AIVC 11763

ELSEVIER

Energy and Buildings 29 (1998) 65–75

ENERGY AND BUILDINGS

A study of energy use and satisfactory zone ventilation of different outdoor air ventilation strategies for terminal reheat variable air volume systems

T.A. Reddy [a,*], M. Liu [b], D.E. Claridge [b]

[a] Civil and Architectural Engineering, Drexel University, 32nd & Chestnut Streets, Philadelphia, PA 19104, USA
[b] Energy Systems Laboratory, Texas A&M University, College Station, TX 77843, USA

Received 14 February 1997; accepted 23 April 1998

Abstract

Increased building indoor air quality (IAQ) complaints due to reduced outdoor air ventilation rates led to ASHRAE Standard 62-1989. Even though the stipulated standard total outdoor ventilation flow rate may be drawn into the HVAC system, thermal imbalances in the various zones of the building can lead to certain zones being starved of the specified ventilation flow rate, thereby creating localized IAQ problems. The objective of this paper is to compare the differences in energy use and ventilation air flow rates supplied to different zones in the building for three different practical outdoor air ventilation strategies all of which are identical in performance at design conditions but which differ under part-load operation. A simplified simulation methodology (which past studies have demonstrated to be useful for field evaluation of actual buildings) has been used to predict the heating and cooling energy use of a two-zone terminal reheat variable air volume (TRVAV) system during part-load operation specified by varying outdoor temperature and humidity conditions. The trade-off between outdoor air intake and energy use are studied for the following ventilation strategies for a typical 10,000 m^2 commercial building: (i) constant outside air intake based on a value 20% higher than the ASHRAE minimum ventilation rate, (ii) constant ventilation air intake fraction, and (iii) ventilation air intake based on the unfavorable zone requirements (even though the other zone may be over-ventilated). How this trade-off is impacted by building size has also been investigated. Finally, we use bin data for Dallas, TX (a moderately hot and humid location) and Seattle, WA (a mild location) in order to study the differences in energy use and zone ventilation flows of different ventilation strategies due to building location. The effect of economizer cycles and of varying ventilation strategies depending upon diurnal building schedules have not been considered in this study. The results of this study which are based on a simplified HVAC simulation approach are consistent with conclusions reached by other researchers using more detailed simulation models. This suggests that sound and meaningful diagnostic insights of actual building performance and operating strategies can be obtained from such simplified simulations.

Keywords: Terminal reheat variable air volume (TRVAV) system; Air ventilation; Zone

1. Objectives and scope

The importance of indoor air quality (IAQ) has increased significantly during the last decade since demographic studies have shown that currently people in the United States tend to spend up to 90% of their lives indoors [24]. The drive for energy conservation during the 1970s and the advent of variable air volume (VAV) systems that were often not operated properly led to reduced outdoor ventilation air flow rates which brought about an emergence of building air quality complaints resulting in ASHRAE Standard 62-1989 [1]. The standard specifies maximum concentration levels of common indoor contaminants and also establishes specific outdoor air ventilation requirements for various building types. For most office-type applications, the ASHRAE Standard prescribes a minimum ventilation rate of 7.5-10 l/s/person as a means of controlling the various indoor air pollutants.

For typical office buildings, the common American HVAC practice is to design the system for an outdoor air supply fraction of about 10% [14]. Having to increase ventilation air flow to meet outdoor air intake standards would then require conditioning this extra outdoor air flow leading to an increase in HVAC energy use. In several European countries, this issue is moot since from the last decade the norm is to use 100% outdoor air with a heat recovery device between the return and supply air streams. Because the American

* Corresponding author. Tel.: +1 215 8952364; fax: +1 215 8951363.

0378-7788/98/$ - see front matter
PII S0378-7788(98)00027-9

HVAC practice is still widely used in other parts of the world, the issue of energy penalty associated with increased ventilation flow is a problem worthy of study and is the raison d'etre of this paper.

Whether and to what extent annual HVAC energy operating costs are impacted by increased outdoor air ventilation rates have been studied from the last decade using the detailed DOE-2 simulation program [15]. Simulation studies have been performed by Refs. [6,7] to determine impacts of increasing the minimum outdoor air ventilation rates from 2.5 to 17 l/s/person for typical small, medium and large office buildings for 10 locations in the United States. They found that an increase from 2.5 to 10 l/s/person would increase total operating costs (i.e., HVAC plus lights and equipment) by about 5% for small offices and 3% for medium size offices (assuming the buildings have economizer cycles). Independent studies by Refs. [17–20,25] for different building sizes and location also indicate that energy impacts of increased ventilation flows are minimal to modest for VAV systems (about the same magnitude as the Eto studies). However, a similar study by Ref. [22] for a 9290 m^2 office building with a conventional VAV system in three different cities in Florida found higher HVAC energy penalties (10–15%) partly due to the higher latent loads to be found in Florida.

A total building ventilation flow rate being specified or maintained is no guarantee that individual zones are also supplied with the required ventilation flow rates. Thermal imbalances in the various zones of the building served by one air handler unit can lead to certain zones being starved of the specified ventilation flow rate thereby creating localized indoor air quality (IAQ) problems (see, e.g., Refs. [1,8]). The objective of this paper is to compare the energy use and deficiency in ventilation flow rate in individual zones for different practical outdoor air ventilation strategies, all of which are identical in performance at design conditions but which differ under part-load operation.

In order to limit the scope of the study, we shall only consider the generic terminal reheat variable air volume (TRVAV) system because of its widespread use, and also overlook the effect of economizer cycles which are widely adopted to lower energy costs. Though economizer cycles may provide adequate outdoor air flow rates when operating under certain outdoor temperature range, typically 5° to 15°C, they are a second level strategy which will not be studied in the framework of the current study. Further, we shall also not consider scheduling-related ventilation strategies (such as 'slow roll' of air-handler units and demand control of outdoor ventilation air flow rates depending on indoor CO_2 levels). We shall assume a typical medium-to-large office building serviced by a single air handler unit, and subject it to three different ventilation and control strategies. The objective of the first part of this study will be to investigate how these strategies would affect ventilation flow rates in individual zones and HVAC air-side heating and cooling thermal energy use under part-load operation (specified by variation of outdoor dry-bulb temperature). Next, we shall vary the size of the building and study how this impacts energy use and zone ventilation flows under part-load operation. Finally, the effect of building location and size will be studied by assuming bin data from two widely different locations and simulating annual energy use for these strategies. How the results of our simplified simulation approach compare with those of previous studies using detailed and sophisticated simulation models will also be investigated. If the results are consistent, then energy managers of actual buildings could resort to such types of simulations to assess the performance of their HVAC system and evaluate alternative strategies.

More recently, concerns have been raised that using a ventilation strategy based on occupancy levels may not be proper because the effect of nonhuman contaminants is ignored. This is important when building occupancy is low, as in the evenings and weekends. Ventilation strategies based on CO_2 and occupancy based demand have been studied as alternatives [12]. In this study, we shall, however, limit ourselves to compliance with ASHRAE Standard 62-1989 as being the sole criterion for satisfactory ventilation.

2. Simulation model approach and assumptions

The simplified system simulation approach initially developed by ASHRAE [14] and later, because of its simplicity and accuracy, has been found to be very appropriate for (i) enhancing the basic understanding of how HVAC systems perform when subjected to different operating and control conditions, (ii) inverse modeling (i.e., reconciling monitored energy use with engineering models) for retrofit energy savings determination, and (iii) for detecting and assessing the impact of operating changes on energy use and comfort. Numerous papers on this approach are available [11,16,21], and so, we shall but briefly describe the two basic assumptions made and the corresponding implications.

(a) Multi-zone commercial buildings often have several HVAC systems that could be of different sizes and different generic types. The simplified systems approach assumes that all HVAC systems of the same generic type can be effectively treated as a single HVAC system. Note that the objective of this approach is not to obtain accurate predictions of energy use for design purposes, but rather to be able to make meaningful comparative evaluations of various different control and operating strategies in an existing building. Further, heating and cooling energy use (channels which are typically measured when continuous monitoring is done for institutional buildings such as in the LoanSTAR program [3] in the framework of which we have refined and adopted the simplified systems approach) and not the primary energy use will be simulated. The equipment (boilers, pumps and chillers) models as well as their part load efficiencies are not included in the simulation. Finally, it is assumed that the heating and cooling coils are of infinite capacity, i.e., they are able to heat or cool the air streams to the desired temperature levels at all times of the year.

(b) Most commercial buildings have multiple zones that even under idealized conditions may simultaneously call for both heating in the outer zones and cooling in the inner zones. Further, these zones typically experience a certain amount of cross or inter-zone air flow due to small temperature and pressure differences that invariably exist in a building. The simplified systems approach assumes that the commercial building to be conditioned can be partitioned into two zones only, with one exterior or perimeter zone and one interior or core zone. Most office and commercial buildings can be conceptually broken down thus because office spaces are normally designed adjacent to windows and so form a ring around the perimeter about 4–5 m wide. Hallways could be either lumped into the perimeter zone (if office doors are normally left open), or lumped into the core zone. Further, the two zones are assumed to have identical zone set point temperatures and the internal loads are shared between both zones in proportion to the conditioned floor areas. Finally, solar and transmission loads are assumed to affect the perimeter zone only.

Other assumptions made are listed below:

(c) The analysis considers steady-state heat loads of the building;

(d) The thermostat set point temperature T_Z is fixed at a mean yearly value;

(e) Infiltration loads are assumed negligible or considered part of the ventilation loads;

(f) Solar gains are a linear function of outdoor dry-bulb temperature [26];

(g) Daily internal loads consisting of heat gains from lights and equipment and from occupants are approximately constant over the year;

(h) Ducts are perfectly insulated and have no air leakage;

(i) No economizer cycle is present (as already stated in the scope of the paper);

(j) There is no system to humidify the supply air stream if it is too dry. This is common in most office buildings in the southern United States;

(k) Constant cooling coil cold deck conditions (i.e., constant air stream temperature and humidity);

(l) Reheat coils are assumed to be placed in the supply ducts (instead of having hydronic radiators in the spaces as is common in VAV systems).

3. Modeling methodology

3.1. Building loads

We shall assume the sign convention that energy flows are positive for heat gains and vice versa. The building loads include (i) internal loads (sensible including gains from people), (ii) solar loads (both direct and transmission), (iii) shell transmission loads, and (iv) infiltration and ventilation loads (both sensible and latent).

It is usually the electricity used by lights and receptacles inside a building which can be conveniently measured. In the absence of exhaust fans and vented lighting fixtures, this use, q_{LR}, appears as a portion of the total sensible internal loads. Heat gains from people consisting of both sensible and latent portions and other types of latent loads are not amenable to direct measurement and are thus usually estimated. Since the schedule of lights and equipment closely follows that of building occupancy, a convenient and logical manner to include the unmonitored sensible loads is to modify q_{LR} by a constant multiplicative correction factor, k_s (typically in the range 1.05 to 1.3) which accounts for the miscellaneous (i.e., unmeasurable) internal sensible loads. Thus:

$$(i)+(ii)+(iii)=q_{LR}k_sA+a_{sol}+b_{sol}T_o+UA_S(T_o-T_Z) \tag{1}$$

where A is the conditioned floor area of the building.

The slope coefficient, b_{sol}, of the linearized solar function is normally small compared to the UA_S term [11]. The term (UA_S+b_{sol}) can be viewed as an 'effective' building envelope coefficient which includes the linearized solar contribution [14,26]. It is thus more convenient to rewrite Eq. (1) as:

$$(i)+(ii)+(iii)=q_{LR}k_sA+a'_{sol}+(b_{sol}+UA_S)(T_o-T_Z) \tag{2}$$

Usually, the latent load inside the building is much smaller than the latent load from ventilation. Indoor comfort can be maintained by closely controlling the indoor air temperature (which thermostats normally do) and seeing to it that during the equipment design phase the HVAC system is so rated that the indoor air relative humidity levels do not stray outside a broad range (typically between 30 and 60% relative humidity, [2]). Hence, indoor humidity is not a variable which is usually controlled on a continuous basis. Thus, a simple manner of treating internal latent loads is to introduce a constant multiplicative factor, k_l, defined as the ratio of internal latent load to the total internal sensible load (k_sq_{LR}) which appears only when outdoor specific humidity, w_o, is larger than that of the conditioned space. Such a model is adopted in order to be as closely consistent with actual HVAC system operation as is possible. Building parameters used for simulation input are shown in Table 1.

3.2. Rated and minimum supply air flow rates

The engineering principles governing energy use in practical HVAC systems as well as algorithms for simulating the hourly performance of such systems are well documented in the published literature [2,14], and we shall assume that the reader is familiar with them. Only a brief description of the modeling equations is given below.

Table 1
Values of various building parameters used for our base-case simulation

Parameters	SI units
A	10,000 m^2
T_Z	22°C
RH_Z	50%
q_{LR}/A	32 W/m^2
k_l	0.2
k_s	1.3
$m_{v,min}$ [a]	0.6×10^{-3} $kg/s/m^2$
U [b]	2.5 $W/m^2/°C$
A_S/A	0.56
A_{int}/A	0.68

[a]Chosen 20% higher than ASHRAE stipulated minimum.
[b]Includes glazing (30%), linearized solar loads and infiltration.

Let A_{int} and A_{ext} be the conditioned floor areas of the interior (or core) and of the exterior (or perimeter) zones, respectively, and A be that of the entire building. The rated supply air mass flow rate per unit conditioned area (m_{Rated}) is determined such that, a supply air stream at $T_{C,design}$ (assumed to be equal to 11°C) and 90% RH can meet the peak cooling loads (sum of sensible and latent) of the entire building. The cooling coil leaving air conditions have been chosen such that the specific humidity of air leaving the cooling coil is slightly less than that of the zone (i.e., $T_Z = 22$°C and $RH_Z = 50\%$). This would assure more or less acceptable specific humidity levels in the zones during year-round operation. In this paper, the peak cooling loads are assumed to occur at $T_{o,\,design} = 37$°C and $RH_{o,\,design} = 50\%$ (see Table 2). The value of $m_{Rated} = 0.00558$ $kg/s/m^2$ for our base case building shown in Table 2 has been determined in this manner.

The minimum supply air flow rate per unit conditioned area, m_{min}, to the conditioned space cannot be assumed to be the minimum outdoor air flow rate required to meet indoor air quality constraints. Indoor comfort requires a minimum supply air circulation rate which is larger than the minimum outdoor ventilation rate. For office spaces, [2] stipulates a value of about $3.6–5.4 \times 10^{-3}$ $kg/s/m^2$ for the minimum supply air circulation rate. We note from Table 2, which presents the inputs to our simulations, that while $(m_{v,\,min}/m_{Rated}) = 0.1075$, $(m_{min}/m_{Rated}) = 0.60$. Note that this value is appropriate for occupied hours. During unoccupied hours, some building energy managers force the air handlers to operate at a lower level, e.g., at $(m_{min}/m_{Rated}) = 0.3$. Such an operation, referred to as 'slow roll', reduces energy use, but, as stated earlier, such strategies are outside the purview of this paper.

Table 2
Additional parameters used in simulating the TRVAV system for the base case building

1. Design conditions: Outdoor air: $T_{o,design} = 37$°C and $RH_{o,design} = 50\%$; Cold deck: $T_{C,design} = 11$°C and $RH_C = 90\%$.
2. Rated building supply air flow rate per unit conditioned building area for all three strategies: $m_{Rated} = 0.00558$ $kg/s/m^2$.
3. Minimum allowable supply air flow rate to rated flow rate for all three strategies: $(m_{min}/m_{Rated}) = 0.60$.
4. Minimum ventilation air flow rate to rated flow rate for Strategy 1 (the former is chosen 20% higher than ASHRAE minimum): $(m_{v,min}/m_{Rated}) = 0.1075$.
5. Part-load system operation simulated by varying T_o only from −12°C to 37°C assuming $RH_o = 0.5$ and constant internal loads.

3.3. Supply air flow rates

The supply air flow rates to each zone are determined as follows. For the interior zone:

$$m_{int} A_{int} = \max\left[m_{min} A_{int}, \frac{q_{LR} k_s A_{int}}{c(T_Z - T_C)}\right] \quad (3)$$

and flow rate through the exterior zone:

$$m_{ext} A_{ext} = \max\left[m_{min} A_{ext}, \frac{a'_{sol} + q_{LR} k_s A_{ext} + (b_{sol} + UA_S)(T_o - T_Z)}{c(T_Z - T_C)}\right] \quad (4)$$

Thus the total supply air flow rate per unit area:

$$m = \frac{(m_{int} A_{int} + m_{ext} A_{ext})}{A} \quad (5)$$

3.4. Heating and cooling energy

Heat and mass balances at the air recycle point will yield values for T_m and w_m. The expression for cooling energy is made up of sensible cooling and latent cooling:

$$E_C = mA[c(T_m - T_C)^+ + h_v (w_m - w_c)^+] \quad (6)$$

where w_C is the specific humidity of air at $T_C = 11$°C and $RH = 90\%$ (see Table 2).

The sign convention $()^+$ signifies that the term within the parenthesis should be set to zero if negative. The expression for the heating energy is:

$$E_H = m_{int} A_{int} c(T_{s,int} - T_C)^+ + m_{ext} A_{ext} c(T_{s,ext} - T_C)^+ \quad (7)$$

where $T_{s,int}$ and $T_{s,ext}$ are computed from sensible heat balances on the individual zones assuming supply air flow rates determined by Eqs. (3) and (4).

4. Different ventilation strategies studied

A number of papers have discussed the issue of how to operate and control VAV systems for acceptable building ventilation (e.g., Refs. [4,8–10,13,23]). In this paper, we shall not consider the practical issues of how to implement the various ventilation strategies chosen but will limit ourselves to providing a theoretical evaluation. The ventilation air flow rate needs to be known in order to use the heating and cooling expressions shown earlier. In fact, the basic objective of this paper is to study how energy use and zone ventilation air flow rates vary when different ventilation air flow strategies are implemented. Since the ventilation standards are specified in terms of unit occupant while we wish to perform our simulation on the basis of unit conditioned area, we shall assume a widely used value of 17 m^2/occupant. The various ventilation strategies studied in this paper are described below.

4.1. S1: Constant outside air flow rate based on total fresh air requirement

We shall assume here that the outdoor ventilation air flow rate is constant throughout the year and satisfies the minimum outdoor air flow rate of 7.5 l/s/occupant stipulated by ASHRAE Standard 62-1989 [1]. If an occupancy density of 17 m^2/occupant is assumed, the recommended minimum standard value would be $m_{v,min}* = 0.50 \times 10^{-3}$ kg/s/m^2. Hence, in this study, we shall assume a conservative value of $m_{v,min} = 0.60 \times 10^{-3}$ kg/s/m^2 which is higher than the ASHRAE minimum by about 20%. Note that in a TRVAV system where the total supply air flow rate is modulated depending on the operating conditions, the outdoor recycle fraction (i.e., the fraction of ventilation air to supply air flow rates) is no longer a constant.

4.2. S2: Constant outside air intake fraction

The ventilation strategy S1 is difficult to follow in practical systems because it requires relatively sophisticated control of both fresh air and return air dampers simultaneously. If these dampers were left uncontrolled altogether during the operation of a VAV system, the ventilation air during different times of the year would change along with the variable supply air flow rate such that the ratio of outdoor ventilation air flow rate to supply air flow rate is more or less constant. This is what usually occurs in practical systems, and a ratio of 0.1 is typical. When older HVAC systems operated under constant air volume (CAV) operation are retrofitted to variable air volume (VAV) operation, the retrofits usually involve installing variable frequency drives and terminal boxes. Very often, the modification of outside air dampers is overlooked or deemed too problematic to perform. Under such circumstances, a constant outside air intake fraction (m_v/m) (which provided sufficient ventilation air when the HVAC system was operated as a constant air volume system with $m = m_{Rated}$) would result in the TRVAV system being starved of ventilation air during winter (when m is low). This problem is very often not realized even when the HVAC system is rebalanced and recommissioned after retrofit since this is usually done in summer when m is high. Hence, strategy S2 simulates a practical problem often encountered when HVAC retrofits from CAV to VAV are done. While simulating this ventilation strategy, we shall assume a value of (m_v/m) = 0.1075 in order to be consistent with S1 under design conditions. However, the equivalence does not hold during part-load operation since, under strategy S2, the outdoor air flow rates are reduced in proportion to supply flow rates.

4.3. S3: Outside air intake based on the unfavorable zone requirements (even though the other zone may be over-ventilated)

The previous two cases merely assumed that drawing in the required total ventilation air flow rate into the HVAC system would satisfy ventilation standards of individual zones. Because the individual zone to total supply air stream fractions of both zones are usually not equal, one zone may be starved of ventilation air while the other may be over-ventilated. Strategy S3 will guarantee that each zone is supplied by, at least, the minimum fresh air flow rate even if the other zone is over-ventilated as a result. The minimum ventilation flow rate per unit conditioned area of the particular zone will be chosen to be 0.6×10^{-3} kg/s/m^2 in order to be consistent with S1. How to model such a strategy is documented in Ref. [1]. In order to make our analysis more conservative, we have neglected the effect of 'vitiated' air. Let us normalize the flows to the interior and exterior zones as follows:

$$f_{m,int} = (m_{int}/m) \text{ and } f_{m,ext} = (m_{ext}/m). \tag{8}$$

Because m is defined as flow rate per unit conditioned area, $f_{m,int}$ or $f_{m,ext}$ can be greater than 1.0. Also note that $(f_{m,int} + f_{m,ext}) \neq 1$, rather $(f_{m,int} + f_{m,ext}) m = m_{int} A_{int} + m_{ext} A_{ext}$. Under S3, we would choose the ventilation flow rate as follows:

$$m_v = \min \{ m_{Rated}, [m_{v,min} \max (f_{m,int}, f_{m,ext})] \} \tag{9}$$

Thus, m_v for S3 will take in excess ventilation air such that neither zone is starved of the stipulated ventilation air flow rate per unit area, provided, of course, that the corresponding total supply air flow rate does not exceed the rated flow rate. Such a condition had to be imposed because the supply fan is chosen based on the rated flow, and once installed is incapable of handling a higher flow rate. However, in all our simulation runs, such an eventuality did not occur.

During our simulations, we would like to keep track of the *extent* to which the starved zone is deficient in ventilation flow rate and not merely flag the occurrence. This measure is provided by the following factor:

$$F_{IAQ} = \min (f_{m,int}, f_{m,ext}) m_v / m_{v,min}* \tag{10}$$

where $m_{v,min}*$ is the ASHRAE stipulated minimum ($= 0.5 \times 10^{-3}$ kg/s/m^2). Thus, a value of unity or above signifies that the ASHRAE standard has been met, while, say $F_{IAQ} = 0.9$ indicates that the starved zone is supplied by a ventilation flow rate per unit area equal to 90% of the ASHRAE minimum. Note that S1 assumes $(m_v/m_{v,min}*) = 1.2$, i.e., we allow 20% excess ventilation air into the system as a precaution against an individual zone being starved.

5. Results and discussion

5.1. Selection of inputs

We shall assume for our simulations the values for building parameters listed in Tables 1 and 2. Note that all values of mass flow rates and heating and cooling energy use presented in this paper are on the basis of unit conditioned area. Area of the building is stipulated in order to get realistic values of (A_{int}/A) and (A_S/A) fractions. The base case building has been assumed to be of square geometry with an area of 10,000 m^2 with 3 floors of total height of 10 m. The external zone is assumed to be 5 m wide for determining (A_{int}/A). Such a selection provided us with the values of (A_{int}/A) and (A_S/A) fractions listed in Table 1. Increasing or decreasing the building size merely varies these two fractions (see Table 3). As mentioned earlier, m_{Rated} for each building size is determined such that peak cooling loads (assumed to occur at $T_{o,design}=37°C$ and $RH_{o,design}=50\%$) can be met with a cold deck temperature $T_C=11°C$ and $RH_C=90\%$.

5.2. Inter-comparison of various ventilation strategies for our base case building

How heating and cooling energy use vary with T_o is shown in Fig. 1 for all three ventilation strategies. We note that E_H is identical for all three strategies which is obvious given that in terminal reheat systems the cooling coil separates the effect of heating coils from the mixed air condition. Cooling energy for S3 (where zone ventilation standard is always satisfied in both zones) is higher than that of S1 and S2 for higher T_o values ($T_o > T_Z = 22°C$) and lower than S2 for lower T_o values. In fact, E_C for S3 is only slightly higher than that of S1 in the lower T_o range. The cooling energy for S2 is higher at low T_o values because this strategy takes in less ventilation flow under such conditions resulting in higher T_m values and thus more cooling.

The variation of $f_{m,int}$ and $f_{m,ext}$ defined by Eq. (8) with T_o is shown in Fig. 2. Note that this is independent of the ventilation strategy chosen since the supply flow rate splits depending only on the load distribution ratio of the two zones.

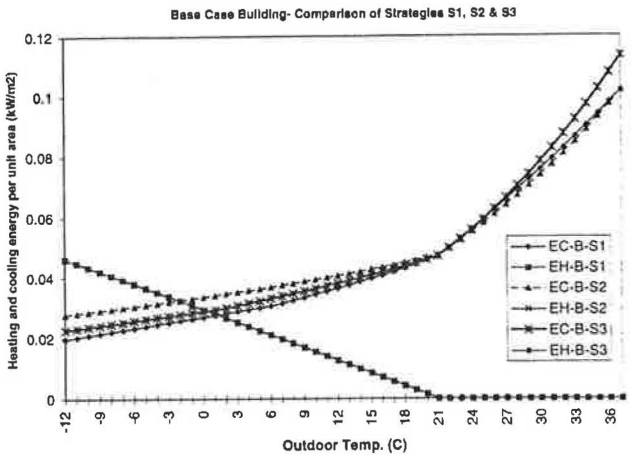

Fig. 1. Variation of heating and cooling thermal energy use with outdoor temperature for the base case building operated under the three ventilation strategies considered. Note that the heating energy use is not affected by ventilation strategy.

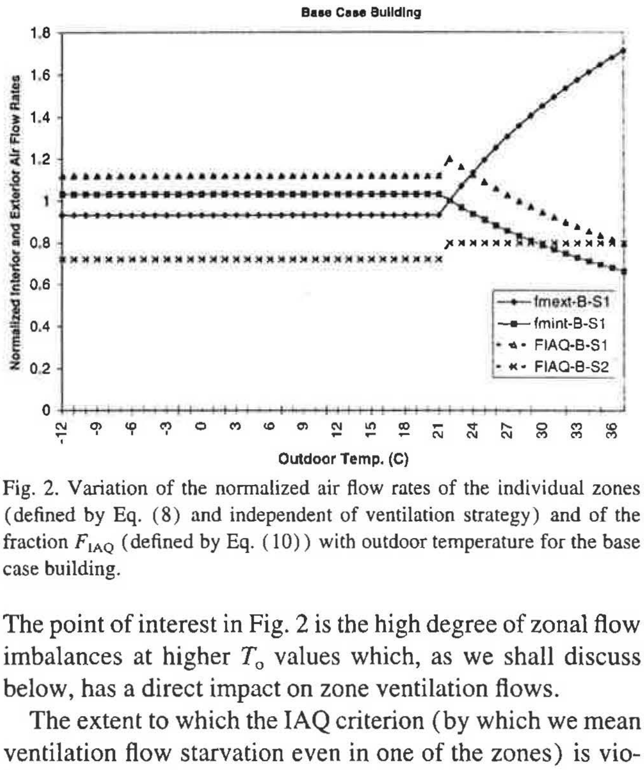

Fig. 2. Variation of the normalized air flow rates of the individual zones (defined by Eq. (8) and independent of ventilation strategy) and of the fraction F_{IAQ} (defined by Eq. (10)) with outdoor temperature for the base case building.

The point of interest in Fig. 2 is the high degree of zonal flow imbalances at higher T_o values which, as we shall discuss below, has a direct impact on zone ventilation flows.

The extent to which the IAQ criterion (by which we mean ventilation flow starvation even in one of the zones) is violated is given by F_{IAQ} (defined by Eq. (10)). How F_{IAQ} varies with T_o for S1 and S2 is also shown in Fig. 2. We note that in S2, the ventilation standard is *never* satisfied since

Table 3
Pertinent input simulation data for the three sizes of building simulated

	A (m^2)	U ($W/m^2/°C$)	A_S/A	A_{int}/A	m_{min}/m_{Rated}	m_{Rated} ($kg/s/m^2$)
Base case (B)	10,000	2.5	0.56	0.68	0.600	0.00558
Smaller bldg. (S)	5,000	2.5	2.66	0.57	0.266	0.01261
Larger bldg. (L)	20,000	2.5	0.49	0.77	0.626	0.00535

The building is square with 3 floors and total height of 10 m. Exterior zone is assumed to be 5 m wide for determining fraction of interior to total area. $m_{min}=0.00335$ $kg/s/m^2$ for all three building sizes.

$F_{IAQ} = 0.8$ for $T_o > 22°C$ and equal to 0.75 for lower T_o values. This implies that the starved zone is always fed by about 20% less ventilation air than the ASHRAE minimum. F_{IAQ} for S1, on the other hand, is less than unity only for $T_o > 27°C$ while ventilation requirements of both zones are satisfied for lower T_o values. Note the similarity of this behavior with that of the flow imbalance variation.

How the ventilation flow rates vary with T_o for the three ventilation strategies is shown in Fig. 3. Recall that S1 assumes a constant value of $m_v = 0.6 \times 10^{-3}$ kg/s/m^2 (20% higher than ASHRAE minimum) throughout the simulation range. Since S2 assumes a constant value of (m_v/m), and the VAV system modulates the flow with decreasing temperature, there is a decrease in m_v. The fraction $(m_v/m) = 0.1075$ has been chosen such that at peak cooling conditions (i.e., $T_o = 37°C$), the ventilation flows of S1 and S2 are equal. How m_v varies for S3 is noteworthy. It is about 45% higher at peak cooling condition but ramps down and reaches a minimum which is in between those of S1 and S2. Notice that the fresh air intake for S3 at its minimum is just above the ASHRAE minimum of 0.5×10^{-3} kg/s/m^2, while that of S2 is lower meaning that the ventilation standard is not satisfied.

5.3. Effect of building size

Using the simulation inputs listed in Table 3, we have generated the heating and cooling energy use plots for the three building sizes chosen. There is more load imbalance between both zones at higher T_o values for the small building as shown by the variation of $f_{m,int}$ and $f_{m,ext}$ in Fig. 4. The variation of the lesser of the two normalized flows, $f_{m,int}$ and $f_{m,ext}$, dictates F_{IAQ} (see Eq. (10)), and so one would, looking at Fig. 4, deduce that much more severe ventilation starvation would be experienced by the smaller building at high T_o values, while the base case building and the larger building would be similar. How F_{IAQ} varies with T_o for the three different building sizes for S1 and S2 can be seen in Fig. 5. The starved zone of the smaller building under S2 gets less than 40% of the ASHRAE minimum, which is about half of that received by the base case building and the large building under the same ventilation strategy. Even S1 applied to the small building will provide less than stipulated minimum ventilation air to the starved zone when $T_o > 22°C$. The same ventilation strategy S1 for the base case building and for the large building will result in inadequate ventilation requirements only when T_o exceeds 28°C or so.

5.4. Effect of location

Fig. 6 depicts the bin temperature data for Dallas, TX (a moderately hot and humid locations) and for Seattle, WA (a mild and less humid location) taken from Ref. [5]. We have used this data to simulate the performance of the TRVAV system when the three different strategies are applied to each of the three building sizes assumed. Table 4 gives the values of m_{Rated} for each case which have been determined from peak cooling loads of the particular location using cold deck specifications given in Table 2. (Note that these are different from those in Table 3 because the $T_{o,design}$ values are differ-

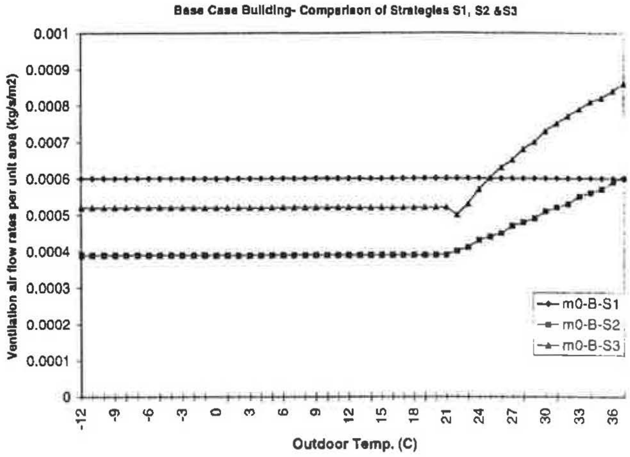

Fig. 3. Variation of ventilation flow rates with outdoor temperature for the base case building operated under the three ventilation strategies.

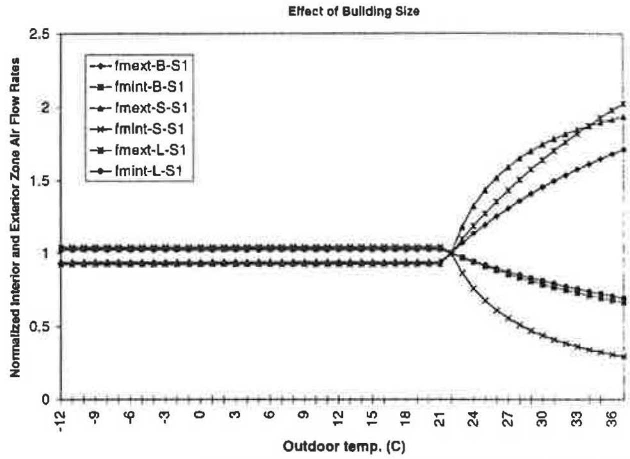

Fig. 4. Variation of normalized air flow rates (defined by Eq. (8)) with outdoor temperature for all three building sizes. The variations are independent of ventilation strategy.

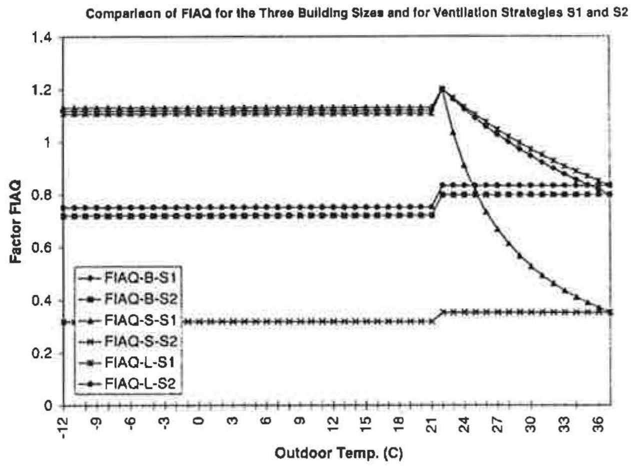

Fig. 5. Variation F_{IAQ} (defined by Eq. (10)) for the three building sizes operated under ventilation strategies S1 and S2.

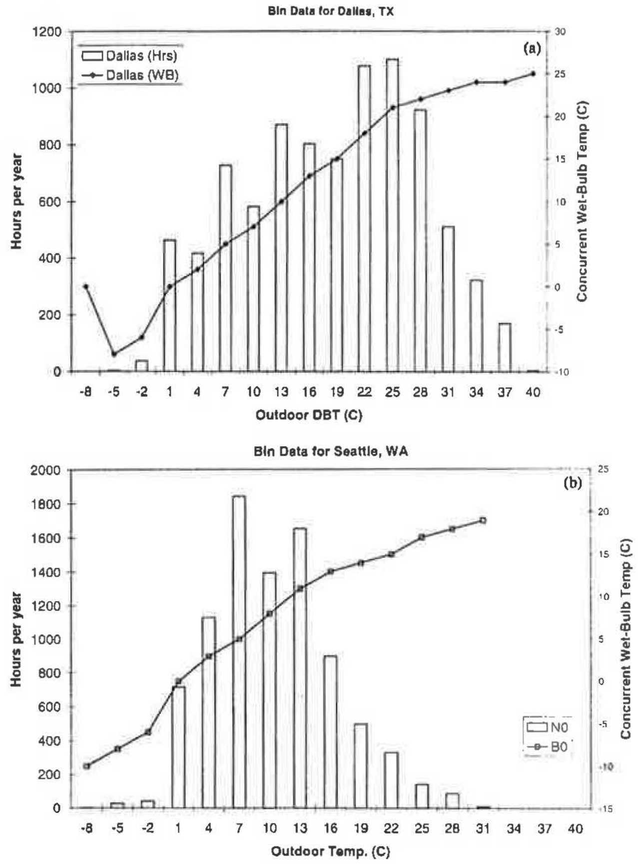

Fig. 6. Number of hours of outdoor dry-bulb temperature and concurrent wet-bulb temperature for the two cities used for annual simulations.

Table 4
Inputs for annual energy use simulation

Parameters		Dallas, TX	Seattle, WA
$T_{o,design}$		40°C	31°C
$WB_{o,design}$		25°C	19°C
Range of simulation		− 5° to 40°C	− 8° to 31°C
m_{Rated} (kg/s/m²)	Base case bldg.	0.00596	0.00483
	Small bldg.	0.01439	0.00905
	Large bldg.	0.00568	0.00469

Other inputs are listed in Table 3.

ent). Table 5 assembles the simulation results of annual heating and cooling energy use for the various cases simulated. Also included is the bin-hour-weighted annual fraction $\tilde{u}_{IAQ}$ when stipulated minimum ventilation flow rate to either zone is *not satisfied*:

$$u_{IAQ,i} = \delta N_i/(24 \times 365) \quad (11)$$

and:

$$\tilde{u}_{IAQ} = \sum_{i=1}^{n} u_{IAQ,i} \quad (12)$$

where:

$$\delta = 0 \text{ when } F_{IAQ} > 1, \text{ and 1 otherwise} \quad (13)$$

N_i is the number of hours in the particular bin i, and n is the number of temperature bins for the particular location selected.

The objectives of the study are better served by looking at the ratio of annual energy use of different strategies and building sizes with respect to that of the base case building rather than absolute values listed in Table 5. Fig. 7 shows these ratios for both locations under S1. We note that cooling energy use in a mild location such as Seattle is almost independent of building size while in Dallas the smaller building consumes about 25% more energy per unit area. What is striking in Fig. 7 is the fivefold increase in heating energy use per unit area for the smaller building in both locations.

Fig. 8 shows how F_{IAQ} (defined by Eq. (10)) varies with T_o for the three building sizes when subjected to ventilation strategies S1 and S2. Note that F_{IAQ} is always equal to 1 for

Table 5
Simulation results of annual thermal heating and cooling energy use per unit area in kWh/m²/yr for different ventilation strategies and for the two locations selected

		Dallas, TX			Seattle, WA		
		S1	S2	S3	S1	S2	S3
Base case bldg.	EC	460.49	467.91	472.58	333.09	356.58	346.07
	EH	65.35	65.38	65.36	133.63	133.66	133.65
	$\tilde{u}_{IAQ}$	0.22	1.00	0.00	0.01	1.00	0.00
Small bldg.	EC	583.13	596.56	603.45	335.23	391.27	347.53
	EH	330.92	331.16	330.96	670.83	671.21	670.92
	$\tilde{u}_{IAQ}$	0.35	1.00	0.00	0.03	1.00	0.00
Large bldg.	EC	458.35	465.00	476.25	336.43	357.25	350.21
	EH	58.12	58.14	58.13	118.58	118.61	118.60
	$\tilde{u}_{IAQ}$	0.11	1.00	0.00	0.00	1.00	0.00

$\tilde{u}_{IAQ}$ is the bin-hour-weighted annual fraction of the number of hours in the year when IAQ is not satisfied (defined by Eq. (12)).

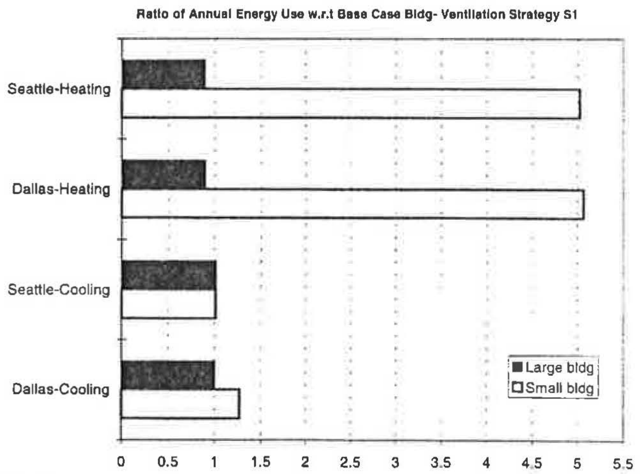

Fig. 7. Ratio of annual heating and cooling energy use of the small and large buildings with respect to the base building operated under ventilation strategy S1.

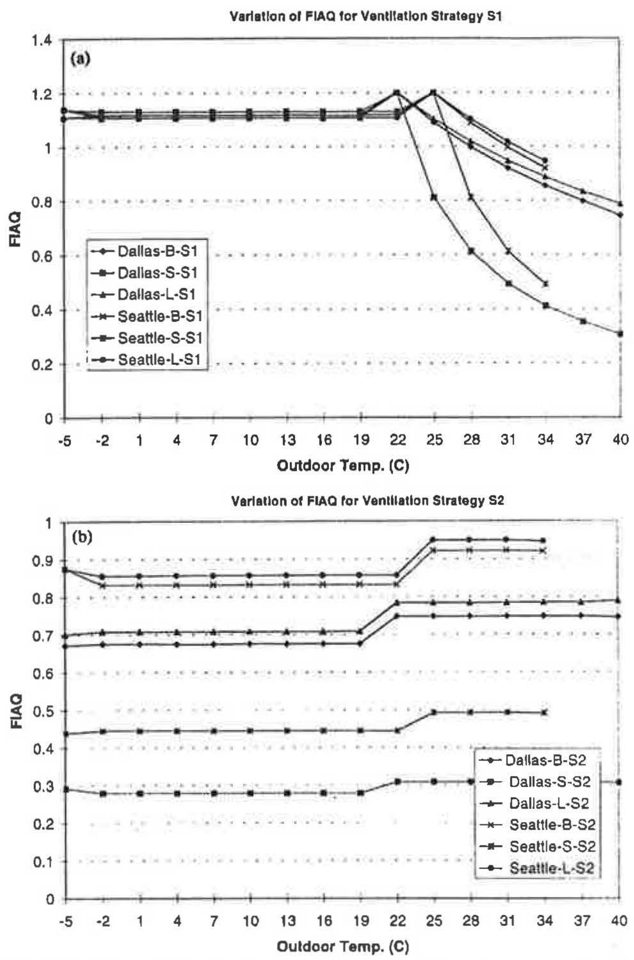

Fig. 8. Variation of F_{IAQ} for the three different building sizes operated under ventilation strategies S1 and S2 in Dallas and Seattle.

S3 and so is not shown. We note that adopting S2 results in unsatisfactory zonal ventilation under all operating conditions and for all three building sizes, being much more acute for the small building. In the case of S1, such unsatisfactory IAQ conditions prevail only for higher T_o values when the lines intersect the unity abscissa. As pointed out earlier, these T_o points are lower for the smaller building and about 5–6°C higher for the base case and large buildings.

The ratios of annual thermal energy use for different locations and building sizes for the three ventilation strategies are shown in Fig. 9. As pointed out in Fig. 7 for S1, heating energy use is almost independent of location, building size and ventilation strategy. Cooling energy use for S2 is higher than that of S1 for all three building sizes, it being more pronounced for Seattle (mild location) and for smaller building size. Since we have found earlier that zonal imbalance of ventilation flow is adversely affected at higher T_o values, we would expect adequate ventilation supply to each zone to be better satisfied in a mild location such as Seattle just because the associated number of hours is less. From Table 5, we note that $\tilde{u}_{IAQ}$ values of S1 are very low (i.e., satisfactory zonal ventilation flow) for Seattle and in the range of 0.11–0.35 for Dallas. On the other hand, $\tilde{u}_{IAQ}$ values are unity for all cases when operated under S2. Thus ventilation strategy S1 is far superior to S2 in terms of both energy and required ventilation flow to each zone.

Looking at Fig. 9b, we note that cooling energy used by S3 is less than 4% higher than that of S1 while being far superior in terms of ventilation. Thus from Table 5, we see that the penalty in excess energy for eliminating problems of zone ventilation flow starvation in the base case (reducing $\tilde{u}_{IAQ}$ values from 0.22 to 0.00) in Dallas is less than 3%. In the case of Seattle, it takes relatively more cooling energy (about 4%) to reduce $\tilde{u}_{IAQ}$ values from 0.01 to 0.00! Though the excess cooling energy use between S3 and S1 is very low, there seems to be a very pronounced location-dependent effect on the synergy between meeting zonal ventilation requirements and energy use. Coming finally to Fig. 9c, we notice that for Seattle, S3 uses less cooling energy than does S2 while eliminating adverse IAQ effects completely. Even for Dallas there seems to be only a 1–2% increase in cooling energy.

6. Conclusions

Several important conclusions have been reached from the results of the present simulation study.

(a) The most-often adopted ventilation strategy of maintaining a constant fresh air intake fraction (strategy S2) always leads to fresh air flow rates to the starved zone which are far lower than the ASHRAE stipulated minimum. Contrary to popular belief, the zonal ventilation deficiency problem in VAV systems may be more acute in summer when the flow distribution to the two individual zones is more nonuniform than in winter when the total ventilation flow is lower than that during summer. This conclusion, it must be pointed out, is limited to the case studied in this paper where a year-round constant cold deck temperature was assumed (as against an outdoor air temperature reset schedule).

(b) Adopting a strategy where the fresh air intake flow rate is constant (strategy S1) and 20% higher than the ASHRAE minimum does not necessarily eliminate the problem of individual zone ventilation deficiency specially in hot locations and in smaller buildings. While S1 is satisfactory in a location such as Seattle, it is unsatisfactory in a hotter location such as Dallas since we found that in a medium sized building (area of 10,000 m^2), the ventilation requirement is compromised during 22% of the hours. In a smaller building, this fraction is 35%.

(c) The heating energy use, as expected of a TRVAV system, is almost independent of the ventilation strategy used.

(d) Completely eliminating the problem of improper zone ventilation (as when S3 is adopted) may require, in fact, *less* energy than S2 in a moderate location such as Seattle, while even in a hotter location such as Dallas, the cooling energy use is only 1–2% higher. Though cooling energy use with S3 is higher than that using S1, the increase is only 2–4% in both locations considered. The interaction between cooling energy

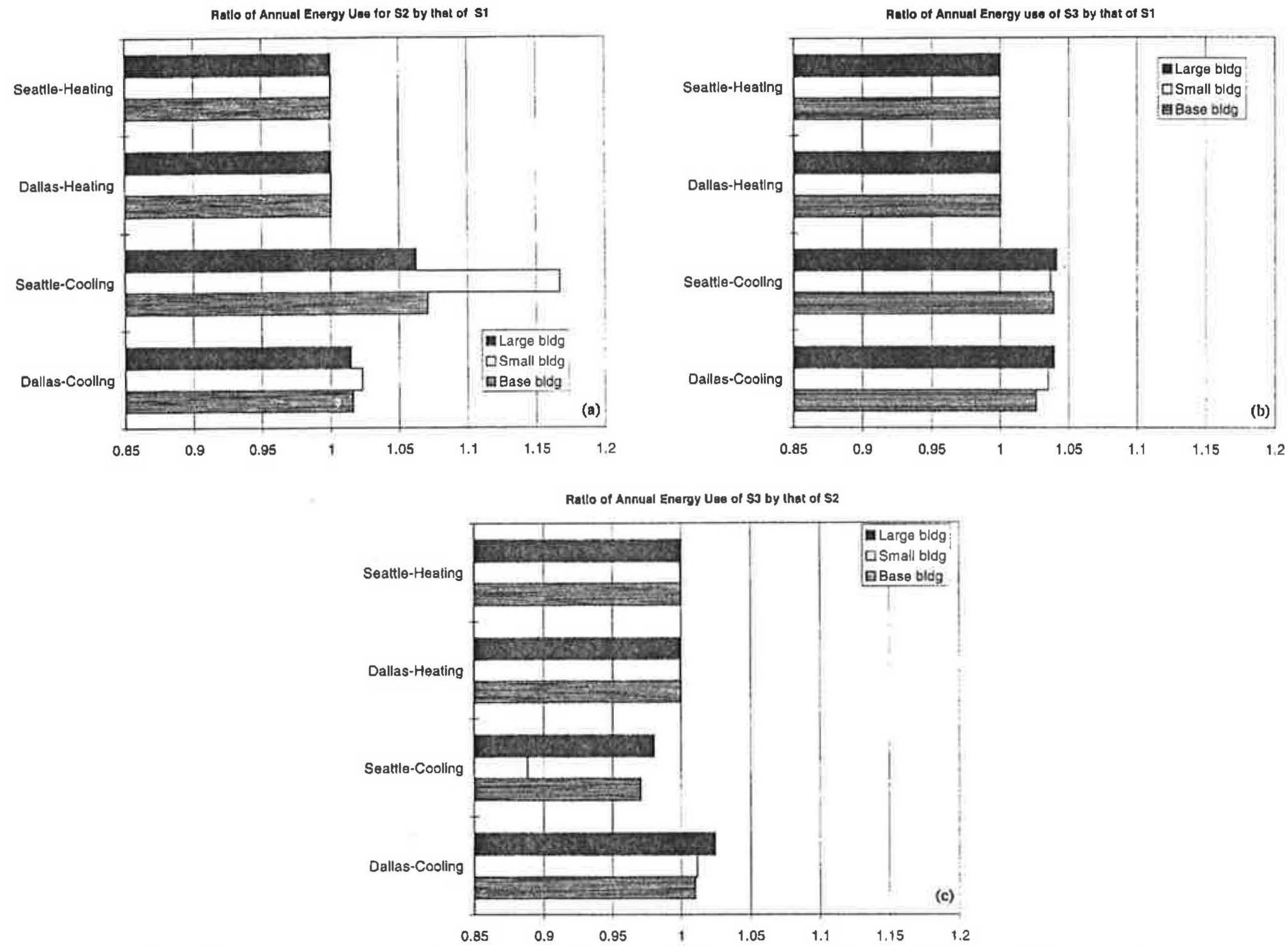

Fig. 9. Ratios of annual energy use for different locations and buildings sizes operated under different ventilation strategies.

use and zone ventilation is location-dependent. It takes S3 about 3% more cooling energy to eliminate this IAQ problem as compared to S1 in a medium sized building in Dallas where $\tilde{u}_{IAQ} = 22\%$. For the same building in Dallas, the extra cooling energy is about 4% to eliminate zone ventilation deficiency problems during 1% of the time.

Though the conclusions are striking and of practical interest to building managers and HVAC designers, these should be treated as preliminary. Further, they are applicable within the framework of the adopted methodology (like a two-zone building, a single air handler unit, no economizer, etc.) and assumptions made (internal load distribution between zones, cold deck setting, etc.). However, the general conclusions are consistent qualitatively and quantitatively with other studies, especially that of Ref. [18] which is close in scope to this study while relying on detailed and sophisticated building simulation programs. Hence, the use of simplified simulation methodology adopted here can be used to provide sound and meaningful diagnostic insights into how actual buildings should be operated so as to minimize energy use while avoiding the problem of inadequate flows in individual zones.

7. Nomenclature

A	Conditioned floor area of building
A_S	Surface area of building
c	Specific heat at constant pressure
E	Whole-building HVAC system thermal energy use
F_{IAQ}	Factor given by Eq. (10) which quantifies the extent to which the starved zone is deficient of the required ventilation flow rate
f	Normalized flow rate defined by Eq. (8)
h_v	Heat of vaporization
k_l	Ratio of internal latent loads to total internal sensible loads of building
k_s	Multiplicative factor for converting q_{LR} to total internal sensible loads
m	Total supply air flow rate per unit conditioned floor area of building
m_{min}	Minimum supply air flow rate per unit conditioned floor area of building
m_v	Ventilation or outdoor air flow rate per unit conditioned floor area

$m_{v,min}$ Minimum ventilation air flow rate per unit conditioned area
$m_{v,min}^*$ ASHRAE recommended minimum ventilation air flow rate
N_i Number of hours in bin i
n Number of temperature bins of the location
q_{LR} Monitored building lights and receptacles electricity use per unit area
RH Relative humidity
T Dry-bulb temperature
U Overall building shell heat loss coefficient
WB Wet-bulb temperature
w Specific humidity
$\bar{u}_{IAQ}$ Bin-weighted annual fraction of the number of hours in the year when IAQ is not satisfied, defined by Eq. (12)

Subscripts

a Air
C Cooling, cold deck
ext Exterior zone
H Heating, hot deck
int Interior zone
m Mixed air
min Minimum
o Outdoor
Rated Rated
s Supply air
sol Solar
v Ventilation
z Zone set point

Greek

δ Indicator variable defined by Eq. (13)

Acknowledgements

This research was performed as part of the analysis and O&M effort of the Texas LoanSTAR Monitoring and Analysis Program funded by the State Energy Conservation Office. Comments by the anonymous reviewers of this paper are acknowledged.

References

[1] ASHRAE, Ventilation For Acceptable Indoor Air Quality, Standard 62-1989, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA, 1989.
[2] ASHRAE, Fundamentals, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA, 1993.
[3] D. Claridge, J. Haberl, W. Turner, D. O'Neal, W. Heffington, C. Tombari, S. Jaeger, Improving Energy Conservation Retrofits with Measured Savings, ASHRAE Journal, October 1991.
[4] T. Cohen, Providing Constant Ventilation in Variable Air Volume Systems, ASHRAE Journal, May 1994, 38–40.
[5] L.O. Degelman, Development of Bin Weather Data, ASHRAE RP-385, American Society of Heating, Refrigeration and Air-Conditioning Engineers, Atlanta, GA, 1984.
[6] J. Eto, C. Meyer, The HVAC costs of fresh air ventilation in office buildings, ASHRAE Transactions 94 (2) (1988) .
[7] J. Eto, The HVAC Costs of Increased Fresh Air Ventilation Rates in Office Buildings, Part 2, Proceedings of Indoor Air 90: The Fifth International Conference on Indoor Air Quality and Climate, Toronto, Canada, 1990.
[8] M.J. Filardo, Outdoor air: how much is enough, ASHRAE Journal, January 1993, 34–38.
[9] K.E. Gill, IAQ and air handling unit design, HPAC Journal, January 1996, 49–54.
[10] G.J. Janu, D.J. Wenger, C.G. Nesler, Strategies for outdoor airflow control from a systems perspective, ASHRAE Transactions 101 (1995) Pt. 2, paper SD-95-2-4.
[11] S. Katipamula, D.E. Claridge, Use of simplified systems models to measure retrofit savings, ASME Journal of Solar Energy Engineering 115 (1993) 57–68.
[12] Y.P. Ke, S.A. Mumma, Using carbon dioxide measurements to determine occupancy for ventilation controls, ASHRAE Transactions 103 (1997) Pt. 2, paper BN-97-1.
[13] J.P. Kettler, Minimum ventilation control for VAV systems: fan tracking vs. workable solutions, ASHRAE Transactions, paper SD-95-2-3, 1995.
[14] D.E. Knebel, Simplified Energy Analysis Using the Modified Bin Method, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA, 1983.
[15] LBL, DOE-2 Supplement, Ver 2.1D, Lawrence Berkeley Laboratory Report No. LBL-8706 Rev. 5 Supplement, DOE-2 User Coordination Office, LBL, Berkeley, CA, 1989.
[16] M. Liu, D.E. Claridge, Application of calibrated HVAC system models to identify component malfunctions and to optimize the operation and control schedules, In: W.B. Stine, T. Tanaka, D.E. Claridge (Eds.), Solar Engineering, ASME/JSME/JSES International Solar Energy Conference, Maui, Hawaii, March 1995.
[17] D. Mudarri, J. Hall, Increasing outdoor air flow rates in existing buildings, Proceedings of Indoor Air '93, Helsinki 5 (1993) 21–26.
[18] D. Mudarri, J.D. Hall, E. Werling, D. Meisegeier, Impacts of increased outdoor air flow rates on annual HVAC energy costs, Proceedings of the 1996 Summer Study on Energy Efficiency in Buildings, Conference of the American Council for an Energy-Efficient Economy, Asilomar, CA, August 1996.
[19] D. Mudarri, J.D. Hall, E. Werling, Energy Costs and IAQ performance of ventilation systems and controls, Proceedings of IAQ 96: Paths to Better Building Environments, Conference of the American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, 1996.
[20] S.A. Mumma, J.C. Hwang, Outdoor Air Control in All Air VAV Systems that Utilize Floor to Floor Air Handling Units, Final report to CABS, Penn State University, College Park, PA, 1992.
[21] T.A. Reddy, S. Katipamula, J.K. Kissock, D.E. Claridge, The functional basis of steady-state thermal energy use in air-side HVAC equipment, ASME Journal of Solar Energy Engineering 117 (1995) 21.
[22] K. Rengarajan, D.B. Shirey, R.A. Raustad, ASHRAE Transactions 102 (1996) paper 3949.
[23] H.J. Sauer, R.H. Howell, Estimating the indoor air quality and energy performance of VAV systems, ASHRAE Journal, July 1992, 43–50.
[24] SMACNA, Indoor Air Quality, 2nd edn., Sheet Metal and Air Conditioning Contractors National Association, Chantilly, VA, July 1993.
[25] S.T. Taylor, Determining ventilation rates: revisions to standard 62-1989, ASHRAE Journal, Feb. 1996.
[26] M. Vadon, J.F. Kreider, L.K. Norford, Improvement of the solar calculations in the modified bin method, ASHRAE Transactions 97 (1991) 204–211 Part 2.