The Impact of Demand-Controlled and Economizer Ventilation Strategies on Energy Use in Buildings

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ABSTRACT

The overall objective of this work was to evaluate typical energy requirements associated with alternative ventilation control strategies for constant-air-volume (CAV) systems in commercial buildings. The strategies included different combinations of economizer and demand-controlled ventilation, and energy analyses were performed for four typical building types, eight alternative ventilation systems, and twenty U.S. climates. Only single-zone buildings were considered so that simultaneous heating and cooling did not exist. The energy savings associated with economizer and demand-controlled ventilation strategies were found to be very significant for both heating and cooling. In general, the greatest savings in electrical usage for cooling with the addition of demand-controlled ventilation occur in situations where the opportunities for economizer cooling are less. This is true for warm and humid climates and for buildings that have relatively low internal gains (i.e., low occupant densities). As much as 20% savings in electrical energy for cooling were possible with demand-controlled ventilation. The savings in heating energy associated with demand-controlled ventilation were generally much larger but were strongly dependent upon the building type and occupancy schedule. Significantly greater savings were found for buildings with highly variable occupancy schedules and large internal gains (i.e., restaurants) as compared with office buildings. In some cases, the primary heating energy was virtually eliminated by demand-controlled ventilation as compared with fixed ventilation rates. For both heating and cooling, the savings associated with demand-controlled ventilation are dependent on the fixed minimum ventilation rate of the base case at design conditions.

INTRODUCTION

ANSI/ASHRAE Standard 62-1989 (ASHRAE 1990) provides guidance to maintain adequate indoor air quality in buildings. While there are a number of approaches to implementation of the standard, the most common approach is to dilute indoor pollutants through ventilation, defined as the intentional introduction of air from outside the building. The standard recommends the minimum ventilation airflows necessary to maintain satisfactory indoor air quality. The minimum requirement depends upon the type of building and the occupancy. Typically, for CAV systems, the ventilation flow rate is determined based upon design occupancy for the specific building type and outdoor dampers are set to maintain a constant ventilation airflow. This approach conforms to the Ventilation Rate Procedure of the standard. However, with this strategy, the ventilation rate exceeds the minimum when the building is not fully occupied. The energy requirements to heat and cool a building can often be reduced if the ventilation airflow is adjusted in response to the number of occupants. Using the Indoor Air Quality Procedure of the standard, an adjustable ventilation airflow can often be implemented by controlling the ventilation to maintain a specific CO_2 level within the building. This strategy is referred to as demand-controlled ventilation.

The energy savings associated with demand-controlled ventilation (as compared with minimum ventilation based upon design occupancy) depend upon several factors, including the building characteristics, occupancy schedule, and climate. In addition, the savings depend upon the type of economizer that is employed. When the outdoor conditions are suitable, an air-side economizer opens the outdoor air dampers

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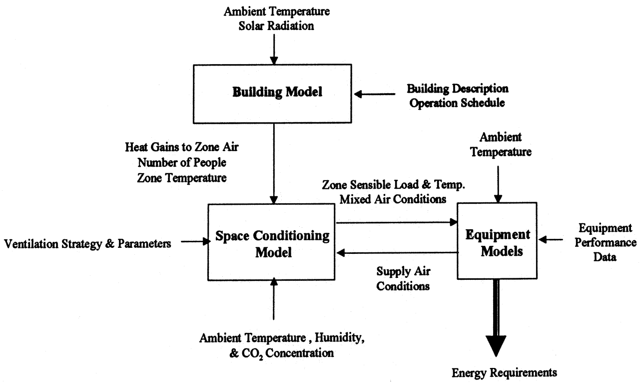

Figure 1 *Modeling approach.*

from their minimum position (minimum ventilation air) to meet building cooling loads with cool outdoor air. Two different types of switchover are typically used: (1) dry-bulb and (2) enthalpy. With a dry-bulb economizer, the switchover occurs when the ambient dry-bulb temperature is less than a specified value, typically between 55°F (12.8°C) and 70°F (21.1°C). With an enthalpy economizer, the switchover happens when the outdoor enthalpy (or wet-bulb temperature) is less than the enthalpy (or wet-bulb temperature) of the return air. Although the enthalpy economizer yields lower overall energy consumption, it requires wet-bulb temperature or dry-bulb and relative humidity measurements.

While the benefits of economizer operation are well understood, there has been relatively little previous analysis of integrated control of outdoor airflow for both economizer and demand-controlled ventilation. Much of the previous analysis of CO_2-based demand-controlled ventilation has focused on ventilation control (Vaculik and Plett 1993; Federspiel 1997) or the effect of HVAC system design in multizone buildings on pollutant transport and energy use (Knoespel et al. 1991; Haghighat et al. 1993; Emmerich et al. 1994; Carpenter 1996). There have also been several field studies of demand-controlled ventilation in commercial buildings (i.e., Donnini et al. 1991). Few of the studies have included multiple locations or systems with economizers, and the basis for reported energy savings varies widely among studies. A notable exception is Rock and Wu (1998), who examined the effect of air-side economizer and demand-controlled ventilation for a small office building in ten U.S. locations. They concluded that, in many cases, demand-controlled ventilation increased energy use by reducing free cooling associated with air-side economizers. However, they did not explore the option of combining demand-controlled ventilation with economizer control.

This paper presents cooling and heating system energy use associated with different combinations of demand-controlled ventilation and economizer strategies for a range of typical buildings, systems, and climates. Energy savings relative to fixed ventilation rates (no economizer and minimum flow based on design occupancy) are also presented.

ANALYSIS APPROACH

The analysis has been performed for commercial building served by packaged, single-zone, CAV heating and cooling equipment. Ventilation is provided by mixing outdoor air with recirculated air from the zones. A simulation model was developed for estimating the energy requirements of this system for alternative ventilation control methods. Figure 1 shows a flow diagram for the modeling approach. The model predicts hourly energy requirements for specified buildings, equipment, controls, and weather. The building and zone air models are separated for computational efficiency. The building model predicts the heat gains to or from the zone air based upon transient heat transfer from the building structure and internal sources (i.e., lights, people, and equipment). The space air-conditioning model solves energy and mass balances for the zone and air distribution system and determines mixed air conditions supplied to the equipment. The zone recirculated and outdoor air are mixed according to the ventilation strategy employed. The zone temperatures are outputs from the building model, whereas the zone and return air humidities and CO_2 concentrations are calculated by the space-conditioning model. The equipment model uses entering conditions and the sensible cooling requirement to determine the average supply air conditions. The entering and exit air conditions for the air distribution and equipment models are determined iteratively at each timestep of the simulation using a nonlinear equation solver developed for this project. Details of each of the component models are described in the following subsections.

Building Model

A dynamic model for building heat gains is essential to properly determine the heating, cooling, and ventilating requirements for a building. For this work, DOE2.1E (Winkelman et al. 1993) was used to calculate space loads based on the building physical characteristics, operating schedule, occupancy patterns, and space setpoints. Hourly outputs from the model include sensible heat gains to the zones, number of occupants, and zone temperatures.

Space Conditioning Model

During the occupied period, the flow rate of air to the zone is constant and the equipment cycles on and off as necessary to maintain the zone temperature setpoint. During the unoccupied period, the fan cycles on and off with the equipment, but the airflow rate is constant when the system is on. This operation is typical packaged CAV rooftop equipment. The zone sensible heat gain or loss determines the required average supply air temperature. Given the supply airflow rate and the supply air temperature, the heating and cooling requirements for the equipment are determined by the mixed air conditions, which are in turn determined by the outdoor air fraction through the ventilation control strategies. The energy use of the equipment to meet the requirements is determined by the equipment model, described below.

When demand-controlled ventilation is enabled, a minimum flow rate of ventilation air is determined that will keep the CO_2 concentration in the zone at or below a specified level. In the absence of demand-controlled ventilation, the minimum ventilation flow rate is a fixed value and is determined using ANSI/ASHRAE Standard 62-1989 based upon the design occupancy.

At any given time, the ventilation flow can be greater than the minimum due to air-side economizer operation. Two different economizer options are included: dry-bulb and enthalpy. With either of these options, outside air is used to provide cooling whenever the outdoor air conditions are deemed to be appropriate. A dry-bulb economizer uses outdoor air for cooling whenever the outdoor air temperature is less than a specified value, typically between 55°F (12.8°C) and 70°F (21.1°C). (Higher temperatures are used in drier climates.) For this analysis, the economizer value was selected to be 60°F. The enthalpy economizer is engaged whenever the enthalpy of the ambient air is less than the enthalpy of the air in the return duct. In both economizer modes, the ventilation flow rate is modulated between the minimum and maximum (wide open) values to maintain a specified set point (i.e., 55°F) for the mixed air temperature supplied to the equipment.

For known ventilation flow, zone temperature, and ambient conditions, steady-state mass and energy balances are applied to the zone and air distribution system to determine average values over each timestep for the return and zone air CO_2 concentration and humidity ratio. These calculations are based on a fully mixed zone model, modified by an air exchange effectiveness to account for partial short-circuiting of the supply air to the ceiling return.

Equipment Models

The model considers packaged rooftop equipment with simple on/off control. Specifically, the analysis includes air conditioners with gas furnaces and heat pumps with electric auxiliary heat. The fan is on during all hours of occupancy, and the compressor or heater cycles on and off to maintain the zone temperature at its set point. Models for a direct expansion air conditioner and heat pump were taken from the ASHRAE *HVAC2 Toolkit* (Brandemuehl et al. 1993) and adapted for this project. The secondary toolkit contains a library of subroutines and functions that have been debugged and documented. The direct expansion and heat pump models are based upon correlations used in DOE 2.1E. These models estimate capacity (cooling or heating) and power consumption as a function of mixed air and ambient conditions for typical devices. The outputs are scaled according to capacity and efficiency values that are specified for ARI rating conditions. For cooling, both sensible and total cooling capacities are determined. Iteration with the space-conditioning model is required, since the space humidity level is determined by the moisture removal rate of the equipment, which is affected by the mixed air humidity.

TABLE 1
Ventilation Strategies

Case Name	IAQ Strategy	Economizer Strategy
Base case	Fixed minimum outside air	No economizer
Temp	Fixed minimum outside air	Dry bulb
Enth	Fixed minimum outside air	Enthalpy
BaseIAQ	Demand-controlled ventilation	No economizer
TempIAQ	Demand-controlled ventilation	Dry bulb
EnthIAQ	Demand-controlled ventilation	Enthalpy

CASE STUDIES

Simulations were performed for a number of different combinations of building types, locations, and ventilation control strategies. Table 1 defines the combinations of different economizer and indoor air quality (IAQ) ventilation strategies that were considered and names that are used to refer to the results.

Four different types of buildings were considered in this study: office, large retail store, school, and sit-down restaurant. Descriptions for these buildings were obtained from prototypical descriptions of commercial buildings developed at a U.S. national laboratory (Huang and Franconi 1995). A summary of the building characteristics is given in Table 2. Each of the buildings was simulated with night setup (for cooling) and setback (for heating) of the zone thermostats and fan shutdown during unoccupied times. It is assumed that there is no infiltration.

Simulations were performed for 20 locations, selected to provide a good cross section of climates within the United States. The weather data are for Typical Meteorological Years (TMY2 data). Table 3 lists the cities considered in this study.

Hourly simulations were performed for all combinations of ventilation strategies and buildings specified in Tables 1 and 2 for all locations. It was also necessary to define several system parameters for the simulations. Table 4 lists parameters used in the simulations that were independent of the building type. It should be noted that the assumed CO_2 generation rate generally applies to adults at light activity, such as standing in a relaxed position or seated performing filing or typing. In fact, CO_2 generation will be greater for adults walking and shopping in a retail store and will be less for children seated in a classroom. In addition, outdoor CO_2 concentration varies by location and throughout the year. A constant value has been assumed for this analysis.

It was also necessary to specify the minimum ventilation flow rates for the cases where demand-controlled ventilation

TABLE 2
Prototypical Building Characteristics

Characteristic	Office	Large Retail	School	Sit-Down Restaurant
Floor area, ft^2 (m^2)	6600 (613)	80,000 (7432)	9,600 (892)	5250 (488)
Floors	1	2	2	1
Percent glass	15	15	18	15
Window R-value, $h \cdot ft^2 \cdot °F/Btu$ ($m^2 \cdot °C/W$)	1.6 (0.28)	1.7 (0.30)	1.7 (0.30)	1.5 (0.26)
Window shading coeff.	0.75	0.76	0.73	0.80
Wall R-value, $h \cdot ft^2 \cdot °F/Btu$ ($m^2 \cdot °C/W$)	5.6 (0.99)	4.8 (0.86)	5.7 (1.00)	4.9 (0.87)
Roof R-value, $h \cdot ft^2 \cdot °F/Btu$ ($m^2 \cdot °C/W$)	12.6 (2.22)	12.0 (2.11)	13.3 (2.34)	13.2 (2.32)
Wall material	Masonry	Masonry	Masonry	Masonry
Roof material	Built-up	Built-up	Built-up	Built-up
Weekday hours (hrs/day)	11	12	Varies	17
Weekend hours (hrs/day)	4	5	Varies	17
Equipment power, W/ft^2 (W/m^2)	0.5 (0.05)	0.4 (0.04)	0.8 (0.08)	2.0 (0.19)
Lighting power, W/ft^2 (W/m^2)	1.7 (0.16)	1.6 (0.15)	1.8 (0.17)	2.1 (0.20)
Heat/cool setpoints, occupied, °F (°C)	70/75 (21/24)	72/75 (22/24)	75/78 (24/26)	72/75 (22/24)
Heat/cool setpoints, unoccupied, °F (°C)	55/90 (13/32)	60/85 (16/30)	65/85 (18/30)	68/85 (20/30)

TABLE 3
Locations for Simulations

East	Mid-East	Midwest	West
Boston	Madison	Minneapolis	Seattle
New York	Chicago	Topeka	Sacramento
Washington, D.C.	Pittsburgh	Denver	Los Angeles
Atlanta	Nashville	Ft. Worth	Albuquerque
Miami	Lake Charles	Houston	Phoenix

TABLE 4
Building-Independent System Parameters for Simulations

Parameter	Value
Set point for return air CO_2 concentration	1000 ppm by volume
Ambient CO_2 concentration	300 ppm by volume
CO_2 generation rate per person	0.30 L/min
Ventilation effectiveness	0.85
Latent energy gains per person	200 Btu/h (58.6 W)
Switchover temperature for dry-bulb economizer	60°F (15.6°C)
Mixed air temperature set point for economizer	55°F (12.8°C)
Furnace efficiency	0.85
Design supply airflow rate per unit cooling capacity	450 cfm/ton (60.4 L/s·kW)
EER of air conditioner at ARI rating condition	10
Sensible heat ratio of air conditioner at ARI rating condition	0.75

TABLE 5
Parameters for Estimating Fixed Minimum Ventilation Requirements

	Values			
Parameter	**Office**	**Retail***	**School**	**Restaurant**
Minimum ventilation flow rate per person, cfm (L/s)	20 (9.44)	10 (4.72)	15 (7.08)	20 (9.44)
Design occupancy density for estimating minimum ventilation flow, ft^2/person (m^2/person)	150 (13.94)	40 (3.72)	40 (3.72)	30 (2.78)

*Retail store minimum ventilation is based upon an average of 0.25 cfm/ft^2 (1.27 L/s·m^2) for upper and lower floors.

was not employed. Table 5 gives parameters used to estimate the fixed minimum ventilation rates according to building type. These values were determined from the estimated maximum occupancy and outdoor air requirements from Table 2 of ANSI/ASHRAE Standard 62-1989. Savings associated with demand-controlled ventilation are particularly sensitive to the parameters of Table 4 and the occupancy schedule.

The occupancy schedules used within the load simulations were obtained from Huang and Franconi (1995). Separate schedules apply for weekdays, weekends, and holidays. In addition, the school has separate occupancy schedules for school holidays and summer. In general, these schedules are representative of average occupancy and do not necessarily approach the design occupancies associated with Table 5. Figure 2 shows the average occupancy schedules for occupied days relative to the design values (determined from Table 5) for the four buildings. The average occupancies are much less than the peak expected occupancy for the office, retail store, and restaurant. This is particularly true for the retail store, for which occupancy can be highly variable. On the other hand, the school has a relatively small variation in its occupancy schedule during the weekdays. For this case, the occupancy used in the simulations is relatively close to the value used for determining the minimum ventilation flow during much of the occupied period.

The large difference between average occupancy profiles of Figure 2 and the peak occupancy implicit in Table 5 is largely traced to the maximum occupancy and ventilation requirements of ANSI/ASHRAE Standard 62-1989. For example, for street level retail stores, Table 2 of the standard gives estimated maximum occupant density of 30 people per 1000 ft^2 of net occupiable space. Assuming approximately 82% utilization of gross floor area for the 80,000 ft^2 retail store, peak occupancy will be about 2000 people. However, average retail occupancy during the year will be dramatically less than this estimated peak.

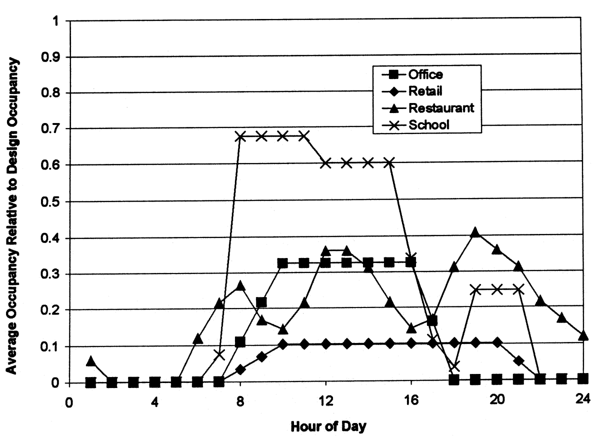

Figure 2 *Average occupancy schedule (for occupied days) relative to design occupancy.*

The analyses described here assume that ventilation based on maintaining 1000 ppm of CO_2 will ensure adequate indoor air quality. This assumption is valid presuming that there are no dominant sources of other contaminants. For some applications, this assumption implies that significant pollutant sources are separately vented, i.e., cooking equipment in restaurants, art and craft rooms in schools, copy centers in office buildings. By comparison, in retail buildings, there are often significant contaminants released by the products in the store sales area. Ventilation rates must be maintained to ensure that such contaminants are maintained at acceptable levels. In many cases, these contaminants may require additional ventilation above that required to maintain 1000 ppm of CO_2.

The design supply airflow rate was determined as the product of the design equipment cooling load and the design cfm per ton from Table 4. The design cooling load was estimated using a psychrometric analysis of the maximum value of the sum of the sensible zone load from DOE 2.1E simulations, the latent gains from people (gains per person from Table 4 and occupancy from DOE 2.1E), and the ventilation load (minimum ventilation flow determined from parameters in Table 5). DOE 2.1E provided design fan power for each building and location.

RESULTS

Hourly simulations were performed for 480 different cases (6 ventilation strategies × 4 buildings × 20 locations). To systematically examine these results, we focus first on the results for three representative locations: Madison, Atlanta, and Albuquerque. Madison represents a north-central climate with significant cooling and large heating requirements, Atlanta is a warm and humid climate with lesser heating needs, and Albuquerque is a warm and dry climate.

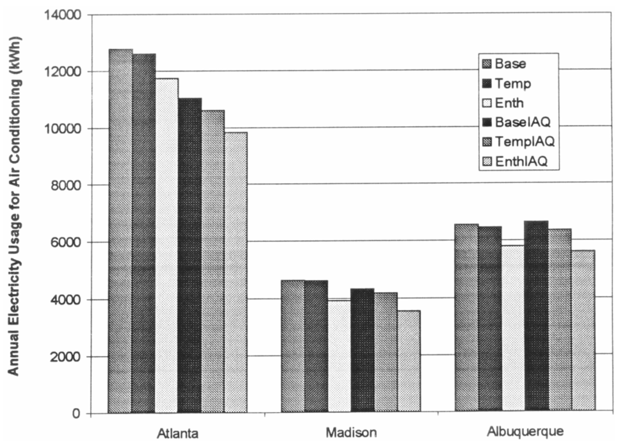

Figure 3 *Annual electrical air conditioning requirements for the office building (not including fan energy).*

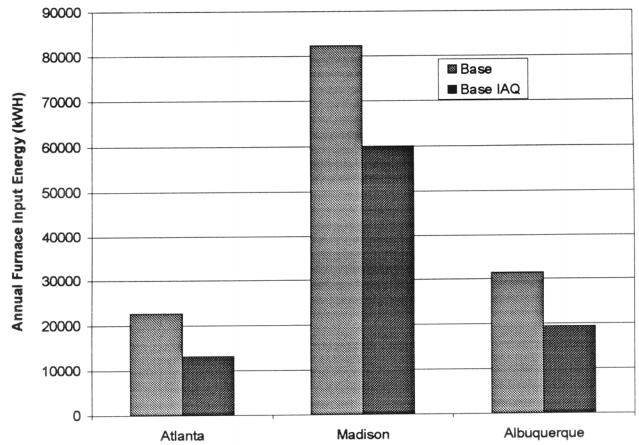

Figure 4 *Annual furnace input requirements for the office building (not including fan energy).*

Figure 3 shows annual electrical energy requirements for air conditioning (not including fan energy) the office building in the three representative locations for all six combinations of ventilation strategies. The air-conditioning requirements are much larger for Atlanta than for Madison and Albuquerque, leading to more significant absolute savings for economizer and demand-controlled ventilation strategies. For all three locations, the dry-bulb economizer (Temp) alone provided only modest savings, while the savings for use of the enthalpy economizer (Enth) were substantial. The additional savings associated with adding demand-controlled ventilation depend upon the location. Demand-controlled ventilation has a much smaller impact in Albuquerque than Atlanta because economizer operation is possible during a significant portion of the cooling season. In fact, demand-controlled ventilation with no economizer (BaseIAQ) resulted in greater energy usage than for fixed minimum ventilation and no economizer (Base) for Albuquerque. This is because the lower ventilation rate associated with demand-controlled ventilation reduces the net “free” cooling provided by the ambient. The greatest savings, in all cases, occurred for the combination of enthalpy economizer and demand-controlled ventilation. The maximum savings were about 23% for Atlanta and Madison and 14% for Albuquerque when compared to no economizer and demand-controlled ventilation.

The savings potential associated with demand-controlled ventilation is much more significant for heating than for cooling, since economizer operation does not play a role. Figure 4 shows the annual furnace energy requirements for the office building in the three locations with and without demand-controlled ventilation. As expected, the heating requirements are much more significant in Madison than for the other two locations, leading to greater absolute savings with demand-controlled ventilation. The reduced ventilation requirements directly decrease the heating loads and lead to very significant savings. The percent reductions in annual heating input energy (BaseIAQ relative to Base) were 27%, 38%, and 42% for Madison, Albuquerque, and Atlanta, respectively. The largest relative savings occurred in Atlanta because the ventilation load was a larger fraction of the total heating load.

The savings associated with different ventilation strategies are strongly dependent upon the building type. Figure 5 shows the impact of building type on annual electricity requirements for air conditioning in Madison. The electricity usage relative to the base case usage (no economizer and fixed minimum ventilation) for each building are presented for the alternative ventilation strategies. The savings for economizer and demand-controlled ventilation were significantly greater for the retail store (36%), restaurant (45%), and school (47%) than for the office building (23%). These buildings have larger internal gains (greater occupant density) and, therefore, longer periods of time where economizer operation can be used to reduce mechanical cooling requirements. However, the impact of demand-controlled ventilation is much smaller with increased internal gains because of the increased opportunities for economizer operation. In fact, demand-controlled ventilation with no economizer (BaseIAQ) resulted in significantly greater energy usage than for fixed minimum ventilation and

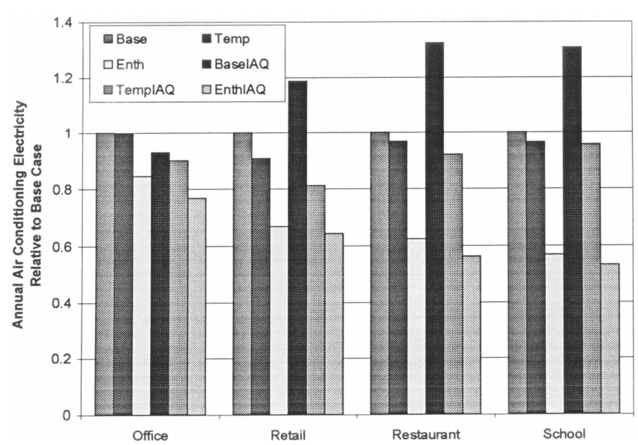

Figure 5 *Annual air conditioning electricity relative to base case for Madison.*

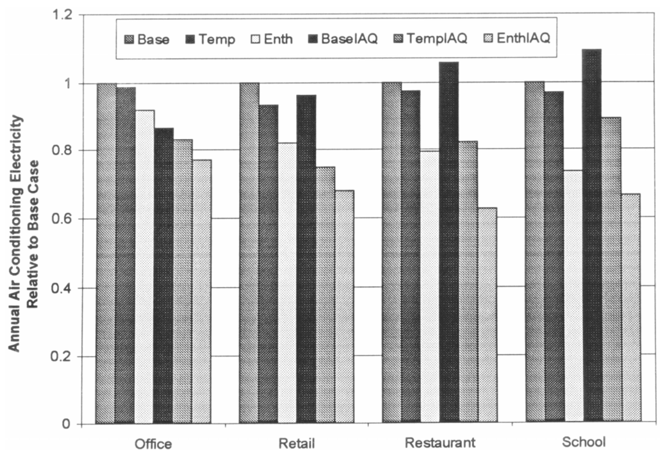

Figure 6 *Annual air conditioning electricity relative to base case for Atlanta.*

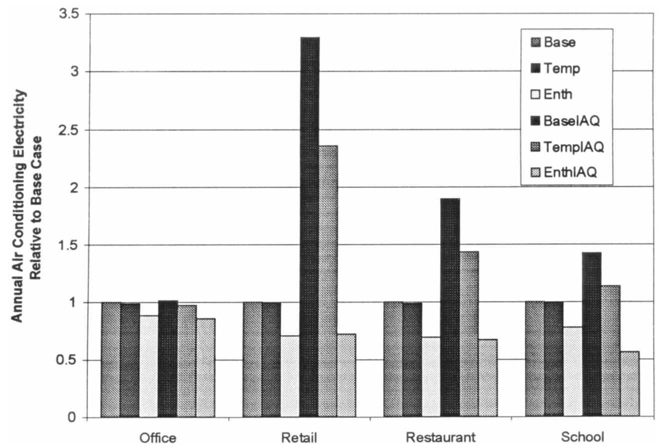

Figure 7 *Annual air conditioning electricity relative to base case for Albuquerque.*

no economizer (Base) for the retail store, restaurant, and school. The lower ventilation rate associated with demand-controlled ventilation reduces the net "free" cooling provided by the ambient. As noted previously, these savings are based on the assumption that occupant-generated pollutants dictate ventilation rates. Savings will be less if other contaminants dictate higher ventilation, as may occur in retail stores.

Figures 6 and 7 show air conditioning results for the different buildings in Atlanta and Albuquerque. The results for Atlanta show trends that are similar to those for Madison. However, the savings associated with demand-controlled ventilation (BaseIAQ) were somewhat larger (or there were smaller penalties) due to reduced benefits of economizer operation. Again, the combination of demand-controlled ventilation and enthalpy economizer resulted in very significant savings. In Albuquerque, the use of demand-controlled ventilation without economizer operation resulted in large energy penalties for the restaurant, retail store, and school. In these cases, the opportunities for economizer cooling are substantial (high occupant densities and dry climate) and reducing the minimum ventilation flow is not advantageous. For Albuquerque, the combination of demand-controlled ventilation and an enthalpy economizer resulted in small savings compared with the economizer-only options. However, the use of demand-controlled ventilation with a dry-bulb economizer resulted in energy use penalties for the retail store, restaurant, and school. Curiously, if there are other dominant contaminant sources (i.e., retail store) that increase the demand-controlled ventilation rate, savings will increase.

Demand-controlled ventilation results in extremely large savings for heating energy since economizer operation does not play a role. Figure 8 shows furnace energy savings for the four buildings in the three locations. Demand-controlled ventilation almost eliminated heating needs for the restaurant and retail store and gave significant savings for the office and school. The extremely large savings for the retail store and restaurant were a result of their occupancy schedule and large internal gains. These buildings had relatively large internal gains, so that the ventilation heating loads for the fixed minimum flow rate were a large fraction of the total heating loads. Demand-controlled ventilation reduces heating requirements due to ventilation during times of low occupancy.

Similar results were obtained for 17 additional U.S. locations. Energy consumption data for the base case ventilation strategy are given in Table 6, which includes both gas and electric energy use for the four building types in each of the 20 locations. The electric energy use includes fan consumption during both heating and cooling seasons. As noted previously, the greatest energy savings occur with the implementation of demand-controlled ventilation with enthalpy economizer as the ventilation strategy. Table 7 gives the gas and electric energy savings associated with this ventilation control strategy. These energy savings are shown graphically in Figures 9 and 10. Figure 9 gives the gas energy savings and Figure 10 gives the electrical energy savings.

The general trend of energy savings is similar to those discussed for Madison, Atlanta, and Albuquerque. Demand-controlled ventilation delivers dramatic energy savings during the heating season. Greater savings occur when the

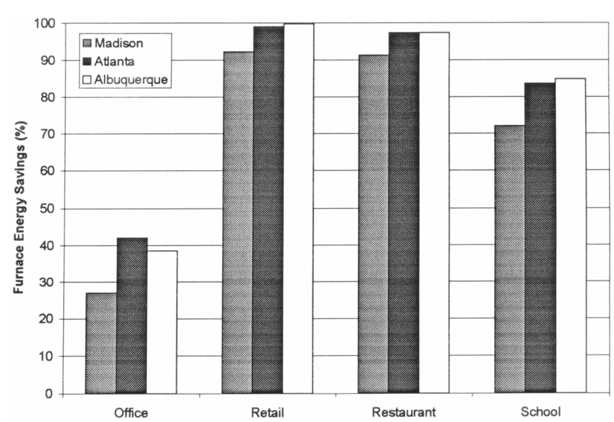

Figure 8 *Annual furnace energy savings for demand-controlled ventilation relative to base case.*

TABLE 6
Annual Energy Consumption for Base Case

	Gas Energy (kWh)				Electric Energy (kWh)			
	Office	Retail	Rest	School	Office	Retail	Rest	School
Boston	61,249	570,387	172,892	85,918	12,540	378,627	33,958	29,703
New York	50,260	468,457	148,341	71,405	14,713	429,647	38,974	35,692
Washington	43,841	404,922	139,588	65,834	19,786	529,424	51,890	45,224
Atlanta	22,570	215,753	92,280	37,478	24,628	621,483	64,208	52,112
Miami	121	3,308	8,404	967	49,538	934,408	104,437	89,801
Pittsburgh	65,349	602,932	177,050	91,350	13,436	392,420	36,737	31,454
Chicago	63,351	618,391	179,296	95,915	16,576	470,502	45,250	40,800
Madison	82,230	776,895	214,760	117,106	17,183	561,768	59,647	45,733
Nashville	26,736	260,999	104,288	43,021	23,925	602,332	61,016	52,832
Lake Charles	6,095	63,064	46,014	12,510	36,092	782,447	82,363	72,572
Minneapolis	96,008	887,856	236,455	137,586	15,327	442,895	44,114	37,524
Topeka	48,186	485,900	150,928	73,860	22,700	578,861	60,565	50,997
Fort Worth	14,069	150,245	71,363	24,834	31,735	712,902	78,462	64,906
Houston	4,867	57,109	42,013	10,282	37,232	796,804	84,822	73,942
Denver	53,101	743,630	182,769	91,822	7,390	132,604	15,141	21,622
Albuquerque	31,376	460,473	131,947	64,548	11,097	159,377	18,052	27,031
Seattle	53,854	401,558	159,915	67,653	7,918	244,809	17,134	16,427
Sacramento	17,201	125,092	84,661	26,068	22,701	512,177	47,070	43,493
Los Angeles	4,858	20,841	45,270	7,130	15,748	429,284	30,339	31,042
Phoenix	3,429	42,246	39,829	7,446	42,256	720,631	79,426	71,534

heating requirements associated with ventilation are a large fraction of the total. Savings for the retail store and restaurant were greater than 85% for all locations; savings for the school were greater than 70% in all locations. The heating loads for these buildings are dominated by ventilation loads. The office building is mostly dominated by envelope loads and shows considerably less savings. For most locations with significant heating requirements, the savings for the office building were approximately 30%.

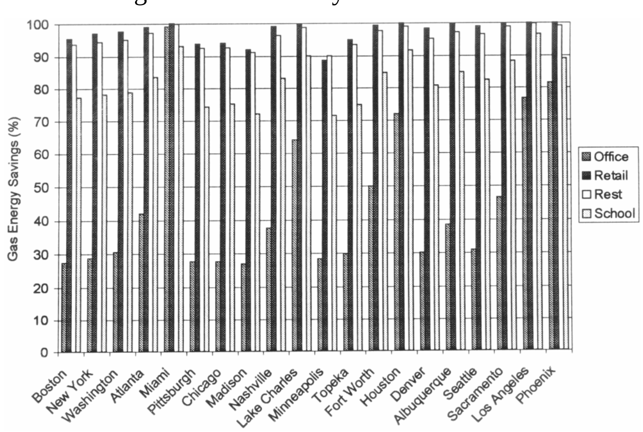

Figure 9 *Annual gas energy savings for demand-controlled ventilation.*

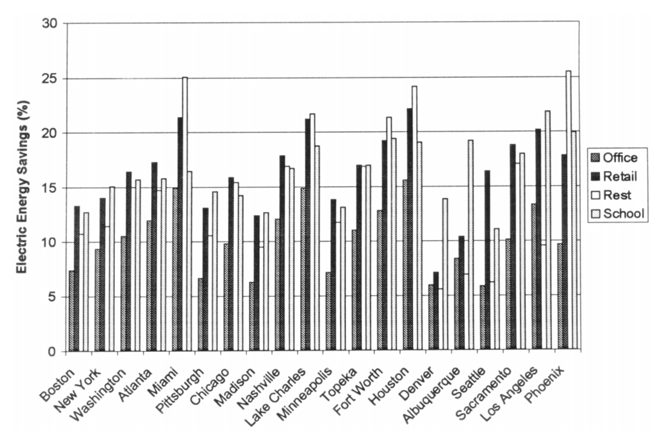

Figure 10 *Annual electric energy savings for enthalpy economizer and demand-controlled ventilation (EnthIAQ).*

TABLE 7
Annual Energy Consumption with Enthalpy Economizer and Demand-Controlled Ventilation (EnthIAQ)

	Gas Energy (kWh)				Electrical Energy (kWh)			
	Office	**Retail**	**Rest**	**School**	**Office**	**Retail**	**Rest**	**School**
Boston	44,418	25,215	10,699	19,548	11,613	328,195	30,306	25,927
New York	35,819	13,265	8,116	15,526	13,340	369,291	34,520	30,317
Washington	30,437	9,345	6,817	13,868	17,700	442,635	44,093	38,131
Atlanta	13,079	2,042	2,515	6,128	21,687	514,117	54,746	43,883
Miami	0	0	9	65	42,138	734,878	78,279	75,050
Pittsburgh	47,290	36,685	13,032	23,515	12,544	340,962	32,854	26,858
Chicago	45,782	36,658	13,015	23,791	14,949	395,802	38,261	34,997
Madison	60,005	60,240	18,721	32,763	16,107	492,242	53,985	39,951
Nashville	16,673	2,425	3,666	7,218	21,044	494,905	50,718	44,026
Lake Charles	2,192	93	547	1,233	30,724	616,769	64,518	59,000
Minneapolis	68,744	100,547	23,245	39,144	14,233	381,703	38,928	32,593
Topeka	33,811	23,758	9,581	18,538	20,195	480,868	50,363	42,349
Fort Worth	7,033	897	1,574	3,777	27,659	576,299	61,727	52,326
Houston	1,365	27	437	846	31,428	620,628	64,323	59,871
Denver	37,106	11,721	8,618	17,730	6,951	123,206	14,297	18,628
Albuquerque	19,304	1,043	3,548	9,768	10,167	142,788	16,800	21,850
Seattle	37,172	3,853	5,290	11,842	7,455	204,759	16,073	14,606
Sacramento	9,187	156	962	3,028	20,403	416,199	39,038	35,688
Los Angeles	1,133	0	78	239	13,644	342,871	27,443	24,265
Phoenix	632	0	342	814	38,168	592,624	59,202	57,277

Electrical energy savings for the most energy-efficient ventilation strategy (enthalpy economizer and demand-controlled ventilation) varied between 6% and 22%, depending on location and building type. The greatest savings tend to occur in locations with high cooling requirements, i.e., Houston and Lake Charles, though significant savings were also found for northern locations such as Minneapolis. Of the four building types, the office building showed the smallest relative savings in every location, largely because ventilation was a smaller fraction of the building cooling load.

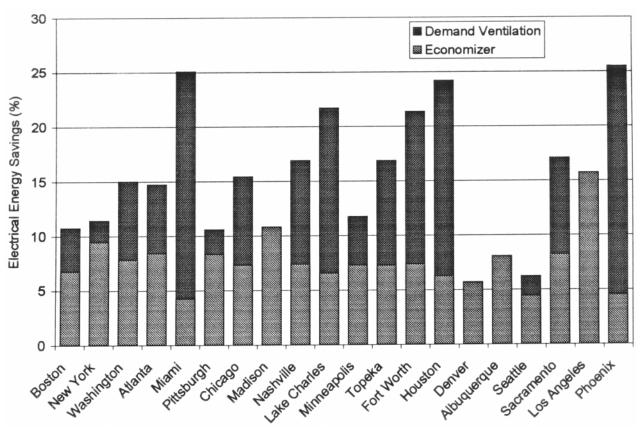

Figure 11 *Incremental energy savings for enthalpy economizer and demand-controlled ventilation (EnthIAQ) for restaurant compared to base case.*

The electrical energy savings of Figure 10 include both the effects of economizer and demand-controlled ventilation. As noted previously, much of these savings could be due to the effect of free economizer cooling rather than the occupancy-based ventilation control. Figure 11 shows the breakdown of electrical energy savings between the two features. The results show that the savings in many locations are largely due to economizer cooling. While many of these locations are in the western U.S., both Pittsburgh and Madison show small incremental savings from demand-controlled ventilation. By comparison, the greatest incremental savings for demand-controlled ventilation occur in the southeastern U.S., where high humidity reduces the benefits of economizer cooling.

Savings with demand-controlled ventilation are due to reduced ventilation airflow in response to reduced occupancy. In this study, occupancy schedules for occupied and unoccu-

TABLE 8
Restaurant Energy Use with Reduced Design Occupancy

	Gas Energy (kWh)			Electric Energy (kWh)			
	Original Base	Base	BaseIAQ	Original Base	Base	Enth	EnthIAQ
Boston	172,892	75,965	10,696	33,958	35,870	30,664	30,306
New York	148,341	63,866	8,114	38,974	41,590	34,479	34,520
Washington	139,588	59,430	6,815	51,890	52,198	45,336	44,093
Atlanta	92,280	35,420	2,513	64,208	65,416	55,598	54,746
Miami	8,404	2,780	8	104,437	93,693	88,317	78,279
Pittsburgh	177,050	79,934	13,030	36,737	38,602	32,760	32,854
Chicago	179,296	81,038	13,013	45,250	45,726	39,670	38,261
Madison	214,760	100,304	18,719	59,647	63,268	52,863	53,985
Nashville	104,288	41,981	3,663	61,016	60,653	52,558	50,718
Lake Charles	46,014	15,072	546	82,363	78,376	69,777	64,518
Minneapolis	236,455	112,480	23,243	44,114	45,087	39,481	38,928
Topeka	150,928	66,641	9,579	60,565	59,513	52,394	50,363
Fort Worth	71,363	26,311	1,573	78,462	75,483	65,993	61,727
Houston	42,013	13,677	436	84,822	79,460	70,823	64,323
Denver	182,769	73,673	8,616	15,141	18,881	14,228	14,297
Albuquerque	131,947	48,823	3,546	18,052	22,752	16,429	16,800
Seattle	159,915	63,690	5,285	17,134	19,594	16,200	16,073
Sacramento	84,661	28,570	960	47,070	49,359	40,670	39,038
Los Angeles	45,270	10,292	78	30,339	37,810	25,712	27,443
Phoenix	39,829	12,934	341	79,426	73,134	66,340	59,202

pied days were used that are representative of average days and did not include the range of occupancies that might normally exist. As Figure 2 shows, the peak occupancies of the average occupied days were considerably less than the design occupancies for the office, restaurant, and retail store. As a result, a large portion of the simulated energy savings associated with demand-controlled ventilation was the result of setting a fixed minimum ventilation rate for the base case that provided adequate ventilation for the worst-case building occupancy. If the design occupancy were close to the peak in the average occupancy, then savings for demand-controlled ventilation would be due to variations in average occupancy only. In order to separate these two effects, results were generated for the restaurant at reduced design occupancy. The design occupancy was cut in half, so that the maximum average occupancy was 80% of the design occupancy. For consistency, it was assumed that the furnace and air-conditioning equipment were the same size as for the original base case and the total system airflow was unchanged. Table 8 gives gas and electrical usage for the original base case (original peak occupancy) and for the reduced design occupancy (Base, BaseIAQ, Enth, and EnthIAQ).

The results of the analysis at reduced design occupancy show that the reported savings in gas consumption attributed to demand-controlled ventilation were largely due to the high minimum outdoor airflow rate of the base case. For every location, only about one-third of the gas energy savings reported in Figure 9 and Table 7 could be attributed to the occupancy variations over the day, with the remaining two-thirds due to the difference between average and design occupancy. However, as shown in Table 8, demand ventilation continues to demonstrate energy savings of approximately 90% over the base case with reduced design occupancy.

The analysis of electricity savings is more complex because of the interactions between economizer cooling and demand ventilation. As noted previously, demand ventilation can increase energy use if it precludes the use of economizer cooling. The same effect can be observed by uniformly reducing ventilation rates. As shown in Table 8, reducing the design occupancy (i.e., reducing the minimum outdoor airflow of the base case) resulted in greater electrical energy consumption in 12 of the 20 locations compared to the original base case with high outdoor airflow. Figure 12 shows the incremental electrical energy savings for reduced design occupancy with the

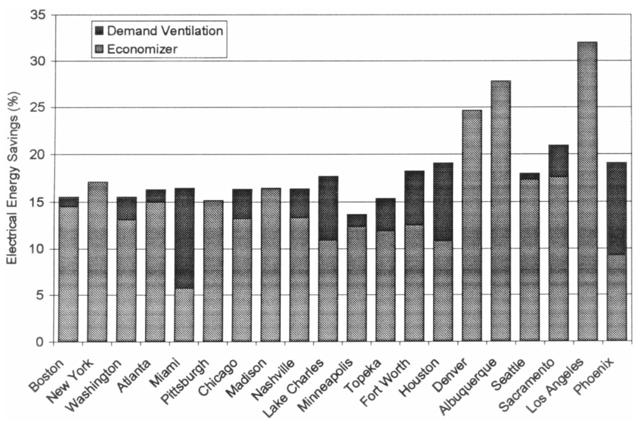

Figure 12 *Incremental energy savings for economizer and demand-controlled ventilation for restaurant with reduced minimum ventilation.*

addition of economizer cooling and demand ventilation, using the data in Table 8. Adding the economizer restores these losses and provides additional savings. Relative to the new base case, economizer savings are a greater percentage than with the original base case (Figure 11). However, the incremental savings of demand ventilation are substantially reduced. The greatest incremental saving is less than 10%, and half the locations show incremental savings of 2% or less.

CONCLUSIONS

This study demonstrated that the energy savings associated with economizer and demand-controlled ventilation strategies could be very significant. The greatest savings in electrical usage for cooling with the addition of demand-controlled ventilation occur in situations where the opportunities for economizer cooling are less. This is true for warm and humid climates and for buildings with relatively low average occupant densities compared to design occupant densities. For cooling, it is extremely important to use demand-controlled ventilation in combination with an air-side economizer. For dry climates, the use of demand-control ventilation can actually increase energy use for cooling in the absence of any economizer strategies. The savings in heating energy associated with demand-controlled ventilation can be more significant than for cooling but are strongly dependent upon the occupancy schedule. Significantly greater savings are possible for buildings with large variability in occupancy and with relatively high internal gains. It is important to note that the savings associated with demand-controlled ventilation are very dependent on the schedule of occupancy and its relationship to the design occupancy used to determine the fixed minimum ventilation rate of the base case. In this study, the design occupancies were estimated using ANSI/ASHRAE Standard 62-1989. However, in practice, minimum ventilation rates could be estimated using less conservative design occupancy estimates. In addition, the results are dependent on the assumption that occupant-generated contaminants dictate ventilation rates. Specifically, retail applications may require additional ventilation to maintain other contaminants at acceptable levels.

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