Ventilation Control Strategy Using the Supply CO_2 Concentration Setpoint

Nabil Nassif
Student Member ASHRAE

Stanislaw Kajl, PhD
Member ASHRAE

Robert Sabourin, PhD

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This paper proposes a new ventilation control strategy applied to multiple spaces subject to variable occupancy. The strategy specified for real-time, online ventilation control takes advantage of uninitiated air from some overventilated spaces to be used as fresh outdoor air in order to reduce system energy use while maintaining the indoor air quality (IAQ) in each space. This proposed strategy maintains a supply CO_2 concentration setpoint low enough to dilute CO_2 generated by full occupancy in critical zones. The supply CO_2 concentration setpoint could be determined online using the monitored zone airflow rates. It is tested and evaluated by making comparisons with other known control strategies. An existing VAV system installed at the École de technologie supérieure is used to evaluate this new strategy. The outdoor air fraction and associated energy use of investigated ventilation control strategies are calculated using the VAV system component models that are developed and validated against the monitored data.

INTRODUCTION

The quality of the air inside buildings and the associated energy use has been of significant growing concern over the last 20 years. Many publications have discussed different ventilation control strategies (Elovitz 1995; Janu et al. 1995; Wang and Xu 2002). The CO_2-based demand-controlled ventilation (CO_2-DCV) is one of the strategies that could reduce energy use by reducing the unnecessary overventilation of buildings. This strategy is investigated in several studies (Alalawi and Krarti 2002; Persily 1993). The energy consumption reduction benefits derived through the use of this strategy have been demonstrated for many different applications (Carpenter 1996; Schell 1998). When this strategy is applied for multiple spaces by detecting the CO_2 concentration in return air, poor air quality may result inside certain zones. To overcome this problem, ASHRAE Standard 62-2001 (ASHRAE 2001) proposes the multiple space equation (MSE), which corrects the fraction of outdoor ventilation air in a supply system in order to minimize energy use while maintaining a proper indoor air quality (IAQ) in all zones, including the critical one. Since the amount of outdoor air is based on the design number of occupants, this amount could be more than required by the actual number (off-design) and result in waste in energy use. The MSE has been discussed and implemented by several researchers (Ke and Mumma 1999; Ke and Mumma 1997a; Kettler 2003; Mumma and Bolin 1994).

The "supply CO_2-based demand-controlled ventilation" (S-CO_2-DCV) technique proposed in this paper is a compromise between the MSE and the CO_2-DCV, taking into account (1) the actual occupancy of the building and (2) the critical zone ventilation requirement. This proposed strategy allows online control of the outdoor air in response to actual building occupancy (as in CO_2-DCV) while ensuring the ventilation requirements of each individual zone, including the

Nabil Nassif, **Stanislaw Kajl**, and **Robert Sabourin** are at École de Technologie supérieure, Montréal, Canada.

critical zone (as in MSE). It is tested and evaluated through a comparison with other conventional control strategies. The existing VAV system installed at the École de technologie supérieure (ÉTS) is used to evaluate this new strategy. The outdoor air fraction and associated energy use of the investigated ventilation control strategies are calculated using the VAV system component models that are developed and validated against the monitored data.

VENTILATION CONTROL STRATEGIES

The ventilation control strategy aims to provide an acceptable IAQ with minimum energy consumption. The two criteria, "the IAQ quality" and "energy use," are used to evaluate the five different control strategies investigated in this paper. The following are the control strategies studied:

1. Strategy S_1: fixed minimum outdoor air percentage;
2. Strategy S_2: fixed minimum outdoor air rate;
3. Strategy S_3: CO_2-based demand-controlled ventilation (CO_2-DCV);
4. Strategy S_4: multiple space equation (MSE);
5. Strategy S_5: proposed supply CO_2-based demand-controlled ventilation (S-CO_2-DCV).

All these strategies use the outdoor, return, and exhaust dampers to control the proportion of outdoor air in the supply air. Figure 1 shows a schematic diagram of the AHU air distribution system for the investigated VAV system, with the key points used. The controller output specified by the implemented ventilation control strategy is used to modulate the three-coupled dampers in order to provide an adequate outdoor airflow rate to meet the ventilation requirement.

At the design phase, the amount of outdoor air to be introduced into each zone i ($\dot{Q}_{o,z,dsg,i}$) or entire building ($\dot{Q}_{o,b,dsg}$) is based on providing enough outdoor air to meet the ventilation requirements at full occupancy, i.e., the ventilation rate per person (15 cfm). The fraction of outdoor air introduced into zone i ($Y_{z,dsg,i}$) and building ($Y_{b,dsg}$) at the design phase are given.

$$Y_{z,dsg,i} = \frac{\dot{Q}_{o,z,dsg,i}}{\dot{Q}_{z,dsg,i}} \tag{1}$$

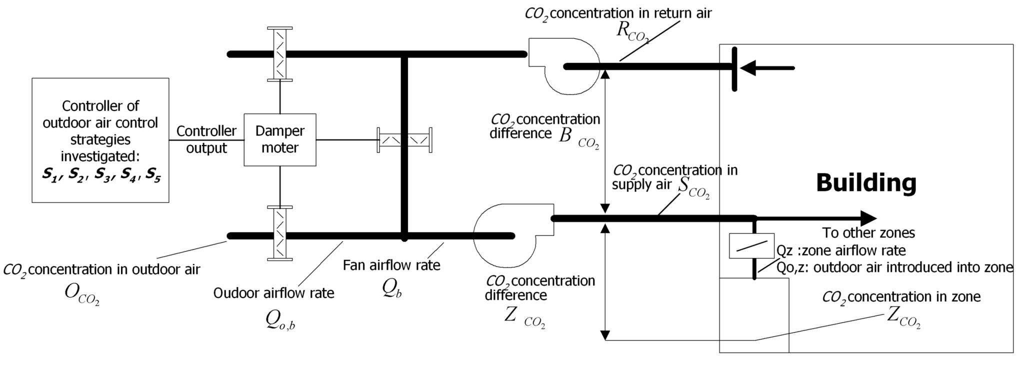

Figure 1. Schematic diagram of the AHU air distribution system for VAV system with the key points used.

$$Y_{b,dsg} = \frac{\dot{Q}_{o,b,dsg}}{\dot{Q}_{b,dsg}} \tag{2}$$

At design conditions, the simple relationship between the occupant-generated CO_2 (ventilation load, *VL*) and the indoor CO_2 concentration (Z_{CO_2}) and outdoor air ($\dot{Q}_o$) is presented in Appendix C of Standard 62-2001 in the following steady-state mass balance equation.

For zone *i*:

$$VL_{z,dsg,i} = \dot{Q}_{o,z,dsg,i} \cdot (Z_{CO_2,i} - O_{CO_2}) \tag{3}$$

For the building:

$$VL_{b,dsg} = \dot{Q}_{o,b,dsg} \cdot (R_{CO_2} - O_{CO_2}) \tag{4}$$

The last equation can be rewritten for off-design conditions in terms of the airflow rate introduced into the building (fan airflow rate, $\dot{Q}_b$) and the CO_2 concentration difference ($B_{\Delta CO_2}$) between the return (R_{CO_2}) and supply (S_{CO_2}) air.

$$VL_b = \dot{Q}_b \cdot B_{\Delta CO_2} \tag{5}$$

where

$$B_{\Delta CO_2} = R_{CO_2} - S_{CO_2} \tag{6}$$

Equation 3 could also be represented using the zone airflow rate ($\dot{Q}_{z,i}$) and the CO_2 concentration difference between the zone (Z_{CO_2}) and supply (S_{CO_2}) air.

$$VL_{z,i} = \dot{Q}_{z,i} \cdot (Z_{CO_2,i} - S_{CO_2}) \tag{7}$$

ASHRAE Standard 62-2001 indicates that comfort criteria with respect to human bioeffluents are likely to be satisfied if the ventilation rate is 15 cfm per person. This corresponds to an indoor/outdoor CO_2 differential of 700 ppm. It means that the standard requires a maximum CO_2 concentration of 1000 ppm in zones, based on the assumption of outdoor air being 300 ppm. Equations 3 and 4 become, for design conditions (full occupancy and fan airflow rate):

$$VL_{z,dsg,i} = Y_{z,dsg,i} \cdot \dot{Q}_{z,dsg,i} \cdot 700 \tag{8}$$

$$VL_{b,dsg} = Y_{b,dsg} \cdot \dot{Q}_{b,dsg} \cdot 700 \tag{9}$$

Four terms, $R_{v,b}$, $R_{a,b}$, $R_{v,z}$, $R_{a,z}$, are introduced here in order to take into account the online operation (off-design conditions). These are defined by the following equations:

$$\left\{ \begin{array}{ll} R_{a,b} = \dfrac{\dot{Q}_b}{\dot{Q}_{b,dsg}} & R_{a,z,i} = \dfrac{\dot{Q}_{z,i}}{\dot{Q}_{z,dsg,i}} \\ R_{v,b} = \dfrac{VL_b}{VL_{b,dsg}} = \dfrac{N_b}{N_{b,dsg}} & R_{v,z,i} = \dfrac{VL_{z,i}}{VL_{z,dsg,i}} = \dfrac{N_{z,i}}{N_{z,dsg,i}} \end{array} \right\} \tag{10}$$

The terms N_b and $N_{z,i}$ represent the actual occupancy in the building and in each zone i, respectively.

The airflow part-load ratios of the building and zone i ($R_{a,b}$, $R_{a,z,i}$, respectively) are defined as the ratio of the actual-to-design airflow rate. The ventilation part-load ratios of the building and zone i ($R_{v,b}$, $R_{v,z,i}$, respectively) are defined as the ratio of the actual-to-design number of occupants.

In order to calculate the actual CO_2 concentration difference between supply and return air ($B_{\Delta CO_2}$), Equations 6, 9, and 10 are used.

$$B_{\Delta CO_2} = \frac{Y_{b,dsg} \cdot 700 \cdot R_{v,b}}{R_{a,b}} \tag{11}$$

The same logic applicable for zones, the actual CO_2 concentration difference between supply and return air in zone i ($Z_{\Delta CO_2,i}$), is given.

$$Z_{\Delta CO_2,i} = \frac{Y_{z,dsg,i} \cdot 700 \cdot R_{v,z,i}}{R_{a,z,i}} \tag{12}$$

Both (CO_2-DCV) and the proposed (S-CO_2-DCV) strategies determine the amount of ventilation outdoor airflow in response to actual building occupancy (real time, online ventilation load) by detecting the CO_2 concentration. However, the other control strategies ($\boldsymbol{S_1}$, $\boldsymbol{S_2}$, and $\boldsymbol{S_4}$) do not respond to occupancy change, and the outdoor air is determined according to full occupation. Figure 2 shows the investigated ventilation control strategies, which are described in the following paragraphs.

Strategy S_1: Fixed Minimum Outdoor Air Percentage

This strategy uses a fixed percentage of outdoor air to supply air when the economizer is not activated. This fixed percentage is based on providing enough outside air to meet the requirements of the building at full occupancy ($R_{v,b}$ = 1) and at design airflow rate ($R_{a,b}$ = 1). This strategy is commonly achieved by a fixed minimum outdoor air damper position (Ke and Mumma 1999). The actual fraction of outdoor airflow (Y_b) when the economizer is not activated is always constant and equal to the design value. It is then presented as

$$(Y_b)_{S_1} = \frac{\dot{Q}_{o,b,dsg}}{\dot{Q}_{b,dsg}} = Y_{b,dsg} \,. \tag{13}$$

Figure 2 shows this control strategy. During the occupied period and when the economizer is not activated, the controller output is used to maintain the damper at a constant position. The main drawback of this control is the low outdoor air available with low supply air, which leads to a poor IAQ at both the building and individual zone levels.

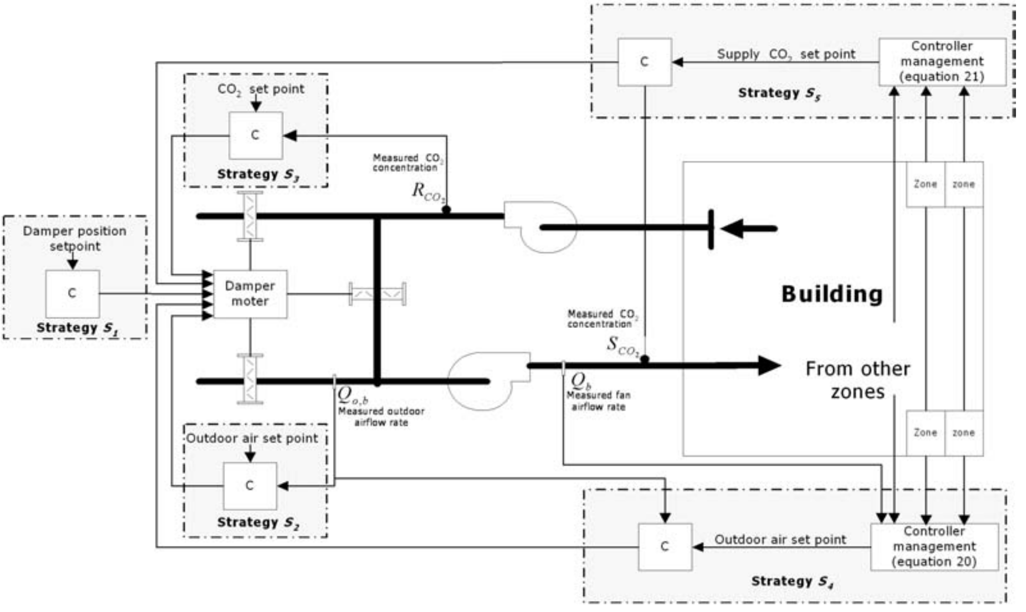

Figure 2. Investigated ventilation control strategies.

Strategy S_2: Fixed Minimum Outdoor Air Rate

This control strategy maintains a constant amount of outdoor airflow rate determined at full occupancy. The online fraction of outdoor airflow (Y_b) becomes

$$(Y_b)_{S_2} = \frac{\dot{Q}_{o,b,dsg}}{\dot{Q}_b} = \frac{Y_{b,dsg}}{R_{a,b}} . \tag{14}$$

This strategy takes into account the actual value of the fan airflow presented by the term ($R_{a,b}$). This strategy is commonly achieved by using the measurement of the outdoor airflow rate, as shown in Figure 2. During the occupied period, and when the economizer is not activated, the controller output is used to adjust the position of three-coupled dampers in order to maintain the measured outdoor airflow rate at its setpoint value.

Strategy S_3: CO_2-Based Demand-Controlled Ventilation

The CO_2-based demand-controlled ventilation (CO_2–DCV) strategy allows an HVAC system to control the amount of outdoor air supplied to entire system. It detects carbon dioxide in the return duct as an indictor of occupancy density and adjusts the outdoor air based on this CO_2 concentration. Since this strategy maintains the CO_2 concentration in the return duct at the threshold setpoint limit while overlooking the individual zone CO_2 concentration, some zones may be over- or underventilated. The controller output of this strategy, which is determined by comparing the CO_2 concentration in the return air with its setpoint value, is used to modulate the three-coupled dampers accordingly (see Figure 2). The resulting amount of outdoor airflow should be enough to maintain the return CO_2 concentration setpoint.

The online fraction of outdoor airflow in the supply air could be calculated using the balance method of return, outdoor, and supply CO_2 concentrations.

$$Y_b = \frac{R_{CO_2} - S_{CO_2}}{R_{CO_2} - O_{CO_2}} \tag{15}$$

Using Equations 6 and 11 while maintaining the return CO_2 concentration at a 1000 ppm set-point and with an outdoor CO_2 concentration of 300 ppm, the equation above becomes, for strategy S_3,

$$(Y_b)_{S_3} = Y_{b,dsg} \cdot \frac{R_{v,b}}{R_{a,b}} . \tag{16}$$

The online outdoor air fraction of the CO_2-DCV strategy, which varies with the online occupancy and fan airflow rate, is decreased at lower occupancy (decrease in $R_{v,b}$), while it is fixed for strategy $\boldsymbol{S_1}$ and is only a function of the online fan airflow rate for strategy $\boldsymbol{S_2}$. The decrease in the outdoor airflow rate used in this strategy at low occupancy is due to the term ($R_{v,b}$), which considers the actual online occupancy.

When there is wide variance between the design and off-design ventilation loads, the CO_2-DCV ventilation provides a significant energy savings potential but with possible poor IAQ in certain zones, especially when the multiple spaces are subject to variable ventilation and airflow part-load ratios. The multiple space equation described below can provide a good air quality inside all zones, including a critical zone, by introducing higher outdoor air determined by the critical zone ventilation requirement at full occupancy.

Strategy S_4: Multiple Space Equation

According to the ASHRAE Standard 62-2001, “Where more than one space is served by a common supply system, the ratio of outdoor to supply air required to satisfy the ventilation and thermal control requirements may differ from space to space.” As a result, the critical space requiring the highest percentage of outdoor airflow drives the percentage of outdoor air required by the entire system. The outdoor airflow rate can be reduced by taking advantage of uninitiated air from non-critical spaces to be used as fresh outdoor air. The corrected fraction of outdoor ventilation air in the supply system (Y_b), as given in ASHRAE Standard 62-2001, is used.

$$Y_b = \frac{X}{1 + X - Z} \tag{17}$$

Since the online occupancy patterns change arbitrarily from what is originally assumed, the online fraction of the outdoor ventilation air in the supply system is always determined at full occupancy.

The term X is the uncorrected fraction of outdoor ventilation air required at full occupancy in the supply system (the ratio of the sum of outdoor ventilation airflow rates for all zones to the fan airflow rate). The term Z is the ratio of the required outdoor air at full occupancy to the primary air in the critical zone.

$$X = \frac{\dot{Q}_{o,b,dsg}}{\dot{Q}_b} = \frac{Y_{b,dsg}}{R_{a,b}} \tag{18}$$

$$Z = \left\{ \frac{\dot{Q}_{o,z,dsg}}{\dot{Q}_z} \right\}_{critical} = \left\{ \frac{Y_{z,dsg}}{R_{a,z}} \right\}_{critical} \tag{19}$$

Using Equations 18 and 19, Equation 17 becomes

$$(Y_b)_{S_4} = \frac{\dfrac{Y_{b,dsg}}{R_{a,b}}}{1 + \dfrac{Y_{b,dsg}}{R_{a,b}} - \left\{ \dfrac{Y_{z,dsg}}{R_{a,z}} \right\}_{critical}} . \tag{20}$$

This strategy takes into consideration the actual values of the fan and critical zone airflow rates by using the ($R_{a,b}$) and ($R_{a,z}$) factors. The strategy is achieved as shown in Figure 2. The controller management determines the outdoor airflow rate setpoint through Equation 20, using the measured fan and zone airflow rates. The controller output determined by comparing the measured outdoor airflow rate with its setpoint value is used accordingly to modulate the three-coupled dampers. The resulting amount of outdoor airflow should be enough to provide the proper IAQ in all zones. Compared to the CO_2-DCV strategy, the MSE strategy needs much higher outdoor air—and, consequently, higher energy use—when the online occupancy pattern is significantly decreased from what is originally assumed at design. The main drawback of this control strategy is its inability to respond to changes in occupancy, as does the CO_2-DCV. This seems to be due to the term ($R_{v,b}$) not being included in Equation 20. The proposed "supply CO_2-based demand-controlled ventilation" provides good IAQ in zones, while taking into account the actual building occupancy as described below.

Strategy S_5: Supply CO_2-Based Demand-Controlled Ventilation

The supply CO_2-based demand-controlled ventilation strategy (S-CO_2-DCV) maintains the supply CO_2 concentration at a setpoint value ($S_{CO_2,set}$) determined using the monitored zone airflow rates, which has recently become possible with the use of direct digital control terminal boxes. The supply CO_2-concentration setpoint is calculated by assuming that full occupancy is faced in the critical zone. This permits the diluting of the highest possible CO_2 generation. Since the setpoint considers the full occupancy in the critical zone, overventilation may occur in the critical as well as in the noncritical zones when the actual occupancy is lower than the full design value. Overventilation in the different zones leads to a lower return CO_2 concentration and a resulting lower outdoor flow rate to maintain the required supply CO_2-concentration setpoint.

Since the supply CO_2 setpoint must maintain the CO_2 concentration in the critical zone at a lower level than the required threshold ($Z_{CO_2,th}$), it could be assumed that the CO_2 concentration in the critical zone is always at the threshold level (i.e., 1000 ppm). This further ensures that the CO_2 concentration in the critical zone does not exceed the threshold level, and the steady-state balance in the critical zone could be used to determine the supply CO_2 concentration setpoint. Thus, by assuming that there is full occupancy, the steady-state balance presented in Equation 7 could be rewritten for each zone as

$$S_{CO_2,set} = \min \left\{ Z_{CO_2,th,i} - \frac{VL_{z,dsg,i}}{\dot{Q}_{z,i}} \right\} . \tag{21}$$

A term within the “min” is calculated for each zone i, representing the required supply CO_2 concentration to dilute the full occupancy CO_2 generation, and the lowest value is used to determine the supply CO_2 concentration setpoint. The critical zone is the zone requiring the lowest value of supply CO_2 concentration. The resulting online CO_2 concentration in the critical zone, which may have either a design or an off-design ventilation load, is equal to or lower than the threshold point. It should be noted that the term ($VL_{z,dsg,i}$) is determined from known design conditions of ventilation requirements (full occupancy) given by Equation 8 or by Equation 3 using the design outdoor airflow rate condition. Using Equation 8 to replace ($VL_{z,dsg,i}$) in Equation 21, the critical zone is the zone with the highest value of $Y_{z,dsg}/R_{a,z}$ (as used in Equations 19 and 22 next).

A comparison of the proposed strategy (S-CO_2-DCV) with the MSE and CO_2-DCV leads to the following conclusions:

1. The S-CO_2-DCV takes into account the online overventilation that occurs in the building while considering the critical zone ventilation requirement.
2. The CO_2-DCV takes into account the online overventilation in the building without considering the critical zone ventilation requirement.
3. The MSE takes into account only the design overventilation in the different zones.

At a low building ventilation load, the ventilation flow rate may be lower than the outdoor air required to dilute building-source contaminants and to maintain space pressurization. To overcome these problems, the minimum outdoor airflow rate is limited in order to provide adequate outdoor air for other occupancy purposes. The building-source contaminant, which does not vary with online occupancy variations, could be estimated at the design condition.

For the ventilation requirement in the critical zone to be satisfied (i.e.,15 cfm per person), the threshold point ($Z_{CO_2,th}$) must be lower than or equal to 1000 ppm, based on the assumption of outdoor air being 300 ppm. Using the assumption above, and inserting Equations 8, 11, and 21 into the CO_2 concentration balance Equation 15, it can be concluded that the one-line outdoor fraction for S-CO_2-DCV strategy is given as

$$(Y_b)_{S_5} = \frac{Y_{b,dsg} \cdot \dfrac{R_{v,b}}{R_{a,b}}}{1 + Y_{b,dsg} \cdot \dfrac{R_{v,b}}{R_{a,b}} - \left\{\dfrac{Y_{z,dsg}}{R_{a,z}}\right\}_{critical}} . \tag{22}$$

The outdoor fraction is less than that obtained from the multiple space equation (Equation 21) because of the term $R_{v,b}$ (online occupancy). Since the actual number of occupants in each zone is not easy to predict unless there is a CO_2 concentration sensor located in each zone, it is proposed that this strategy maintain a supply CO_2 concentration setpoint low enough to dilute CO_2 generated by full occupancy in the critical zone. As shown in Equation 20, the zone with the highest value of ($Y_{z,dsg}/R_{a,z}$) is considered to be the critical zone.

When it is possible to predict the actual occupancy in critical zone $R_{v,z}$, the outdoor air fraction could be calculated as follows:

$$Y_b = \frac{Y_{b,dsg} \cdot \dfrac{R_{v,b}}{R_{a,b}}}{1 + Y_{b,dsg} \cdot \dfrac{R_{v,b}}{R_{a,b}} - \left\{Y_{z,dsg} \cdot \dfrac{R_{v,z}}{R_{a,z}}\right\}_{critical}} \tag{23}$$

Since there are difficulties faced in predicting occupancy by measuring the CO_2 concentration in all zones (68 zones in the investigated existing system), Equation 23 is not investigated in this paper. The S-CO_2-DCV strategy can be applied through the following two methods:

1. **S-CO_2-DCV strategy with local control loop:** This method employs the local CO_2 control loop to maintain the supply CO_2 concentration below its setpoint (Equation 21) by controlling the outdoor damper. This is the same as with the CO_2-DCV, except that the CO_2 concentration sensor is placed in the supply rather than in the return duct. Figure 2 shows the S-CO_2-DCV strategy with the local control loop. The controller management calculates the supply CO_2 concentration setpoint through Equation 21, using the monitored zone airflow rates. The controller output, which is determined by comparing the measured supply CO_2 concentration with its calculated setpoint value, is used to modulate the three-coupled damper accordingly.
2. **S-CO_2-DCV strategy with Equation 22:** The outdoor air fraction in this method is determined by Equation 22. However, the actual occupancy in the building ($R_{v,b}$) must be predicted using the methods presented in several studies (Ke and Mumma 1997b; Wang and Jin 1998). With these methods, the actual number of occupants in the entire building is calculated using the monitored CO_2 concentration in the return and the supply air.

CONTROL STRATEGY OPTIMIZATION

The proposed supply CO_2-based demand-controlled ventilation could be integrated into the online optimization program of the supervisory control strategy (Nassif et al. 2003) such that with each simulation, the setpoints of the HVAC system, including the supply CO_2 concentration setpoint, could be optimized. To reduce the outdoor airflow rate, the supply CO_2 concentration setpoint should be increased by using the higher airflow supplied in the critical zone. This could be achieved by optimizing the zone reheat (supply zone air temperature) and system supply and zone air temperature setpoints. The S-CO_2 –DCV as well as the CO_2-DCV could be applied online without using the CO_2 control loop, allowing Equations 16 and 22 to be used. In that case, the actual occupancy in the entire building could be determined based on the measured CO_2 concentrations in the supply and return air.

EVALUATION OF VENTILATION CONTROL STRATEGIES

The investigated ventilation control strategies are evaluated and tested for the existing HVAC system installed at the ETS campus. Only two air-handling units (AHU-6 and AHU-4) working alongside several others to provide conditioned air to the ETS campus are investigated. Figure 1 shows a schematic diagram of the investigated VAV system. The AHU-6 meets the thermal loads for 70 internal zones on the second floor, while the AHU-4 meets the thermal loads for 68 southwest perimeter zones. The investigated ventilation control strategies are evaluated and tested for following three cases:

1. Evaluation based on three operating conditions of a VAV system. In this evaluation, the potential of the proposed strategy to introduce an exact outdoor air for ventilation comfort are presented.
2. Evaluation based on real monitored data of the AHU-6 system. The performances of the VAV system working with the strategies ($\boldsymbol{S_3}$, $\boldsymbol{S_4}$, and $\boldsymbol{S_5}$) are discussed. How the proposed strategy minimizes the chiller energy use by introducing an exact outdoor air for ventilation comfort is presented and discussed.
3. Evaluation based on bin weather data of different locations tested on the AHU-4 system. The effect of the actual occupancy (Rv) and the locations on annual energy use of all investigated strategies are studied.

For these evaluations, the three modeling methodologies (MM) presented below were developed. It should be noted that the ventilation control strategy for the AHU-4 and AHU-6 systems uses a fixed minimum air damper position while the return CO_2 concentration is less than 600 ppm; otherwise, the damper position is modulated to maintain a return CO_2 concentration less than 800 ppm. The design outdoor airflow fractions of all zones could be approximated to be the same as 16% ($Y_{z,dsg,i}$ = 0.16), and the design building outdoor airflow fraction is 23% ($Y_{b,dsg}$ = 0.23). The two design building and zone fractions are not the same because the sum of the design zone airflow rates equals about 0.76% of the design fan airflow rate (diversity factor).

MODELING METHODOLOGY

To study the energy use and satisfactory zone ventilation of the investigated strategies, the component models required for simulation calculations, such as the fan, damper, cooling coil, and chiller models, are developed and validated against the monitored data (Nassif et al. 2004).

Three modeling strategies (MM-1, MM-2, and MM-3) required for the three evaluations presented above are used:

MM-1 Modeling strategy for evaluation based on three operating conditions of a VAV system operation. The outdoor air fractions of investigated strategies are determined by Equations 13, 14, 16, 20, and 22.

MM-2 Modeling strategy for AHU-6 system evaluation. The chiller energy use and indoor air quality index (CO_2 concentration) are determined using the measured data and validated component models. The fan airflow rate (to meet the thermal load) is calculated from monitored data, whereas the CO_2 concentrations, outdoor air fraction, and chiller energy use are then determined by the detailed and validated component models. This modeling strategy is illustrated in Figure 3 and described below.

MM-3 Modeling strategy for AHU-4 system evaluation. The annual energy use and indoor air quality index (CO_2 concentration) are determined using the weather bin temperature data and validated component models. The thermal loads at given zone temperatures are simplified as a function of outdoor temperature and heat gain from occupants, whereas the CO_2 concentrations, outdoor air fractions, and chiller and heating energy uses are then determined by the detailed and validated component models. This modeling strategy is illustrated in Figure 4 and described below.

Modeling Strategy for AHU-6 System Evaluation (MM-2 real monitored data)

The chiller energy use and outdoor air fraction are determined using validated component models and the measured data for one year (2002) culled from the AHU-6 VAV system. The monitored data are provided at each minute and saved to a data file throughout the year. The periods of June, July, August, and September are only investigated when the economizer may be at minimum air mode and ventilation control strategy is required for application. This limited period is only investigated for the following reasons, concluded from the monitoring of the AHU-6 (internal zones):

- The outdoor damper is often modulated in winter and mid-season weather (free cooling) in order to maintain supply air temperature setpoints without a ventilation control strategy requirement.
- The valve position of the heating coil is opened only for a few days a year (during very cold weather). In addition, there are no zone reheats working with the AHU-6 system that provides conditioned air to the internal zones (cooling loads). Thus, the effect of ventilation control strategies on annual heating energy use could be neglected.

The modeling methodology (see Figure 3) is divided into two calculation parts: (1) pre-calculation and (2) VAV system model. The airflow and ventilation part-load ratios ($R_{a,b}$, $R_{a,z}$, and $R_{v,b}$) are determined in the first one, while the outdoor airflow rate and energy use are obtained in the second.

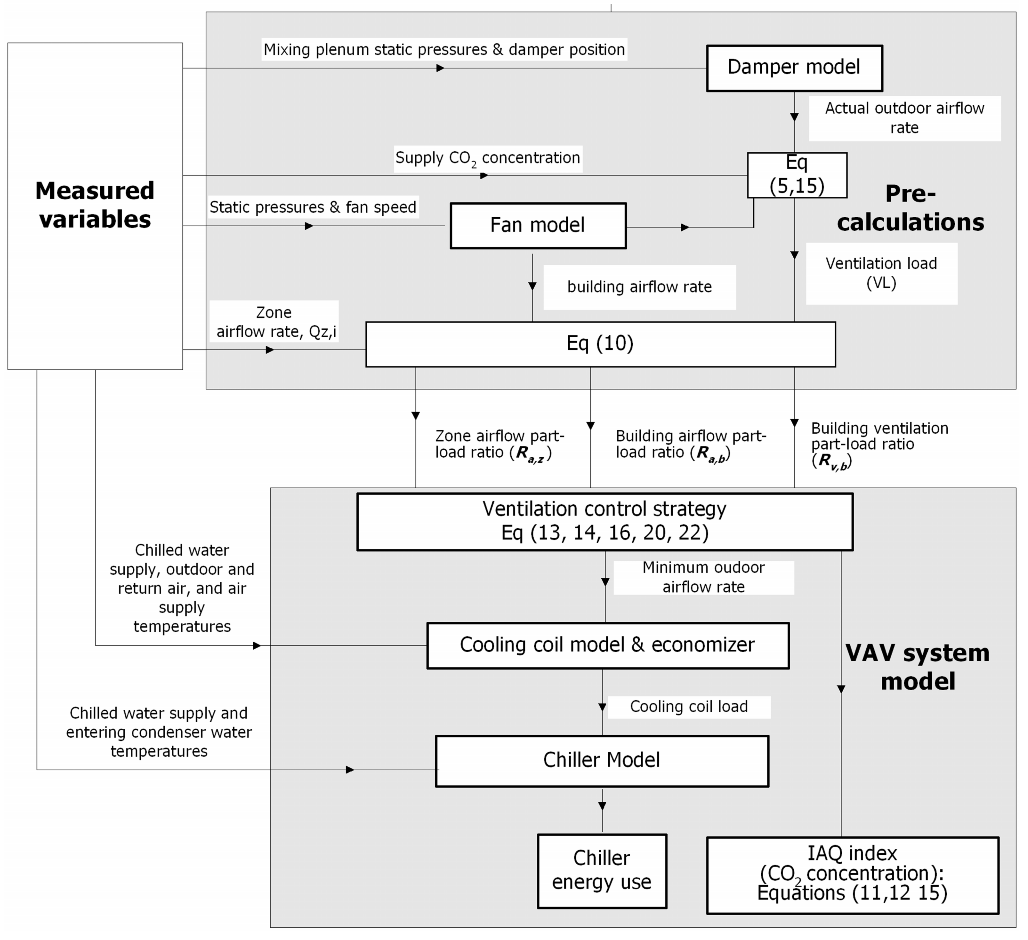

Figure 3. Modeling strategy used for AHU-6 system evaluation.

In the precalculation part, the airflow and ventilation part-load ratios ($R_{a,b}$, $R_{a,z}$, and $R_{v,b}$) are determined by Equation 10 using (1) the fan airflow rate and building ventilation loads (VL_b) calculated from the measured data and (2) the measured zone airflow rates. The fan airflow rate is not measured but, rather, is calculated by the validated fan model based on the dimensionless coefficients of flow, pressure, and head. The inputs of the fan model consist of the measured static pressures and fan speed. The building ventilation load (VL_b) is determined by Equations 5 and 15 using the calculated outdoor airflow rate and the measured return CO_2 concentration. The outdoor airflow rate in the existing system is also not measured but calculated using the damper model based on exponential relations. The inputs of the damper model consist of the measured mixing duct static pressure and damper position. When the airflow and ventilation part-load ratios are determined, the outdoor airflow rate and energy use are determined by a VAV system model.

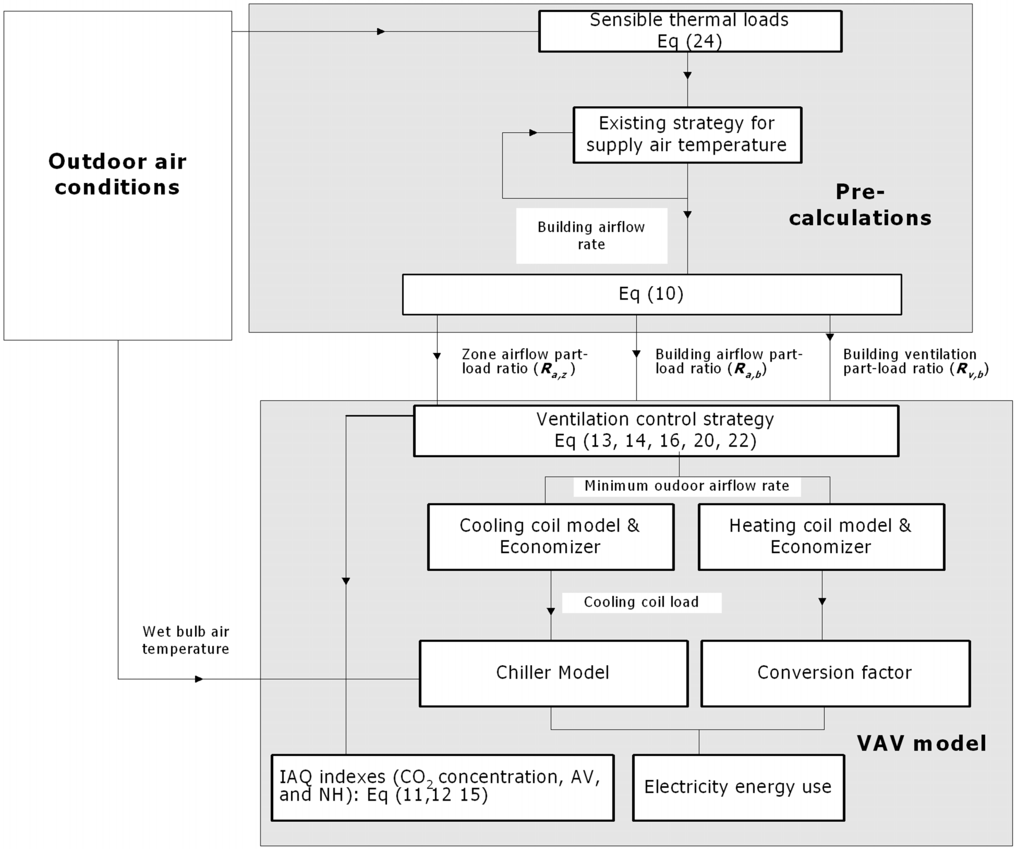

Figure 4. Modeling strategy used for AHU-4 system evaluation.

The VAV system model could be presented through the four following main steps:

Step 1. The outdoor airflow fractions are determined only for CO_2-DCV (***S_3***), MSE (***S_4***), and S- CO_2-DCV (***S_5***) strategies using Equations 16, 20, and 22, respectively. The highest value of $Y_{z,,dsg}/R_{a,z}$ (critical zone) is used in Equations 20 and 22.

Step 2. This step represents the iteration process during which the initial value of the cooling coil leaving air humidity ratio is assumed, and the new value is calculated and reused. This iterative process continues calculating through the loop several times until the values of the cooling coil leaving air humidity ratio stabilize within a specified tolerance. The cooling coil model developed from the ASHRAE HVAC 2 Toolkit (Brandemuel et al. 1993) and validated against the monitored data (Nassif et al. 2004) is used. The initial value of the cooling coil leaving air humidity ratio is assumed, and that for the return air is determined using building latent loads calculated from actual occupancy ($R_{v,b}$). The measured return air temperature can be used because the building's thermal cooling load and supply air temperature setpoint do not vary with changes in the outdoor airflow rate (strategies used). The mixing plenum air temperature and humidity ratio are determined using the energy balance, knowing the outdoor and fan airflow rates and outdoor and return conditions. The cooling coil load and cooling coil leaving air humidity ratio are determined through a detailed cooling coil model using the input variables: the calculated fan airflow rate, calculated mixing plenum air conditions (air condition entering the cooling coil), and measured chilled water and air supply temperatures. The cooling coil leaving air humidity ratio is then compared to the previous value. This iterative process continues, calculating through the loop several times until the values of the cooling coil leaving air humidity ratio stabilize within a specified tolerance.

Step 3. The chiller energy is determined by the chiller model using the performance curves of electric chillers as functions of variables, such as the calculated cooling coil load, measured entering condenser water temperature, and measured chilled water supply temperature (NRCC 1999).

Step 4. The CO_2 concentrations in the critical zones as indexes of indoor air quality are determined. The rise in the CO_2 concentration throughout the building is determined by Equation 11. The supply and return CO_2 concentrations are determined using Equation 15. The CO_2 concentration in the critical zone is determined by adding the term ($Z_{\Delta CO_2,i}$) calculated by Equation 12 to the supply CO_2 concentration.

Modeling Strategy for AHU-4 System Evaluation (MM-3-Weather Bin Temperature Data)

The objective of this evaluation is to study the effect of the actual occupancy (*Rv*) and the weather locations on annual energy use with the investigated strategies.

The annual energy use (considering the heating period) and indoor air quality in zones are determined at different values of *Rv* and at different locations. Since the objective 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 at different ventilation load ($R_{v,b}$) and building locations, the thermal loads are determined as functions of bin weather temperature and actual occupancy. These evaluations are made on an AHU-4 system using the bin temperature data of Montreal, Seattle, and Dallas. Thus, the monitored real data for one year are only used to determine the constant parameters of Equation 24 below, which is developed for determining the building and zone thermal loads as a function of outdoor air temperature and occupancy.

The thermal load as a function of the bin weather temperature with the simplified simulation approach (modified bin method) was investigated by Reddy et al. (1998) in order to obtain sound and meaningful diagnostic insights of actual building performance and operating ventilation control strategies. The results of these control ventilation strategy studies, which are based on a simplified HVAC simulation approach, are consistent with the conclusions reached by other researchers using more detailed simulation models (Reddy et al. 1998).

As mentioned above, it assumed that the total sensible thermal building ($q_{s,b}$, %) and zone loads for fixed interior temperatures are functions of internal thermal loads and outdoor temperature. To separate the sensible thermal load from the actual occupants (c_b, $R_{v,b}$), all internal thermal loads other than occupancy are considered as constant values during the occupancy period. Thus, the building (or zone) sensible thermal loads are given by

$$q_{s,b} = a_b \cdot t_o + b_b + c_b \cdot R_{v,b} . \tag{24}$$

The building (or zone) sensible thermal loads of the existing VAV system (AHU-4), which are equal to the energy removed from the space when the space temperature is constant, are calculated, and the parameters of the equations above are determined.

The modeling strategy (see Figure 4) is also divided into two calculation parts: (1) precalculation and (2) VAV system model, which is the same as that described above. The airflow and ventilation part-load ratios ($R_{a,b}$, $R_{a,z}$, and $R_{v,b}$) are determined in the first, and the outdoor airflow rate and energy use are obtained in the second.

In the precalculation part, the thermal sensible building or zone loads are determined by Equation 24 for each bin temperature. To calculate the fan and zone airflow rates from the sensible thermal loads, the supply air temperature must be known. The supply air temperature setpoint is determined by the strategy applied in the existing system (it is a function of the outdoor temperature and the fan airflow rate). The strategy is described as follows: the supply air temperature setpoint changes linearly within the 13°C to 18°C range as the outdoor temperature varies within the –20°C to 20°C range. The supply air temperature calculated above is corrected by adding a value that varies linearly from –2°C to +2°C, corresponding progressively to the variation of the fan airflow rate ratio from 50% to 90%. The supply air temperature setpoint is always limited within the 13°C to 18°C range. Since the supply air temperature setpoint is a function of the fan airflow rate, the iteration process is applied using the initial fan airflow rate value.

The building and zone airflow part-load ratios ($R_{a,b}$, $R_{a,z}$) are determined by Equation 10 using the calculated fan and zone airflow rates. It should be noted that the $R_{v,b}$ is assumed for each calculation.

In the VAV system model, the energy use and indoor air quality are determined using the VAV model similar to that described above, while taking into account the AHU-4 specifications. The chilled water supply temperature of 7.2°C is assumed as in an actual system. The condensing water temperature, higher than the outdoor wet temperature by 3°C, is assumed. The summer and winter zone temperatures are 23°C and 22°C, respectively. The zone reheats are used and calculated when the zone airflow rates are at their minimum in order to maintain zone air temperatures. The heating coil and dehumidification models are also used. The humidification strategy applied in the existing HVAC system is used. The humidification valve position is modulated to maintain the return air relative humidity setpoint determined by the control strategy of the AHU-4 system. The conversion factor of non-electrical energy consumption (heating coil and humidifiers) to electricity was assumed to be 0.21 (Ke and Mumma 1997a). The fan energy use calculated by the fan model is considered for annual energy use. The inputs of the fan model are thus the calculated fan airflow rate and the fan static pressure. The fan static pressure is determined by an equation representing the operation curve, which is a function of the fan airflow rate and the supply duct pressure setpoint of 300 Pa (Nassif et al. 2004).

EVALUATION RESULTS

Results of Evaluation Based on Three Operating Conditions (MM-1)

In order to evaluate the ventilation control strategies described above ($\boldsymbol{S_1}$, $\boldsymbol{S_2}$, $\boldsymbol{S_3}$, $\boldsymbol{S_4}$, $\boldsymbol{S_5}$), the outdoor air fraction and CO_2 concentration in the return, supply, and critical zones of each investigated ventilation control strategy are determined and presented in Table 1 for three operating conditions:

1. *Design fan airflow rate and occupancy*. In this case, all part-load factors ($R_{a,b}$, $R_{v,b}$, $R_{a,z}$, $R_{v,z}$) are equal to 1.
2. *Off-design fan airflow rate and full occupancy*. In this case, the following are assumed: (1) full occupancy (design building ventilation load, $R_{v,b}$ = 1), (2) off-design fan airflow rate assumed to be 50% of design rate ($R_{a,b}$ = 0.5), and (3) airflow rate introduced into critical zone falls to its minimum limit (30% of design); at full occupancy, that means $R_{a,z}$ = 0.3 and $R_{z,v}$ = 1.
3. *Off-design fan airflow rate and occupancy*. In this case, the following are assumed: (1) off-design occupancy ($R_{v,b}$ = 0.5) and off-design fan airflow rate ($R_{a,b}$ = 0.5), and (2) airflow rate introduced into critical zone falls to its minimum limit (30% of design) at full occupancy; that means $R_{a,z}$ = 0.3 and $R_{z,v}$ = 1.

Table 1. Results of the Evaluation Based on Three Operating Conditions ($R_{a,b}$, $R_{v,b}$, $R_{a,z}$, $R_{v,z}$)

	Strategy	Design Fan Airflow Rate and Occupancy	Off-Design Fan Airflow Rate and Full Occupancy	Off-Design Fan Airflow Rate and Occupancy
		A	B	C
Part-load ratios	System	$R_{a,b}$ = 1, $R_{v,b}$ = 1	$R_{a,b}$ = 0.5, $R_{v,b}$ = 1	$R_{a,b}$ = 0.5, $R_{v,b}$ = 0.5
	Critical zones	$R_{a,z}$ = 1, $R_{v,z}$ = 1	$R_{a,z}$ = 0.3, $R_{v,z}$ = 1	$R_{a,z}$ = 0.3, $R_{v,z}$ = 1
Outdoor air fraction Y	S_1	10	10	10
	S_2	10	20	20
	S_3	10	20	10
	S_4	10	23.1	23.1
	S_5	10	23.1	13
CO_2 concentration in critical zone, ppm	S_1	1000	1733.3	1163.3
	S_2	1000	1093.3	813.3
	S_3	1000	1093.3	1163.3
	S_4	1000	1000	766.3
	S_5	1000	1000	1000
CO_2 concentration in supply air, ppm	S_1	930	1560	930
	S_2	930	860	580
	S_3	930	860	930
	S_4	930	766.6	533
	S_5	930	766.6	766.6
CO_2 concentration in return air, ppm	S_1	1000	1700	1000
	S_2	1000	1000	650
	S_3	1000	1000	1000
	S_4	1000	906	603
	S_5	1000	906	836.6

The design building and zone outdoor airflow fractions in this evaluation are assumed to be 10% ($Y_{z,dsg,i}$ and $Y_{b,dsg}$ are 0.1). The outdoor air fraction is determined using Equations 13, 14, 16, 20, and 22. The CO_2 concentration rise throughout the building is determined by Equation 11. The supply and return CO_2 concentrations are determined using Equation 15. The CO_2 concentration in the critical zone is determined by adding the term ($Z_{\Delta CO_2,i}$) calculated by Equation 12 to the supply CO_2 concentration.

At design conditions (Case A), all the strategies perform similarly, as shown in the third column of Table 1. At off-design fan airflow rate and full occupancy (Case B), strategies $\boldsymbol{S_4}$ and $\boldsymbol{S_5}$ perform similarly, as shown in the fourth column. Although strategies $\boldsymbol{S_1}$, $\boldsymbol{S_2}$, and $\boldsymbol{S_3}$ introduce less outdoor air fraction, the fraction is not sufficient to meet the ventilation requirements in all zones. At off-design fan airflow rate and occupancy (Case C), $\boldsymbol{S_5}$ introduces a much lower outdoor air fraction than $\boldsymbol{S_4}$ while maintaining good indoor quality in all zones.

A comparison of the three strategies S_3, S_4, and S_5 shows for Case C that S_3 (CO_2-based demand-controlled ventilation) introduces the least—but inadequate—outdoor air fraction to meet ventilation requirements, causing poor indoor air quality in certain zones (CO_2 concentration is 1163.3 ppm). However, the multiple space equation (S_4) introduces an unnecessarily high outdoor air fraction required for actual occupancy. In this case, the overventilation is seen in both the critical (CO_2 concentration is 766.6 ppm) and noncritical zones and, consequently, unnecessary energy is used to ventilate this high outdoor air. The supply CO_2-based demand-controlled ventilation (S_5) introduces an exact outdoor air fraction to meet ventilation requirements in the critical zone based on online occupancy in the building (CO_2 concentration is 1000 ppm), thus minimizing the energy use required for better indoor air quality. To enhance the indoor air quality (compared to strategy S_3 in Case C), the supply CO_2 demand-controlled ventilation provides only 3% additional outdoor air to satisfy all zones, whereas the multiple space equation provides 13.1%.

Results of Evaluation Based on Real Monitored Data of AHU-6 System (MM-2)

The CO_2-DCV (S_3), MSE (S_4), and S- CO_2-DCV (S_5) strategies are evaluated and compared with the actual operation scenario, using real monitored data for one year (2002) culled from the AHU-6 of the VAV system.

As mentioned above, the outdoor air fraction and associated chiller energy of the three investigated strategies are determined for the period of June, July, August, and September. The airflow and ventilation part-load ratios are first determined using the modeling strategy shown in Figure 3. The VAV system model is then used to calculate the system performance response due to theses values and the ventilation control strategy used. Figure 5 shows the building airflow and ventilation part-load ratios of the AHU-6 for July 25-31, 2002. It is clear that the ventilation part-load ratios (actual occupancy) for Saturday and Sunday are lower than those for other days. The outdoor airflow fractions of the investigated strategies tested on the AHU-6 are calculated by the VAV system model and illustrated for July 25-31 in Figure 6. In the first three days

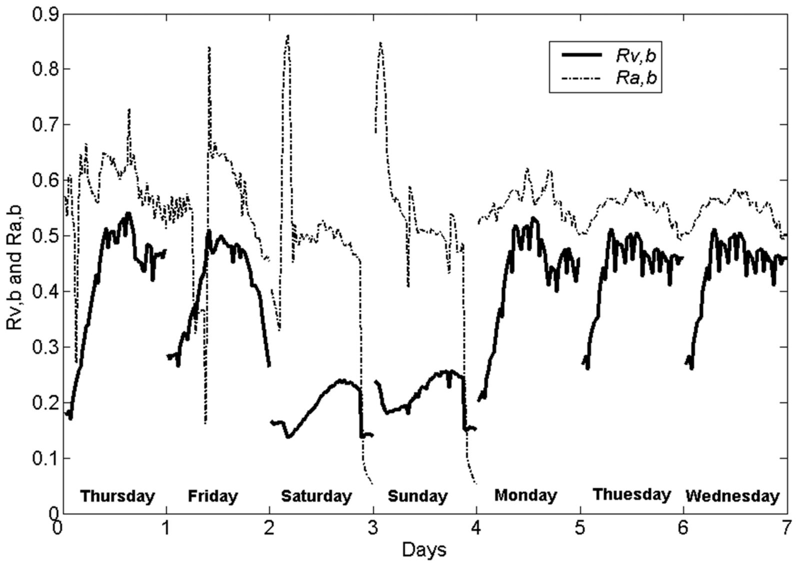

Figure 5. Building airflow and ventilation part-load ratios of AHU-6 for July 25-31, 2002.

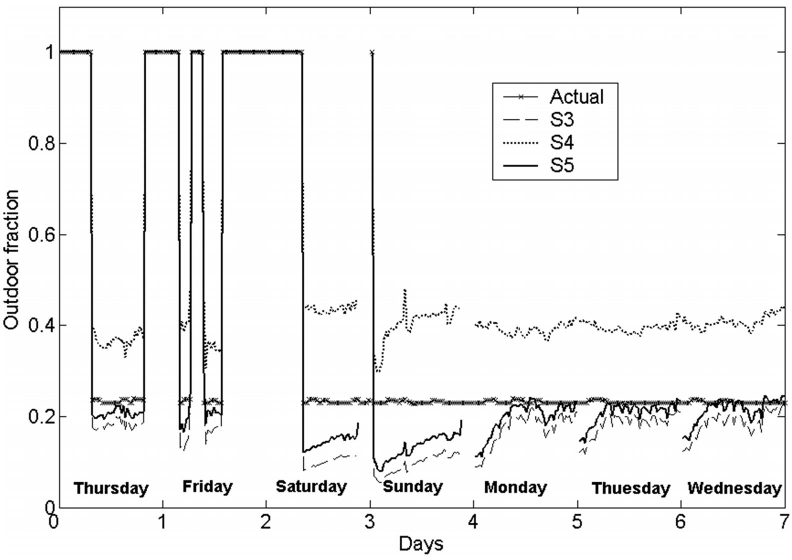

Figure 6. Outdoor airflow fractions of investigated strategies tested on AHU-6 for July 25-31, 2002.

(Thursday, Friday, and Saturday), the outdoor air enthalpy is close to the return one, and the outdoor damper resulting from the economizer strategy thus varies between the full and minimum positions. Since strategies $\boldsymbol{S_3}$ and $\boldsymbol{S_5}$ consider the actual occupancy, their outdoor fractions are lower for Saturday and Sunday than for other days. The actual outdoor fractions (close to strategy $\boldsymbol{S_1}$) are calculated by the damper model and shown in Figure 6. The fan energy use is not considered because the fan airflow rate and static pressure of all the investigated strategies have the same values. The chiller energy demands are calculated by the VAV system model and illustrated for July 27-31 in Figure 7.

The chiller energy demands required for Thursday and Friday are not shown in Figure 7 because the outdoor air enthalpy is quite close to the return one, and the chiller energy demands for all investigated strategies are about the same.

The CO_2 concentrations in critical zones, representing the indoor air quality of each strategy, are calculated using step 4 of the VAV system model. The critical zone is the zone with the highest value of $Y_{z,dsg}/R_{a,z}$. Since the design outdoor airflow fractions of all zones could be approximated to be the same as 16% ($Y_{z,dsg,i}$ = 0.16), the critical zone in this case is the zone with the lowest value of ($R_{a,z}$), which is limited for the investigated system to $R_{a,z}$ = 0.3. There are several zones having ($R_{a,z}$ = 0.3), which are considered as the critical zones. Figure 8 shows the CO_2 concentration in the critical zones of the investigated strategies for July 25-31, 2002.

Strategy $\boldsymbol{S_3}$ employs the least outdoor air and uses the least chiller energy, but poor indoor air quality in the critical zones (1100-1300) is obtained most of the time. To improve the indoor air quality in the critical zones, strategy $\boldsymbol{S_4}$ proposes an unnecessarily high outdoor air fraction. In this case, we are faced with overventilation and high chiller energy demands. Strategy $\boldsymbol{S_5}$ proposes an exact outdoor fraction to meet the ventilation requirement (1000 ppm).

For the period of June, July, August, and September, the chiller energy use when strategy $\boldsymbol{S_4}$ is used is 18% higher than that for strategy $\boldsymbol{S_3}$, but only 2.5% higher than for strategy $\boldsymbol{S_3}$, when the strategy $\boldsymbol{S_5}$ is used while meeting the ventilation requirements in the critical zones. However, the actual chiller energy use is 6% higher than $\boldsymbol{S_3}$.

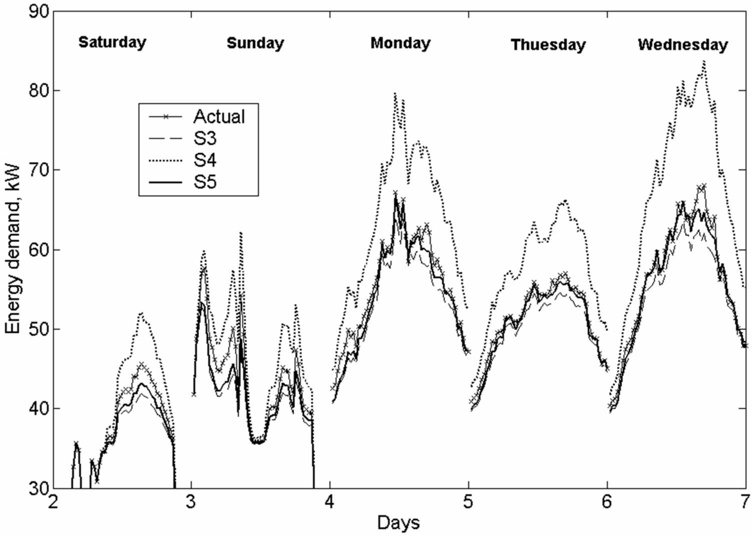

Figure 7. Chiller energy demands of investigated strategies tested on AHU-6 for July 27-31, 2002.

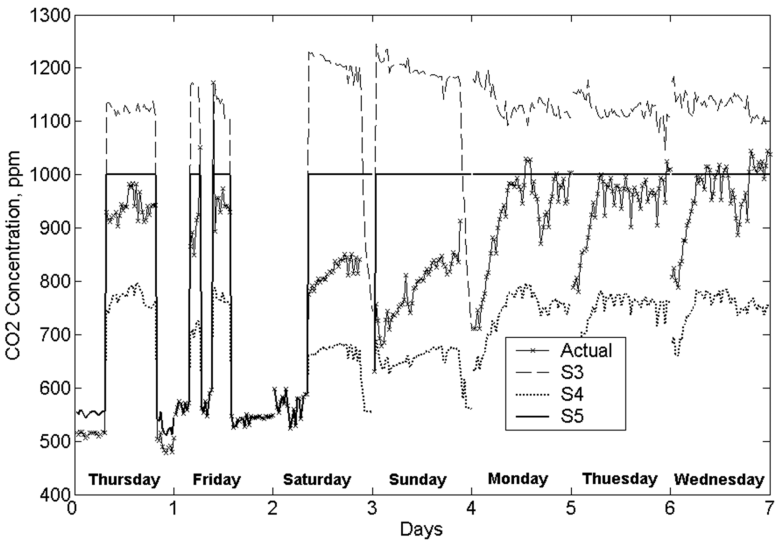

Figure 8. CO_2 concentration in the critical zones of investigated strategies tested on AHU-6 for July 25-31, 2002.

It is concluded that strategy $\boldsymbol{S_5}$ can potentially minimize chiller energy use while respecting indoor quality in all zones, including the critical ones. However, further savings could be obtained by $\boldsymbol{S_3}$ but with scarifying the ventilation in certain zones.

In Figure 5, the building airflow part-load ratios ($R_{a,b}$) are close to the ventilation values ($R_{v,b}$) for Monday, Tuesday, and Wednesday. From Equations 13 and 16, it is shown that the outdoor air fractions of strategy $\boldsymbol{S_3}$ are close to that for the actual strategy (as in $\boldsymbol{S_1}$) as well as to that for strategy $\boldsymbol{S_5}$ (see Figure 6), and their chiller energy demands are not widely different for these days (see Figure 7). Although the actual strategy performs well in the AHU-6 system, it poses a considerable problem for the AHU-4 system (perimeter zones), as will be discussed in the next section.

Results of Evaluation Based on Weather Bin Temperature (MM-3)

The objective is to compare the differences in annual energy and ventilation airflow rates supplied to different zones in the building situated in different locations. The cooling and heating building thermal loads are determined for different bin temperature data for Montreal (cold locations), Seattle (mild and less humid locations), and Dallas (moderately hot and humid locations). The concurrent wet-bulb temperatures are also used. These bin temperature data are presented by Reddy et al. (1998) and taken originally from Degelman (1984). The investigated strategies are tested on the AHU-4 for different assumed values of $R_{v,b}$ and selected weather locations. To study the effect of $R_{v,b}$ on annual energy use, the assumed actual building occupancy pattern ($R_{v,b}$) for the occupied period (from 8:00 to 22:00) is considered to be constant during the occupied day and year.

The outdoor air fraction, the CO_2 concentration in the critical zones, and the annual energy use for the investigated strategies are determined through the modeling strategy shown in Figure 4. Figures 9 and 10 show the outdoor air fractions and CO_2 concentrations in the critical zones for the Montreal location and when the $R_{v,b}$ is 0.6, respectively.

Below a 0°C bin temperature, the values for the outdoor air fractions and CO_2 concentrations in critical zones do not vary due to nonvariations of $R_{v,b}$, $R_{a,b}$, and $R_{a,z}$. The fan and zone airflow ratios ($R_{a,b}$, $R_{a,z}$) freeze at their minimum values (40% and 30%, respectively). Around 15°C, the economizer is used, introducing the outdoor airflow rate required for free cooling.

In order to evaluate the indoor air quality (IAQ) in the critical zones, two indexes are used: (1) the number of hours (NH*)* when the critical zones are above 1000 ppm and (2) the average value (AV*)* of CO_2 concentration in critical zones for the period when the CO_2 concentration is above 1000 ppm. As mentioned above, ASHRAE Standard 62-2001 requires a maximum CO_2 concentration of 1000 ppm in zones. Thus, the indexes measure how long (NH) and how much (AV) the critical zones could be underventilated. The NH is calculated as the percentage of the total investigated time. The total investigated time is the number of annual hours for the occupied period. The AV is the average CO_2 concentration in critical zones for the NH period. For instance, when strategy $\boldsymbol{S_1}$ is used in the Montreal location and $R_{v,b}$ is 0.8, the average CO_2 concentration in critical zones, as presented in Table 2, is 1450 ppm during 75.4% of the total investigated time.

Taking the annual energy use of strategy $\boldsymbol{S_3}$ as a reference, the ratios of annual energy use for the investigated strategies at different occupancies ($R_{v,b}$) and different locations are calculated and illustrated in Figure 11.

As shown in Figure 9, when the $R_{v,b}$ equals 0.6 for Montreal, the $\boldsymbol{S_1}$ introduces very low outdoor air in winter days, when the building airflow rate is low and frequently at a minimum level. Certain zones, including critical ones, are underventilated by an average CO_2 concentration value of 1334.3 ppm during 61% of the total investigated time (see Table 2). There could be some improvement of the IAQ in the critical zone when this strategy works in hot weather (Dallas) due to high thermal loads (and $R_{a,b}$) and, consequently, outdoor air. In Dallas, the critical

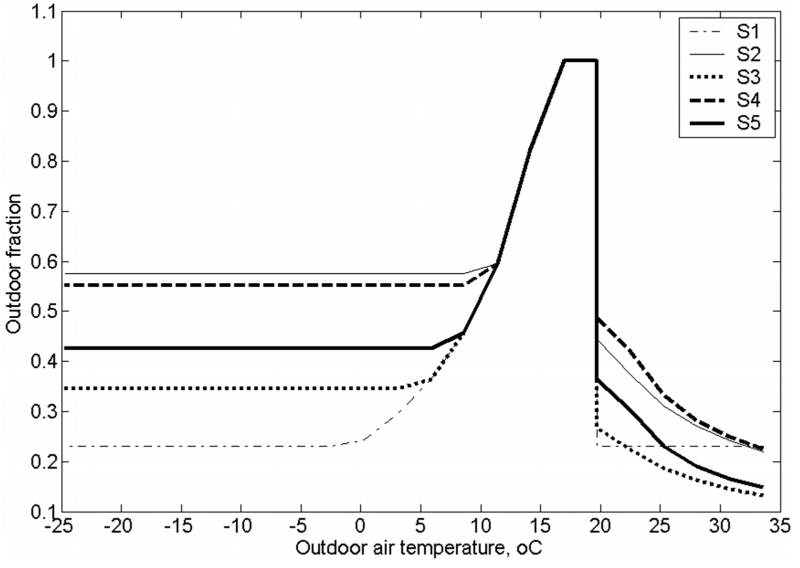

Figure 9. Outdoor air fraction of investigated strategies for each Montreal bin temperature and when the $R_{v,b}$ during the occupied period is 0.6.

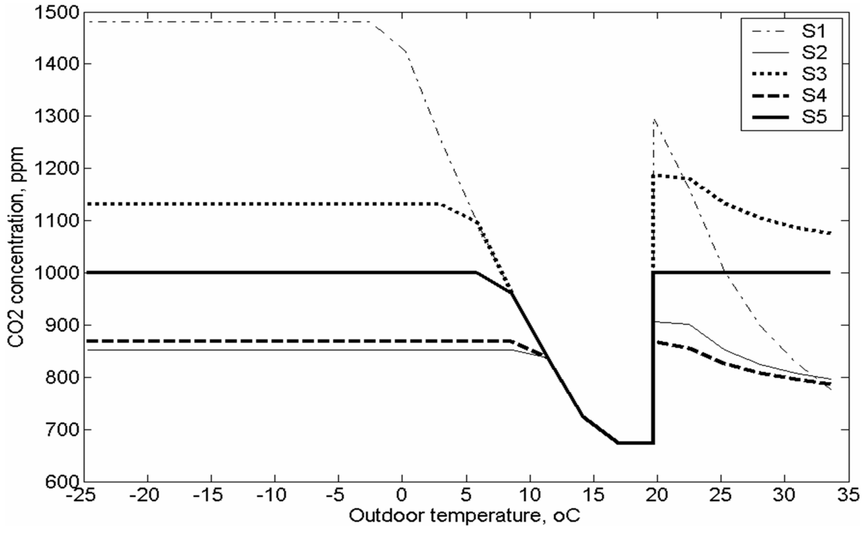

Figure 10. CO_2 concentration in the critical zones of investigated strategies for each Montreal bin temperature when the $R_{v,b}$ during the occupied period is 0.6.

Table 2. Results of IAQ Indexes for the Investigated Ventilation Control Strategies Tested on AHU-4

			S_1	S_2	S_3	S_4 and S_5
Montreal	$R_{v,b} = 1$	NH[a]	78.4	28.4	28.4	0
		AV[b]	1604.3	1060.3	1060.3	0
	$R_{v,b} = 0.8$	NH	75.4	0	80	0
		AV	1450	0	1070.6	0
	$R_{v,b} = 0.6$	NH	61	0	72.5	0
		AV	1334.3	0	1139.3	0
Seattle	$R_{v,b} = 1$	NH	70.6	11.6	11.6	0
		AV	1392.2	1064.6	1064.6	0
	$R_{v,b} = 0.8$	NH	70	15	76	0
		AV	1306	1016	1099	0
	$R_{v,b} = 0.6$	NH	53.8	0	74.7	0
		AV	1184.6	0	1104.5	0
Dallas	$R_{v,b} = 1$	NH	76.2	57.5	57.5	0
		AV	1400.6	1068.2	1068.2	0
	$R_{v,b} = 0.8$	NH	54	0	54.7	0
		AV	1340	0	1064.4	0
	$R_{v,b} = 0.6$	NH	64.6	0	76	0
		AV	1190.1	0	1146.5	0

a. Number of hours, %: percentage hour number of investigated time when the CO_2 concentration is above 1000 ppm. Investigated time is the number of annual hours for occupied period.
b. Average value, ppm: average value of CO_2 concentration for the period when the CO_2 concentration is above 1000 ppm.

zones are underventilated for 64.6% of the total investigated time, but the average value of CO_2 concentration in critical zones is 1190.1 ppm. Thus, the annual energy uses of strategy $\boldsymbol{S_1}$ are not at their lowest when $R_{v,b}$ is 0.6 at the Dallas location. For this location, the annual energy uses are 96.7%, 98.7%, and 100.6% of the reference strategy, $\boldsymbol{S_3}$, for $R_{v,b}$ values of 1, 0.8, 0.6, respectively. However, for other investigated locations, strategy $\boldsymbol{S_1}$ causes very poor IAQ but consumes the least annual energy uses, which are 92.9%, 95.7%, and 98.2% of reference strategy $\boldsymbol{S_3}$ for Montreal and 94%, 97.2%, and 99.3% for Seattle at $R_{v,b}$ values of 1, 0.8, and 0.6, respectively.

At full occupancy ($R_{v,b} = 1$), strategy $\boldsymbol{S_2}$ performs exactly the same as $\boldsymbol{S_3}$ (see Equations 13 and 14) and strategy $\boldsymbol{S_4}$ performs the same as $\boldsymbol{S_5}$ (see Equations 20 and 22). The performance of strategies $\boldsymbol{S_2}$ and $\boldsymbol{S_3}$ differs slightly from that for strategies $\boldsymbol{S_4}$ and $\boldsymbol{S_5}$ due to thermal load diversity ($R_{a,b}$ is not the same value of $R_{a,z}$). When strategies $\boldsymbol{S_2}$ and $\boldsymbol{S_3}$ are used, the *NH*s are 28.4, 11.6, and 57.5 and the *AV*s are 1060.3, 1064.6, and 1068.2 ppm for the Montreal, Seattle, and Dallas locations, respectively. If the thermal load distributions between zones vary significantly (high difference between $R_{a,b}$ and $R_{a,z}$), then when strategies $\boldsymbol{S_2}$ and $\boldsymbol{S_3}$ are used, the critical zones will be very underventilated for long periods.

With respect to the IAQ criteria, strategies $\boldsymbol{S_4}$ and $\boldsymbol{S_5}$ are the best. There are no zones that are underventilated for all locations and values of $R_{v,b}$. Strategy $\boldsymbol{S_5}$ performs much better than $\boldsymbol{S_4}$

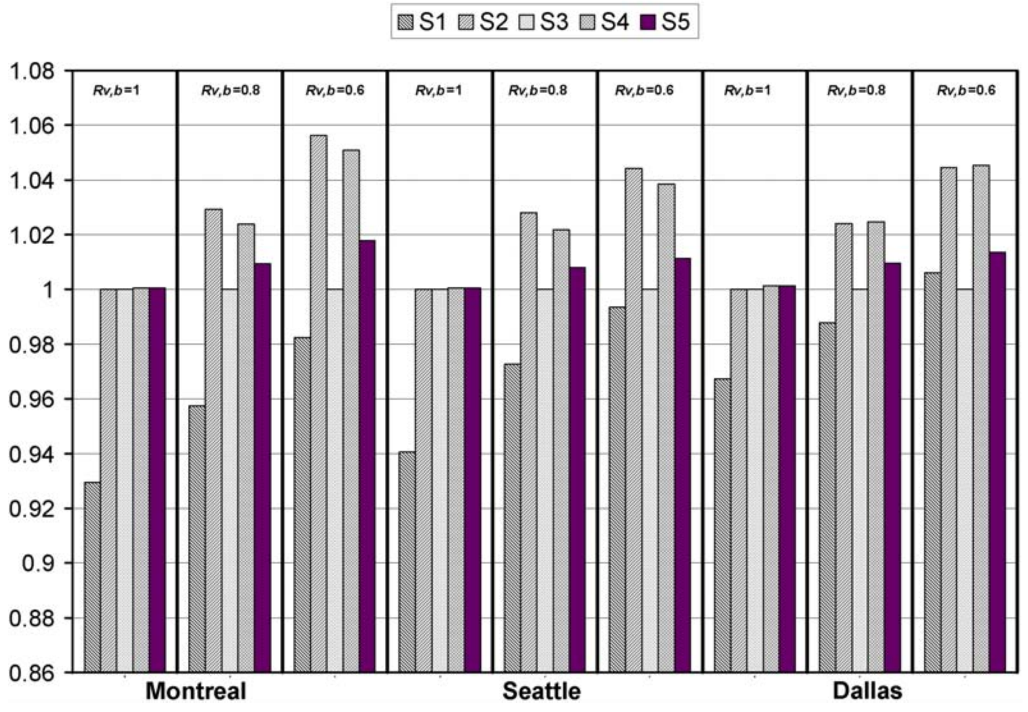

Figure 11. Ratio of annual energy use of investigated strategies to reference strategy S_3.

with respect to the two criteria: IAQ and energy use. Annual energy savings of 3.1%, 2.6%, and 3%, respectively, are obtained using strategy $\boldsymbol{S_5}$ instead of $\boldsymbol{S_4}$ for Montreal, Dallas, and Seattle when the actual occupancy ($R_{v,b}$) is 0.6. Annual energy savings is decreased when the actual occupancy is increased, and no saving is obtained when the actual occupancy is still the same as the design value ($R_{v,b}$ = 1).

Comparing $\boldsymbol{S_3}$, $\boldsymbol{S_4}$, and $\boldsymbol{S_5}$ shows that $\boldsymbol{S_3}$ introduces the least outdoor air and consumes the least annual energy use for all cases, but that leads to a very poor IAQ in critical zones. In this case, when $R_{v,b}$ is 0.6, the *NH*s are 72.5%, 74.7%, and 76% and the *AV*s are 1139.3, 1104.5, and 1146.5 ppm for the Montreal, Seattle, and Dallas locations, respectively. However, when $R_{v,b}$ is 0.8, the *NH*s are 80%, 76%, and 54.7% and the *AV*s are 1070.6, 1099, and 1064.4 ppm. To improve the IAQ, $\boldsymbol{S_4}$ and $\boldsymbol{S_5}$ could be used. The annual energy uses of $\boldsymbol{S_4}$, with $R_{v,b}$ of 0.6, are 5%, 3.8%, and 4.5%, respectively, higher than for strategy $\boldsymbol{S_3}$ for Montreal, Seattle, and Dallas, whereas the annual energy uses of $\boldsymbol{S_5}$ are only 1.7%, 1.1%, and 1.3% of that for strategy $\boldsymbol{S_3}$. That confirms that strategy $\boldsymbol{S_5}$ consumes lower additional energy than strategy $\boldsymbol{S_4}$ in improving the poor IAQ of strategy $\boldsymbol{S_3}$. $\boldsymbol{S_5}$ performs better than $\boldsymbol{S_3}$ respecting the IAQ criteria and better than $\boldsymbol{S_4}$ respecting two criteria: energy use and IAQ.

CONCLUSION

A new supply CO_2-based demand-controlled ventilation strategy is proposed in order to minimize energy use while ensuring proper indoor air quality in all zones, including critical zone(s). This proposed strategy maintains a supply CO_2 concentration setpoint that is low enough to dilute CO_2 generated by full occupancy in the critical zone. The supply CO_2 concentration setpoint could be determined online using the monitored zone airflow rates, which has recently become possible with the use of direct digital control terminal boxes. The strategy is a compromise between the multiple-space equation and the CO_2-based DCV, taking into account (1) the building's actual occupancy and (2) the critical zone ventilation requirement. Thus, the S-CO_2-DCV ($\boldsymbol{S_5}$) takes into account the online overventilation occurring in buildings, while considering the critical zone ventilation requirement. The CO_2-DCV ($\boldsymbol{S_3}$) takes into account the same, but without considering the critical zone ventilation requirement, and the MSE ($\boldsymbol{S_4}$) takes into account only design overventilation in the different zones. The simulation results applied to two existing AHU systems showed that the S-CO_2-DCV ($\boldsymbol{S_5}$) needs less outdoor air and consumes less energy than the MSE by taking greater advantage of online overventilated spaces. Although the CO_2-DCV strategy uses the lowest outdoor air and energy, there is poor indoor air quality in some zones. The simulation results also show that energy use could be saved as compared to the actual ventilation control strategy by implementing the proposed S-CO_2-DCV strategy while ensuring a good indoor air quality in all zones, including the critical one.

NOMENCLATURE

Acronyms

CO_2-DCV = CO_2-based demand-controlled ventilation
ÉTS = École de technologie supérieure
MSE = multiple space equation
S-CO_2-DCV = supply CO_2-based demand-controlled ventilation

Symbols

$B_{\Delta CO_2}$ = CO_2 concentration difference between supply and return air, ppm
N = number of occupants
O_{CO_2} = outdoor CO_2 concentration, ppm
$\dot{Q}$ = airflow rate, L/s
q_s = ratio of sensible thermal load, %
R_{CO_2} = CO_2 concentration in return air, ppm
R = part-load ratio
S_{CO_2} = CO_2 concentration in supply air, ppm
$S_{CO_2,set}$ = CO_2 concentration setpoint in supply air, ppm
$\boldsymbol{S_1}$ = fixed minimum outdoor air percentage
$\boldsymbol{S_2}$ = fixed minimum outdoor air rate
$\boldsymbol{S_3}$ = CO_2–based demand-controlled ventilation
$\boldsymbol{S_4}$ = multiple space equation
$\boldsymbol{S_5}$ = supply CO_2-based demand-controlled ventilation
t_o = outdoor air temperature, °C
VL = ventilation load, (L/s).(ppm)
X = uncorrected fraction of outdoor ventilation air required at full occupancy in the supply system
Y = online fraction of outdoor airflow
Z = ratio of the required outdoor air at full occupancy to the primary air in the critical zone.
$Z_{\Delta CO_2,i}$ = CO_2 concentration difference between supply and return air in zone i, ppm
$Z_{CO_2,th}$ = threshold point of CO_2 concentration in zone, ppm

Subscripts

a = air
b = building
i = ith zone
dsg = design condition
o = outdoor air
v = ventilation
z = zone

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