Extending the Concept of Energy Delivery Efficiency (EDE) of HVAC Systems

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ABSTRACT

An earlier study advanced the concept of energy delivery efficiency (EDE) of HVAC systems by which energy mixing in large commercial buildings can be evaluated against an absolute energy standard, namely, the energy needs of the building if operated as a one-zone building. The study also pointed out that this EDE(1-zone) is an upper limit, and that the efficiency of the two-zone building, which we term EDE(2-zone), would constitute a more realistic energy standard in order to evaluate the efficiency of actual HVAC systems. The mathematical treatment, previously limited to dealing with sensible heat flows, has been expanded in this study to cover supply air latent effects as well as the influence of economizer operation. Appropriate equations are presented for the minimum heating and cooling energy use for ideal one-zone and two-zone buildings with and without economizer operation.

An energy standard based on a two-zone model may still be an unrealistic standard when comparing the energy efficiency of actual HVAC systems. This argument has a parallel in thermodynamics where the efficiency of an actual steam power plant is more realistically compared with the Rankine efficiency rather than with the Carnot efficiency. An energy standard based on ideal HVAC system performance would then be the logical basis of comparing actual system performance. Consequently, we have suggested an HVAC configuration that ensures the stipulated ventilation airflow rates, necessary for satisfactory IAQ, to each of the multiple zones of the building. Subsequently, we have defined the operation and control of such an ideal HVAC system, which will minimize thermal energy use while being subject to some of the fundamental restrictions of system operation and human comfort under which an actual HVAC system operates. Simulations, assuming typical two-zone building and system parameters, are used to illustrate the extent to which EDE(2-zone) and EDE of ideal HVAC systems differ from EDE(1-zone). Other studies have applied the concepts developed in this paper to year-long monitored data from several buildings and illustrated the usefulness of the EDE approach as a diagnostic tool to evaluate HVAC retrofit performance and Operation & Maintenance measures.

INTRODUCTION

HVAC systems of large commercial buildings consume energy in excess of the sum total of the building loads. This excess energy use is due to the fact that a single air-handler unit in a HVAC system, having to provide conditioned air at different supply temperatures to multiple zones in the building, can do so only by resorting to either (a) a certain amount of mixing of cold and hot airstreams as in dual-duct systems or (b) to terminal reheating as in single-duct systems. This mixing of cold and hot airstreams, or terminal reheating, results in an energy penalty that can be minimized by converting a constant-air-volume (CAV) system to a variable-air-volume (VAV) system; however, it cannot be entirely eliminated.

A previous paper (Reddy et al. 1994) proposed an index, called energy delivery efficiency (EDE), which characterizes the excess energy penalty and rates the energy performance of HVAC systems on an absolute scale. This approach is akin to the concept of Carnot efficiency as a way of defining the theoretical limit of heat engines, as well as rating the relative performance of different engines. The index would serve as a means of evaluating different generic HVAC system types (for example, dual-duct constant-air-volume (DDCAV) or dual-duct variable-air-volume (DDVAV) systems), as well as permitting the energy performance of a particular HVAC system in a specific building to be assessed against an absolute standard. The mathematical basis of EDE and its power as a diagnostic tool in providing insights as to how well an HVAC retrofit in a particular building is performing, has also been illustrated with monitored data from two buildings (Reddy et al. 1994).

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The basis of the EDE concept may be described as follows. Taking the control volume to include both the HVAC system and the building, and viewing internal loads such as lighting to be generated inside the control volume, an instantaneous heat balance neglecting transient effects associated with thermal mass yields

$$Q_B = E_C - E_H \quad (1)$$

where

Q_B = net building heat gains or net cooling load,

E_C = measured whole-building cooling thermal energy supplied by the cooling coil, and

E_H = measured whole-building thermal heating energy supplied by the heating coil.

The value (E_C - E_H) can be viewed as the amount of comfort energy that would be required had no mixing of cold and hot airstreams taken place. Thus, this amount is a sort of absolute thermodynamic minimum. In reality, the building consumes total thermal energy, amounting to (E_C + E_H). Consequently, the EDE that rates the amount of simultaneous heating and cooling is defined as

$$\text{EDE} = \frac{\textit{Thermodynamic minimum energy use}}{\textit{Actual energy use}}$$

$$= (E_C - E_H)/(E_C + E_H) \quad (2a)$$

Alternatively, in order not to have negative values of efficiency when the building requires more heating energy than cooling energy, absolute values can be taken as follows:

$$\text{EDE} = |(E_C - E_H)|/(E_C + E_H) \quad (2b)$$

The EDE of an actual building defined by Equation 2b would lie between 0 and 1, the latter limit indicating no simultaneous heating and cooling and that energy is consumed most efficiently in the building. The building can then be viewed as operated at its thermodynamic efficiency limit. Either definition of EDE can be used, depending on whichever is more appropriate under the specific circumstance. Note that the EDE concept, in its current development, is limited to the thermal efficiency of the air-side system. It does not consider primary system conversion efficiencies, transport losses in the ducting, or possible changes to existing building material and to equipment and lights.

The appropriate time scale for analyzing heating and cooling energy-use data to deduce EDE has also been investigated. Using daily time scales (against hourly time scales) tends to minimize heat capacity effects of the building shell and furnishings but progressively introduces more error in the assumption that thermodynamic minimum energy use is equal to (E_C - E_H). A study by Deng (1997) has determined that EDE values computed by using either time scale produces similar results. Consequently, daily time scales are implicitly assumed throughout this paper.

Actual commercial buildings have more than one zone, and so EDE(1-zone) is not the proper efficiency standard. Therefore, EDE for a two-zone building was suggested (Reddy et al. 1994). The rationale for a two-zone treatment is that although actual commercial buildings may have multiple zones, a two-zone treatment with one exterior or perimeter zone and one interior or core zone is a good compromise between accuracy and simplification (Knebel 1983; Katipamula and Claridge 1993; Liu and Claridge 1995). An energy standard based on a two-zone model may still be an unrealistic standard when comparing the energy efficiency of actual HVAC systems. This argument has a parallel in thermodynamics, where the efficiency of an actual steam power plant is more realistically compared against the Rankine efficiency rather than the Carnot efficiency. An energy standard based on ideal HVAC system performance would then be the logical basis of comparing actual system performance, a concept developed in this paper.

The objectives of this paper are threefold. First, we shall extend the theoretical development of the EDE concept for one-zone and two-zone buildings to include supply-air latent effects and economizer operation. Secondly, we will define how ideal HVAC systems should be operated and controlled so that energy use is minimized while meeting the stipulated requirements for outdoor ventilation flow. Finally, we shall evaluate, by means of simulations, using typical building and system parameters, the extent to which heating and cooling energy use, as well as EDE, of such ideal HVAC systems differ from EDE(1-zone) and EDE(2-zone).

EXPRESSIONS FOR IDEAL BUILDING LOADS

The ideal HVAC system should only consume the required amount of energy necessary to offset the net building heat gains and to condition the minimum outdoor air intake stipulated by indoor air quality requirements. We shall assume steady-state operation and make the following assumptions:

- The thermostat set point temperature T_Z is fixed at a mean yearly value;
- Infiltration loads are assumed negligible or considered part of the ventilation loads;
- Solar gains are a linear function of outdoor dry-bulb temperature (Vadon et al. 1991, Knebel 1983);
- The ventilation airflow rate, i.e., the fresh air intake is the same for CAV and VAV systems. This is the flow rate required to satisfy indoor air quality (IAQ) constraints. Note that it is the airflow rate to the space that should be modulated in a VAV system, not the fresh air or ventilation flow rate;
- Ducts are perfectly insulated (i.e., no heat losses) and have no air leakage. Alternately, duct losses are considered to be part of envelope loads.

Some of the above assumptions have been made to keep the EDE analysis simple and thus be amenable to practical use. The

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reader could make appropriate modifications to the expressions derived below if he wishes not to observe some of the above assumptions.

One-Zone Model Without Economizer

An expression for the idealized hourly total heat gains ($Q_{B,1\text{-}zone}$) of a one-zone space (i.e., a space where simultaneous heating and cooling does not occur) will be derived below. Assuming the sign convention that energy flows are positive for heat gains and vice versa,

$$Q_{B,\,1-zone} \tag{3}$$

= internal loads (sensible including gains from people) (a)
+ solar loads (both direct and transmission) (b)
+ shell transmission loads (c)
+ infiltration and ventilation loads (both sensible and latent) (d)

The objective of this study is to be able to analyze monitored data in the framework of the EDE approach rather than to evaluate alternative design options. Hence, the equations should be formulated according to how the monitoring of building energy use is usually done. It is the electricity used by lights and receptacles inside a building that 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 thus, are usually estimated. A first order approximation is to assume that the schedule of lights and equipment closely follows that of building occupancy (this is especially valid at the daily time scale). A convenient and logical manner of including the unmonitored sensible loads is to modify q_{LR} by a constant multiplicative correction factor, k_s (typically in the range 1.05 to 1.2), which accounts for the miscellaneous (i.e., unmeasurable) internal sensible loads. Note that this is a simplified treatment since, in reality, not all lights and equipment are shut off when people leave commercial buildings. Instead of a proportionality constant, a linear relationship would perhaps be more accurate for hourly time scales. However, this paper assumes a proportionality constant in the equations derived below although the expressions can be extended to satisfy the more general linear relationship.

The sum of three of the four terms of Equation 3 is given by

$$(a)+(b)+(c) = q_{LR}k_sA + a_{sol} + b_{sol}T_o + UA_s(T_o - T_z) \tag{4a}$$

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 Equation 4a as

$$(a)+(b)+(c) = q_{LR}k_sA + a'_{sol} + (b_{sol} + UA_s)(T_o - T_z) \tag{4b}$$

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 due to the outdoor air intake. Indoor comfort can be maintained by closely controlling the indoor air temperature (which thermostats normally do) and ensuring 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 [Mitchell 1983]). Hence, indoor humidity is not a variable that is usually controlled on a continuous basis in actual HVAC systems.

Proper design of air-conditioning systems permits humidifying the supply air if the air is too dry. However, many buildings, especially in hot and humid locations, do not need humidification most of the year, and it is common not to install humidification systems in such locations. For example, other than medical buildings, most of the buildings in the Texas LoanSTAR program (Claridge et al. 1991) do not have humidification systems. These HVAC systems do not directly control dehumidification other than that provided by the cooling coil. However, the cooling coil is so designed that the specific humidity of air leaving the cooling coil is close to the middle of the allowable indoor relative humidity range. For example, a typical cooling coil with leaving air conditions of 12°C and 90% RH has a specific humidity equal to that corresponding to a zone condition of 22°C and 50% RH. 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, in the case of an HVAC system without a humidifier, 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 possible. The appropriateness of this model has been verified against detailed HVAC simulations using commercially available computer software (Deng et al. 1997).

In our theoretical treatment of the ideal building loads, we are forced to separately consider the case of a building with an ideal humidification/dehumidification system, and one in which no such arrangement exists. Humidification of air involves injecting hot water or steam into the airstream and, consequently, this increases the heating energy consumed by the HVAC system (Stoecker and Jones 1982).

Case (a) For a building with ideal humidity control, where both T_Z and w_Z are perfectly controlled, the infiltration and ventilation load, i.e., term (d) of Equation 3, is given by $m_{v,\,min}A(h_o - h_z)$, where h_z is the enthalpy of the specified indoor air comfort state. Note that energy requirements in an idealized process depend only on the end states and not on actual HVAC equipment installed. (This is analogous in thermodynamics to path independent quantities such as properties of the work-

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ing fluid). Thus, from Equations 3 and 4 and including the latent load generated inside the building, we have

$$Q_{B,\,1-zone} = q_{LR}k_s(1+k_l)A + a'_{sol} + (b_{sol} + UA_s + m_{v,\,min}Ac)(T_o - T_z) + m_{v,\,min}Ah_v(w_o - w_z) \tag{5}$$

Case (b) For a building with no humidification system, and when $w_o < w_z$, we should not include the energy required to humidify the ventilation air nor the energy associated with internal latent loads because the latter is a benefit in terms of increasing the humidity in the conditioned space. Note, however, that the necessary energy required for dehumidification of the supply air should be included. Then,

$$Q_{B,\,1-zone} = q_{LR}k_s(1+k_l\delta)A + a'_{sol} + (b_{sol} + UA_s + m_{v,\,min}Ac)(T_o - T_z) + m_{v,\,min}Ah_v\delta(w_o - w_z) \tag{6}$$

where δ is an indicator variable associated with the latent effect of the ventilation air such that

$$\delta = 1, \quad \text{when } w_o > w_z \qquad (7)$$
$$= 0 \quad \text{otherwise}$$

Note that the only difference between equations 5 and 6 is the presence of δ in the latent load terms associated with internal loads and ventilation loads.

In a one-zone building, simultaneous heating and cooling need not occur. Let T_b be the balance point temperature, i.e., the outdoor temperature at which $Q_{B,1\text{-}zone} = 0$ (Mitchell 1983). Note that T_b, unlike the conventional definition, is now a function of w_o. Thus, for case (b), we have from equation 6:

$$T_b(w_o) = T_z - \frac{a'_{sol} + q_{LR}k_s(1+k_l\delta)A + m_{v,\,min}Ah_v\delta(w_o - w_z)}{b_{sol} + UA_S + m_{v,\,min}Ac} \tag{8}$$

Subsequently, the ideal cooling and heating loads of a one-zone building are given by

$$E_c(\text{Ideal, 1-zone}) = Q_{B,\,1-zone} \quad \text{when } T_o \geq T_b(w_o) \qquad (9a)$$
$$= 0 \quad \text{otherwise}$$

$$E_h(\text{Ideal, 1-zone}) = |Q_{B,\,1-zone}| \quad \text{when } T_o < T_b(w_o) \qquad (9b)$$
$$= 0 \quad \text{otherwise}$$

Finally, EDE as defined by equation 2a is given by

$$\text{EDE (1-zone)} = 1 \text{ when } T_o \geq T_b(w_o) \qquad (10a)$$

$$= -1 \text{ when } T_o < T_b(w_o) \qquad (10b)$$

One-Zone Model With Economizer

Under economizer operation, the treatment of idealized building loads should be modified to take into account the fact that outdoor air intake m_v is no longer a constant for all T_o values. How m_v should be varied under an enthalpy-controlled economizer cycle (the other common control is the temperature-controlled economizer, which, though less energy efficient, is cheaper and less failure prone) is well described in the literature (i.e., Knebel 1983). Let $T_e(w_o)$ be the outdoor temperature at which $h_o = h_z$. Recognizing that $T_e(w_o)$ is in the majority of cases greater than T_b, we have

$$E_C(\text{Ideal, 1-zone}) = Q_{B,1\text{-}zone} \quad \text{when } T_o \geq T_e(w_o) \qquad (11a)$$
$$= 0 \quad \text{otherwise}$$

and

$$E_H(\text{Ideal, 1-zone}) = |Q_{B,1\text{-}zone}| \quad \text{when } T_o < T_b(w_o) \qquad (11b)$$
$$= 0 \quad \text{otherwise}$$

As expected, E_H(Ideal, 1-zone) is not affected by the economizer cycle, i.e., equations 9b and 11b are identical. The EDE is undefined in the range $T_b(w_o) \leq T_o \leq T_e(w_o)$ because both E_C and E_H are zero.

Two-Zone Model Without Economizer

The one-zone model is generally applicable to residences and small buildings. However, 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. In order to render our discussion more realistic, we need to extend the one-zone treatment to a multizone building.

In addition to the assumptions made earlier, the following were also assumed.

- The building is regular in geometry (a rectangle can be assumed for further simplification) with one exterior or perimeter zone, and one interior or core zone. Most office and commercial buildings can be conceptually broken down this way (Knebel 1983). Offices are normally designed adjacent to windows and form a ring around the perimeter. Corridors could be either lumped into the perimeter zone (if office doors are normally left open) or lumped into the core zone.
- The two zones have identical zone setpoint temperatures.
- Ventilation and the internal loads are shared between both zones in proportion to the conditioned floor areas.
- Solar and transmission loads affect the perimeter zone only.

Let A_{int} and A_{ext} be the floor areas of the interior (or core) and of the exterior (or perimeter) zones respectively. Then the building load on the interior zone for case (b), i.e., a building with no humidification control, is

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$$Q_{B,int} = q_{LR}k_s(1+k_1\delta)A_{int} + m_{v,min}A_{int}[c(T_o - T_z) + h_v\delta(w_o - w_z)] \quad (12)$$

The building load on the exterior zone is

$$Q_{B,ext} = q_{LR}k_s(1+k_l\delta)A_{ext} + a'_{sol} + (b_{sol} + UA_s + m_{v,min}A_{ext}c)(T_o - T_Z) + m_{v,min}A_{ext}h_v\delta(w_o - w_z) \quad (13)$$

Note that equations 12 and 13 are strictly valid for the case when no humidification control system is present. When such a system is provided, one needs to simply consider the enthalpy potential (Stoecker and Jones 1982), i.e., replace the term $[(w_o - w_Z)\delta]$ by $(w_o - w_Z)$ in the above equations and remove the term δ in the internal loads (see equations 5 and 7).

The expressions for $T_{b,int}(w_o)$, the balance point temperature of the interior zone, and $T_{b,ext}(w_o)$, that of the exterior zone, are deduced from equations 12 and 13.

$$T_{b,int}(w_o) = T_z - \frac{q_{LR}k_s(1+k_l\delta) + m_{v,min}h_v\delta(w_o - w_z)}{m_{v,min}c} \quad (14a)$$

and

$$T_{b,ext}(w_o) = T_z - \frac{a'_{sol} + q_{LR}k_s(1+k_l\delta)A_{ext} + m_{v,min}A_{ext}h_v\delta(w_o - w_z)}{b_{sol} + UA_S + m_{v,min}A_{ext}c} \quad (14b)$$

Subsequently, the expressions for cooling and heating energy use are given by

(15a)

$$\begin{aligned} E_C\,(\text{Ideal, 2-zone}) &= Q_{B,int} + Q_{B,ext} && \text{when } T_o > T_{b,ext}(w_o) \\ &= Q_{B,int} && \text{when } T_{b,int}(w_o) \le T_o \le T_{b,ext}(w_o) \\ &= 0 && \text{when } T_o < T_{b,int}(w_o) \end{aligned}$$

and

(15b)

$$\begin{aligned} E_H\,(\text{Ideal, 2-zone}) &= 0 && \text{when } T_o > T_{b,ext}(w_o) \\ &= |Q_{b,ext}| && \text{when } T_{b,int}(w_o) \le T_o \le T_{b,ext}(w_o) \\ &= |Q_{B,int} + Q_{B,ext}| && \text{when } T_o < T_{b,int}(w_o) \end{aligned}$$

Recognizing that simultaneous heating and cooling need occur only when the expression for the ideal EDE for a two-zone building is obtained by manipulating equations 12 through 14:

EDE(2-zone) = (16)

$$\begin{aligned} &(1 - Q_{B,ext}/Q_{B,int})/(1 + Q_{B,ext}/Q_{B,int}) && T_{b,int}(w_o) \le T_o \le T_{b,ext}(w_o) \\ &= 1 && \text{otherwise} \end{aligned}$$

where

$$\frac{Q_{B,ext}}{Q_{B,int}} = \left(\frac{A_{ext}}{A_{int}}\right)\frac{T_{b,ext}(w_o) - T_o}{T_o - T_{b,int}(w_o)}\left(1 + \frac{UA_S + b_{sol}}{m_{v,min}cA_{ext}}\right)$$

Two-Zone Model With Economizer

In this case, we shall assume that two separate ducts provide outdoor air to the two zones so that ventilation airflow to each zone can be independently controlled. As in the one-zone case, the ideal heating energy use is not affected by the economizer cycle if the energy required to humidify outdoor air is not considered. The expression for the cooling energy use, however, gets modified. Again, let us assume an enthalpy-controlled economizer cycle, with the difference from the one-zone case being that two separate sets of economizer cycles are provided, one for each zone. Let $T_e(w_o)$ be the outdoor temperature at which $h_o = h_Z$. Then

$$\begin{aligned} E_C\,(\text{Ideal,2-zone}) &= Q_{B,int} + Q_{B,ext} && \text{when } T_o > T_e(w_o) \quad (17) \\ &= 0 && \text{otherwise} \end{aligned}$$

As for the one-zone case with economizer, the EDE is undefined in a certain range, which in this case is $T_{b,ext}(w_o) < T_o < T_e(w_o)$.

TYPICAL VARIATION OF IDEAL BUILDING LOADS

The objective of this section is to illustrate the variation of ideal E_C, E_H and EDE as described by the previous equations.

Building Loads and Ideal Systems Energy Use

Figure 1 illustrates how ideal heating and cooling energy use per unit-conditioned area, assuming no indoor humidity control, vary with T_o for three different values of RH_o for ideal one-zone and two-zone buildings, without economizer cycles. Indoor humidity control is assumed in Figure 2, but without an economizer cycle. Values of all relevant parameters used to generate these plots are given in Table 1. When heating and cooling thermal energy use in a commercial building is monitored, an inverse parameter identification scheme, which allows these parameters to be determined, is described by Deng et al. (1997). In the framework of this study, however, representative but arbitrary values of these parameters have been selected. We assume a constant minimum outdoor ventilation airflow rate of 10 L/s (20 cfm) per occupant as stipulated by *ANSI/ASHRAE Standard 62-1989* (ASHRAE 1989). If an occupancy density of 16 m^3 per occupant is selected, $m_{v,min} = 0.76 \times 10^{-3}$ kg/s/m^2 as shown in Table 1. We note the following from Figures 1 and 2:

a. Variation of RH_o results in a fan-like variation in E_C in Figure 1 (due to increasing contribution of the latent loads), while E_H is independent of RH_o when no humidification control is present. However, when humidification control is required, both E_C and E_H depend on RH_o (see Figure 2) and the fan-like behavior (due to the relative effect of latent loads with respect to the total) in E_C is markedly reduced.

b. In a one-zone building (Figure 1a and Figure 2a), no simultaneous heating and cooling occurs, but this is not so for a two-zone building (Figure 1b and Figure 2b). Though values of E_C for a two-zone building are small, below about T_o = 10°C, these are not negligible and are due to the fact that the ventilation air brought into the interior zone is not cool enough to offset the internal

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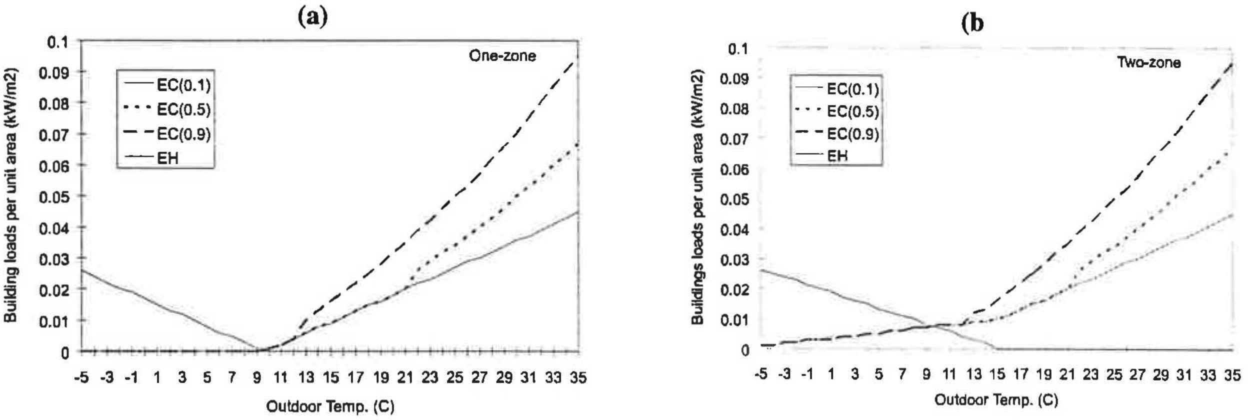

Figure 1 *Variation of ideal heating and cooling building loads with outdoor temperature and humidity (RH_o of 0.1, 0.5, 0.9) assuming no economizer cycle and no indoor humidification control. Input parameters are listed in Table 1. Note that heating energy use is independent of RH_o.*

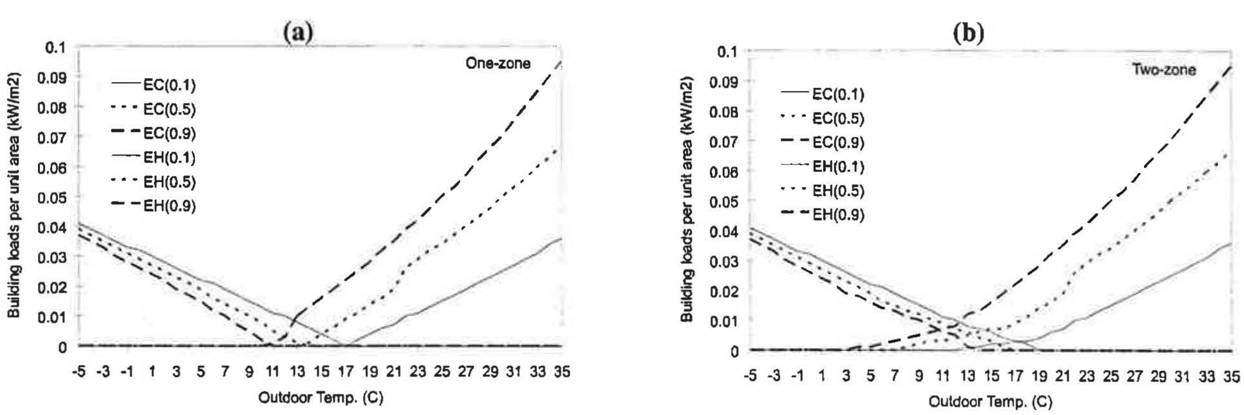

Figure 2 *Same as Figure 1, but with indoor humidification control.*

TABLE 1
Realistic Range of Variation of Building Parameters Along with a Set of Typical Values Used for Simulations

Parameters	Units	Range of Variation	Typical Values
T_z	°C	21 - 24	22
RH_z	%	30 - 60	50
q_{LR}	W/m^2	10 - 40	20
k_l	—	0.1 - 0.4	0.2
k_s	—	1.05 - 1.20	1.08
$m_{v,min}$	kg /s/ m^2	$0.3 - 7 \times 10^{-3}$	0.76×10^{-3}
UA_S/A	W/m^2 °C	0.5 - 3	1.0
A_{int}/A	—	0.3 - 0.8	0.60

loads, thereby requiring cooling even when T_o is very low.

c. The E_C plots for the two-zone building (Figure 1b) show a change point behavior, i.e., a change in slope at T_o values close to where $E_H = 0$. Such change point behavior is commonly seen in monitored data from commercial buildings (Kissock et al. 1992), and change point linear regression models (Ruch and Claridge 1993; Kissock 1993; Katipamula et al. 1994) have been developed to capture such behavior.

Figure 3 contains the same information as that in Figures 1 and 2, but it serves to better illustrate the relative differences in one-zone and two-zone building loads with T_o (for RH_o = 0.5), with and without humidification control. The extent to which E_H with humidification control is higher than without humidification control is clearly seen in Figure 3. The reverse holds for cooling energy since evaporative cooling can provide both cooling and humidification to the supply air.

Effect of Humidity on EDE

The extent to which humidification effects impact EDE (2-zone) is now considered. For the case with no economizer and no humidification control, EDE (2-zone), defined by Equation 2a, can be computed from Equation 16. Figure 4a illustrates this variation with T_o for three different RH_o values, and the necessary parametric values are listed in Table 1. It is clear that RH_o has almost no effect on EDE (2-zone). However, it is seen from Figure 4b that for extreme cases of 100% outdoor air intake and very high latent to sensible loads inside the conditioned space (i.e., high k_l values), RH_o does

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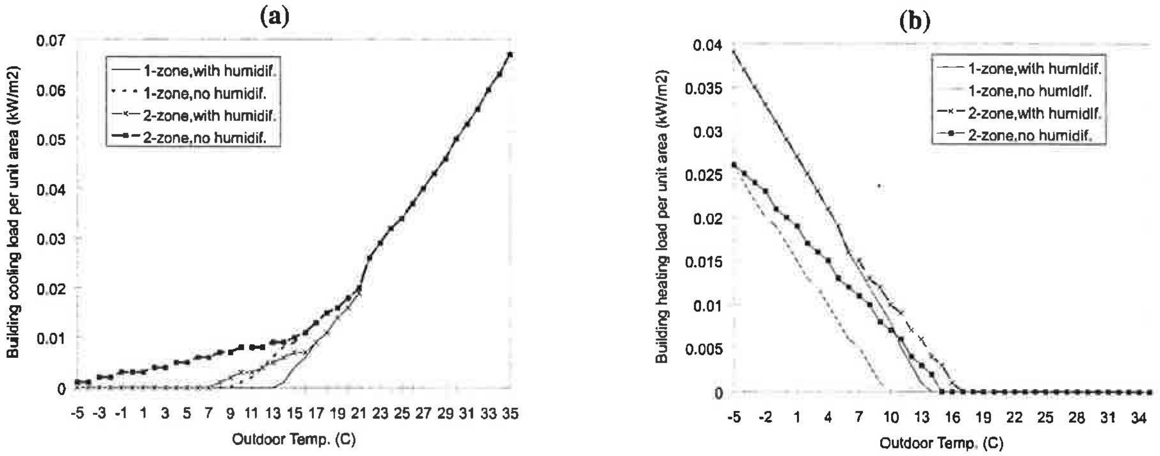

Figure 3 *Variation of ideal building loads with outdoor temperature for $RH_o = 0.5$ assuming no economizer cycle for the case of with and without humidification control. Input parameters are listed in Table 1.*

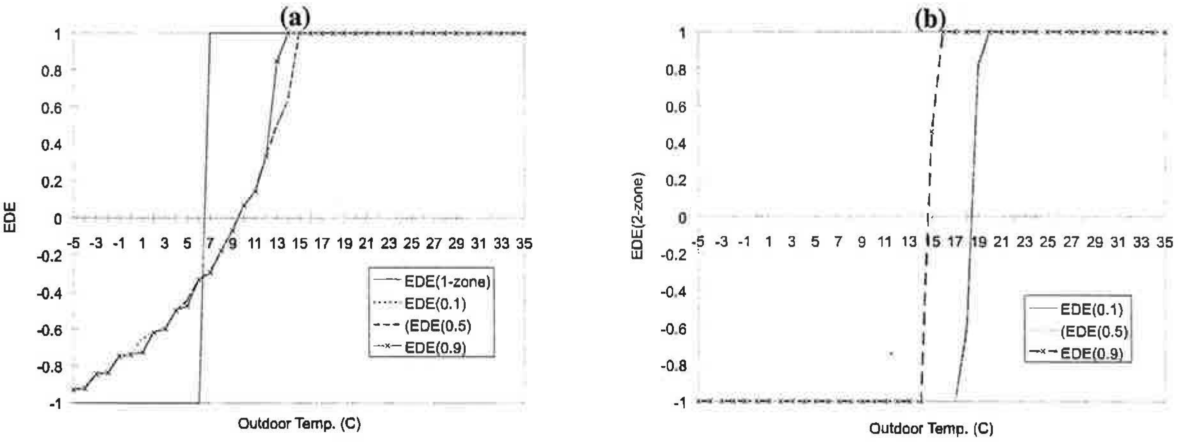

Figure 4 *Effect of outdoor temperature on EDE (2-zone) for three different RH_o values (0.1, 0.5, 0.9) when no humidity control is present. Input parameters are listed in Table 1. Figure (a) assumes an outdoor ventilation flow rate of 0.00076 kg/s/m^2 and $k_l = 0.2$, while figure (b) assumes a ventilation flow rate of 0.005 kg/s/m^2 and $k_l = 0.4$ representative of very high latent load contribution.*

seem to impact EDE to a small extent. However, when perfect indoor humidification control is required, note from Figure 5 that EDE(2-zone) is strongly dependent on RH_o, and, consequently, the EDE dependence on outdoor humidity cannot be neglected in this case.

ENERGY USE BY IDEAL HVAC SYSTEMS

This section derives expressions for the minimum heating and cooling energy use by ideal HVAC systems servicing a two-zone building. A sketch of the system is shown in Figure 6 along with the notation used for various state points. Note that in order to ensure that each zone is supplied by the minimum outdoor ventilation airflow as stipulated by indoor air quality constraints (ASHRAE 1989), an ideal system has to be provided with separate outdoor air ducts to each zone. Merely taking in the total ventilation air into the HVAC system at the recycle point (as is done in actual systems) does not guarantee the stipulated ventilation airflow rate to each zone (Reddy et al. 1996). Determining how such HVAC systems should be controlled and operated, such that seasonal or annual fan power and E_C and E_H at a specific location are minimized,

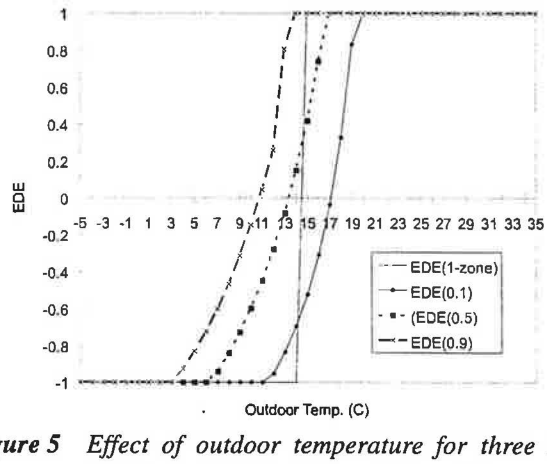

Figure 5 *Effect of outdoor temperature for three RH_o levels on EDE(2-zone) with humidity control. Input parameters are listed in Table 1 and no economizer cycle is assumed.*

TABLE 2
Additional Parameters and Operating Conditions Used in Ideal HVAC System Simulation

1. Minimum allowable supply air flow rate per unit-conditioned building area $m_{min} = 3.6 \times 10^{-3}$ kg/s/m² (0.6 cfm/ft²)
2. The zone supply flow rates per unit area: $m_{ext} = m_{int} = m_{min}$
3. The ventilation airflow rates per unit area: $m_{v,ext} = m_{v,int} = m_{v,min}$
4. Outdoor relative humidity taken to be 50%.
5. No economizer cycle operation.
6. Energy associated with having to humidify the supply air streams to each zone is taken into consideration in order that dry-bulb temperature and relative humidity of both zones are explicit design parameters to be maintained constant year-round.

requires global optimization. This paper adopts an engineering approach that would be intuitively appealing to practitioners. 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; Katipamula and Claridge 1993; Liu and Claridge 1995).

Assumptions made earlier while deriving expressions for building heating and cooling energy use for one-zone and two-zone buildings will still apply. The ideal HVAC system, though "idealized" in terms of control and operation, should be subject to some of the fundamental restrictions in terms of human comfort and operation under which practical HVAC systems operate. The proper formulation of these restrictions is the crux in defining ideal systems. In essence, for the HVAC system layout sketched in Figure 6, separate ducts ensure that the stipulated minimum ventilation airflow rate is supplied to each of the zones. There is another major difference between ideal building operation and ideal HVAC systems: the minimum stipulated ventilation airflow rate from IAQ considerations to the conditioned space can no longer be assumed to be the supply airflow rate to the conditioned zones (m). A more realistic value of minimum supply airflow per unit area (m_{min}) has to be assumed because indoor comfort requires a minimum air circulation rate that is larger than the minimum ventilation rate (ASHRAE 1993). For this simulation study, we have taken a value of $m_{min} = 3.6 \times 10^{-3}$ kg/s/m² (0.6 cfm/ft²) (see Table 2), while the assumed value for $m_{v,min}$ remains unaltered at 0.76×10^{-3} kg/s/m². Further, the supply flow rates per unit area for each zone are assumed to be constant year-round and equal to $m_{int} = m_{ext} = m_{min}$.

The two cooling coils, one for each zone and assumed to be of infinite capacity, lower the temperatures of the mixed airstreams to the appropriate supply air temperatures. In most actual HVAC systems, dehumidification is performed by the cooling coil. Though one could control the cooling coil temperature in such a manner that the necessary dehumidification to the supply airstreams can be provided, such a control increases both cooling energy use and reheat energy use. There are components, such as heat pipes, which are being used in actual HVAC systems that reduce this energy penalty. Hence, in this framework of idealized HVAC system operation, we chose to consider a system where dehumidification is separate from the cooling coil and is assumed to be process independent.

The expression for cooling energy is made up of sensible and latent cooling for each airstream:

$$E_c = m_{ext}\cdot A_{ext}\cdot[c\cdot(T_{m,ext}-T_{s,ext})^{+} + h_v\cdot(w_{m,ext}-w_{s,ext})] + m_{int}\cdot A_{int}\cdot[c\cdot(T_{m,int}-T_{s,int})^{+} + h_v\cdot(w_{m,int}-w_{s,int})] \qquad (18)$$

where $w_{s,int}$ and $w_{s,ext}$ are the specific humidities of the airstreams entering the zones which are determined from zonal moisture balances, and $T_{s,int}$ and $T_{s,ext}$ are computed from sensible heat balances on the individual zones and supply air temperatures. Values for $T_{m,ext}$ and $T_{m,int}$ and $w_{m,ext}$

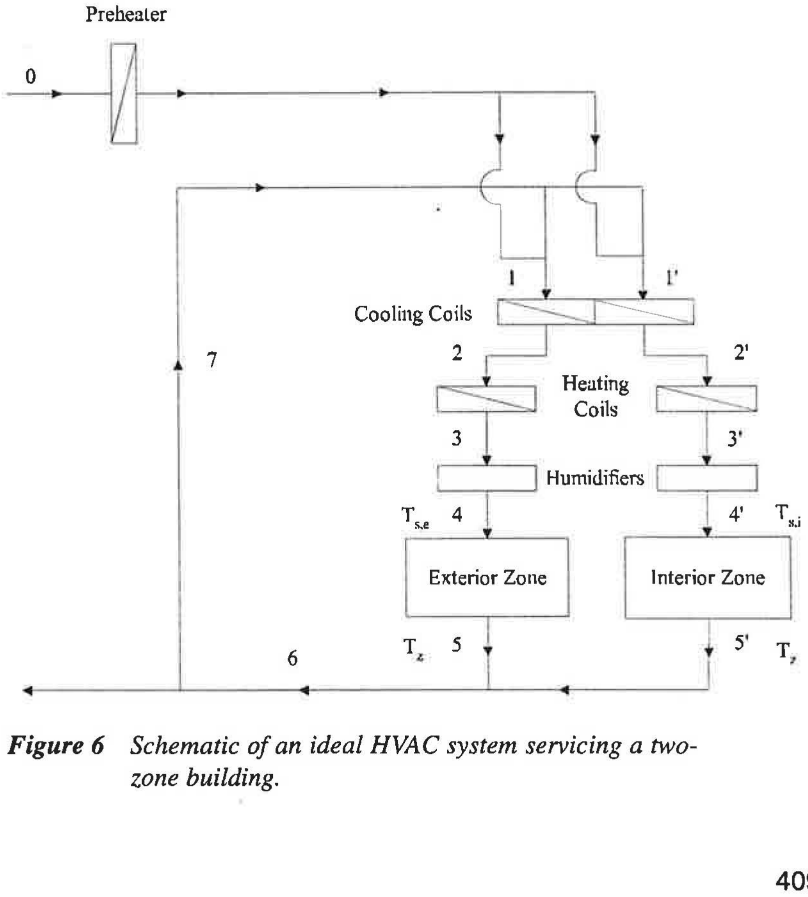

Figure 6 *Schematic of an ideal HVAC system servicing a two-zone building.*

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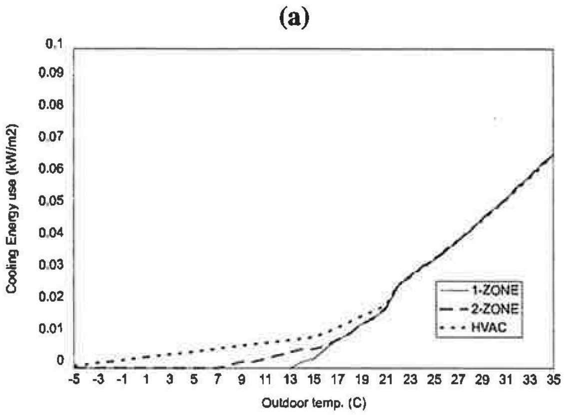

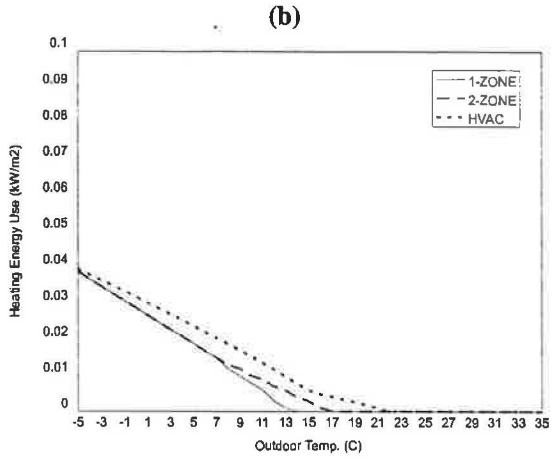

Figure 7 *Comparison of cooling and heating energy use of ideal 1-zone and 2-zone building loads and ideal HVAC systems. All inputs to the simulations are given in Tables 1 and 2.*

and $w_{m,int}$ (which may or may not be equal for both airstreams if ventilation airflow rates to both zones are different when IAQ requirements so dictate) are determined from heat and mass balances at the air recycle points (points 1 and 1' in Figure 6).

The expression for the sensible heating energy is

$$E_{H,sen} = m_{int} \cdot A_{int} \cdot c \cdot (T_{s,int} - T_{m,int})^+ + m_{ext} \cdot A_{ext} \cdot c \cdot (T_{s,ext} - T_{m,ext})^+ \quad (19)$$

The expression for humidification energy is

$$E_{H,humid} = m_{int} \cdot A_{int} \cdot h_v \cdot (w_{s,int} - w_{m,int})^+ + m_{ext} \cdot A_{ext} \cdot h_v \cdot (w_{s,ext} - w_{m,ext})^+ \quad (20)$$

The total heating energy use, E_H,is the sum of $E_{H,sen}$ and $E_{H,humid}$.

Note that the expressions for minimum heating and cooling energy use for a HVAC system derived above are valid when both dry-bulb temperature and relative humidity of both zones are explicitly controlled, the humidification system controlling the latter parameter. Appropriate modifications to these equations, when such a humidification system is not available, are straightforward. Further, the above treatment of how ideal HVAC systems should operate is general enough that it can be extended to multizone buildings as well as to HVAC operation under economizer cycle, if required.

DISCUSSION OF SIMULATION RESULTS

The same typical values for building parameters listed in Table 1 are assumed for the simulations. Table 2 lists the criteria and parameters assumed for the ideal HVAC system.

One would expect heating and cooling energy use of the one-zone building to be less (or equal over some T_o range) to the two-zone energy use, which in turn will be less (or equal over some T_o range) to the energy use of the ideal HVAC system. Figure 7 shows this to be true. The one-zone and two-zone energy uses are different only in the outdoor temperature range 7° C < T_o <17° C, the limiting points representing the balance point temperatures of the two zones. The ideal HVAC energy use is similar to the ideal loads only when T_o > 22° C (the zone setpoint temperature). Both the heating and cooling energy use of the ideal HVAC system are higher than those of the ideal building loads by the same amount. This excess is the penalty associated with the control and operation of the HVAC system as described in the previous section.

Figure 8 shows the variation of total energy use (i.e., sum of heating and cooling energy use) with T_o for ideal one-zone and two-zone building loads and for the ideal HVAC system. The extent to which heating and cooling energy use for the two-zone building and for the ideal HVAC system is higher than for the one-zone building, is clearly illustrated.

Figure 9 depicts how absolute values of EDE of the ideal HVAC system compare with those of the ideal one-zone and two-zone building loads. As discussed above, the EDE(1-zone) has an absolute value of unity under all operating conditions. The EDE plots for the two-zone building is less than unity in the range 7° C < T_o <17° C as noted earlier. The EDE of the ideal HVAC system servicing a two-zone building is less than unity in the range of T_o < 22° C. The extent to which excess energy use (in the form of heating and cooling) is used by the ideal HVAC system is well captured by the EDE index. For example, from Figure 9, at T_o = 9° C, EDE(2-zone) = 0.72 and EDE(HVAC) = 0.42. We can then infer that an HVAC system, even if operated ideally, would need (0.72/0.42) = 1.71 times more energy than for the building loads of the same two-zone building. Thus, the EDE of an actual HVAC system in a particular building, compared to that of an ideal HVAC system, would provide a direct and absolute measure of the energy use efficiency and the possible scope of energy conservation in that building.

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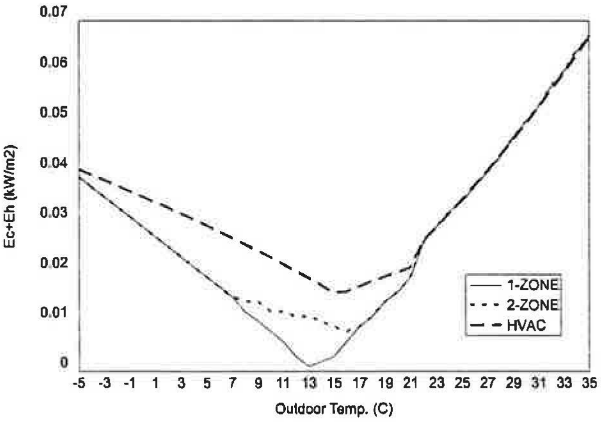

Figure 8 *Variation of total energy use (i.e., sum of heating and cooling energy) with outdoor temperature for ideal 1-zone and 2-zone building loads and for the ideal HVAC system using input data from Tables 1 and 2 as simulation inputs.*

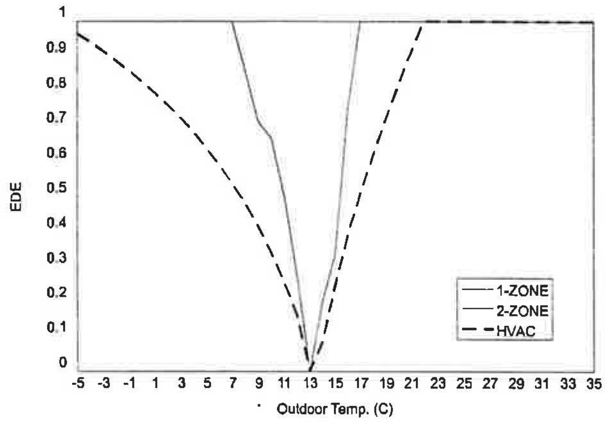

Figure 9 *Variation of absolute EDE with outdoor temperature for ideal 1-zone and 2-zone building loads and for the ideal HVAC system using input data from Tables 1 and 2 as simulation inputs.*

SUMMARY

HVAC systems of large commercial buildings consume energy in excess of the sum total of the building loads as a result of mixing the hot and cold airstreams. This energy penalty can be characterized by the energy delivery efficiency (EDE) concept, which was proposed in an earlier paper (Reddy et al. 1994). One of the objectives of the present study was to extend the theoretical development of the EDE concept by including humidity effects, as well as economizer cycle operation in the expressions for the minimum heating and cooling energy use of one-zone and two-zone buildings.

An energy standard based on a two-zone model may still be unrealistic when comparing the energy efficiency of actual HVAC systems. This argument has a parallel in thermodynamics, where efficiency of an actual steam power plant is more realistically compared with the Rankine efficiency rather than with the Carnot efficiency. An energy standard based on ideal HVAC system performance would then be the logical basis of comparing actual system performance. Consequently, we have suggested an HVAC configuration and operation that ensures the stipulated ventilation airflow rate necessary for satisfactory IAQ in the multiple zones while recirculating the minimum airflow rate necessary for human comfort. Expressions for the heating and cooling energy for such an ideal HVAC system have also been presented in this paper.

The extent to which heating and cooling energy use, as well as EDE of such an ideal HVAC system, differ from those of one-zone and two-zone buildings has been discussed by means of simulations, using "typical" building and system parameters. The appropriateness of the models for minimum energy use and how to determine the various physical parameters from monitored heating and cooling energy use data has been discussed by Deng et al. (1997). The usefulness of the EDE approach as a diagnostic tool to evaluate HVAC retrofit performance in actual buildings, where year-long monitored energy use data are available, has been illustrated by Deng (1997). Note that the EDE concept, in its current development, is limited to the thermal efficiency of the air-side system and does not consider primary system conversion efficiencies or transport losses in the ducting or possible changes to existing building material and to equipment and lights.

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NOMENCLATURE

A	= conditioned floor area of building
A_S	= surface area of building
c	= specific heat at constant pressure
E	= whole-building HVAC system energy use or load
h	= enthalpy
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 airflow rate per unit-conditioned floor area of building
m_{min}	= minimum supply airflow rate per unit-conditioned floor area of building
$m_{v,min}$	= minimum ventilation airflow rate per unit-conditioned area
Q_B	= building thermal loads

q_{LR}	=	monitored electricity use per unit area of lights and receptacles inside the building
RH	=	relative humidity
T	=	temperature
U	=	overall building shell heat loss coefficient per unit area
w	=	specific humidity

Subscripts

a	=	air
b	=	balance point
C	=	cooling, cold deck
e	=	economizer
ext	=	exterior zone
H	=	heating, hot deck
h	=	humid, humidification
int	=	interior zone
m	=	mixed air
min	=	minimum
o	=	outdoor
s	=	supply air
sen	=	sensible
sol	=	solar
v	=	ventilation
z	=	zone

Greek

δ	=	indicator variable defined by Equation 7

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