeling because this DOE commercial reference facility type uses a single zone (SZ) constant air volume (CAV) system (Deru et al., 2011). Liu et al. (2012) and Lau (2012) are skeptical of the use of DCV systems in multizone (MZ) air handling systems. Liu et al. (2012) and Lau (2012) state that SZ systems are the most viable candidates for DCV system usage. Many of the other common DOE commercial reference facility types employ variable air volume systems.

Additionally, the small office building is chosen because it is a common facility type at almost every United States Air Force base. Office space accounts for over 64.6 million square feet of Air Force property (AFCEC Real Estate Transactions, 2013). Small offices, defined as less than or equal to 5,000 square feet in footprint, account for 3.4 million square feet of Air Force property (AFCEC Real Estate Transactions, 2013). The small office building is a 511 m^2 office space with five different HVAC zones, one core and four perimeter zones (Deru et al., 2011). Figure 14 shows a summary of the facility zones' size (DOE, 2012d).

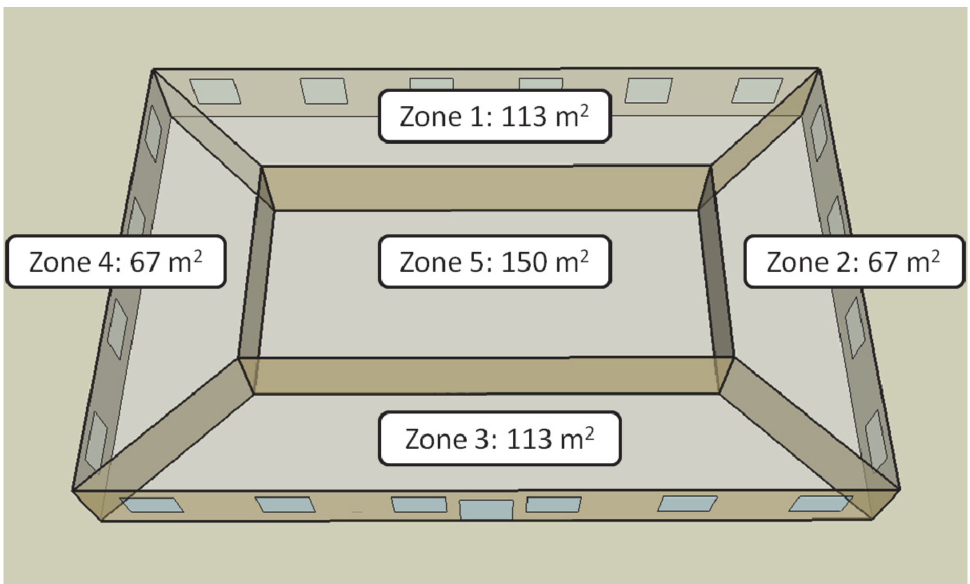

Figure 14. DOE Reference Facility Small Office Zones (DOE, 2012d)

An individual CAV system serves each zone (Deru et al., 2011). A natural gas furnace heats the facility, and an electric packaged air conditioning unit (PACU) cools the facility (Deru et al., 2011). This model makes a crucial assumption regarding the operation of the PACU. In the DOE small office commercial reference building model, the PACU does not actively control humidity; this model does allow for the control of humidity (Deru, 2013). Meeting dry-bulb temperature set-point within the zone is the only controlling factor for the DOE model's PACU (Deru, 2013).

Even though the DOE's PACU does not actively control zone humidity, it does remove moisture from outside air. Cooling coils (as with the ones in the DOE's PACU) remove both sensible energy and latent energy from the outside air when the coils' surface temperature is below the dew point of the air passing over the coils (ASHRAE, 2008). PACU cooling coils are sized according to design criteria—typically extreme cooling day dry-bulb temperature and extreme moisture content. In the case of the DOE's PACU, system performance was generalized for a typical PACU—cooling coil performance is system-specific (Deru, 2013). The researchers used system performance curves for the PACU and had to make assumptions about latent energy performance (Deru, 2013).

In the case of this research, it is assumed that the cooling coils are designed to accommodate latent cooling capacity needed for extreme humidity conditions. This research's PACU, similar to the DOE's PACU, only controls zone conditions based on meeting a dry-bulb set-point within the zone. Even though it will not actively control humidity, the model's PACU will remove moisture any time that the moisture content of outside air is greater that the internal set-point humidity and the dry-bulb temperature of the outside air is above the temperature of the cooling coils. This assumption is the same one made by Persily et al. (2004). By choosing the latent cooling capacity of the PACU coils based on extreme humidity conditions, the assumed PACU will remove enough moisture from the air to meet the building's internal set-points. Building set-points are described later in this chapter.

Additionally, the small office building simulation does not include an economizer, a technology that uses outside air for heating or cooling purposes (Deru et al., 2011; ASHRAE, 2005). This leads to the assumption that the Air Force facility does not have an economizer, which may or may not be a good assumption. When designing Air Force facilities, economizer technologies must always be considered (DoD, 2010). In areas where the wet-bulb temperature is 19 degrees Celsius or higher for over 3,000 hours annually or 23 degrees or higher for 1,500 hours annually, economizers will not typically be used (DoD, 2010). This chapter will state further assumptions in the "model application" subsections as they become relevant.

2. Model Application: Selected Bases for Analysis

Initially, all major air force bases were considered for analysis. Weather data should accurately reflect location conditions. For this reason, some areas are removed from analysis due to insufficient weather data. Bases are eliminated if they are within a 30 mile proximity to another base. Also, bases without historical weather data within a 30 mile proximity are eliminated.

After determining the availability of weather data, each base receives an IECC climate calculation based on the location of its county or multiple counties (Baechler et al., 2010). Base climate zones are then compared to the corresponding DOE reference facility's IECC climate zone classification (DOE, 2012e). Because each facility model is designed for its specific climate zone, bases without an equivalent DOE Commercial Reference Facility model are eliminated (DOE, 2012e). This stipulation leads to the elimination of any base within the Zone 3, Hot-Humid climate zone. A lack of extreme weather information eliminates McChord AFB. Table 9 summarizes what bases this research eliminates from analysis and what rationale leads to their elimination.

Table 9. Air Force Bases Excluded from Modeling

Base	Reason for Elimination
Bolling AFB, DC	Proximity to Andrews AFB
Brooks City-Base, TX	Proximity to Lackland AFB
Cheyenne Mountain AFS, CO	Proximity to Peterson AFB
Creech AFB, NV	Proximity to Nellis AFB
Duke Field AFB, FL	Proximity to Hurlburt Field
Onizuka AFS, CA	Proximity to Travis AFB
Randolph AFB, TX	Proximity to Lackland AFB
Schriever AFB, CO	Proximity to Peterson AFB
USAFA, CO	Proximity to Peterson AFB
Arnold AFB, TN	No viable TMY3 Data Files
Barksdale AFB, LA	No Zone 3, Hot-Humid Model
Charleston AFB, SC	No Zone 3, Hot-Humid Model
Eglin AFB, FL	Proximity to Hurlburt Field
Hanscomb AFB, MA	No viable TMY3 Data Files
Maxwell-Gunter AFB, AL	No Zone 3, Hot-Humid Model
McChord AFB, WA	No design weather data in TMY3 folder
Patrick AFB, FL	Proximity to Cape Canaveral AFS
Robins AFB, GA	No Zone 3, Hot-Humid Model

A source of historical weather data was typical meteorological year three (TMY3) data. TMY3 is the newest of two previous weather data sets, with over 1,400 different sites (Wilcox and Marian, 2008). TMY3 data represents the expected weather and solar conditions for a given location for a normal year (Wilcox and Marian, 2008). The National Renewable Energy Lab predicts these expected conditions from a minimum 15 year period (some locations have more historical data) from 1991-2005 and use statistical weighting to find the most “appropriate” value (Wilcox and Marian, 2008). For several bases, TMY3 weather data was not available for the immediate location. For those locations, this research uses the nearest TMY3 weather site for analysis. No TMY3 weather site is more than 12 miles from the selected base. Table 10 provides a list of bases using nearby locations for weather data. All other weather sites were located on the selected base. Ultimately, 52 different locations are used for analysis.

Table 10. Analyzed Locations Using Different TMY3 Weather Sites

Location	**Nearby TMY3 Location Used**
Cape Canaveral AFS, FL	NASA Shuttle Landing Facility
F. E. Warren AFB, WY	Cheyenne Municipal Airport
Goodfellow AFB, TX	San Angelo-Mathis Airport
Hickam AFB, HI	Honolulu International Airport
Kirtland AFB, NM	Albuquerque International Airport
Lackland AFB, TX	San Antonio/Kelly Field
Los Angeles AFB, CA	Los Angeles International Airport
Malmstrom AFB, MT	Great Falls International Airport
Sheppard AFB, TX	Wichita Falls Municipal Airport
Vandenberg AFB, CA	Lompoc Automated Weather Observing System

3. Generalized Model: Energy Required to Treat Outdoor air

The energy model's primary goal is to determine the energy differential between a facility with a DCV system and a facility without a DCV system. The possible differences between the two cases are the energy required to treat outside air and the energy required to supply outside air. The factors considered are fan energy, cooling energy, and heating energy.

A CAV system requires an equal amount of fan energy for both a facility with DCV control and a facility without DCV controls. An outdoor air damper controls the outdoor air flow rate. The energy differential is determined by comparing the energy required by the HVAC coils (heating and cooling) to meet sensible and latent loading. Sensible loading is the energy required to lower or raise the dry-bulb temperature; latent loading is how much energy the system needs to dehumidify the air. Equation 1 shows the components of sensible loading; Equation 2 shows the components of latent loading (ASHRAE, 2005). A negative sensible load indicates cooling; a positive sensible load indicates heating. A negative latent load indicates dehumidification. These equations assume air properties at ASHRAE standard air conditions. A potential limitation of this portion of the model is because the model assumes no economizer use, which would affect whether or not the HVAC system would treat incoming outdoor air.

$$q_s = 1.2 * Q * \Delta T \tag{1}$$

where q_s = sensible heat load (kW)
Q = airflow rate (m^3 / s)
ΔT = temperature difference between indoors and outdoors

$$q_l = Q * \rho * \Delta W * (2501 + 1.805 * T_{avg}) \tag{2}$$

where q_l = latent heat load (kW)
Q = airflow rate (m^3 / s)
ρ = air density (kg / m^3), Approximately 1.2
ΔW = humidity ratio difference between indoors and outdoors (kg_{water} / $kg_{dry\ air}$)
T_{avg} = mean between indoor and outdoor temperature (degrees Celsius)

3. Model Application: Energy Required to Treat Outdoor Air

The application of the generalized model uses Equation 1 to find sensible heating and cooling requirements and Equation 2 to calculate required energy for dehumidification. For all model instances, it must first be determined if the system is in "cooling mode" or "heating mode." No codified method exists for determining when a cooling or heating season begins or ends. Therefore, for complete analysis, an assumption must be made regarding heating and cooling seasons. To compare the effectiveness of a DCV system, the model bases the air conditioning system mode on the mean of the three-day outdoor air temperature from 0700 to 1900 hours. If the mean outdoor air temperature is below 20 degrees Celsius, the model assumes the facility is in heating mode; otherwise, the model assumes the facility is in cooling mode.

When the facility is in cooling mode, the model only counts negative latent requirements and negative sensible requirements. As previously discussed, this model assumes that the PACU's cooling coils were modeled to accommodate extreme humidity conditions, allowing the coil to remove excess moisture throughout a location's humid, cooling season. Moisture will only be removed from outside air when the air temperature is above the cooling coil temperature and the humidity content is above that of the room's set-point.

This stipulation also assumes that the system will not heat during cooling mode. When the system is in heating mode, the model only counts positive sensible loading because it assumes only sensible heating occurs during this time. Another factor for consideration in the model is the effectiveness of the furnace unit. The DOE commercial reference building model assumes the furnace unit is 80 percent effective (Deru et al., 2011). This assumption means any heating requirement requires 1.25 times as much energy as Equation 1 indicates.

4. Generalized Model: Outdoor Conditions and Indoor Set-Points

To determine how much energy needs to be add or removed to outdoor air for proper treatment, the outdoor dry-bulb temperature, outdoor humidity ratio, indoor dry-bulb set-point, and indoor humidity ratio set-point must be considered. Real world raw data are ideal to find the outdoor bulb temperature and outdoor humidity ratio. If that information is unavailable, historical weather data like TMY3 data can be used. Indoor set-points are the relative humidity and sensible temperature goals a designer sets for a zone. ASHRAE codifies the indoor air conditions are most likely to be acceptable to occupants in ASHRAE Standard 55 (2010). ASHRAE bases comfort level on indoor temperature, humidity ratio, activity level, and amount of clothing worn as shown in Figure 15 (ASHRAE, 2010a).

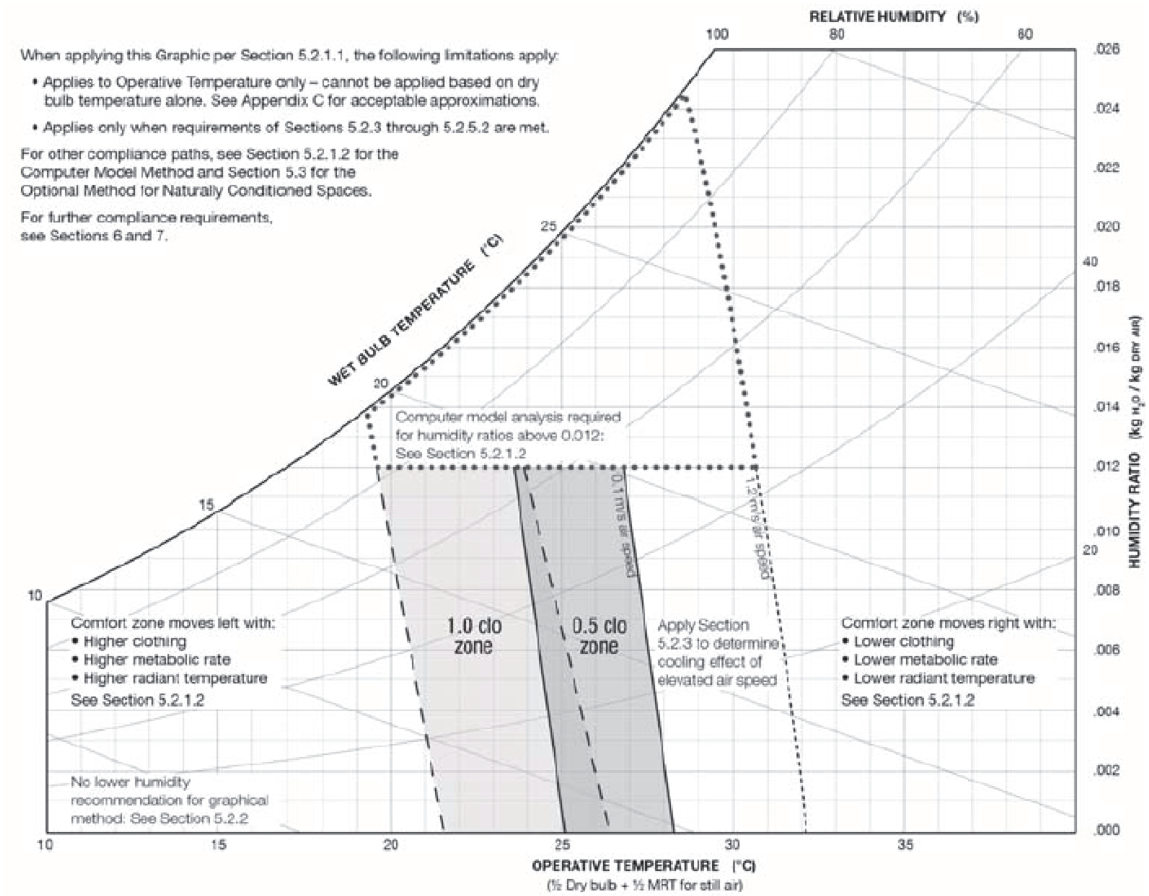

Figure 15. Thermal Comfort Range (ASHRAE, 2010a)

4. Model Application: Outdoor Conditions and Indoor Set-Points

For model application, the model uses TMY3 weather data to determine outdoor air atmospheric conditions. The model specifically uses the dry-bulb temperature and the corresponding dew point temperature (DOE, 2012e). TMY3 weather data provides these values as hourly averages for every month over the length of a year (DOE, 2012e).

Once these values are established, the next step is to find humidity ratio by calculating the water vapor saturation pressure for the corresponding dew point temperature. This value can be found by using first Equation 3 for temperatures between

-100 and 0 degrees Celsius or Equation 4 for temperatures between zero and 200 degrees Celsius (ASHRAE, 2005).

$$p_{ws} = \exp[C_1 / T + C_2 + C_3 * T + C_4 * T^2 + C_5 * T^3 + C_6 * T^4 + C_7 * \ln(T)] \tag{3}$$

$$p_{ws} = \exp[C_8 / T + C_9 + C_{10} * T + C_{11} * T^2 + C_{12} * T^3 + C_{13} * \ln(T)] \tag{4}$$

where p_{ws} = saturation pressure (Pa)

T = temperature (Kelvin)	C_7 = 4.1635019 E+00
C_1 = −5.6745359 E+03	C_8 = −5.8002206 E+03
C_2 = 6.3925247 E+00	C_9 = 1.3914993 E+00
C_3 = −9.6778430 E–03	C_{10} = −4.8640239 E−02
C_4 = 6.2215701 E−07	C_{11} = 4.1764768 E−05
C_5 = 2.0747825 E−09	C_{12} = −1.4452093 E−08
C_6 = −9.4840240 E−13	C_{13} = 6.5459673 E+00

Once water vapor saturation pressure is found using Equation 3 or Equation 4, the atmospheric pressure must be found using the elevation listed on the TMY3 data file and Equation 5 (DOE, 2012e; ASHRAE, 2005).

$$p_{atm} = 101325 * (1 - 2.5577 * 10^{-5} * Z)^{5.2559} \tag{5}$$

where p_{atm} = atmospheric pressure (Pa)
Z = elevation above sea level (m)

Now that water saturation pressure and atmospheric pressure have been found, Equation 6 can find the outside humidity ratio. The model assumes that the dew point water saturation pressure is the same as the dry-bulb water vapor pressure, consistent with a typical ASHRAE assumption. (ASHRAE, 2005)

$$W = 0.62198 * \frac{p_w}{p_{atm} - p_w} \tag{6}$$

where W = humidity ratio (kg_{water} / kg_{air})
p_w = water vapor pressure (Pa)
p_{atm} = atmospheric pressure (Pa)

The DoD's Unified Facilities Criteria for HVAC prescribes the indoor set-points for this model application (2010). The maximum (most conservative) cooling temperature is 26 degrees Celsius and 50 percent relative humidity (DoD, 2010). The lowest (most conservative) heating temperature is 20 degrees Celsius in areas of low physical activity (DOD, 2010). The model assumes that when the air conditioning system is in heating mode, the heating indoor temperature requirement is the indoor set-point. Conversely, the model assumes when the air conditioning system is in cooling mode the indoor cooling temperature and indoor relative humidity are indoor set-points.

To find the corresponding humidity ratios for heating and cooling, Equation 4 must be used, using the dry-bulb temperature instead of the dew point. The water vapor saturation pressure for the dry-bulb design temperature must now be used in Equation 7 (ASHRAE, 2005). After finding water vapor pressure, Equation 5 can now calculate the atmospheric pressure, and Equation 6 can calculate indoor design humidity ratio for heating and cooling (ASHRAE, 2005).

$$p_w = \phi * p_{ws} \tag{7}$$

where p_w = pressure of water vapor (Pa)
ϕ= relative humidity (Percent)
p_{ws} = saturation pressure of water vapor at dry-bulb design condition (Pa)

5. Generalized Model: Baseline Ventilation Outdoor Airflow

Another factor for consideration in the generalized model is the volume of outdoor air the HVAC system delivers. ASHRAE Standard 62.1 provides three different methods for determining the minimum outdoor air requirement within a zone— the IAQ Procedure, the Natural Ventilation Procedure, and the Ventilation Rate Procedure (2010b). Depending on the method used to determine the required outdoor air flow rate, the minimum outdoor air flow rate may be different for the same zone.

The IAQ Procedure is a performance-based design method. It requires designers to consider the concentration and origin of different indoor air contaminants by completing a mass balance analysis to determine the required indoor airflow rate for proper dilution of each zone (ASHRAE, 2010b). Once the facility has its designed HVAC system, there is a requirement for a zone IAQ assessment by a "subjective evaluation" (ASHRAE, 2010b). This evaluation means that the occupants take a survey to determine whether or not they find the air quality of the zone questionable or not (ASHRAE, 2010b).

The Natural Ventilation Procedure prescribes design criteria for designers who want to use natural ventilation to dilute a zone with outdoor air (ASHRAE, 2010b). This procedure requires designers to include mechanical ventilation systems designed in accordance with the IAQ Procedure or the Ventilation Rate Procedure with a few exceptions. These exceptions include: an engineer-designed natural ventilation system must be approved by the authority having jurisdiction, natural ventilation openings must be unable to be closed during periods of estimated occupancy, or if the zone is unconditioned (ASHRAE, 2010b).

With the in-depth requirements of the IAQ Procedure and the Natural Ventilation Procedure, the Ventilation Rate Procedure is the most simple and prescriptive of the three methods. The first step of the Ventilation Rate Procedure in determining the minimum outdoor airflow is to calculate the breathing zone outdoor airflow, $V_{bz,}$ show in Equation 8 (ASHRAE, 2010b).

$$V_{bz} = R_p * P_z + R_a * A_z \tag{8}$$

where V_{bz} = breathing zone outdoor airflow rate (L/s)
R_p = outdoor airflow rate per person (L / (s * person))
P_z = zone peak population (people)
R_a = outdoor airflow rate required per unit area (L / (s * m^2))
A_z = zone occupiable floor area (m^2)

The breathing zone outdoor airflow rate is the amount of air required to be distributed into a zone. The breathing zone outdoor air flow does not consider the effectiveness of the air distribution system. In order to incorporate this effectiveness, zone outdoor airflow must be calculated as shown in Equation 9 (ASHRAE 2010b). Table 11 lists typical values of zone distribution effectiveness.

$$V_{oz} = \frac{V_{bz}}{E_z} \tag{9}$$

where V_{oz} = zone outdoor airflow

V_{bz} = breathing zone outdoor airflow rate (L/s)

E_z = zone air distribution effectiveness

Table 11. Typical Values of Zone Air Distribution Effectiveness (ASHRAE 2010b)

Air Supply System Distribution Configuration	**E_z**
Ceiling supply of cool air	1.0
Ceiling supply of warm air and floor return	1.0
Ceiling supply of warm air 8 deg-C or more above space temperature and ceiling return	0.8
Ceiling supply of warm air less than 8 deg-C above space temperature and ceiling return (0.8 m/s reaches within 1.4m of floor level)	1.0
Floor supply of cool air and ceiling return (0.8 m/s supply air reaches at least 1.4m of floor level)	1.0
Floor supply of cool air and ceiling return (unidirectional flow and thermal stratification)	1.2
Floor supply of warm air and floor return	1.0
Floor supply of warm air and ceiling return	0.7
Makeup supply drawn in on the opposite side of the room from the exhaust or return	0.8
Makeup supply drawn in near the exhaust and/or return	0.5

Outdoor intake flow (V_{ot}) is the actual value of outdoor air required to enter through the outdoor air inlet for the entire supplying HVAC system. In the case of single zone systems, outdoor intake flow equals zone outdoor airflow ($V_{ot} = V_{oz}$). In the case of 100 percent outdoor air systems, the outdoor intake flow is equal to the sum of all zones' outdoor airflow ($V_{ot} = \sum_{allzones} V_{oz}$). (ASHRAE, 2010b)

5. Model Application: Baseline Ventilation Outdoor airflow

The outdoor air flow rate in the DOE small office commercial reference building uses the Ventilation Rate Procedure, using a flow rate of 10 L/s person. The assumed occupancy rate is 18.58 m^2/person, and the zone air distribution effectiveness is 1.0 (Deru et al., 2011). Table 12 summarizes the design volumetric outdoor air flow rate based on the DOE reference design. Because air distribution varies based on occupancy, the outdoor airflow rate is constant for all hours of HVAC system operation. See Appendix B for minimum outdoor air flow schedule.

Table 12. Baseline Outside Ventilation Air Flow

Zone	Outdoor Air Flow Rate
	(m^3/s)
1	0.0611
2	0.0362
3	0.0611
4	0.0362
5	0.0805

6. Generalized Model: Demand Control Ventilation Outdoor airflow

As an alternate to the three baseline methods, ASHRAE Standard 62 allows the option of using dynamic reset. ASHRAE only allows DCV in zones where people are the only source of CO_2 and zones where there are no CO_2 removal systems in place. Another DCV system operation stipulation is to ensure that outdoor air flow is never less than the product of the outdoor airflow rate required per unit area and the zone occupiable floor area ($R_a * A_z$). The final two major stipulations for DCV operation are that the system shall provide each zone with a minimum airflow (V_{bz}) based on occupancy and that the coincident total exhaust outdoor air flow shall be less than or equal to the total outdoor air intake for the system. HVAC designers using DCV must select a control scheme to ensure that the HVAC system achieves all of these criteria. (ASHRAE, 2010b)

6. Model Application: Demand Control Ventilation Outdoor airflow

Because of the DOE commercial benchmark facility's occupancy density, the model application uses the proportional control method (Schell et al., 1998). Equation 10 shows a modified version of the control algorithm Schell et al. introduce (1998). Schell et al. developed this equation before ASHRAE 62.1 started using two separate terms in the Ventilation Rate Procedure—one for zone population and one for zone area. The original equation includes a term "base ventilation rate for non-occupant-related sources;" the area-based ventilation requirement replaces it. Additionally, the original equation includes the difference between the "design ventilation rate" and the non-occupant required ventilation rate; the population-based ventilation requirement replaces it. As Equation 10 shows, the outdoor air flow rate to the zone will not fall below the product of the zone occupiable floor area and the outdoor airflow rate required per area $(A_z * R_a)$ dictated by ASHRAE 62.1 (2010b).

$$V_{DCVbz}(t) = (A_z * R_a) + P_z * R_p * \frac{(C(t) - C_{amb}) + C_{LSP}}{C_{USP} + C_{LSP}} \tag{10}$$

where $V_{DCVbz}(t)$ = the DCV-based breathing zone outdoor airflow rate at time "t" (L/s)
A_z = zone occupiable floor area (m^2)
R_a = outdoor airflow rate required per unit area (L / (s * m^2))
P_z= maximum possible zone population
R_p = outdoor airflow rate per person (L / (s * person))
$C(t)$ = the concentration of CO_2 in the zone at time "t"
C_{amb} = the CO_2 concentration in outdoor air (ppm)
C_{LSP} = the lower set-point (ppm above outdoor air concentration)
C_{USP} = the upper set-point (ppm above outdoor air concentration)

Because this is a single zone system, the relationship between DCV-based ventilation and CO_2 concentration can be found using Equation 11, using a system effectiveness of 1.0. This effectiveness is the same as the baseline outdoor air flow rate assumes (Deru et al., 2011).

$$V_{DCVot}(t) = \frac{V_{DCVbz}(t)}{E_z} \tag{11}$$

Where

$V_{DCVot}(t)$ = total DCV outdoor air flow (L/s)
$V_{DCVbz}(t)$ = breathing zone DCV outdoor air flow requirement
E_z = zone effectiveness (see Table (x))

The model calculates CO_2 concentration in 15 minute increments over the course of a year. The model does this based on the sensor reading rate for a fully loaded multipoint sensor suite of 20-30 sensors (Aircuity, 2006b). This reading is several minutes slower than a sensor bank with five sensors (Wedding, 2013). It is assumed the impact of using a smaller, more realistic time increment would have a negligible effect on the overall energy requirement of the DCV system.

7. Generalized Model: Carbon Dioxide Generation Rate

Facility occupant CO_2 generation critically drives how much outdoor air a CO_2-based DCV system requires. This criticality occurs because the CO_2 generation rate directs the amount of CO_2 present in a zone, which in turn determines how much outdoor air a room needs for dilution back to an acceptable level. American Society for Testing and Materials (ASTM) D6245 (2012) provides the necessary guidance to determine CO_2 generation rate on a per-person basis. First, the model uses Equation 12 to compute human oxygen consumption (ASTM, 2012). Table 13 shows how metabolic rate per unit of surface area varies based on the activity vigorousness.

$$V_{O2} = \frac{0.00276 * A_D * M}{(0.23 * RQ + 0.77)} \tag{12}$$

where V_{O2} = rate of oxygen consumption (L/s)

A_D = Du Bois surface area (m^2).

M = metabolic rate per unit of surface area, met (1 met = 58.2 W / m^2)

RQ = respiratory quotient

Table 13. Metabolic Rate per Surface Area for Various Activities (ASTM, 2012)

Activity	met
Seated, quiet	1.0
Reading and writing, seated	1.0
Typing	1.1
Filing, seated	1.2
Filing, standing	1.4
Walking, at 0.89 m/s	2.0
House cleaning	2.0-3.4
Exercise	3.0-4.0

For typically sized adults, the Du Bois surface area is 1.8m^2 (ASTM, 2012). For children, it ranges from 0.8 to 1.4 m^2 (ASTM, 2012). Equation 13 can be used for direct calculation of Du Bois surface area (EPA, 2011b).

$$A_D = 0.024265H^{0.3964} * W^{0.5378} \tag{13}$$

where A_D = Du Bois surface area (m^2)
H = body height (m)
W = body weight (kg)

Respiratory quotient (*RQ*) is the ratio of the volumetric rate of CO_2 generation to oxygen consumption. For average adults, *RQ* is usually 0.83 for light activity and 1.0 for heavy physical activity. Once the a researcher finds oxygen consumption rate, rate of CO_2 creation can be found by using Equation 14 (ASTM, 2012).

$$V_{CO2} = RQ * V_{O2} \tag{14}$$

where VCO_2 = rate of CO_2 generation (L/s)
RQ = respiratory quotient
V_{O2} = rate of oxygen consumption (L/s)

Once the model calculates CO_2 generation on a per-person basis, the total population of the zone must be found for various periods throughout the day. Real-world occupancy data would be ideal in estimating total CO_2 generation throughout the day for a specific building. In lieu of real data, occupancy rates can be coupled with occupancy schedules to estimate occupancy at specific times. Default occupancy rates for a variety of different facility types can be found in ASHRAE 62.1 (2010b). Occupancy schedules can be found in the 1989 edition of ASHRAE 90.1, or with minor revisions in the ASHRAE 90.1 User's Guide (Deru et al., 2011). More recent occupancy schedules have not been discovered (Deru et al., 2011; Mukhopadhyay et al, 2011).

Another option for simulating occupancy is to incorporate an occupancy model. Using complex mathematical algorithms to simulate facility occupancy is a challenging task. Fortunately, these kinds of models have become a more popular area of research with increased computing power and a heightened interest in accurate models. The work of Page (2008) and the work of Liao and Barooah (2011) present two different methods for generating occupancy models based on building occupant surveys.

7. Model Application: Carbon Dioxide Generation Rate

This model application uses the occupancy rate and schedule from the DOE's Small Office Commercial Reference Building (Deru et al., 2011). The model application assumes all occupants are adults engaged in activity similar to "seated filing" and with a respiratory quotient of 0.83. See Appendix B for the occupancy schedule.

8. Generalized Model: Carbon Dioxide Concentration

CO_2 concentration in a conditioned zone is dependent on three primary factors: the CO_2 concentration of air entering the zone, the flow rate of the air entering or leaving the zone, and the rate at which occupants create CO_2 in the zone. Depending on how a model considers and simulates flow rates, CO_2 concentration in the space can be determined using a CO_2 balance for each time step of the model. Figure 16 shows the interaction between the considered inputs to CO_2 generation. Because the model creates a CO_2 mass balance based on flow rates and corresponding CO_2 concentration, overall zone CO_2 concentration can be found. An example of a CO_2 mass balance was created by Emmerich and Persily (2001), providing a detailed summary of CO_2 calculations.

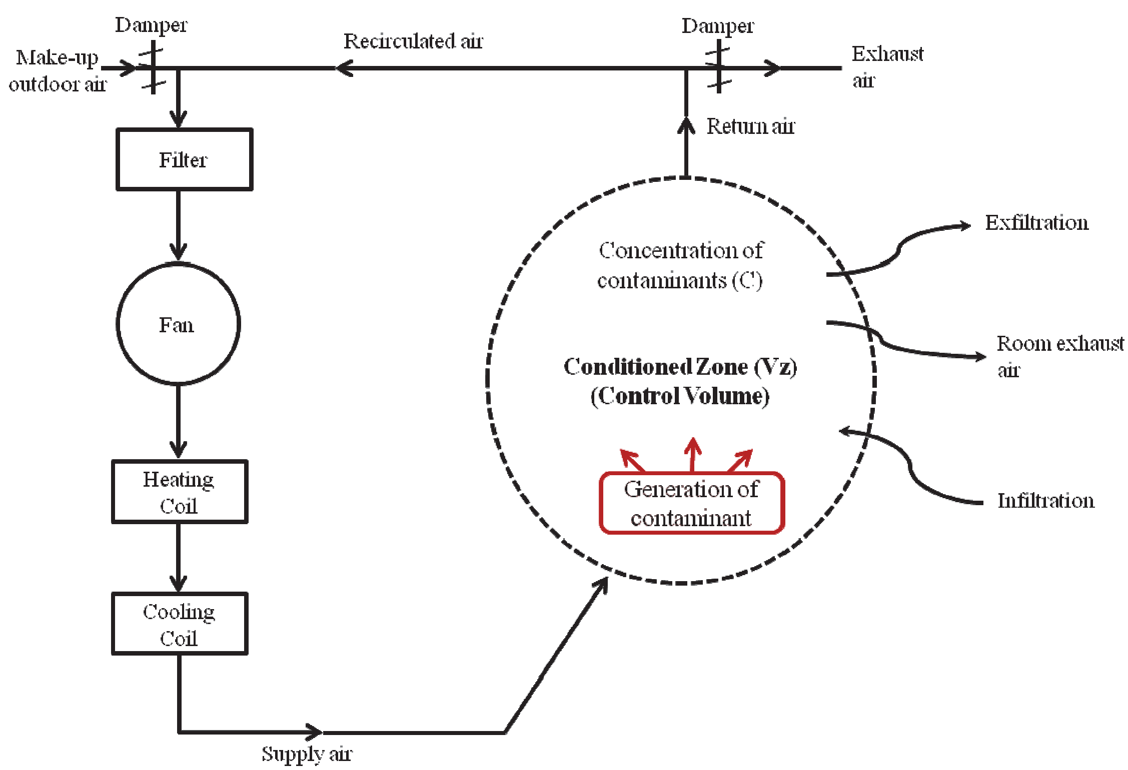

Figure 16. Airflows to be Considered in a Typical HVAC System (McQuiston et al., 2005)

8. Model Application: Carbon Dioxide Concentration

The DOE small office commercial benchmark facility makes several assumptions when considering airflow. One such assumption is that each zone is independent of the other zones. Another assumption is that there is no dedicated mechanical exhaust ventilation in any zone of the building (Deru et al., 2011). Since there is no mechanical exhaust, the model application assumes air volume added to the zone by outdoor air causes some air in the zone to exfiltrate from the room. Without this assumption, the model application would simulate no dilution in the facility, which is not feasible. The CAV system recirculates remaining zone air to the air handler.

In addition to considering the ventilation system, the DOE commercial reference building model assumes infiltration occurs at a constant volumetric flow rate per surface area exposed to the outside. Deru et al. (2011) base this assumption on a building air tightness testing option in a proposed addendum to ASHRAE 90.1 and uses an assumed pressure difference and flow exponent to model constant infiltration. Additionally, the DOE commercial reference building model assumes infiltration rate is 75% less when the HVAC system operates. Please see Appendix B for an infiltration schedule.

Air infiltration and exfiltration are complex processes, dependent upon many factors including wind speeds, internal and external temperature difference, building construction tightness, and pressure distribution in the building (Deru et al., 2011). The creators of the DOE commercial reference model facilities acknowledge that this is a "gross simplification," but assumed that the constant flow rate "average[s] effects over the year and in different locations" (Deru et al., 2011). Figure 17 shows all considered air flows in the model application, adapted from the graphic from McQuiston et al. (2005).

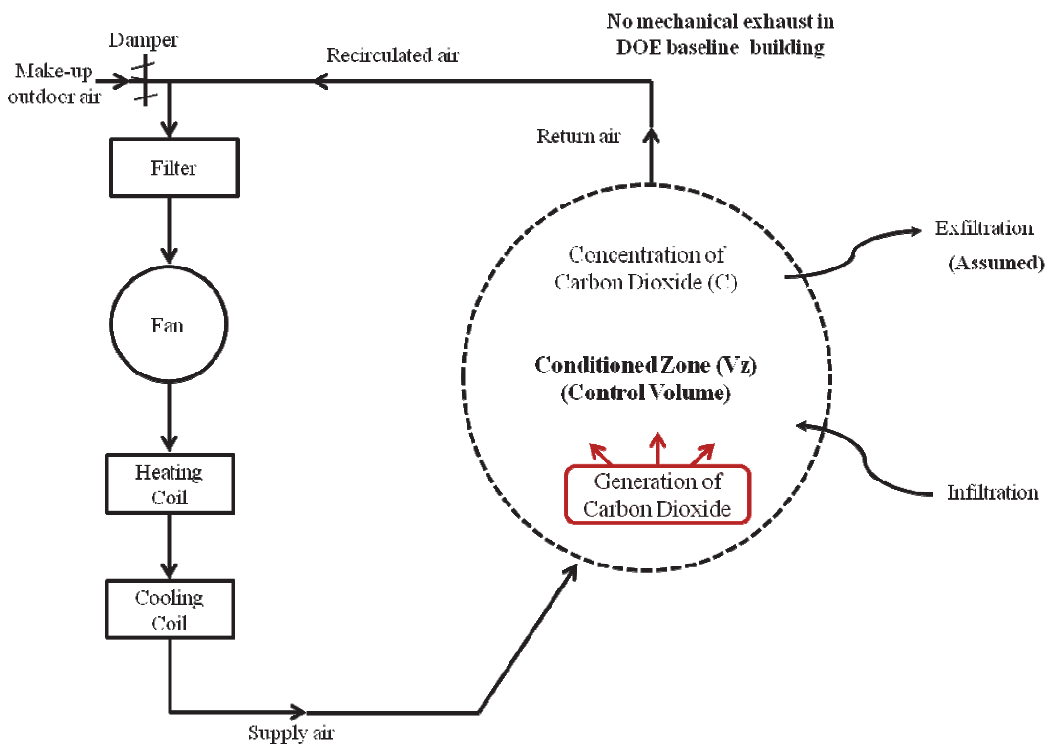

Figure 17. Airflows Considered in the Modeled Small Office Building

The concentration at time “t” in the model application can be found by solving a CO_2 mass flow rate balance. This CO_2 mass balance is developed specifically for this model application. In creating the CO_2 mass balance equation, the model application considers all air volumes to affect the CO_2 concentration from the previous time step—infiltration air, mechanically ventilated outside air, and CO_2 generated by zone occupants. The CO_2 concentration of each air flow contributes to the overall zone CO_2 concentration. The CO_2 mass balance includes the fraction of the overall zone volume that still retains the same zone CO_2 concentration as the previous time step. Equation 15 shows the result of the CO_2 mass balance. Equation 16 bases the volume of CO_2 in the zone at time “t-1” on the CO2 concentration in the zone at time “t-1.” Equation 17 shows the fraction of air remaining in the zone from time "t-1" to time "t" by considering all other airflows present in the zone from time "t-1" to time "t." The amount of air contributed to the zone by infiltration, CO2 generation, or mechanical ventilation can be found by multiplying the volumetric flow rate at the beginning of time "t" by the time increment.

$$C(t) = \frac{10^6}{V_z}(V_{zCO2}(t-1) * F(t) + V_{\inf CO2}(t) + V_{mechCO2}(t) + V_{genCO2}(t)) \quad (15)$$

where $C(t)$ = Concentration of CO_2 in the zone at time "t" (ppm)
V_z = Zone volume (m^3)
$V_{zCO2}(t-1)$ = Volume of CO_2 in the zone at time "t-1" (m^3)
$F(t)$ = The fraction of original air remaining in the zone from time "t-1" to time "t"
$V_{infCO2}(t)$ = Volume of CO_2 in the air infiltrated to the zone from time "t-1" to time "t" (m^3)
$V_{mechCO2}(t)$ = Volume of CO_2 in the outdoor air mechanically ventilated into the zone from time "t-1" to time "t" (m^3)
$V_{genCO2}(t)$ = Volume of CO_2 generated by occupants from time "t-1" to time "t" (m^3)

$$V_{zCO2}(t-1) = C(t-1)\frac{V_z}{10^6} \quad (16)$$

where $V_{zCO2}(t-1)$ = Volume of CO_2 in the zone at time "t-1" (m^3)
$C(t-1)$ = the concentration of CO_2 in the zone at time "t-1" (ppm)
V_z = Zone volume (m^3)

$$F(t) = \frac{V_z - [V_{\inf}(t) + V_{genCO2}(t) + V_{mechOA}(t)]}{V_z} \quad (17)$$

where V_z = Zone volume (m^3)
$V_{inf}(t)$ = Volume of air infiltrated to the zone from time "t-1" to time "t" (m^3)
$V_{genCO2}(t)$ = Volume of CO_2 generated in the zone from time "t-1" to time "t" (m^3)
$V_{mechOA}(t)$ = Volume of outdoor air mechanically ventilated into the zone from time "t-1" to time "t" (m^3)

9. Generalized Model: Cost

Once the annual facility energy requirements are determined, the costs associated with the overall heating, cooling, and air treatment of the facility for both the facility with a DCV system and without a DCV system can be considered. These costs will be dependent upon facility location utility rates and the fuel source for each HVAC operation. Once annual cost savings are calculated, initial and recurring DCV system costs need to be considered against the annual DCV system cost savings. Depending on the type of economic analysis, a discount factor may be applied to consider the time-value of money.

9. Model Application: Cost

For the model application, DCV system implementation is considered from the perspective of the federal government evaluating a prospective energy conservation project. This evaluation means that the project cost model will follow the Federal Energy Management Plan (FEMP) energy project guidelines of NIST Handbook 135 (Fuller and Petersen, 1996). DCV implementation follows an "accept/reject project" economic decision (Fuller and Petersen, 1996).

The cost model assumes annual costs are a cash flow at the end of each year, starting in December 2012, for a 25-year study period. All utility-associated costs must follow the appropriate commercial annual cost factor for the current year (Fuller and Petersen, 1996). These commercial utility costs are then multiplied by a FEMP uniform present value factor that can be found in the corresponding NIST Handbook Supplement (Fuller and Petersen, 1996; DOC, 2011).

Natural gas and electricity costs for 2011 on a state-by-state basis for the commercial sector are used for this economic analysis (EIA, 2013b; EIA, 2011). The cost model adjusts the 2011 prices to 2012 prices using energy price indices that project the future cost of each fuel based on census sector (DOC, 2011). In considering natural gas consumption, the heat content per volume of natural gas must also be considered. The model application uses natural gas heat content on a state-by-state basis for 2011, and it is assumed to be constant throughout the life cycle of the proposed DCV project (EIA, 2013a).

Equations 18 and 19 are developed specifically for this model application using previously defined terms. Equation 18 incorporates DCV energy savings, the 2011 price of natural gas, the 2011 to 2012 price adjustment factor for natural gas, the energy content of natural gas, and a constant term to convert various units. Equation 19 incorporates DCV energy savings, the cost of electricity, the price adjustment factor from 2011 to 2012, and a constant term to convert various units.

$$S_{gas} = \frac{947.8 * (E_{Baselineheat} - E_{DCVheat}) * C_{gas} * e_{gas2012}}{G_{eff}} \tag{18}$$

$$S_{elec} = 2.777 * (E_{Baselinecool} - E_{DCVcool}) * C_{elec} * e_{elec2012} \tag{19}$$

where S_{gas} = Annual gas cost savings by using DCV (USD_{2012})
S_{elec} = Annual electricity cost savings by using DCV (USD_{2012})
$E_{Baselineheat}$ = Annual energy required to heat outdoor air in baseline case (GJ)
$E_{DCVheat}$ = Annual energy required to heat outdoor air in DCV case (GJ)
G_{eff} = Energy effectiveness of natural gas (BTU / ft^3)
C_{gas} = Cost of natural gas (USD_{2011} / 1000 ft^3)
$e_{gas2012}$ = Price adjustment for natural gas (USD_{2012} / USD_{2011})

$E_{Baselinecool}$ = Annual energy required to cool and dehumidify outdoor air in baseline case (GJ)
$E_{DCVcool}$ = Annual energy required to cool and dehumidify outdoor air in DCV case (GJ)
C_{elec} = Cost of electricity ($Cents_{2011}$ / kW hr)
$e_{elec2012}$ = Price adjustment for electricity (USD_{2012} / USD_{2011})

Once Equation 18 and Equation 19 calculate annual cost savings of a DCV system in 2012 dollars, the life cycle component of the project is considered. To find life cycle costs savings for DCV usage, the 2012 fuel cost savings are multiplied by a FEMP uniform present value factor that incorporates time-value of money and projected fuel price escalation but not overall inflation. The Department of Commerce distinguishes FEMP uniform present value factors by census region (DOC, 2012).

Once the model accounts for life cycle cost savings, the model must incorporate the cost of the OptiNet DCV system. A discount rate of three percent is applied to all non-energy costs associated with the project (DOC, 2012). A one-time initial cost of $5,000 per sensor occurs in year zero; this facility will require five sensors (Wedding, 2013). Also, a reoccurring annual fee of $2,300 occurs to replace sensor suite components and recalibrate as necessary, regardless of geographic location (Wedding, 2013). Equation 20 shows how net savings are calculated for the DCV system (Fuller and Petersen, 1996). This equation is modified from the net savings equation in NIST Handbook 135; it now reflects only all relevant costs (Fuller and Petersen, 1996).

$$NS_{DCVlife} = -C_{initial} - C_{recurring} * UPV_{25\,years,3\%} + S_{gas} * UPV_{gas} + S_{elec} * UPV_{elec} \tag{20}$$

where $NS_{DCVlife}$ = Life cycle cost of the system (USD_{2012})
$C_{initial}$ = Initial cost of system (USD_{2012})
$C_{recurring}$ = Annual recurring costs associated with DCV system

(USD$_{2012}$)

$UPV_{25 years, 3\%}$ = Unit present value factor for 25 years and 3 percent

S_{gas} = Annual gas cost savings by using DCV (USD$_{2012}$)

UPV_{gas} = Unit present value factor for natural gas over 25 years including escalation rate

S_{elec} = Annual electricity cost savings by using DCV (USD$_{2012}$)

UPV_{gas} = Unit present value factor for electricity over 25 years including escalation rate

Conclusion

In the preceding sections, this chapter shows readers a generalized model that incorporates outdoor air ventilation rates, outdoor air conditions, internal set-points, occupancy, and facility characteristics to determine potential energy and costs savings of a DCV system. Additionally, Chapter III shows readers an application of this generalized model for a specific building across 52 unique locations. Chapter IV presents the results of this model application.

IV. Results

This chapter presents the results of the methodology discussed in Chapter III. This chapter presents the modeled demand control ventilation (DCV) air flow rates' frequency of occurrence for each zone over the course of the year. The chapter then presents the frequency of occurrence for carbon dioxide (CO_2) concentration in each zone. The chapter then displays the top 15 locations by annual energy reduction and the top 15 locations based on annual cost reduction. Finally, the chapter discusses the overall net savings or net present worth of the energy project for each location.

DCV Outdoor air Flow Rates

This subsection presents the calculated outdoor air flow rates for each zone of the small office building modeled. Figures 18 through 20 present the results for the five zones of the small office building modeled, zones one and three, zones two and four, and zone five, respectfully. Because equally sized zones produce identical results, same size zones' results are combined in the same figure. Each graph shows the outside air flow rate in the baseline system, the DCV system, and the minimum allowable outdoor airflow rate during facility operation—the area-based factor defined in ASHRAE 62.1 (2010b). Instances where there is a flow rate of 0.0 m^3/s indicate that the facility is not operational.

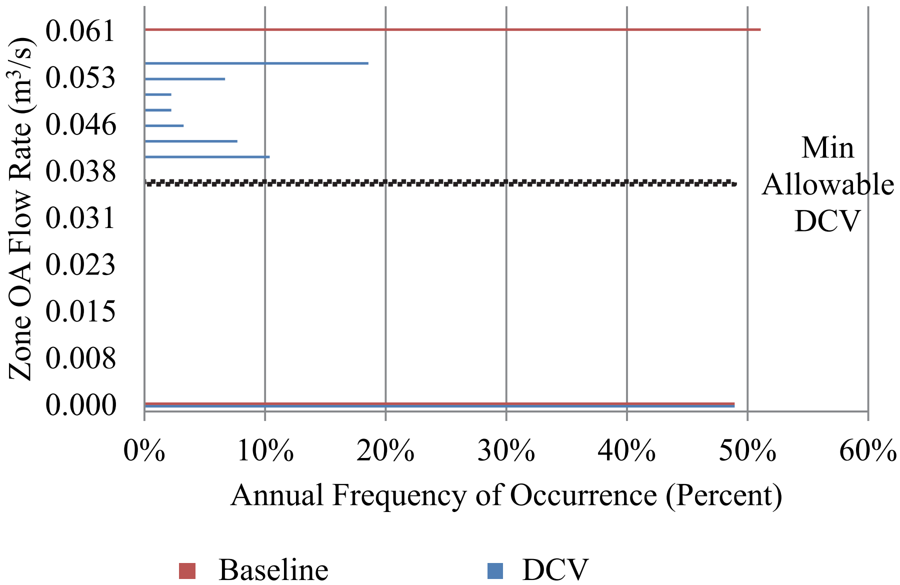

Figure 18. Zone 1 and Zone 3 Annual Outdoor Air Flow Rates

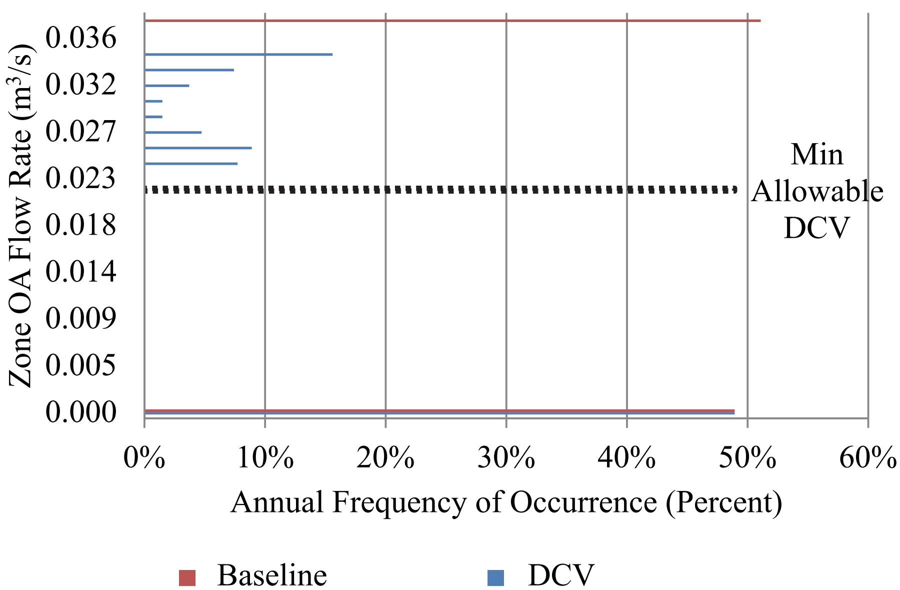

Figure 19. Zone 2 and Zone 4 Annual Outdoor Air Flow Rates

As seen in Figure 18 and Figure 19, the DCV outdoor air flow rate never reaches the baseline flow rate and remains above the minimum allowable air flow rate at all times during building operation. Facility infiltration diluted the perimeter zones' CO_2 concentration. These results comply with ASHRAE Standard 62.1 and show a reduced airflow compared to the baseline. Additionally, because the DCV system's outside air flow rate is lower than the baseline's, the DCV system usage results in energy savings.

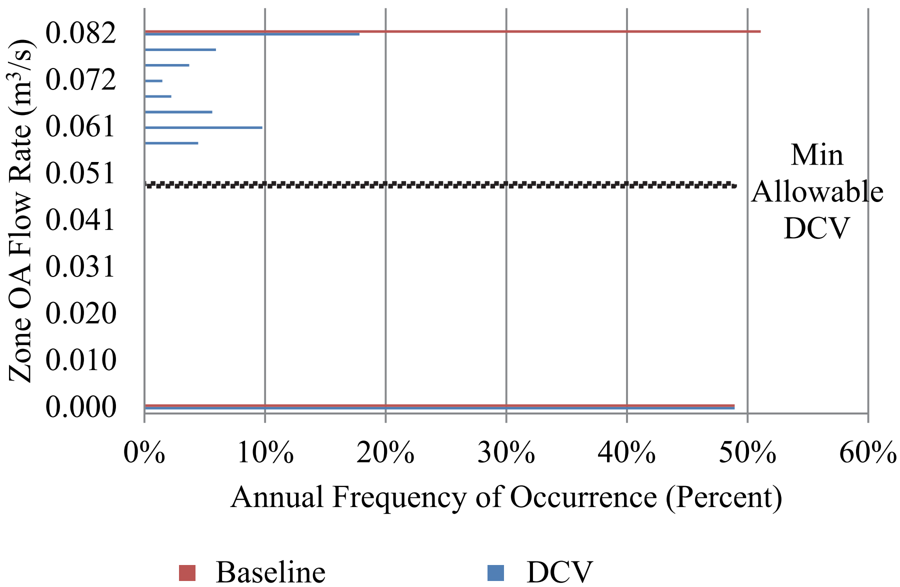

Figure 20. Zone 5 Annual Outdoor Air Flow Rate

Figure 20 shows a slightly different result. Though it remains above the minimum required outdoor air flow rate, the DCV flow rate occasionally reaches the baseline required flow rate. Because this is a core zone with no infiltration, only mechanical ventilation dilutes the room's CO_2 concentration. Even without the diluting effects of infiltration, the proportional control scheme still keeps the DCV air flow rate below the baseline for most of the year. Additionally, because the DCV system's outside air flow rate is less than the baseline system, DCV system usage will result in energy savings—though not as large of savings as the zones with outside air infiltration.

DCV Carbon Dioxide Concentrations

This subsection presents the calculated CO_2 concentration for each zone of the small office building modeled. Figures 21 through 23 present results for zones one and three, zones two and four, and zone five, respectfully. Identical zones are paired together because they have identical results. In each graph, the CO_2 concentration stays well below the recommended design concentration (ASHRAE, 2010b). These graphs show that the DCV system will save energy and still provide acceptable indoor air conditions for facility occupants.

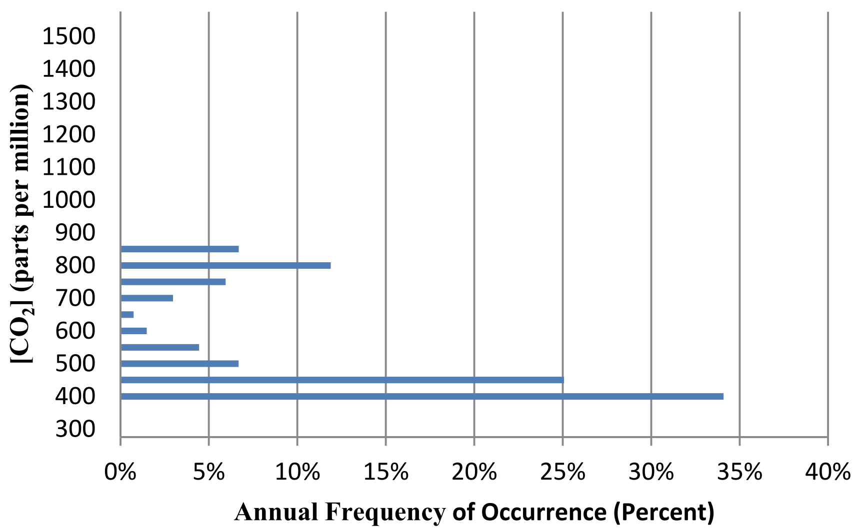

Figure 21. Zone 1 and Zone 3 Annual CO_2 Concentrations

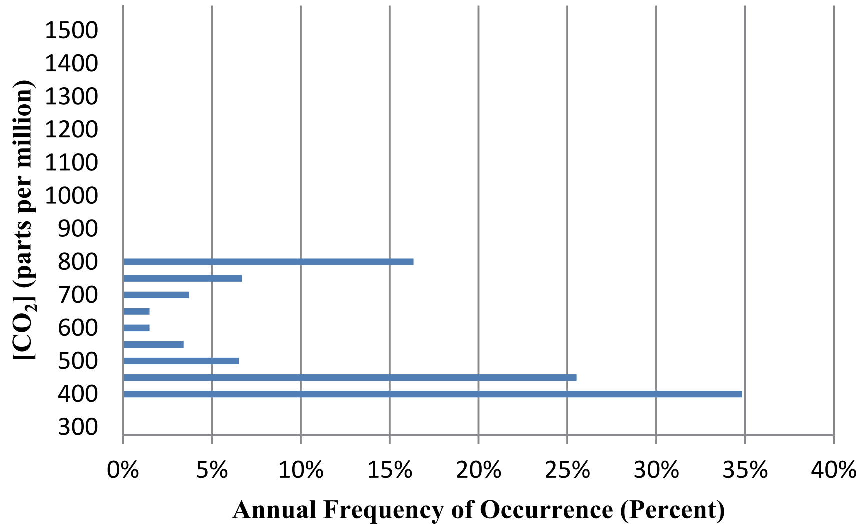

Figure 22. Zone 2 and Zone 4 Annual CO_2 Concentrations

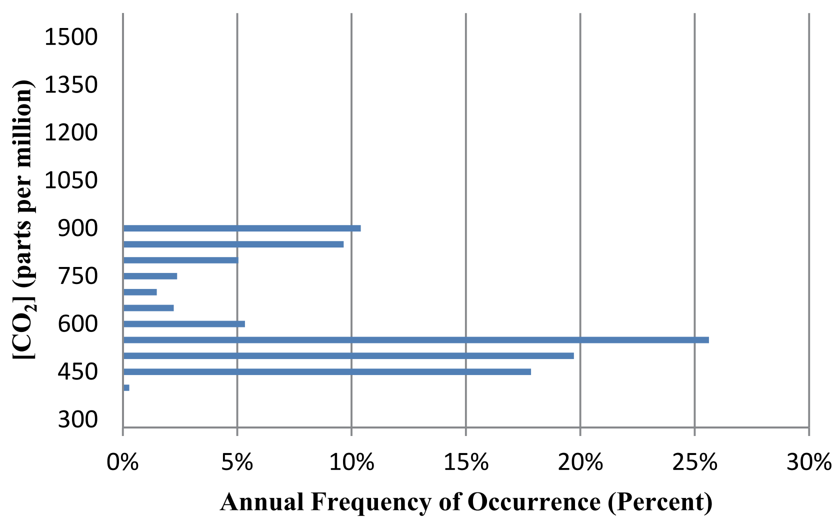

Figure 23. Zone 5 Annual CO_2 Concentration

Top 15 Locations for Energy Reduction

Table 14 presents the top 15 locations out of the 52 total modeled with the greatest potential energy savings. The highest potential energy savings occur in areas where the heating load is the dominant thermal load. The top annual energy savings are in a subarctic climate; two through four are in very cold climates. Nearly all of the other top 15 locations are in cold climates. To view the results for all locations, please see Appendix C.

Table 14. Top 15 Locations Ranked by Annual Energy Reduction

Location	Climate Zone	Annual HVAC Operations Energy Savings	Annual HVAC Cost Savings
		[GJ]	[Dollars]
Eielson AFB	8	21.4	$ 173.56
Elmendorf AFB	7	16.4	$ 133.26
Grand Forks AFB	7	15.7	$ 96.22
Minot AFB	7	15.6	$ 94.65
Ellsworth AFB	6B	13.9	$ 91.61
Malmstrom AFB	6B	13.5	$ 108.32
Fairchild AFB	5B	13.3	$ 127.52
F. E. Warren AFB	6B	12.8	$ 85.03
Offut AFB	5A	12.7	$ 102.90
Scott AFB	4A	11.6	$ 122.64
Buckley AFB	5B	11.0	$ 81.42
Wright Patterson AFB	5A	11.0	$ 99.41
Hill AFB	5B	11.0	$ 73.01
Mountain Home AFB	5B	11.0	$ 85.09
Peterson AFB	5B	10.7	$ 76.93

Top 15 Locations for Cost Reduction

Table 15 shows the top 15 locations in rank order that present the greatest potential cost savings for DCV implementation. Both the magnitude of the energy requirement and the location's energy prices influence these results. Comparing the magnitude of the energy reductions in Table 14, it is evident that the cost of utilities is a much more influential factor than energy savings. The highest cost savings occurs in Hawaii, and the second through fourth highest cost savings occur in Florida. The remaining highest saving locations are typically in the southern United States. To view the results for all locations, please see Appendix C.

Table 15. Top 15 Locations Ranked by Annual Cost Savings

Location	Climate Zone	Annual HVAC Cost Savings	Annual HVAC Operations Energy Savings
		[Dollars]	[GJ]
Hickam AFB	1A	$ 632.49	7.0
MacDill AFB	2A	$ 216.79	8.9
Cape Canaveral AFS	2A	$ 212.68	8.6
Hurlburt Field	2A	$ 206.43	9.6
Tyndall AFB	2A	$ 183.86	8.9
Keesler AFB	4A	$ 176.14	8.9
Eielson AFB	8	$ 173.56	21.4
Dover AFB	4A	$ 159.95	10.6
Columbus AFB	3A	$ 156.46	10.2
Moody AFB	2A	$ 155.55	8.0
Little Rock AFB	3A	$ 144.39	10.6
Shaw AFB	3A	$ 137.30	8.8
Seymour Johnson AFB	3A	$ 134.58	9.9
Elmendorf AFB	7	$ 133.26	16.4
Lackland AFB	2A	$ 132.55	7.1

Comparing energy reduction to cost reduction, cost reduction appears to be the heaviest weighted factor for selecting DCV candidate locations. One of the four primary factors affecting Air Force Energy Plan implementation is funding (SAF/IE, 2010). A subfactor of funding includes "budgetary priorities drive the degree of investments in energy initiatives" (SAF/IE, 2010). With the budgetary woes the Department of Defense and other government agencies are facing, cost savings will be the first consideration for selecting potential energy reduction projects.

Net Savings

Net savings incorporate the time-value of money to consider energy cost savings over system life cycle with initial and recurring CO_2 sensor costs. Based on the results of all 52 locations, multipoint sensors are not cost effective for the small office building modeled. To see the calculated net savings for every location, please see Appendix D.

This negative result does not necessarily indicate that multipoint sensors are cost prohibitive in all cases. This result only shows that multipoint sensors are cost prohibitive for the specific facility and specific assumptions modeled. Additionally, these results do not mean that all CO_2-based DCV systems are cost prohibitive for a small office building. These results only indicate that one particular kind of CO_2 sensor is cost prohibitive.

Conclusion

The preceding sections of this chapter show readers the resulting outdoor air flows and CO_2 concentration for all zones in this model application. The chapter also shows readers which locations present the greatest potential energy savings and cost savings for CO_2-based DCV control in a small office building. Finally, the chapter presents readers with an assessment of the overall potential savings of implementing a certain CO_2-based DCV control type—multipoint monitoring.

V. Conclusions

This chapter presents the ramifications of the previous chapters' work. First, the chapter will give readers a review of this research effort's accomplishments. Then, the chapter will show readers how the research adds to the current body of knowledge. After highlighting the strengths of this research, the chapter will present a discussion showing limitations of this research. Finally, the chapter will suggest avenues for future potential research stemming from this original research effort.

Review of Findings

Chapter I proposed two primary research questions and three secondary research questions. The primary research questions asked if a generic model could be developed to predict demand control ventilation (DCV) performance and if this generic model could be compared to a baseline facility. Chapter III presented a generic model to predict DCV performance. Additionally, this research applied the generic model to a small office building at 52 different Air Force installations, answering the second primary research question.

The three secondary research questions asked how environmental factors affected the decision to incorporate DCV systems, how much energy and money could be saved by implementing DCV, and which locations were the best candidates for DCV systems. Chapter II answered the first secondary research question by reviewing pertinent background information on heating, ventilation, and air conditioning (HVAC) systems and a literature review of previous DCV studies. Chapter II defined what types of facilities can be fitted with DCV technology, finding that single zone air handling systems make the best candidates for DCV. Chapter IV answered the second secondary research question when the chapter presented model application results. Chapter IV also answered the third secondary research question when the chapter showed top locations for energy reduction and cost savings. Cold weather-dominant bases provided the most energy savings; Southern state bases provided the highest cost savings.

Significance of Research

There have been several well-defined models discussing a specific application of a carbon dioxide (CO_2) based DCV system, but no found research documented all factors that should be considered in a DCV energy model. The generalized model presented in this research stream serves as a next step in modeling relevant factors for DCV. The model application additionally provides an analysis at more locations than any previously found DCV modeling effort. Additionally, there have been several economic DCV models that have incorporated a few locations either on a state scale or limited nationwide scale. The model application provides a broader variety of sites considered from an annual and life cycle scale than previous works.

Limitations

Although this research presents a unique application to modeling DCV performance, it presents three limitations beyond typical modeling variance. First, the energy model assumes no economizer usage. If a facility uses an economizer, the energy model would over predict energy saving from DCV. Creating a model conditional statement where energy requirements are zero during times meeting economizer usage criteria poses a potential solution.

A second limitation is the model application assumes a constant infiltration rate. This assumption could simulate more or less unintended outside air in the zone than the real world case, leading to overestimating or underestimating DCV performance. Pursuing a more in-depth examination of infiltration for a particular building in a particular location serves as a potential solution to this limitation. This solution makes modeling at multiple locations more challenging than in the presented model application.

A third limitation of the model application is that it does not provide any active dehumidification controls. Although the PACU's coils are designed based on extreme humidity conditions, inherently removing moisture from the air on humid days, this is not an ideal engineering practice. Mechanical dehumidifiers or desiccant dehumidifiers utilizing active humidity monitoring within a zone provide ideal solutions to humidity control (Harrian et al., 2008).

Future Research

One potential avenue of research is analyzing additional benefits and options of certain multipoint monitoring systems, like Aircuity's OptiNet suite. This sensor suite can be outfitted with sensors to monitor additional air quality contaminants or air quality indicators (Aircuity, 2006a). Using these additional sensors in critical zones with more stringent requirements can be better monitored to ensure requirement compliance. Additionally, the OptiNet suite can generate near real time updates of facility conditions (Aircuity, 2006a). These reports could be used to track HVAC performance. Additionally, these updates could be used to monitor system failures, either from the mechanical system's wear-and-tear or even cyber attacks.

A second stream of research, related directly to this work, would be to model additional facility types similarly to the small office building by using assumed facility and environmental parameters for several different locations. Researchers could determine which geographic locations are best suited for a DCV system for a different type of facility modeled at multiple locations.

A third related research stream would be to simulate DCV operation at an existing facility, generating an accurate assessment by using measured data. Occupancy could be directly measured to predict CO_2 generation. Air properties and air flow rates could be measured to calculate energy requirements. Researchers could generate a much more accurate assessment of potential energy and cost savings for an actual facility by using real data.

For both of these two final potential research streams, consideration should be given to the facility type modeled. Laboratories that provide large amounts of outside air for prescriptive ventilation are an excellent candidate for DCV. Also, large auditoriums, gymnasiums, or classrooms, with a highly variable occupancy and large occupancy are also excellent candidates for DCV modeling. Facilities with 100 percent outdoor air, such as certain zones in hospitals, would provide an excellent third facility type for modeling.

Another consideration for these final two research streams is the type of DCV sensor modeled in the research. This analysis used multipoint sensors because cost data were readily available. This research shows that multipoint sensors are not an ideal choice for a small office building. There are several different types of CO_2-based DCV sensors that could also be considered. These sensors can be subdivided into several categories—one of the largest categories is how they sensor detects CO_2 concentration. Common detection methods include infrared, electrochemical, photoacoustic, and photoionization (FEMP, 2004; Aircuity, 2006). Sensors can also be categorized based on placement for CO_2 detection. Some sensors are wall-mounted, installed within HVAC ductwork, or transmit a sample of air to a central sensor suite (FEMP, 2004; Aircuity, 2006). Each type of sensor has strengths and limitations. The sensors used for the model application, multipoint sensors, are a photoionization-based sensor with a central sensor suite. These types of sensors have a higher initial cost, lower maintenance cost, and a favorable economy of scale (Wedding, 2013). Future research efforts could examine a sensor with lower initial cost and economy of scale like wall-mounted infrared CO_2 sensor for use in a smaller building (FEMP, 2004, Wedding 2013). Emmerich and Persily (2001) provide summaries of several case studies of actual DCV implementation.

Summary

This research explored modeling DCV technology. The purpose of this research was to find a way to predict potential energy savings and cost reduction in using DCV control systems. The research methodology created a generalized model and an instance of applying this model. The model application provided a rank order of locations based on energy reduction and cost savings. For the facility modeled in this research, bases in cold climate locations yielded higher energy reduction and certain bases in certain states yielded higher cost savings. The CO_2 sensor modeled was not cost efficient for the facility modeled. The generalized model limited itself to economizer-free constant air volume, single zone systems. The model application was a small office that used a natural gas furnace and an electric packaged air conditioning unit. In summary, a generalized model predicted DCV performance, and an application of this model predicted energy reduction values and cost savings, and found that multipoint sensors were cost prohibitive at a small office building at 52 locations.

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Appendix A: Units of Measurement and Acronyms

ach	Air changes per hour
AFCEC	Air Force Civil Engineer Center
ASHRAE	American Society of Heating, Refrigeration, and Air Conditioning Engineers
ASTM	American Society for Testing and Materials
BTU	British Thermal Unit
CAV	Constant Air Volume
cfm	Cubic feet per minute
CO2	Carbon Dioxide
DCV	Demand Control Ventilation
DoD	Department of Defense
DOE	Department of Energy
EIA	Energy Information Administration
EO	Executive Order
FEMP	Federal Energy Management Program
ft	Foot
FY	Fiscal Year
GJ	Gigajoule
hr	Hour
HVAC	Heating, Venting, and Air Conditioning
kg	Kilogram
kW	Kilowatt
L	Liters
m	Meters
met	Metabolic rate per unit of surface area
MZ	Multizone
NIST	National Institute for Standards and Testing
Pa	Pascals
PACU	Packaged air conditioning unit
ppm	Parts per million
s	Second
SBS	Sick Building Syndrome
SZ	Single Zone
TMY3	Typical meteorological year 3
USD	United States Dollar
VAV	Variable Air Volume
W	Watts

Appendix B: Schedules

Table 16. Schedules (Deru et al., 2011)

Infiltration Schedule			
Hr.	WD	Sat.	Sun.
1	1	1	1
2	1	1	1
3	1	1	1
4	1	1	1
5	1	1	1
6	1	1	1
7	0.25	0.25	1
8	0.25	0.25	1
9	0.25	0.25	1
10	0.25	0.25	1
11	0.25	0.25	1
12	0.25	0.25	1
13	0.25	0.25	1
14	0.25	0.25	1
15	0.25	0.25	1
16	0.25	0.25	1
17	0.25	0.25	1
18	0.25	0.25	1
19	0.25	1	1
20	0.25	1	1
21	0.25	1	1
22	0.25	1	1
23	1	1	1
24	1	1	1

Occupancy Schedule			
Hr.	WD	Sat.	Sun.
1	0	0	0
2	0	0	0
3	0	0	0
4	0	0	0
5	0	0	0
6	0	0	0
7	0.1	0.1	0
8	0.2	0.1	0
9	0.95	0.3	0
10	0.95	0.3	0
11	0.95	0.3	0
12	0.95	0.3	0
13	0.5	0.1	0
14	0.95	0.1	0
15	0.95	0.1	0
16	0.95	0.1	0
17	0.95	0.1	0
18	0.3	0	0
19	0.1	0	0
20	0.1	0	0
21	0.05	0	0
22	0.05	0	0
23	0.05	0	0
24	0.05	0	0

Baseline OA Schedule			
Hr.	WD	Sat.	Sun.
1	0	0	0
2	0	0	0
3	0	0	0
4	0	0	0
5	0	0	0
6	0	0	0
7	0	0	0
8	1	1	0
9	1	1	0
10	1	1	0
11	1	1	0
12	1	1	0
13	1	1	0
14	1	1	0
15	1	1	0
16	1	1	0
17	1	1	0
18	1	1	0
19	1	0	0
20	1	0	0
21	1	0	0
22	1	0	0
23	0	0	0
24	0	0	0

Appendix C: Complete List of Energy Reduction and Cost Savings

Table 17. Complete List of Energy Reduction and Cost Savings by Base

Location	Climate Zone	Annual HVAC Operations Energy Reduction	Annual HVAC Cost Savings
		[GJ]	[Dollars]
Altus AFB	3A	7.6	$ 89.41
Andrews AFB	4A	9.6	$ 121.67
Beale AFB	3B	5.5	$ 63.08
Buckley AFB	5B	11.0	$ 81.42
Cannon AFB	4B	8.7	$ 65.27
Cape Canaveral AFS	2A	8.6	$ 212.68
Columbus AFB	3A	10.2	$ 156.46
Davis Monthan AFB	2B	4.2	$ 71.76
Dover AFB	4A	10.6	$ 159.95
Dyess AFB	3B	6.5	$ 85.38
Edwards AFB	3B	6.4	$ 74.24
Eielson AFB	8	21.4	$ 173.56
Ellsworth AFB	6B	13.9	$ 91.61
Elmendorf AFB	7	16.4	$ 133.26
F. E. Warren AFB	6B	12.8	$ 85.03
Fairchild AFB	5B	13.3	$ 127.52
Goodfellow AFB	3B	6.3	$ 74.40
Grand Forks AFB	7	15.7	$ 96.22
Hickam AFB	1A	7.0	$ 632.49
Hill AFB	5B	11.0	$ 73.01
Holloman AFB	3B	6.5	$ 62.87
Hurlburt Field	2A	9.6	$ 206.43
Keesler AFB	4A	8.9	$ 176.14
Kirtland AFB	4B	7.7	$ 54.56
Lackland AFB	2A	7.1	$ 132.55
Langley AFB	4A	9.9	$ 119.14
Laughlin AFB	2A	6.8	$ 131.77
Little Rock AFB	3A	10.6	$ 144.39
Los Angeles AFB	3B-Coast	3.4	$ 38.67
Luke AFB	2B	5.2	$ 101.74
MacDill AFB	2A	8.9	$ 216.79

Malmstrom AFB	6B	13.5	$	108.32
McConnell AFB	4A	9.9	$	99.81
McGuire AFB	4A	9.9	$	112.59
Minot AFB	7	15.6	$	94.65
Moody AFB	2A	8.0	$	155.55
Mountain Home AFB	5B	11.0	$	85.09
Nellis AFB	3B	6.6	$	96.15
Offut AFB	5A	12.7	$	102.90
Peterson AFB	5B	10.7	$	76.93
Pope AFB	3A	8.7	$	117.83
Scott AFB	4A	11.6	$	122.64
Seymour Johnson AFB	3A	9.9	$	134.58
Shaw AFB	3A	8.8	$	137.30
Sheppard AFB	3A	7.6	$	96.30
Tinker AFB	3A	9.8	$	114.64
Travis AFB	3B	5.3	$	50.78
Tyndall AFB	2A	8.9	$	183.86
Vance AFB	3A	9.8	$	103.78
Vandenberg AFB	3C	6.9	$	52.81
Whiteman AFB	4A	10.3	$	117.07
Wright Patterson AFB	5A	11.0	$	99.41

Appendix D: Net Savings

Table 18. Net Savings for Multipoint Sensing for a Small Office Building

Location	Climate Zone	Total Cost Savings
		[U. S.D$_{2012}$]
Altus AFB	3A	$ (65,126.82)
Andrews AFB	4A	$ (64,489.74)
Beale AFB	3B	$ (65,588.75)
Buckley AFB	5B	$ (65,145.23)
Cannon AFB	4B	$ (65,514.05)
Cape Canaveral AFS	2A	$ (63,026.19)
Columbus AFB	3A	$ (63,948.86)
Davis Monthan AFB	2B	$ (65,488.66)
Dover AFB	4A	$ (63,743.37)
Dyess AFB	3B	$ (65,235.04)
Edwards AFB	3B	$ (65,378.98)
Eielson AFB	8	$ (63,257.78)
Ellsworth AFB	6B	$ (64,999.73)
Elmendorf AFB	7	$ (64,078.39)
F. E. Warren AFB	6B	$ (65,060.65)
Fairchild AFB	5B	$ (64,202.30)
Goodfellow AFB	3B	$ (65,422.75)
Grand Forks AFB	7	$ (64,906.93)
Hickam AFB	1A	$ (55,868.47)
Hill AFB	5B	$ (65,319.01)
Holloman AFB	3B	$ (65,558.87)
Hurlburt Field	2A	$ (63,091.96)
Keesler AFB	4A	$ (63,647.62)
Kirtland AFB	4B	$ (65,702.80)
Lackland AFB	2A	$ (64,430.68)
Langley AFB	4A	$ (64,557.19)
Laughlin AFB	2A	$ (64,449.52)
Little Rock AFB	3A	$ (64,149.20)
Los Angeles AFB	3B-Coast	$ (66,054.65)
Luke AFB	2B	$ (64,974.29)
MacDill AFB	2A	$ (62,948.27)
Malmstrom AFB	6B	$ (64,587.26)
McConnell AFB	4A	$ (64,879.96)

McGuire AFB	4A	$	(64,772.31)
Minot AFB	7	$	(64,935.90)
Moody AFB	2A	$	(63,983.49)
Mountain Home AFB	5B	$	(65,076.82)
Nellis AFB	3B	$	(65,041.13)
Offut AFB	5A	$	(64,826.14)
Peterson AFB	5B	$	(65,226.54)
Pope AFB	3A	$	(64,610.90)
Scott AFB	4A	$	(64,464.37)
Seymour Johnson AFB	3A	$	(64,303.48)
Shaw AFB	3A	$	(64,278.23)
Sheppard AFB	3A	$	(65,029.83)
Tinker AFB	3A	$	(64,652.49)
Travis AFB	3B	$	(65,797.62)
Tyndall AFB	2A	$	(63,483.46)
Vance AFB	3A	$	(64,830.47)
Vandenberg AFB	3C	$	(65,716.35)
Whiteman AFB	4A	$	(64,555.22)
Wright Patterson AFB	5A	$	(64,872.82)

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14. ABSTRACT
Heating, ventilation, and air conditioning (HVAC) use accounts for 43% of commercial energy consumption, with close to 5% used for ventilation purposes. Federal government agencies face both energy consumption reduction mandates and reduced funding. Carbon dioxide (CO2) based demand control ventilation (DCV) is a technology that allows for reduced energy consumption by allowing facility designers to introduce outside air based on facility occupancy, per American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) standards. This research aims to create a generalized methodology assessing energy and cost reductions from CO2–based DCV and then apply it to a specific facility at multiple locations. This research creates a generalized methodology for future researchers to follow based on the present body of knowledge. The model application then applies this methodology to one of the Department of Energy's (DOE) commercial benchmark facility models. The selected DOE model is a small office building with single zone HVAC air systems, assessing DCV impact on energy consumption and costs for 52 United States locations. Although the model application is not life cycle cost effective for the building modeled, it successfully identifies which areas experience the greatest cost and energy savings from DCV.

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