Energy and IAQ Implications of Different Outdoor Air Ventilation Strategies for Terminal Reheat Variable Air Volume Systems

T.A. Reddy, M. Liu, and D.E. Claridge, Energy Systems Laboratory, Texas A & M University

Increased building indoor air quality (IAQ) complaints due to reduced outdoor air ventilation rates led to ASHRAE Standard 62-1989. Even though the stipulated 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 energy and IAQ implications of different practical outdoor air ventilation strategies all of which are identical in performance at design conditions but which differ under part-load operation. A simulation methodology and a computer program have been developed 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 IAQ 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). Another issue which has been investigated is the manner in which the size of the building affects energy use and IAQ. Finally, we use bin data for Dallas, TX (a moderately hot and humid location) and Seattle, WA (a mild location) in order to determine the energy and IAQ implications of different ventilation strategies on 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 will provide energy managers and HVAC designers insights into how to better operate/design TRVAV systems that will satisfy IAQ criteria while minimizing energy use.

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 (SMACNA 1993). 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 (ASHRAE 1989). 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 ventilation rate of 7.5–10 l/s (15–20 cfm) per person as a means of controlling the various indoor air pollutants.

That increased outdoor ventilation flow rates will increase building energy use has been pointed out in several studies (see for example, Taylor 1996; Rengarajan et al. 1996). However, given a total building ventilation flow rate, 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 for example, Filardo 1993). The objective of this paper is to compare the energy and IAQ implications of different practical outdoor air ventilation strategies all of which are identical in performance at design conditions but which differ under part-load operation. Note that in this paper, we shall use the term IAQ in its narrow sense by relating it only to the ventilation air flow requirements. Hence when we say that IAQ is satisfied in a zone, it would merely mean that the ventilation air flow rate to that zone is equal to or higher than the ASHRAE stipulated value.

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 used 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 (40° to 60° F), 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 IAQ and HVAC air-side heating and cooling thermal energy 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 IAQ 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. The results of this study will provide energy managers and HVAC designers insights into how to better operate/design TRVAV systems that will satisfy comfort IAQ criteria while minimizing energy use.

SIMULATION MODEL APPROACH AND ASSUMPTIONS

We decided not to use any of the public domain detailed HVAC system simulation software (such as DOE-2, TRNSYS, BLAST,. . .), but instead, use the simplified system simulation approach initially developed by ASHRAE (Knebel 1983) and later, because of its simplicity and accuracy, 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 (Katipamula and Claridge 1993; Reddy et al. 1995; Liu and Claridge 1995), 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 which 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. 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 (Claridge et al. 1991) 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 which even under idealized conditions may simultaneously call for both heating in the outer zones and cooling in the inner zones. 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 5–7 m (15–22 ft) wide. Corridors 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 (Vadon et al. 1991);

(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).

MODELING METHODOLOGY

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

$$\text{(i)} + \text{(ii)} + \text{(iii)} = q_{LR} k_s A + a_{sol} + b_{sol} T_o + UA_s (T_o - T_z) \tag{1}$$

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 (Katipamula and Claridge 1993). The term $(UA_S + b_{sol})$ can be viewed as an ''effective'' building envelope coefficient which includes the linearized solar contribution (Knebel 1983; Vadon et al., 1991). It is thus more convenient to rewrite eq. (1) as

$$\text{(i)} + \text{(ii)} + \text{(iii)} = q_{LR} k_s A + a'_{sol} + (b_{sol} + UA_s)(T_o - T_z) \tag{2}$$

where $a'_{sol} = a_{sol} + b_{sol} T_z$

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 to 60% relative humidity, ASHRAE 1993). 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 $(k_s \cdot q_{LR})$ which appears *only* when outdoor specific humidity w_0 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.

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 (ASHRAE 1993; Knebel 1983), and we shall assume that the reader is familiar with them. Only a brief description of the modeling equations is given below.

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 or 51.8° F) and 90% RH can meet the

Table 1. *Values of Various Building Parameters Used for Our Base-Case Simulation*

Parameters	SI Units	British Units
A	10,000 m^2	100,000 ft^2
T_z	22° C	71.6° F
RH_z	50%	50%
q_{LR} / A	32 W / m^2	3.2 W / ft^2
k_l	0.2	0.2
k_s	1.3	1.3
$m_{v,min}{}^{+}$	0.6×10^{-3} kg /s/ m^2	0.1 cfm/ft^2
U*	2.5 W / m^2/°C	0.44 Btu/ft^2/hr/°F
A_S / A	0.56	0.56
A_{int} / A	0.68	0.68

$^+$Chosen 20% higher than ASHRAE stipulated minimum.
*Includes glazing (30%), linearized solar loads and infiltration.

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 (98.6° F) and $RH_{o,design} = 50\%$ (see Table 2). The value of $m_{Rated} = 0.00558$ kg/s/m² (0.92 cfm/ft²) 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 (or air velocity) which is larger than the minimum outdoor ventilation rate. For office spaces, ASHRAE (1993) stipulates a value of about $3.6–5.4 \times 10^{-3}$ kg/s/m² (0.6–0.9 cfm/ft²) 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, for example, 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: $T_{o,design} = 37°C$ (98.6° F), $RH_{o,design} = 50\%$,

   Cold deck: $T_{C,design} = 11°C$ (51.8° F), $RH_C = 90\%$.

2. Rated building supply air flow rate per unit conditioned building area for all three strategies:

   $m_{Rated} = 0.92$ cfm/ft² or 0.00558 kg/s/m²

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_0 only from −12° C (10° F) to 37° C (98° F) assuming $RH_o = 0.5$ and constant internal loads.

Supply air flow rates

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

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

and flow rate through the exterior zone

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

Thus the total supply air flow rate per unit area

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

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:

$$E_C = m \cdot A[c \cdot (T_m - T_C)^+ + h_v \cdot (w_m - w_C)^+] \quad (6)$$

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

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

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).

DIFFERENT VENTILATION STRATEGIES STUDIED

A number of papers have discussed the issue of how to operate and control VAV systems for acceptable building ventilation (for example, Sauer and Howell 1992; Filardo 1993; Cohen 1994; Janu et al. 1995; Kettler 1995; Gill 1996). 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 the conceptual basis. The ventilation air flow rate needs to be known in order to use the heating and cooling expressions shown in section 3.4. In fact, the basic objective of this paper is to study how energy use and IAQ vary when different ventilation air flow strategies are chosen. 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 (180 ft^2/occupant). We shall now describe the various ventilation strategies studied in this paper.

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 (15 cfm) per occupant stipulated by ASHRAE Standard 62-1989 (ASHRAE 1989). If an occupancy density of 17 m^2/occupant (180 ft^2/occupant) is assumed, the recommended minimum standard value would be $m_{v,min}* = $ (15 cfm per occupant/180 ft^2 per occupant) $= 83.3 \times 10^{-3}$ cfm/ft^2 or 0.50×10^{-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 (0.1 cfm/ft^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.

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 re-balanced and re-commissioned 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.

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 ventilation air flow rate into the HVAC system would satisfy IAQ requirements 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 (0.1 cfm/ft^2) in order to be consistent with S1. Modeling such a strategy is fairly simple. Let us normalize the flows to the interior and exterior zones as follows:

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

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 $(f_{m,int} + f_{m,ext}) \cdot m = m_{int} \cdot A_{int} + m_{ext} \cdot A_{ext}$. Under S3, we would choose the ventilation flow rate as follows:

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

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:

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

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.

RESULTS AND DISCUSSION

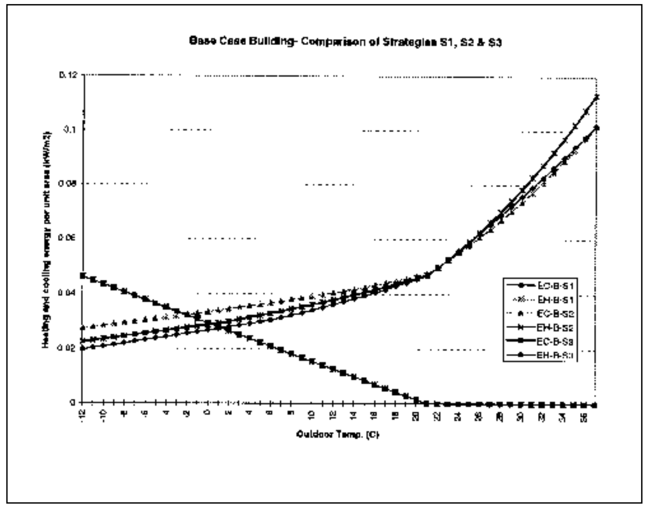
Selection of inputs

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 per unit conditioned area basis. 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 corridor 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 (98.6° F) and $RH_{o,design} = 50\%$) can be met with a cold deck temperature $T_C = 11°$ C (51.8° F) and $RH_C = 90\%$.

Intercomparison 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 IAQ is always satisfied in both zones) is higher

Figure 1. *Variation of heating and cooling 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.*



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.

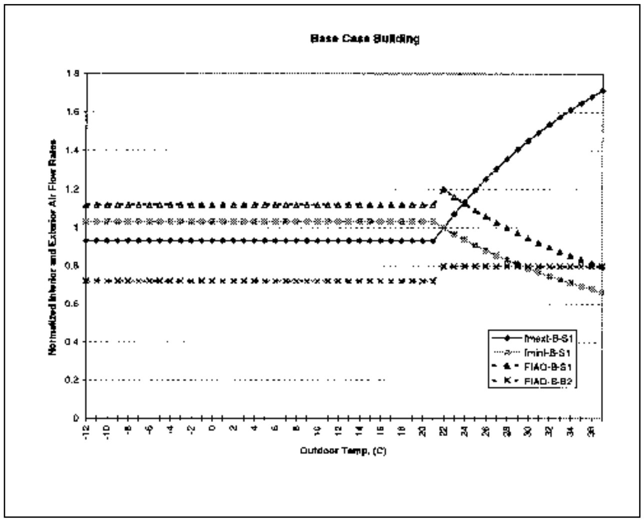
The variation of $f_{m,int}$ and $f_{m,ext}$ defined by eq. (8) with T_o is shown in Fig. 2. Note that this is independent of the ventilation strategy chosen since the supply flow rate splits depending only on the load distribution ratio of the two zones. The point of interest in Fig. 2 is the high degree of zonal flow imbalances at higher T_o values which, as we shall discuss below, has a direct impact on IAQ.

The extent to which the IAQ criteria 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 IAQ is

Table 3. *Pertinent Input Simulation Data for the Three Sizes of Building Simulated. The building is square with 3 floors and total height of 10 m. External corridor is assumed 5 m wide for determining fraction of interior to total area. $m_{min} = 0.00335$ kg/s/m² for all three building sizes.*

	A		U		A_S/A	A_{int}/A	m_{min}/m_F	m_{Rated}
	m^2	ft^2	W/m²/°C	Btu/ft²/hr/°F	—	—	—	kg/s/m²
Base case (B)	10,000	100,000	2.5	0.44	0.56	0.68	0.600	0.00558
Smaller bldg (S)	5,000	50,000	2.5	0.44	2.66	0.57	0.266	0.01261
Larger bldg (L)	20,000	200,000	2.5	0.44	0.49	0.77	0.626	0.00535

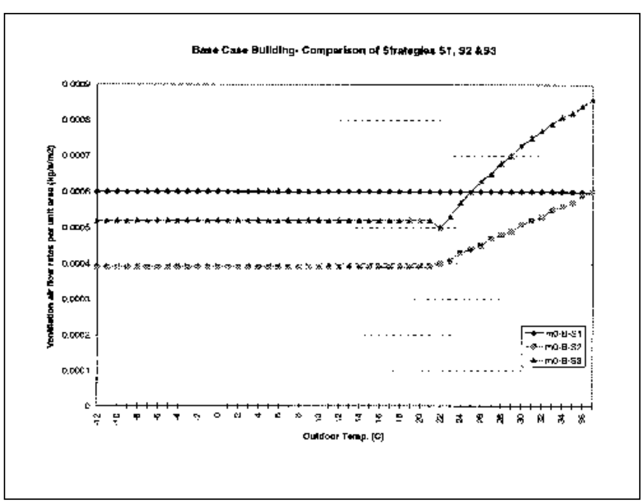
Figure 2. *Variation of the normalized air flow rates of the individual zones (defined by eq. 8 and independent of ventilation strategy) and of the fraction F_{IAQ} (defined by eq. 10) with outdoor temperature for the base case building.*



never satisfied since $F_{IAQ} = 0.8$ 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 $T_0 > 27°$ 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. S2 assumes a constant value of (m_v / m), and so as the VAV system modulates the flow with decreasing temperature, m_v also decreases. 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 most 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 IAQ is not satisfied.

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



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 imbalances 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 IAQ problems would be experienced by the smaller building at high T_o values, while the base case building and the larger building would have 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

Figure 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.*

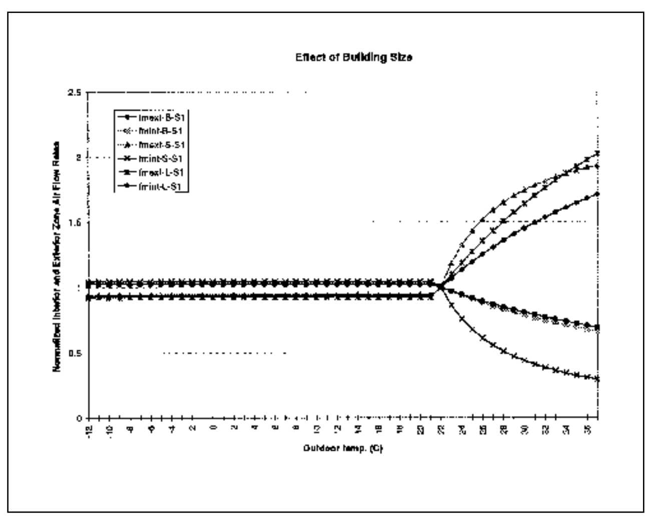
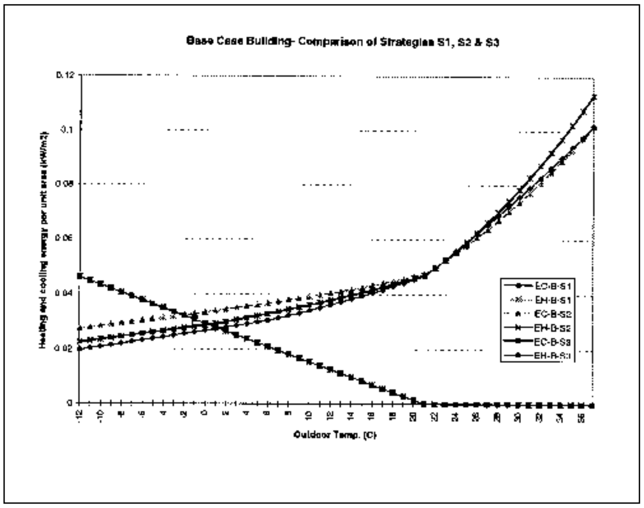


Figure 5. *Variation of F_{IAQ} (defined by eq. 10) for the three different building sizes operated under ventilation strategies S1 and S2.*



large building will violate IAQ requirements only when T_o exceeds 28° C or so.

Effect of location

Figure 6 depicts the bin temperature data for Dallas, TX (a moderately hot and humid locations) and for Seattle, WA (a mild and drier location) taken from Degelman (1984). 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. 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 $\tilde{u}_{IAQ}$ of the number of hours in the year when *IAQ is not satisfied* which is defined below:

$$u_{IAQ,i} = \delta \cdot N_i \,/\, (24x365) \qquad (11)$$

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

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

Table 4. *Inputs for Annual Energy Use Simulation. Other inputs are listed in Table 3.*

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

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

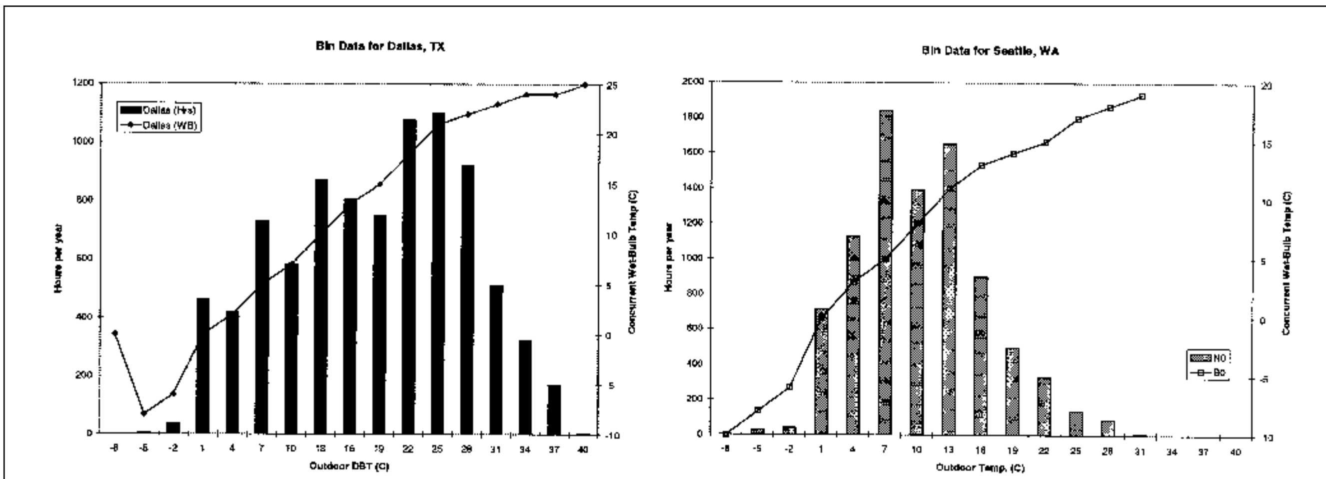


Fig. 6(a) Fig. 6(b)

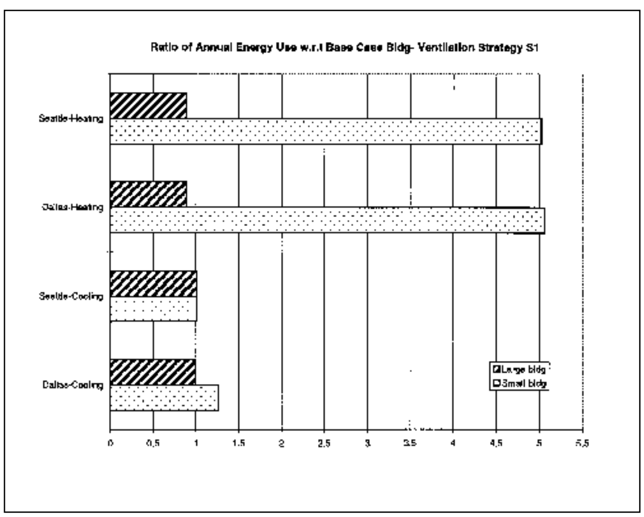
Table 5. Simulation Results of Annual Heating and Cooling Energy Use Per Unit Area in kWh/m²/yr for Different Ventilation Strategies and for the Two Locations Selected. $\tilde{u}_{IAQ}$ *is the bin-hour-weighted annual fraction of the number of hours in the year when IAQ is not satisfied (defined by eq. 12).*

		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

N_i is the number of hours in the particular bin i and n 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. Figure 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.

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

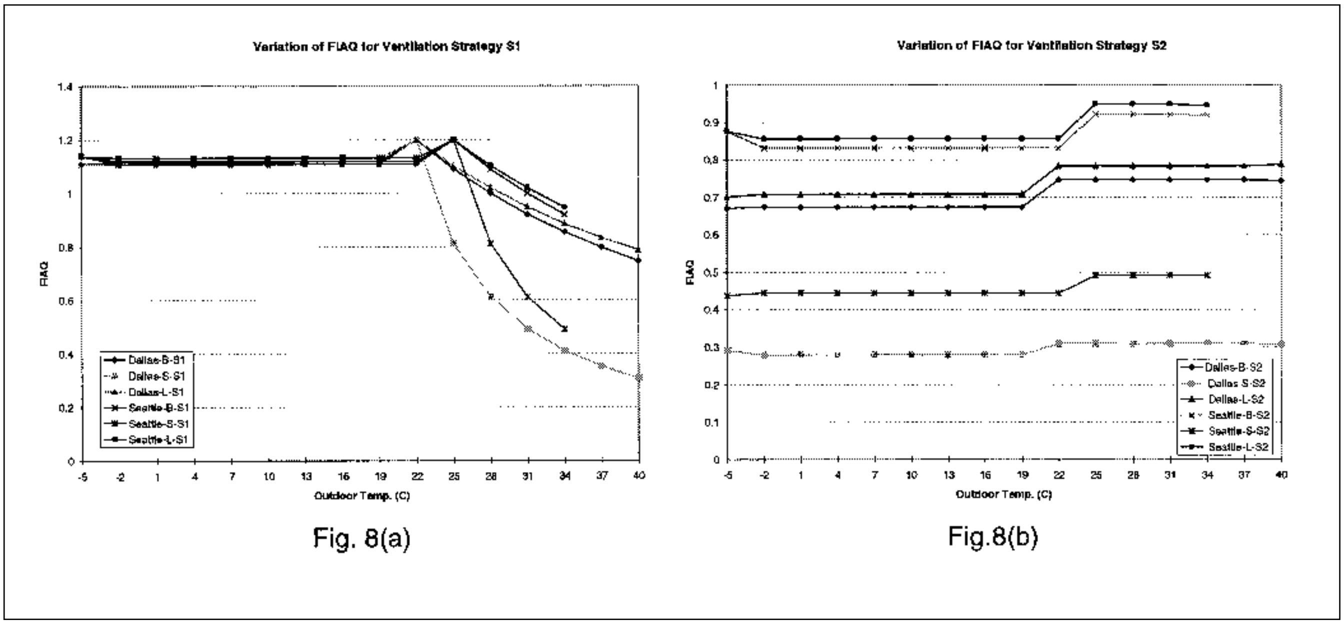


What is striking in Fig. 7 is the five fold increase in heating energy use per unit area for the smaller building in both locations.

Figure 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 S3 and so is not shown. We note that adopting S2 results in unsatisfactory IAQ 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 energy use for different locations and building sizes for the three ventilation strategies is 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 IAQ is adversely affected at higher T_o values, we would expect IAQ to be better 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 IAQ) 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 IAQ.

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



Looking at Fig. 9(b), we note that cooling energy used by S3 is less than 4% higher than that of S1 while being far superior in terms of IAQ. Thus from Table 5, we see that the energy excess penalty for eliminating IAQ problems 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 IAQ and energy use. Coming finally to Fig. 9(c), 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.

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 IAQ problem in VAV systems may be more acute in summer when the flow distribution to the two individual zones is more non-uniform than in winter when the total ventilation flow is lower than that during summer.

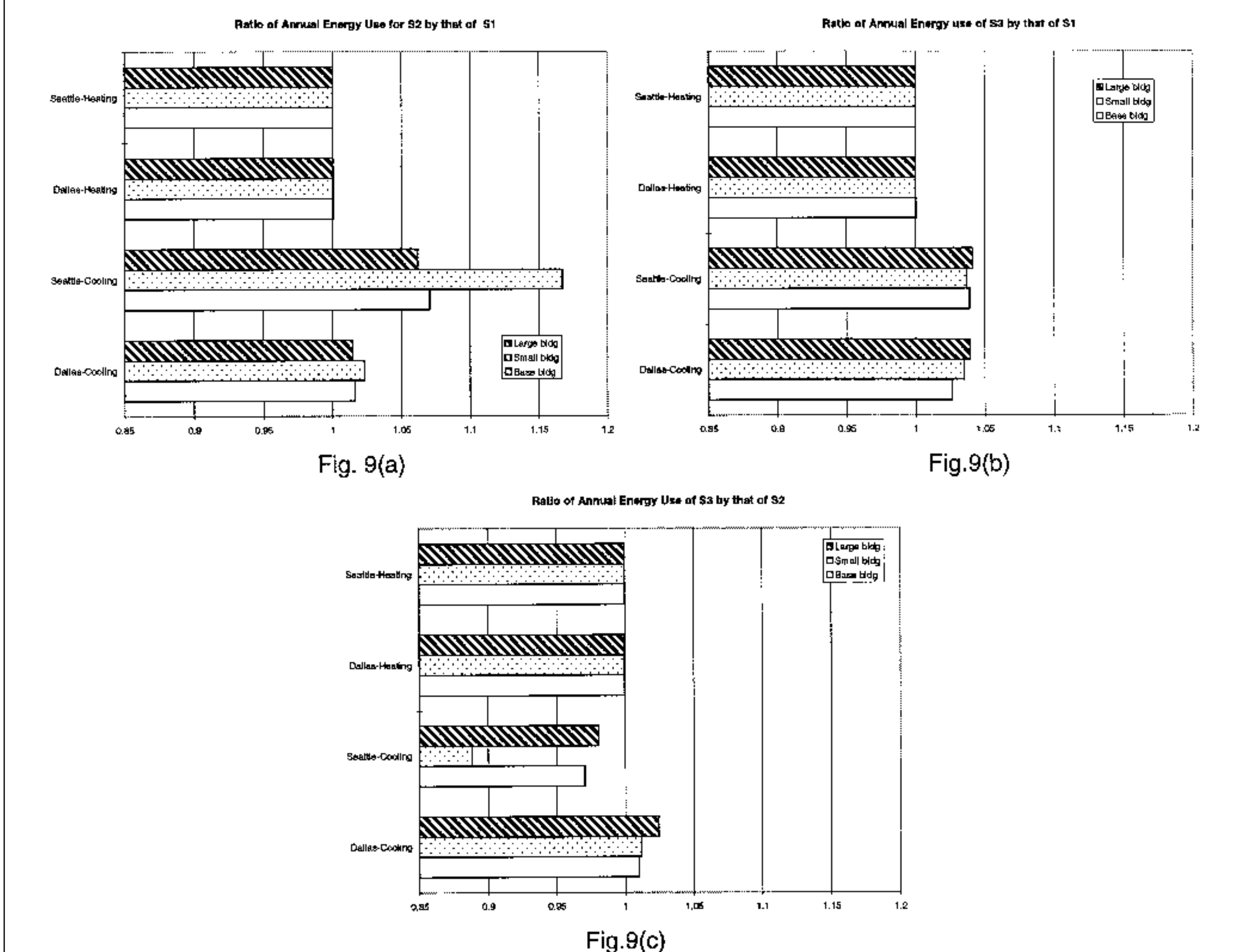
(b) Adopting a strategy where the fresh air intake flow rate is constant (strategy S1) and 20% higher than the ASHRAE minimum does not necessarily eliminate IAQ problems specially in hot locations and in smaller buildings. While S1 is satisfactory in a location such as Seattle, it is unsatisfactory in a hotter location such as Dallas since we found that in a medium sized building (area of 10,000 m^2), the IAQ 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 IAQ problems (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 use and IAQ is location dependent. It takes S3 about 3% more cooling energy to eliminate IAQ problems due to S1 in a medium sized building in Dallas during 22% of the time. For the same building in Dallas, the extra cooling energy is about 4% to eliminate IAQ 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

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



building, a single air handler unit, no economizer, . . .) and assumptions made (internal load distribution between zones, cold deck setting, . . .). More exact quantification of the synergism between IAQ and energy would require a more detailed treatment of building loads, system operation and other complex phenomena such as indoor and outdoor pollution levels and explicit modeling of changes in pollution levels of supply and return air streams.

ACKNOWLEDGMENTS

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

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 fresh 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
$\tilde{u}_{IAQ}$	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)

REFERENCES

ASHRAE, 1989. ‘‘Ventilation For Acceptable Indoor Air Quality, Standard 62-1989,’’ American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, Georgia.

ASHRAE, 1993. *Fundamentals*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, Georgia.

Claridge, D., J. Haberl, W. Turner, D. O’Neal, W. Heffington, C. Tombari and S. Jaeger. 1991. ‘‘Improving Energy Conservation Retrofits with Measured Savings,’’ *ASHRAE Journal*, October.

Cohen, T., 1994. ‘‘Providing Constant Ventilation in Variable Air Volume Systems,’’*ASHRAE Journal*, pp. 38–40, May.

Degelman, L.O., 1984. ‘‘Development of Bin Weather Data,’’ ASHRAE RP-385, American Society of Heating, Refrigeration and Air-Conditioning Engineers, Atlanta, GA.

Filardo, M.J., 1993. ‘‘Outdoor Air: How Much is Enough,’’ *ASHRAE Journal*, pp. 34–38, January.

Gill, K.E., 1996. ‘‘IAQ and Air Handling Unit Design,’’ *HPAC Journal*, pp. 49–54, January.

Janu, G.J., Wenger, D.J and Nesler, C.G., 1995. ‘‘Strategies for Outdoor Airflow Control from a Systems Perspective,’’ *ASHRAE Transactions*, V. 101, Pt. 2., paper SD-95-2-4.

Katipamula, S. and D.E. Claridge. 1993. ‘‘Use of Simplified Systems Models to Measure Retrofit Savings,’’ *ASME Journal of Solar Energy Engineering*, Vol. 115, pp. 57–68, May.

Kettler, J.P., 1995. ‘‘Minimum Ventilation Control for VAV Systems: Fan Tracking vs Workable Solutions,’’ *ASHRAE Transactions*, paper SD-95-2-3.

Knebel, D.E., 1983. *Simplified Energy Analysis Using the Modified Bin Method*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, Georgia.

Liu, M. and Claridge , D.E., 1995. ‘‘Application of Calibrated HVAC System Models to Identify Component Malfunctions and to Optimize the Operation and Control Schedules,’’ Solar Engineering 1995, W.B. Stine, T. Tanaka and D.E. Claridge (Eds.), *ASME/JSME/JSES International Solar Energy Conference*, Maui, Hawaii, March.

Reddy, T.A. , S. Katipamula, J.K. Kissock and D.E. Claridge. 1995 ‘‘The Functional Basis Of Steady-State Thermal Energy Use In Air-Side HVAC Equipment,’’ *ASME Journal of Solar Energy Engineering*, vol. 117, p. 21, Feb.

Rengarajan, K., Shirey, D.B. and Raustad, R.A., 1996. *ASHRAE Transactions*, V. 102, paper 3949.

Sauer, H.J. and Howell, R.H., 1992. ‘‘Estimating the Indoor Air Quality and Energy Performance of VAV Systems,’’ *ASHRAE Journal*, pp. 43–50, July.

SMACNA,1993. *Indoor Air Quality*, 2nd Ed. Sheet Metal and Air Conditioning Contractors National Association, Inc, Chantilly, VA, July.

Taylor, S.T., 1996. ''Determining Ventilation Rates: Revisions to Standard 62-1989,'' *ASHRAE Journal*, Feb.

Vadon, M., J.F. Kreider and L.K. Norford. 1991. ''Improvement of the Solar Calculations in the Modified Bin Method,'' *ASHRAE Transactions*, Vol. 97, Part 2, pp. 204–211.