Article

Development of a Terminal Control System with Variable Minimum Airflow Rate

Young-Hum Cho

School of Architecture, Kumoh National Institute of Technology, Gumi, Gyeongbuk 730-701, Korea;
E-Mail: yhcho@kumoh.ac.kr; Tel.: +82-54-478-7589; Fax: +82-54-478-7609

Abstract: A constant minimum airflow rate is used in conventional Single Duct Variable Air Volume Terminal Box control sequences. This control sequence can cause occupant discomfort or use excessive energy under partial load conditions. If the minimum airflow rate is higher than required; terminal boxes will have significantly more simultaneous heating and cooling; and AHUs will consume more fan power. Buildings will have indoor air quality problems if the minimum airflow rate is less than required. Many engineers and researchers have investigated advanced variable air volume terminal box control algorithms without a system retrofit for thermal comfort; indoor air quality and energy savings. In this study a developed control system with variable minimum airflow rate for Single Duct Variable Air Volume Terminal Boxes was applied and validated using an actual building and evaluated for comfort; indoor air quality and energy consumption. The energy consumption and thermal performance of terminal boxes using the conventional and proposed control algorithms were compared.

Keywords: terminal control system; variable minimum airflow rate; energy consumption; thermal performance; control algorithms

Nomenclature:

A_Z	zone floor area, ft^2 (m^2)
C_p	specific heat of air, Btu/lbm °F (kJ/kg—K)
$E_{d,f}$	design fan power, kW
E_f	fan power, kW
E_{th}	reheating energy consumption, Btu/hr (kW)
E_z	zone air distribution effectiveness

$\dot{m}_a$ mass flow rate, lbm/hr (kg/s)

min OA   minimum outside air requirement (%)

Q_c cooling design load of the room, Btu/hr (kW)

Q_h heating design load of the room, Btu/hr (kW)

R_a outdoor airflow rate required per unit area, (L/s)

R_p outdoor airflow rate required per person, (L/s)

P_z zone population, person

$T_{d,s}$ discharge air temperature, ℉ (℃)

T_{OA} outside air temperature, ℉ (℃)

T_s supply air temperature, ℉ (℃)

$T_{s,clg}$ supply dry bulb temperature for cooling, ℉ (℃)

$T_{s,htg}$ high limit of the supply dry bulb temperature for heating, ℉ (℃)

T_R return air temperature, ℉ (℃)

$\dot{V}_{bz}$ air volumetric flow rate for fresh air requirement, ft^3/min (L/s)

$\dot{V}_{d,max}$ maximum air volumetric flow rate supplied to the room, ft^3/min (L/s)

$\dot{V}_f$ air volumetric flow rate for fresh air requirement at zone level, ft^3/min (L/s)

$\dot{V}_{min,h}$ minimum air volumetric flow rate supplied to the room, ft^3/min (L/s)

$\dot{V}_{min,v}$ minimum air volumetric flow rate supplied to the room for ventilation, ft^3/min (L/s)

$\dot{V}_{oz}$ design zone outdoor airflow, ft^3/min (L/s)

$\alpha_{min,h}$ minimum airflow ratio for heating load, %

$\alpha_{min,v}$ minimum airflow ratio for fresh air requirement, %

$\alpha_{min,sf}$ variable minimum airflow ratio, %

α_{oa} AHU outside air intake ratio, %

α_{sf} supply fan airflow ratio, %

$\eta_{f,d}$ fan efficiency

ρ standard air density, lbm/ft^3 (kg/m^3)

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1. Introduction

Single duct variable air volume (VAV) air-handling units (AHUs) are the most popular system in the USA. Terminal boxes are a critical component in VAV systems. The minimum airflow rate of terminal boxes is a key factor for user comfort, indoor air quality (IAQ) and energy costs.

A constant minimum airflow rate is used in conventional control sequences. This control sequence can cause occupant discomfort or use excessive energy. If the minimum airflow conditions are not optimal in terms of energy efficiency, terminal boxes will have significantly more simultaneous heating and cooling cycles and AHUs will consume more fan power. Buildings will have IAQ problems if the minimum airflow rate is less than required.

Common practice uses simple and easy methods to determine the minimum airflow rate. Consequently, a higher than required value is often selected to avoid air circulation problem, temperature stratification and lack of fresh air while sacrificing energy efficiency.

Fan-powered terminal boxes were developed to solve air circulation issues. Each terminal box is equipped with a small fan that effectively maintains high air circulation rates under low load conditions. However, the disadvantages associated with the use of fan-powered terminal boxes outweigh the benefits. Not only are these terminal boxes more expensive, but they expend a large amount of fan energy and cannot meet the fresh air requirements. Installing fan-powered terminal boxes in existing buildings is very costly and labor intensive. Moreover, they are not ideal in spaces with conditions that need special consideration, such as offices with a noise limit requirement [1,2].

To achieve the same functions as fan-powered terminal boxes without the fan, the self-powered VAV adjustable thermal diffuser was developed. It can maintain high velocity even under low load conditions by varying the cross-sectional area, and it can maintain excellent air mixing. It is good for small and medium size systems for occupants' comfort, but it may cause pressure issues and large systems generate noise.

Many engineers and researchers have investigated advanced VAV terminal box controllers without a system change. Some current controllers can provide occupants' thermal comfort, and IAQ and save energy; however, these controllers' performance should be verified sufficiently by using scientific evidence and confirmed studies, and these minimum airflow rate setpoints should be identified [3,4].

The minimum airflow rate is a variable since both the zone load and fresh air fraction of the supply air varies with time and outside air conditions. It is important to set up the minimum airflow rate to ensure IAQ and thermal comfort and to minimize energy consumption. In principle, the minimum airflow rate should be determined based on the zone load for the temperature control, the fresh air requirement for the IAQ, and the air circulation for uniform air distribution. Optimal control algorithms for single duct VAV terminal boxes were developed to achieve this. The terminal box can automatically identify the minimum heating airflow rate under the actual working conditions with the improved control algorithms. The improved control algorithm ensures excellent room air mixing and temperature control, and it avoids the excessive primary airflow by automatically adjusting the heating airflow. It can improve system reliability and reduce costs. It can stably maintain the room air temperature. The vertical difference of room air temperature is kept lower than the comfort value. The measurements of CO_2 levels show there is no indoor air quality problem when the minimum airflow rate setpoint is reduced; however, it could not control and meet fresh air requirement level by terminal control system and also needs to develop new terminal control system with variable minimum airflow rate [5].

The ASHRAE 2010 *Handbook of Fundamentals* [1] states that discharge temperatures in excess of 90 ℉ (32 ℃) may occur air temperature stratification and ventilation short circuiting. When heating from overhead, the discharge air temperature cannot be more than 15 ℉ (8 ℃) above the room temperature, and the 150 fpm terminal velocity from the diffuser must extend to within 4.5 ft of the floor. This rule avoids ventilation short-circuiting to the ceiling return [6]. ASHRAE Standard 62-2010 states that in order to avoid excessive temperature stratification the room temperature difference should not exceed 15 ℉ (8 ℃). The supply air temperature can be no greater than about 90 ℉ (32 ℃) assuming a 75 ℉(24 ℃) room temperature [7]. Some zones may require higher supply air temperatures to satisfy the peak heating load requirements and still be able to maintain a low minimum airflow rate set point.

The dual maximum control sequence was suggested by Stein [8]. In the dual maximum control sequence, as the load goes from full cooling to full heating, the airflow rate setpoint is reset from the maximum to the minimum before the supply air temperature setpoint is reset from the minimum [e.g., 55 °F (13 °C)] to the maximum [e.g., 90 °F (32 °C)]. If more heat is needed, the airflow rate setpoint is reset from the minimum to the heating maximum. Because it is decoupled from the heating needs, the minimum airflow rate setpoint can be as low as the controllable minimum of the VAV box (as long as ventilation requirements are met). When a dual maximum control sequence is used, the minimum airflow rate setpoint typically is 10 to 20% of the design airflow rate. The simulation models of typical office buildings in Sacramento, CA (California climate zone 12) have shown that switching from a single maximum control approach with a 40% minimum airflow rate setpoint to a dual maximum control approach with a 20% minimum airflow rate setpoint can save 30 cents per square foot per year.

The dual maximum control sequence is similar to the previous dual minimum airflow rate control sequence. The difference is that the supply air temperature is maintained by installing the supply air temperature sensor at the terminal box and having the maximum supply air temperature set at 90 °F (32 °C), which is the ASHRAE Standard 62-2010 recommended value; this is just a different way to control a reheating coil, but supply air temperature sensors are not installed in most the current terminal boxes, which may ignore the buoyancy effect under the low minimum airflow rate condition. The maximum leaving-air temperature (LAT) is 90 °F (32 °C). A LAT above 90 °F (32 °C) will result in stratification and/or short-circuiting. To maintain a LAT of 90 °F (32 °C), the minimum (heating) airflow rate can be adjusted upward. This will have high reheating energy consumption, and AHUs will consume more fan power [9].

The objective of this study is to develop s terminal control system with the variable minimum airflow rate of Single Duct VAV Terminal Boxes and apply and validate it with an actual building and evaluate it for comfort, IAQ and energy consumption performance. The energy consumption and thermal performance of terminal boxes were compared using conventional and the proposed control sequences by dynamic energy simulation and field experiments.

2. Minimum Airflow Rate of VAV Terminal Control System

2.1. Single-Duct VAV Terminal Boxes

According to the ASHRAE *Application Handbook, Supervisory Control Strategies and Optimization* [10], "with a VAV system, a feedback controller regulates the airflow rate to each zone in order to maintain zone temperature setpoints. Zone airflow rates are regulated using dampers located in VAV boxes in each zone. VAV systems also incorporate feedback control of the primary airflow rate through modulation fans. Typically, the inputs to a fan outlet damper, inlet vanes, blade pitch, or variable speed motor are adjusted in order to maintain a duct static pressure setpoint within the supply duct." Figure 1 presents the Schematic diagram of a Single-Duct VAV Terminal system [5].

Figure 1. Schematic diagram of a single-duct VAV terminal system.
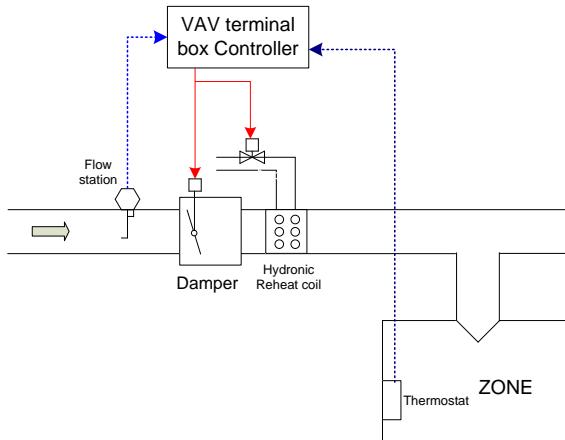
* Dotted lines represent input signal and solid lines represent output signal.

2.2. Determining the Minimum Airflow Rate

Based on the standard control sequence of the Single Duct VAV terminal box, the airflow rate requires a minimum limit, which should determine the constant minimum airflow rate setpoint. There are two categories determining the minimum airflow rate setpoint. One is the principle method based on the standard, and another category is the common practice method.

2.2.1. Principle Method

For VAV reheat terminal boxes that serve exterior zones, the minimum airflow rate setpoint is typically selected by the maximum value of the following:

- The airflow rate required by the room design heating load, or
- The minimum required for ventilation.

In 1989, the ventilation rates established by ASHRAE Standard 62-1989 [11] increased substantially over those previously required by the 1981 version of the standard. In 2010, Standard 62.1-2010 [7] prescribed new minimum breathing zone ventilation rates and a new calculation procedure to find the minimum intake airflow rate needed for different ventilation systems.

Before 2001, ASHRAE determines the required ventilation rates using Equation (1) based on either the number of occupants in the zone (cfm/person) or the floor area of the zone (cfm/ft^2):

$$\dot{V}_{bz} = R_p \cdot P_z \text{ or } \dot{V}_{bz} = R_a \cdot A_Z \tag{1}$$

ASHRAE Standard 62.1-2010 [7] prescribed minimum fresh air breathing zone ventilation rates and a new calculation procedure to find the minimum intake airflow rate needed for different ventilation systems. Equation (2) can be used to determine the design outdoor airflow required in the breathing zone of the occupied space:

$$\dot{V}_{bz} = R_p \cdot P_z + R_a \cdot A_Z \tag{2}$$

Equation (3) can be used to determine the design zone outdoor airflow:

$$\dot{V}_{oz} = \dot{V}_{bz}/E_z \tag{3}$$

Uncertainty occurs as a result of a change of plan layouts, variable zone load conditions, occupancy rates, and the fraction of outside air in the primary air. Therefore, it is hard to determine an accurate constant minimum airflow rate set point with the principle method.

2.2.2. Common Practice Methods

According to ASHRAE Standard 90.1-2010 [12] "zone thermostatic controls shall be capable of operating in sequence the supply of heating and cooling energy to the zone. Such controls shall prevent reheating; recooling; mixing or simultaneously supplying air that has been previously mechanically heated and air that has been previously cooled, either by mechanical cooling or by economizer; and other simultaneous operation of heating and cooling system to the same zone."

Based on the ASHRAE Standard 90.1, the rate of minimum airflow rate can be determined according to the largest of the following: (1) 30% of the maximum rate of airflow rate; (2) the volume of outdoor air required to meet the ventilation requirements.

While common practice methods are still employed by HVAC designers, they may not always guarantee the best results. In fact, some of the values are just plain erroneous. In spite of a lack of any confirmed studies and empirical evidence, for example, many HVAC designers feel that selecting 30% of the peak supply volume adequately maintains thermal comfort, provides good room air mixing, and satisfies the fresh air requirement [13].

In many buildings, engineers are responsible for a large number of terminal boxes with differing load conditions. Time constraints therefore lead many HVAC designers to employ common practice methods. Unfortunately, this makes it difficult to calculate the minimum airflow rate set point. The most economical solution for setting the minimum airflow rate is to choose the lowest value possible. However, this often results in very poor air distribution in the heating mode, ventilation short-circuiting, and temperature stratification.

ASHRAE Standard 55-2010 gives the vertical air temperature difference limitations. According ASHRAE standard 55, the occupied zone is defined as: the region normally occupied by people within a space, generally considered to be between the floor and 1.8 m (6 ft) above the floor and more than 1.0 m (3.3 ft) from outside walls/windows or fixed heating, ventilating, or air-conditioning equipment and 0.3 m (1 ft) from internal walls. The air temperature of an enclosed space generally increases from floor to ceiling. If this increment is sufficiently large, occupants may feel localized head and foot thermal discomfort [14]. In order to prevent such discomfort, "the vertical air temperature difference within the occupied zone measured at 4 inches (0.1 m) and 67 inches (1.7 m), should not exceed 5.4 ℉ (3 ℃) [15]."

3. Numerical Analysis

3.1. Control Options for Minimum Airflow Rate

The minimum airflow rate through terminal boxes is a critical parameter affecting indoor air quality, air circulation and energy consumption. To improve the conventional control sequence, the

optimal minimum airflow rate ratio should be determined. The variable minimum airflow rate ratio can be determined by the room conditions. It should have a range of high and low minimum airflow rate ratios during operation.

Option 1. Existing Minimum Airflow Rate Ratio

The existing minimum airflow rate ratio can be determined by field experiments [16].

Option 2. Constant Minimum Airflow Rate Ratio for Heating Load

Generally, the design supply airflow should be chosen the maximum value among at least the following:

1. Design cooling load;
2. Ventilation according to ACH requirement;
3. Exhaust airflow.

In this paper, we assumed that the cooling design airflow satisfies the ventilation and building pressurization requirement. The maximum airflow for this office can be calculated by the room design cooling load by using Equation (4):

$$\dot{V}_{d,max} = \frac{Q_c}{\rho \cdot c_p \cdot (T_r - T_{s,clg})} \tag{4}$$

The minimum airflow for heating load is the airflow required by the room design heating load by using Equation (5). The minimum airflow ratio to satisfy the building maximum heating load can be calculated by the following Equation (6):

$$\dot{V}_{min,h} = \frac{Q_h}{\rho \cdot c_p \cdot (T_{s,htg} - T_r)} \tag{5}$$

$$\alpha_{min,h} = \frac{\dot{V}_{min,h}}{\dot{V}_{d,max}} \tag{6}$$

Option 3. Constant Minimum Airflow Ratio for Ventilation

The design zone outdoor airflow can be calculated by using Equation (7), which combines Equations (2,3), is for zone levels. AHU Outdoor fresh air intake ratio of economizer operation condition which meets the minimum outside air requirement can be calculated by using Equation (8) at the AHU level. According to ASHRAE standard 62, from zone to AHU, the following steps need to be conducted:

1. Calculate zone primary outside air fraction;
2. Determine system ventilation efficiency based on maximum zone primary outside air fraction;
3. Calculate occupant diversity;
4. Calculate uncorrected outside air intake;
5. Calculate design outside air intake at AHU.

In this paper, we assumed that the outside air intake under the economizer mode satisfies the minimum outside air intake. The minimum airflow rate that satisfies the outside air ventilation requirements can be calculated by the following Equation (9) and its ratio can be calculated by using Equation (10):

$$\dot{V}_f = (R_P \cdot P_z + R_a \cdot A_z) / E_z \tag{7}$$

$$\alpha_{oa} = Min\left(Max\left(\frac{T_R - T_s}{T_R - T_{OA}}, \ min \ OA\right), 1\right) \ when \ T_R > T_{OA} \tag{8}$$

$$\dot{V}_{min,v} = \frac{\dot{V}_f}{\alpha_{oa}} \tag{9}$$

$$\alpha_{min,v} = \frac{\dot{V}_{min,v}}{\dot{V}_{d,max}} \tag{10}$$

Option 4. Variable Minimum Airflow Ratio

The variable minimum airflow ratio can be calculated between the minimum airflow ratio for heating load of Equation (6) and the minimum airflow ratio for ventilation of Equation (10). When the room air temperature is below the set point, airflow will decrease as the damper closes towards the minimum airflow set point for ventilation requirement and discharge air temperature is variable to meet the room temperature setpoint. If the room air temperature continues to decrease the minimum airflow will be reset from the minimum airflow ratio for ventilation requirement of Equation (10) up to that of heating load requirement of Equation (6) to maintain the room air temperature and discharge air temperature is maintained at its constant setpoint. Therefore, the minimum airflow ratio should not be constant, but rather varied during the heating period.

3.2. Determining Parameters for Control Options

The minimum airflow rate ratio is determined for exterior zone boxes by applying the previous method. The building load selected as the same office as in the field experiments [16] was calculated by the Trane computer simulation program. The simulation results show that the cooling load (peak) was 9,042 Btu/hr (2,650 W) and heating load (peak) was 3,146 Btu/hr (920 W).

The maximum airflow rate in the exterior zone calculated by Equation (4) is 411 CFM (194 L/s) when the supply fan airflow rate ratio at AHU is assumed 1.0. The minimum airflow rate for building load calculated by Equation (5) is 143 CFM (67 L/s) and its ratio calculated by Equation (6) is 35%.

To meet the ventilation air requirements [7], we will give about 16 CFM (8 L/s) for fresh air. During winter, when the outside air temperature is −8 °F (−22 °C), the supply air temperature set point is 55 °F (13 °C), and the return air temperature is 75 °F (24 °C), the AHU outside air intake ratio calculated by Equation (8) is 24%. When the AHU outside air intake ratio is 24%, then the minimum airflow rate to satisfy the ventilation requirement calculated by Equation (9) is 62.5 CFM (30 L/s). Therefore, the constant minimum airflow rate ratio for ventilation for calculated by Equation (10) is 15%. Table 1 shows the determined parameters for the various control options.

Table 1. Parameters for the control options.

Option No.	Control option	Minimum airflow rate ratio
Option 1	Existing minimum airflow rate ratio	55%
Option 2	Constant minimum airflow rate ratio for heating load	35%
Option 3	Constant minimum airflow rate ratio for ventilation	15%
Option 4	Variable minimum airflow rate ratio	Variable ratio

3.3. Simulation Condition

To evaluate the control sequence, a simulation model was developed with TRNSYS. This is a modular simulation program which includes a model library of building and energy systems and Fortran modules developed for the control. The model for the simulation is selected as the same office as in the field experiments [16]. The outdoor condition is assumed to follow the standard weather data of Omaha (NE, USA), by TMY2 (Typical meteorological years 2). Heat gains due to people are 70 W/person (sensible heat) and 45 W/person (latent heat), based on the ASHRAE *Handbook Fundamentals*. Heat gain from lighting is assumed to have a maximum 14 W/m^2 by field survey and heat gain from equipment has a maximum 8W/m^2 from ASHRAE *Handbook Fundamentals* [17]. Occupied Hours in weekdays is 8 to 18 hr. Room air temperature set point is 75 ℉ (24 ℃), and supply air temp set point is 55 ℉ (13 ℃). In the minimum airflow rate of conventional control, the results of the field experiments are used as an input condition. Table 2 shows the input data for the simulation.

Table 2. Input data for the simulation.

Item		Input data
Simulation tool		TRNSYS
Simulation model	Location	Omaha, Nebraska
	Model room	North and South office rooms [16]
	Window	(Floor area: 20 m^2, Volume: 44 m^3) Double glazing, 0.6 cm air space (Area: 14 m^2)
Weather condition		TMY2 (Typical meteorological years 2)
People load	Sensible	70 W/person
	Latent	45 W/person
	Number	1 Persons
Lighting load		14 W/m^2
Equipment load		8 W/m^2
PMV variables	Metabolism	1.2 met
	External work	0
	Clothing	1.5 clo
	Air velocity	0.1 m/s
Operation condition	Control	Continuous heating and cooling control
	Schedule	Occupied hour (8: 00 a.m.–6:00 p.m.)
	Heating Set point	75 ℉ (24 ℃)

3.4. Simulation Method

To determine the heating period, the simulation was conducted under natural conditions (no heating and cooling) during the general heating season from November to February. The heating period was determined as November 28 through February 8, when the room air temperature is lower than 71 ℉ (21 ℃). Simulation 1 is conducted to figure out problems with the conventional minimum airflow rate ratio. Simulations 2 and 3 are for a constant minimum airflow rate ratio for the heating load and fresh air requirement. Simulation 4 evaluates the variable minimum airflow rate ratio.

3.5. Simulation Results and Discussion

3.5.1. Evaluation of Control Performance

Simulation 1. Existing Minimum Airflow Rate Ratio

In order to analyze the problems of current minimum airflow rate ratio, Simulation 1 is performed with 55% of the minimum airflow rate ratio. The results of Simulation 1 show that the room air temperature ranged from 73.6 ℉ (23.1 ℃) to 75.4 ℉ (24.1 ℃), and the average room air temperature was 74.4 ℉ (23.5 ℃), in the north zone, as shown in Figures 2, 3 and Table 3. The room air temperature could not be maintained at the set point [75 ℉ (24 ℃)] within control deviation. This is due to the fact that the high minimum airflow rate ratio causes significant simultaneous heating and cooling. The range of the airflow rate ratio was 55.4% ~ 64.7% and the average airflow rate ratio was 56.1% of the design as shown in Figures 4, 5 and Table 3. The room air temperature and airflow rate ratio of the south zone were higher than that of the north zone, because the south zone has solar radiation and internal heat gain during occupied hours. Also, the reheating valve operates simultaneously to meet the heating load. Therefore, the terminal boxes need to adjust the minimum airflow rate ratio according to building operation conditions.

Figure 2. Simulation results of north zone air temperature.
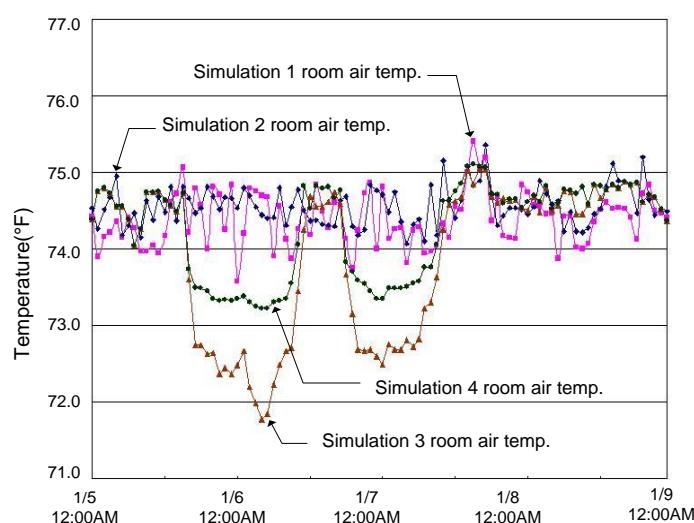
Figure 3. Simulation results of south zone air temperature.
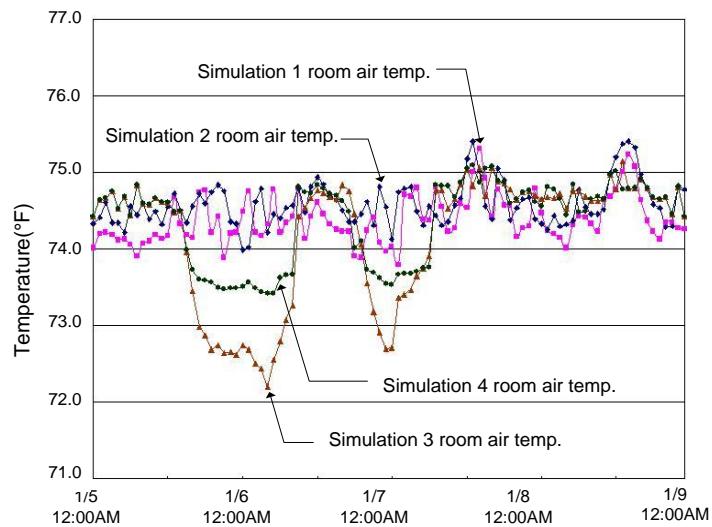
Figure 4. Simulation results of north zone airflow rate ratio.
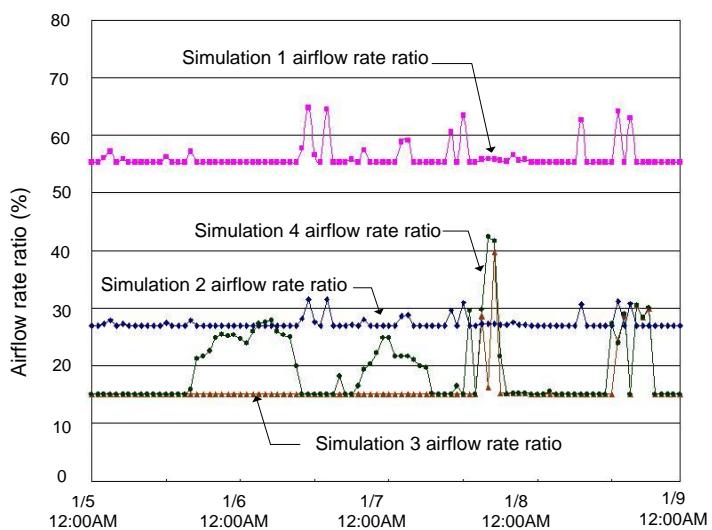
Figure 5. Simulation results of south zone airflow rate ratio.
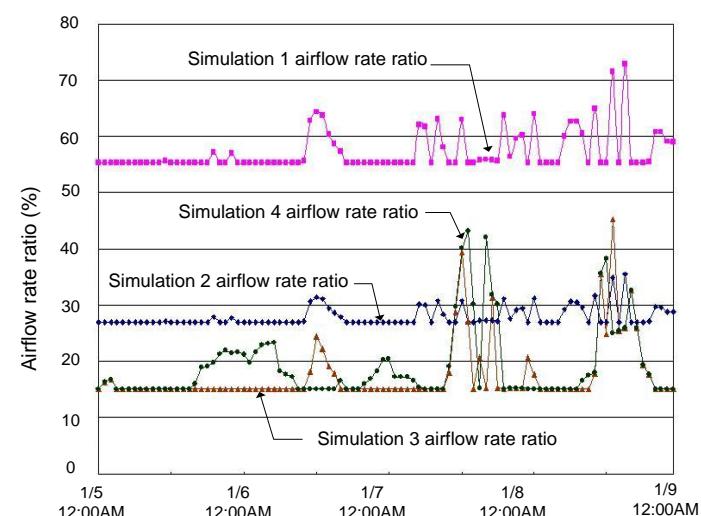
Table 3. Simulation data of room air temperature and airflow rate ratio.

Simulation item	Zone	Room air temperature [°F (°C)]	Airflow rate ratio (%)
Case 1	North	73.6 (23.1)–75.4 (24.1)	55.4–64.7
	South	73.8 (23.2)–75.3 (24.1)	55.4–72.8
Case 2	North	74.0 (23.3)–75.4 (24.1)	35.2–68.7
	South	73.6 (23.1)–75.4 (24.1)	35.2–69.3
Case 3	North	71.8 (22.1)–75.0 (23.8)	15.1–39.8
	South	72.2 (22.3)–75.1 (23.9)	15.1–45.3
Case 4	North	73.2 (22.8)–75.1 (23.9)	15.1–42.4
	South	73.4 (23.0)–75.4 (24.1)	15.1–43.1

Simulation 2. Constant Minimum Airflow Rate Ratio for Heating Load

In order to solve the problems of the current minimum airflow rate, simulation 2 was performed with 35% of the minimum airflow rate ratio for the heating load, which was calculated by Equation (6). The results of simulation 2 show that the room air temperature ranged from 74.0 °F (23.3 °C) to 75.4 °F (24.1 °C), and the average room air temperature was 74.5 °F (23.6 °C), in the north zone, as shown in Figure 2, 3 and Table 3. The room air temperature was maintained at the set point [75 °F (24 °C)] within control deviation. The range of the airflow rate ratio was 35.2%−68.7% and the average airflow rate ratio was 37.6% of the design as shown in Figures 4, 5 and Table 3. The room air temperature and airflow ratio of south zone were higher than those of the north zone. Also, the opening time of the reheating valve is reduced because of decreasing minimum airflow rate. However, minimum airflow ratio was still high during the entire heating season because the minimum airflow ratio was based on the maximum heating load. Therefore, the minimum airflow rate should be reduced further during most of the heating season, especially in the south zone.

Simulation 3. Constant Minimum Airflow Rate Ratio for Fresh Air

Simulation 3 was performed with 15% of the minimum airflow rate ratio for fresh air requirement, which was calculated by Equation (10). The results of simulation 3 show that the room air temperature ranged from 71.8 °F (22.1 °C) to 75.0 °F (23.8 °C), and the average room air temperature was 73.9 °F in the north zone, as shown in Figure 4 and Table 3. The room air temperature could not be maintained at the set point [75 °F (24 °C)] within control deviation. The range of the airflow rate ratio was 15.1%−42.4% and the average airflow rate ratio was 19.1% of the design as shown in Figures 4, 5 and Table 3. The reason why the room temperature of the north zone was kept 3.2 °F (2.0 °C) lower than the set point temperature is because the minimum airflow rate was too low during the time of maximum heating load. The minimum airflow rate cannot eliminate the maximum heating load. To solve this problem, the minimum airflow rate should be increased to meet the maximum heating load. However, the period of the maximum heating load is short during the entire heating season, so this is not a good solution. Therefore, the minimum airflow rate ratio needs to vary between a high limit for the maximum heating load and a low limit for the fresh air requirements according to building operation conditions.

Simulation 4. Variable Minimum Airflow Rate Ratio

In order to solve previous problems from constant minimum airflow rate, simulation 4 was performed with the variable minimum airflow rate ratio between 16% and 35%. The results of simulation 4 show that the room air temperature ranged from 73.2 ℉ (22.8 ℃) to 75.1 ℉ (23.9 ℃) and the average room air temperature was 74.2 ℉ (23.4 ℃), in the north zone, as shown in Figures 2, 3 and Table 3. The room air temperature was maintained at the set point [75 ℉ (24 ℃)] within control deviation. The range of the airflow rate ratio was 15.1%–42.4% and the average airflow rate ratio was 19.1% of the design airflow rate as shown in Figures 4, 5 and Table 3. The results show that the minimum airflow rate ratio increases followed by the heating load to maintain room air temperature. In particular, the rate of increase of minimum airflow rate in the north zone is higher than that of the south zone. The low limit minimum airflow rate ratio should be able to operate during most of the heating season.

3.5.2. Comparison of Energy Consumption

In order to evaluate the energy consumption when optimizing the minimum airflow rate, energy consumption was compared between conventional minimum airflow rate and variable minimum airflow rate. The simulations were divided into four cases, the same as the previous simulation cases for evaluation control performance. General fan efficiency is about 0.5–0.8. But, we assumed no fan efficiency reduction for simple calculation. When the impact of the fan efficiency reduction is neglected, thermal energy consumption and fan power can be calculated by the following Equations (11,12) when design fan power consumption is assumed 2.6 k Wh at maximum cooling load condition:

$$Q_e = \dot{m} \cdot c_p \cdot (T_{d,s} - T_s) \tag{11}$$

$$E_f = E_{d,f} \cdot \alpha_{sf}^{\ 3} / \eta_{f,d} \tag{12}$$

The simulation results for thermal energy consumption given in Table 4 show that the energy consumption in the north zone and south zones of case 4 is less than that of case 1 by 54% and 58%, respectively. The energy consumption in north and south zones of case 4 is less than that of case 2 by 36% and 45%, respectively. The thermal energy consumption of the south zone is less than that of north zone by 7%–22%. The reason is that the south zone has a lower heating load than the north zone, such as solar radiation and internal heat gain during occupied hours. The simulation results for fan power, given in Table 4, show that the energy consumption of case 4 is less than that of case 1 and case 2 by 90% and 77%, respectively. This is due to reduction of the minimum airflow rate ratio. Therefore, to optimize the minimum airflow rate, the terminal box can control thermal room conditions in addition to reducing fan power and saving reheat energy.

Table 4. Simulation results of energy consumption.

Simulation item	Zone	Thermal energy consumption (k Wh)	Fan energy consumption (k Wh)
Case 1	North	946.1	307.8
	South	875.2	333.2
Case 2	North	677.2	132.2
	South	660.5	146.1
Case 3	North	412.5	28.6
	South	359.3	33.7
Case 4	North	432.6	29.2
	South	359.9	33.7

4. Experiments

4.1. Installation

4.1.1. Test room Information

One typical terminal box located at a west exterior office room was chosen in this study for application of the proposed terminal box control algorithms. There are normally four persons occupying this office. The HVAC system operates 24 hours a day, 7 days a week. Table 5 shows the applied test room information. The design maximum airflow rate was 450 CFM (212 L/s), and the design minimum airflow rate was 150 CFM (70 L/s).

Table 5. Test room information.

Item	Information	
Zone	Location	Omaha, Nebraska
	Purpose	Office room
	Size	Floor area: 185.4 ft^2 (18 m^2) Volume: 2039.4 ft^2 (58 m^3)
People	Number	Normally 4 Persons
	Occupied hours	Variable
Operation condition	Schedule	24 hrs, 7 days working
	Primary air temp.	Reset
	Room set point	74 °F (23.3 °C)
Terminal box	Model	Staefa
	Maximum airflow rate	450 CFM (212 L/s)
	Minimum airflow rate	150 CFM (70 L/s)

4.1.2. Terminal Box Design

The terminal box consists of a controller, a room temperature sensor, a reheating coil, a CO_2 sensor, a flow sensor, a discharge air temperature sensor and a modulation damper as shown in Figure 6. It can maintain the ventilation requirement using the CO_2 sensor and provide the acceptable room air circulation using the discharge air temperature sensor. This box can improve thermal comfort, IAQ and save reheating energy.

Figure 6. View of the VAV terminal control system.
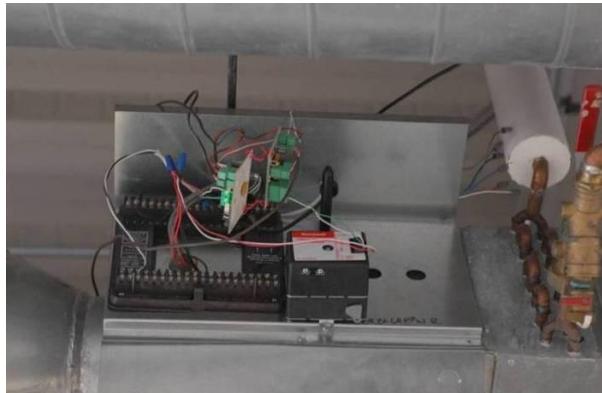
(**a**) Damper actuator
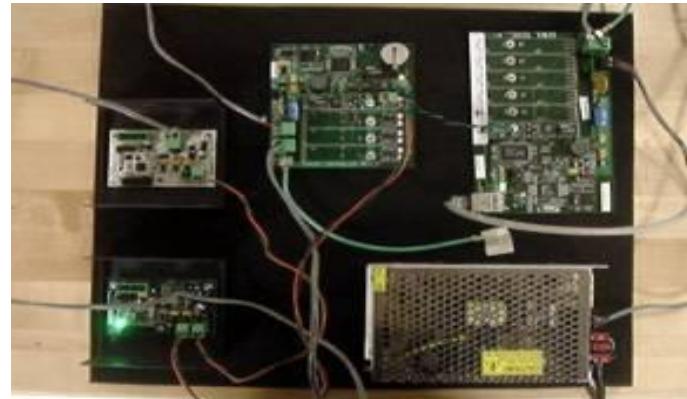
(**b**) Analog input/output modules
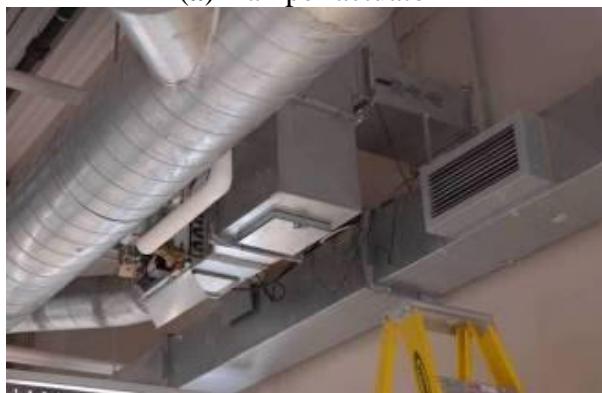
(**c**) Terminal box
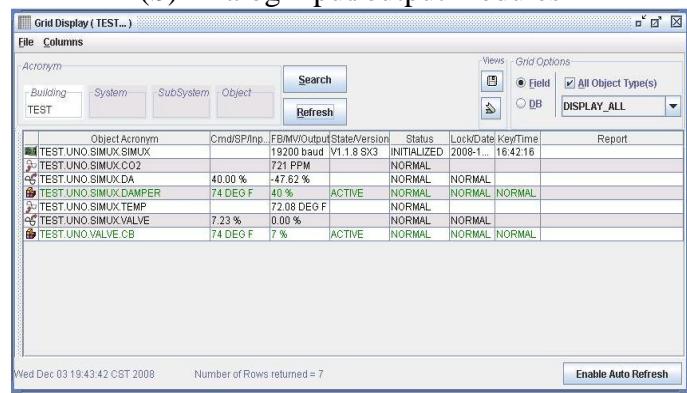
(**d**) EMCS (Energy Management Control System)

4.1.3. Implementation

4.1.3.1. Control Parameters

For the proposed control algorithms, four analog inputs, two analog outputs and four input parameters are required, as shown in Table 6. A discharge air temperature sensor and CO_2 sensor are added as analog input to apply and implement proposed control algorithms compared with conventional control algorithms.

Table 6. Control parameters for proposed control algorithms.

Point type	Point index	Point name	Description
Analog inputs	AI1	T_R	Zone Temperature
	AI2	CFM	Box supply airflow rate
	AI3	T_d	Discharge air temperature
	AI4	CO_2	CO_2 concentration
Analog outputs	AO1	D	Supply damper position
	AO2	V	Reheating coil valve position
Input parameters	IP1	$T_{R,c}$	Room temperature cooling setpoint
	IP2	$T_{R,h}$	Room temperature heating setpoint
	IP3	CFM max	Maximum cooling airflow rate setpoint
	IP4	CFM min	Minimum airflow rate setpoint

4.1.3.2. Control Logic

Definitions:

1. Cooling mode: the reheat valve is closed; the room temperature cooling setpoint is maintained by modulating the damper position.
2. Heating mode: the room temperature heating setpoint is maintained by modulating the damper position. The damper position is set as the minimum heating airflow rate setpoint; the discharge air temperature is maintained by modulating the reheating valve position.
3. Ventilation mode: The CO_2 concentration setpoint is maintained by modulating the damper position; the room temperature setpoint is maintained by modulating the reheat valve [18].

Process:

When the terminal box is activated, the terminal box is set at the cooling mode. The room temperature cooling setpoint is maintained by modulating the damper position. Under the cooling mode, the controller checks two conditions:

1. The CO_2 concentration is higher than the setpoint.
2. The room temperature reaches the heating setpoint.

When the first condition is satisfied, the terminal box is switched to the ventilation mode. In this mode, the controller modulates the damper position to maintain the CO_2 concentration setpoint, and the room temperature setpoint is maintained by modulating the reheat valve. During the ventilation mode, the controller checks the CO_2 concentration. If the CO_2 concentration is below the setpoint, the terminal box is switched from the ventilation mode to the cooling mode.

If the second condition is satisfied, the terminal box is switched from the cooling mode to the heating mode. The room temperature heating setpoint is maintained by modulating the damper position. The damper position is modulated to maintain the minimum heating airflow rate setpoint; the discharge air temperature is maintained by modulated the reheating valve position. The discharge air temperature setpoint is determined by minimum heating airflow rate setpoint.

In the heating mode, the controller checks the following two conditions:

1. The valve is fully closed, and the room temperature reaches the cooling setpoint.
2. The CO_2 concentration is higher than the setpoint.

When the first condition is satisfied, the terminal box is switched from the heating mode to the cooling mode. When the second condition is satisfied, the terminal box is switched to the ventilation mode. Figure 7 shows the proposed control algorithm for exterior zone terminal box.

The proposed control algorithms have a variable minimum airflow rate setpoint and discharge air temperature setpoint. It can automatically identify the minimum heating airflow rate and discharge air temperature under the actual working conditions. The discharge air temperature has a function of minimum airflow rate setpoint to maintain acceptable air circulation and minimum energy usage. It can also meet the ventilation requirement using CO_2 sensor.

The cooling setpoint was set at 74 ℉ (23.3 ℃); the heating setpoint was set at 72 ℉ (22.2 ℃). CO_2 setpoint should be variable based on ASHRAE standard 62.1-2010 [7], but we used a constant setpoint at 1,000 ppm for field test purposes.

Figure 7. Flow chart of proposed control algorithms with variable minimum airflow rate.
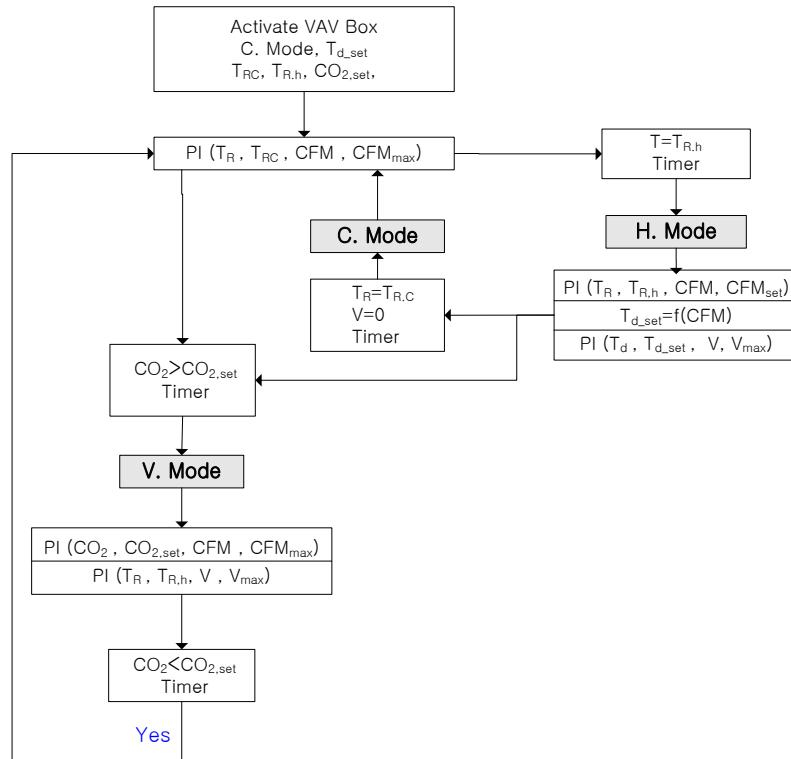
4.1.4. Evaluation

Field experiments were conducted to evaluate the developed control sequence with variable minimum airflow rate. Zone temperature control, fresh air requirement for the IAQ, air circulation for uniform air distribution and energy consumption were analyzed.

Hobo data loggers were positioned at 5 ft locations, which are the conventional room sensor locations (to measure the room air temperature at 1 minute intervals). To verify this data, average room temperatures at five locations in occupied areas were measured. EMCS trending data were used to evaluate the control performance of the room temperature. Figure 8 shows the measurement points used to evaluate the control performance of proposed control algorithms.

Carbon dioxide (CO_2) concentrations were measured using a Telaire 7001 CO_2 monitor (Telaire, Santa Barbara, CA, USA), which was connected to a data logger to record the measurements in the zone at 1 minute intervals. They were located in return duct and occupant positions. To compare this data, the primary CO_2 concentration was measured. EMCS trending data were used to evaluate the control performance of the ventilation mode.

Twenty-five Hobo data loggers were positioned to measure the vertical room air temperature at 1 minute intervals. Five vertical lines have five data loggers with 33 inch difference.

To measure energy consumption, the airflow rate, primary air temperature, discharge air temperature and room air temperature were measured. The TSI Multi-meter was used to trend the air

velocity in the supply ductwork at 5 minute intervals. The airflow rate was calculated using previous calibration data. Hobo data loggers were positioned to measure the temperature at 1 minute intervals.

Figure 8. Measurement points for evaluations.
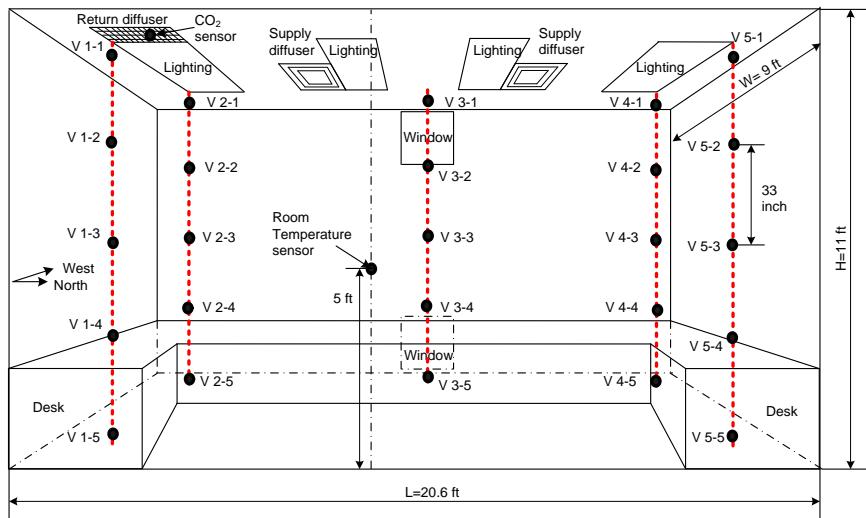
4.2. Results and Discussion

4.2.1. Zone Temperature Control

The range of the room air temperature is 71.6 ℉ (22.0 ℃)–72.8 ℉ (22.7 ℃), and the supply airflow rate is 0–79 CFM (37 L/s), as shown in Figure 9. The room temperature can maintain its setpoint with variable minimum airflow rate and discharge air temperature.

Figure 9. Trending data of room air temperature and airflow rate.
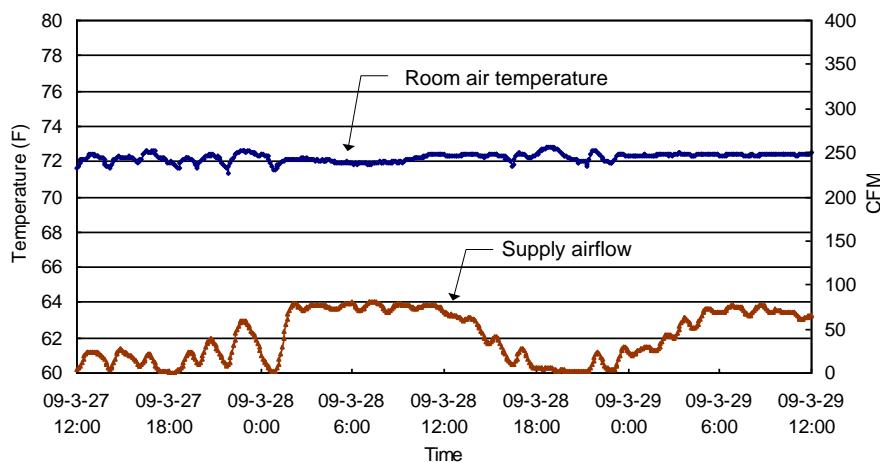
4.2.2. Fresh Air Requirement for IAQ

The CO_2 level was in the range of 480–780 ppm as shown in Figure 10. According to ASHRAE Standard 62-2010 [7], comfort criteria for ventilation can be satisfied with the variable minimum airflow rate.

Figure 10. Trending data of room CO_2 level and airflow rate.
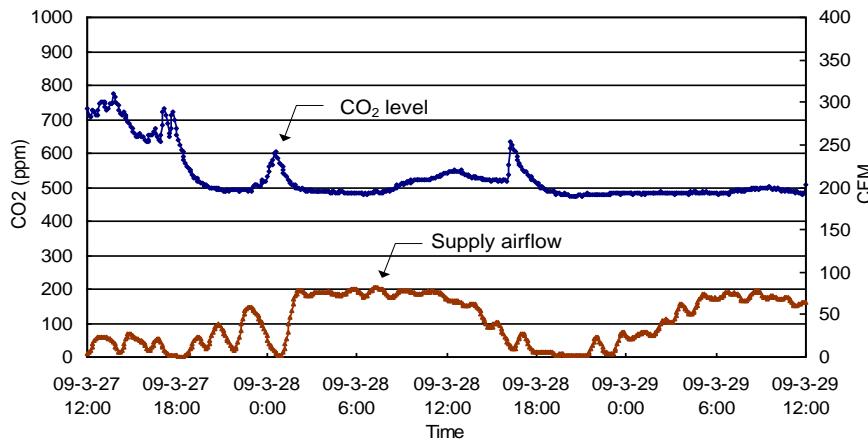
4.2.3. Air Circulation for Uniform Air Distribution

Figure 11 shows the vertical room air temperature profile at five different heights for implementing the proposed control algorithms. The maximum temperature difference between ceiling and bottom was 15 ℉ (8 ℃). The vertical air temperature below 8 ft (2.4 m) from the ceiling of 5 points was a range of 70 ℉ (21.1 ℃)–73 ℉ (22.8 ℃) when the discharge air temperature was a range of 90 ℉ (32.0 ℃)–117 ℉ (47.2 ℃). Around ceiling area, the temperature was about 85 ℉ (29.4 ℃). Average vertical air temperature of five points was a range of 70 ℉ (21.1 ℃)–74 ℉ (23.3 ℃).

Figure 11. Trending data of vertical room air temperature at different height.
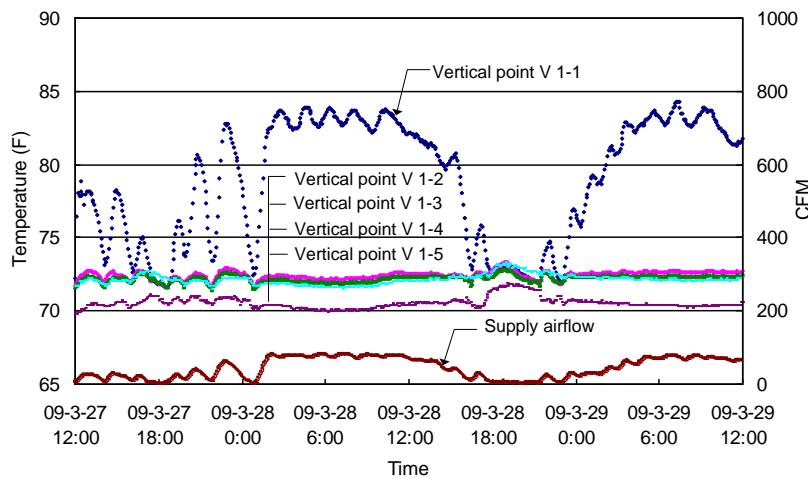
In this case, the room has good air mixing and has acceptable air distribution. The local discomfort due to the vertical difference of room air temperature is not thought to be significant. The vertical distribution is lower than the value proposed by the study on comfort because the vertical temperature difference is below 5.4 ℉ (3 ℃) within the occupied zone, measured at the 4 inch (0.1m) and 67 inch (1.7 m) levels (ASHRAE Standard 55 [14]; Olesen [15], 2002).

4.2.4. Room Load and Reheating Energy Consumption

Figure 12 shows the calculated room load and reheating energy consumption for implementing proposed control algorithms. The minimum airflow rate increases followed by an increase in the heating load to maintain room air temperature. It can save fan power and reheating energy during most of the heating season.

Figure 12. Trending data of room load and reheat energy.
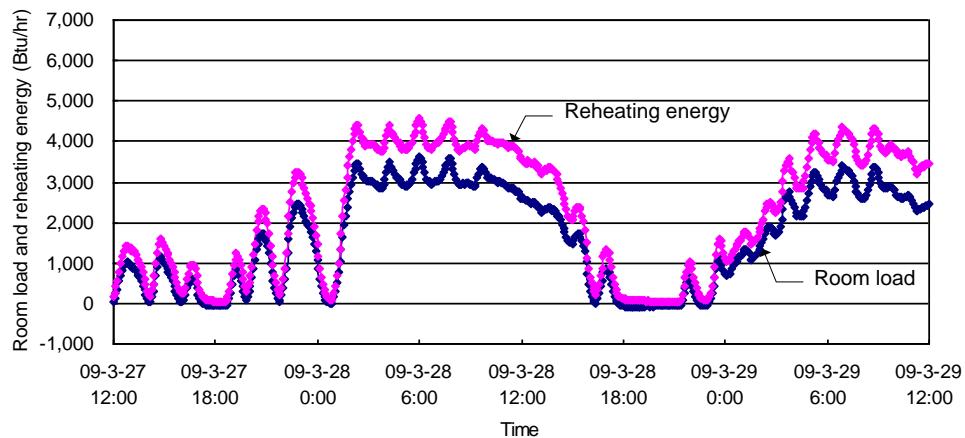
4.2.5. Comparison of Reheating Energy Consumption

To compare energy consumption, we measured the supply airflow rate, the supply air temperature and the zone discharge air temperature. The reheat energy consumption can be determined using Equation (11). Figure 13 shows the comparison of trend data for reheating energy consumption for implementation of proposed terminal box control algorithms. When the room load is 1,000 Btu/hr (293 W), the energy consumption is 2,200 Btu/hr (644 W) for conventional control algorithms. On the other hand, the energy consumption is 1,300 Btu/hr (380 W) for proposed control algorithms. The reheating energy consumption using proposed control algorithms is less than that of using the conventional control algorithms.

Figure 13. Comparison of reheating energy between conventional minimum airflow rate and variable minimum airflow rate.
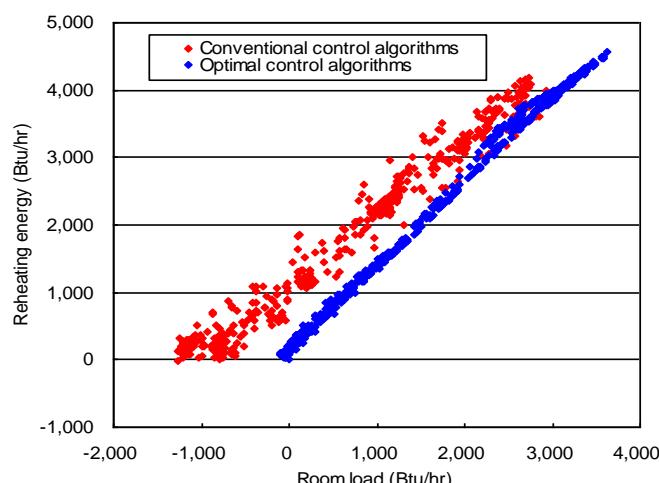
5. Conclusions

In this study, developed control algorithms were applied and validated with an actual building and evaluated for comfort, IAQ and energy consumption. The energy consumption and thermal performance of terminal boxes were compared using conventional and proposed control sequences. The results are as follows:

1. The constant minimum airflow rate ratio causes significant simultaneous heating and cooling cycles. Moreover, fan power usage is often excessive in mild weather. The minimum airflow rate ratio needs to vary between a high limit for the maximum heating load and a low limit for the fresh air requirements, according to building operation conditions.

2. According to the variable minimum airflow rate ratio, the terminal box can maintain room thermal comfort conditions to meet the various load changes in addition to reducing fan power and saving reheat energy.

3. The energy consumption of the variable minimum airflow rate ratio is less than that of the conventional constant minimum airflow rate ratio.

4. Control algorithms were developed to realize potential benefits. The proposed control algorithm for the VAV terminal box automatically identifies and implements the minimum heating airflow rate and discharge air temperature under actual working conditions. It ensures acceptable room air mixing and temperature control and avoids excessive primary airflow rate by automatically adjusting the heating airflow rate. It can improve the system's reliability and reduce costs.

5. The terminal box can automatically identify the minimum heating airflow rate under the actual working conditions using proposed control algorithms.

6. Improved control can stably maintain the room air temperature. The vertical difference of room air temperature is kept lower than the comfort value. Measurements of CO_2 levels show there is no indoor air quality problem when the minimum airflow rate setpoint is reduced.

7. The measured reheating energy consumption with proposed control algorithms is less than that with the conventional control.

Further studies should focus on analysis of vertical distributions of temperature and velocity by experiment and simulation integrated with developed control algorithm and conduct to validate in cooling load condition.

Acknowledgments

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