Evaluation of the Impact of Minimum Airflow on the Energy Consumption of Single Duct VAV Terminal Boxes

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Abstract
Terminal boxes are critical components of variable air volume (VAV) systems. The minimum airflow of terminal boxes is a key factor for comfort, indoor air quality (IAQ) and energy cost. If the minimum airflow is higher than needed, terminal boxes will have significantly more simultaneous heating and cooling, and air-handling units (AHUs) will consume more fan power. Buildings will have IAQ problems if the minimum airflow is less than required. A dynamic building simulation and field experiment were carried out to evaluate the impact of minimum airflow on the energy consumption of single duct VAV terminal boxes. A comparison of the simulation data with the existing constant setpoint revealed 20% annual energy savings in the improved minimum airflow setpoint. The experiment showed that the room air temperature can be maintained stably and the vertical difference in the room air temperature is lower than the comfort value when the improved minimum airflow setpoint is applied. Measurements of the CO₂ levels also revealed no indoor air quality problems.

Keywords: VAV terminal boxes; minimum airflow; field experiment; IAQ; energy consumption; EnergyPlus

1. Introduction
Single duct variable air volume (VAV) air-handling units (AHUs) are the most popular systems in the USA. Terminal boxes are the critical component of VAV systems. The minimum airflow of terminal boxes is a key factor for comfort, indoor air quality (IAQ) and energy cost. A constant minimum airflow is used in conventional control sequences. This control sequence can cause occupant discomfort or use excessive energy. If the minimum airflow is higher than needed, terminal boxes will have significantly more simultaneous heating and cooling, and the AHUs will consume more fan power. On the other hand, buildings will have IAQ problems if the minimum airflow is less than required.

For VAV reheat terminal boxes that serve exterior zones, the minimum airflow setpoint is typically selected by the maximum value of either of the following: (1) the airflow required by the room design heating load, or (2) the minimum required for ventilation. In 1989, the ventilation rates established by ASHRAE Standard 62-1989 were increased substantially over those previously required by the 1981 version of the standard. In 2004, Standard 62.1-2004 prescribed new minimum breathing zone ventilation rates and a new calculation procedure to find the minimum intake airflow needed for different ventilation systems.

Common practice methods, which are selected by the maximum of the following, 30% maximum airflow, the volume of outdoor air required to meet the ventilation requirements, 0.4 cfm/ft² of the conditioned floor area of the zone, and 300 cfm, are still used by HVAC designers. Nevertheless, they may not always guarantee the best results. In fact, some of the values are erroneous. Unfortunately, some engineers apply these values as a standard for every space, regardless of the thermal conditions.

Cho et al. (2009) developed optimal control algorithms for single duct VAV terminal boxes. The terminal box can automatically identify the minimum heating airflow under actual working conditions with improved control algorithms. This can stably maintain the room air temperature. The vertical difference of room air temperature is kept lower than the comfort value. Measurements of CO₂ levels show there is no indoor air quality problem when the minimum airflow setpoint is reduced. Cho et al. (2010) identified the near-optimal room airflow and discharge air temperature that will maintain the room air temperature and save
energy using a CFD simulation. The near-optimal discharge air temperature can prevent short air circulation under different heating load conditions and airflow, and stably maintain the room air temperature and keep the comfort value of vertical difference of room air temperature and indoor air quality. Researchers have worked to derive a representative dynamic model based on a commercially manufactured terminal unit and to analyze the Life Cycle Cost and evaluate the thermal comfort and load for the VAV system.

The aim of this study was to evaluate the impact of energy consumption and estimate the building energy savings on the discharge air temperature and minimum airflow of single duct VAV terminal boxes by conducting a dynamic building simulation and field experiment. A field experiment was carried out to evaluate the comfort, IAQ and energy consumption.

2. Field Experiment

2.1 Test Room Information

One typical terminal box located at a west exterior office room, as shown in Fig.1. and Table 1., was chosen to evaluate the impact of the minimum airflow of single duct VAV terminal boxes. Four persons normally occupy this office. The HVAC system operates 24 hours a day, 7 days a week. The type of terminal box is a single-duct VAV terminal box with a reheat coil. The design maximum airflow was 212 L/s, and the design minimum airflow was 70 L/s. The accuracy and resolution of all the sensors were collected from sensor specifications, as shown in Table 2.10.

![Fig.1. Schematic Diagram of the Test Room](image)

![Fig.2. View of the Terminal Box](image)

2.2 System Information

Fig.2. shows the conventional VAV terminal box controller for a typical exterior office area. The controller consists of a controller, room temperature sensor, reheat coil, flow sensor, and modulation damper. The VAV terminal box controller can supply the required airflow for the air-conditioned area to maintain the room temperature setpoint using the flow station.

| Table 1. Test Room Information |
|-----------------------------|
| Item                  | Information                          |
| Zone                   | Location Purpose Size                |
|                         | Omaha, Nebraska Office room Floor area: 17.2 m² Volume: 57.7m³ |
| People                 | Number Occupied hours Normally 4 Persons Variable |
| Operation condition    | Schedule Primary air temp. Room set point 24 hrs, 7 days working Reset 22 °C |
| Terminal box           | Model Staefa Maximum airflow Minimum airflow 212 L/s 70 L/s |

| Table 2. Accuracy and Resolution of the Measurement Sensors |
|-----------------------------------------------------------|
| Item                    | Accuracy | Resolution | Instrument |
| Temperature transmitter | 0.3 °C   | < 0.02 °C  | HOBO data logger |
| Air velocity transmitter| +/- 2% at 0-82 °C | 0.005~0.01 m/s | TSI Velocity meter |
| CO₂ sensor              | <+- [30 ppm CO₂ + 2% of reading] | 10 ppm CO₂ | Telaire CO₂ monitor |

2.3 Evaluation Cases

Table 3. lists the experiment items used to compare the conventional constant minimum airflow and variable minimum airflow.

| Table 3. Evaluation Cases |
|----------------------------|
| Cases | Condition |
|-------|-----------|
| 1-1   | Constant minimum airflow setpoint - The minimum required for ventilation |
| 1-2   | Constant minimum airflow setpoint - The airflow rate required by the room design heating load |
| 1-3   | Constant minimum airflow setpoint - Maximum number between ventilation (case 1-1) and heating load (case 1-2) required minimum airflow |
| 2     | Variable minimum airflow setpoint - Minimum airflow should not be constant, but rather varied during the heating period. |

Conventional control algorithms with constant minimum airflow setpoint

As shown in Fig.3., Cases 1-1 ~ 1-3 were conducted to determine the problems with the conventional constant minimum airflow setpoint. The following
contained two modes of conventional control algorithms.

- Cooling mode: the reheat valve is closed; the room temperature cooling setpoint is maintained by modulating the damper position.
- Heating mode: The damper position is set as the minimum heating airflow setpoint; the room temperature heating setpoint is maintained by modulating the reheat valve.

Improved control algorithms with variable minimum airflow setpoint

As shown in Fig.4., case 2 evaluates the variable minimum airflow using the improved control algorithms. The following is three modes of control algorithms.

- Cooling mode: the reheat valve is closed; the room temperature cooling setpoint is maintained by modulating the damper position.
- 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.
- Ventilation mode: The \( \text{CO}_2 \) concentration setpoint is maintained by modulating the damper position; the room temperature setpoint is maintained by modulating the reheat valve.

The variable minimum airflow can be calculated between the minimum airflow for the heating load and the minimum airflow for ventilation. When the room air temperature is below the setpoint, the airflow will decrease as the damper closes towards the minimum airflow setpoint for the ventilation requirements, and the discharge air temperature is a 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 for the ventilation requirement up to that of the heating load requirement to maintain the room air temperature, and the discharge air temperature is maintained at its constant setpoint. Therefore, the minimum airflow should not be constant, but rather should be varied during the heating period\(^1\).

3. Experiment Results and Discussion

3.1 Zone Temperature Control

Data loggers were positioned at locations 1.5m apart, which are the conventional room sensor locations (to measure the room air temperature at 1 minute intervals). The experiment was carried out under similar outside air conditions during a range of outside air temperatures of -3 ~ -1°C between January 14~20. No difference in the experimental conditions was observed among the 4 cases.

Fig. 5. shows the trending data of the room air temperature and airflow for each case. The X (horizontal) axis, left Y (vertical) axis and Y (vertical) axis is the experiment time, temperature and airflow respectively. As shown in Fig.5. and Table 4., an analysis of the room air temperature control in an actual system shows that the range of room air temperatures in case 1-1 is 20.1°C ~ 23.3°C, and the supply airflow is 40 L/s ~ 52 L/s. The corresponding values for cases 1-2, 1-3 and 2 are 21.3°C ~ 22.8°C, 70 L/s, and 21.2°C ~ 22.7°C, & 77 L/s.

During the unoccupied period of case 1-1, the lowest room air temperature was 20.1°C, so the system could not maintain the heating setpoint (22°C). If the outside air condition is extremely cold and the airflow is lower than the required minimum airflow, the system will not properly control the zone temperature because the minimum airflow is too low during the time of the maximum heating load conditions. On the other hand, the low minimum airflow can maintain the room temperature setpoint during most of the heating period.

Fig.3. Flow Chart of Conventional Control Algorithms with Constant Minimum Airflow

Fig.4. Flow Chart of the Improved Control Algorithms with Variable Minimum Airflow
because the period of the maximum heating load is short compared to the entire heating season.

The room air temperature of case 1-2 could maintain its heating setpoint when the minimum airflow was higher than required. On the other hand, it might waste fan power and reheating energy. The room air temperature of case 1-3 could maintain its heating setpoint when the minimum airflow was the required value but it might waste fan power and reheating energy during low load conditions. The minimum airflow was still high during the entire heating season because the design minimum airflow was based on the maximum heating load condition. The room air temperature of case 2 can maintain its setpoint with variable minimum airflow. If the outside air condition is extremely cold, the system will respond quickly based on the room air conditions, so the controller resets the minimum damper and maximum valve positions to maintain the room heating temperature setpoint.

Fig.6. shows the trending data for the correlation between the room air temperature and airflow for each case. The X (horizontal) axis and Y (vertical) axis is the airflow and room air temperature, respectively. As shown in Fig.6., the room air temperature of case 2 can be maintained within the control deviation at its setpoint (22°C). On the other hand, case 1-2 shows that the range of room air temperature has greater control deviation (~2°C) than the other cases because of the higher than design minimum airflow. The narrow control deviation of case 2 can give an occupant thermal comfort and save thermal energy.

3.2 Fresh Air Requirement for IAQ

The carbon dioxide (CO$_2$) concentrations were measured using a CO$_2$ monitor, which was connected to a data logger to record the measurements in the zone at 1-minute intervals. The measurement of CO$_2$ in occupied spaces has been used widely to evaluate the amount of outdoor air supplied to the indoor spaces. To verify the indoor air quality due to the reductions in minimum airflow, the CO$_2$ levels were measured using indoor air quality meters.

Fig.7. shows the trending data for the correlation between the CO$_2$ level and airflow for each case. The X (horizontal) axis was the airflow and the Y (vertical) axis was the CO$_2$ level. As shown in Fig.7. and Table 4., the CO$_2$ level of case 1-1, case 1-2, case 1-3, and case 2 ranged from 580 ~ 1,060 ppm, 560 ~ 620 ppm, 560 ~ 820 ppm, and 510 ~ 940 ppm, respectively.

According to ASHRAE Standard 62, if the ventilation results in the indoor CO$_2$ concentrations are higher than approximately 1,000 ppm, it may cause an indoor air quality issue. During the occupied periods, the CO$_2$ level of case 1-1 was approximately 1,060 ppm because the minimum airflow was lower than the required minimum airflow. On the other hand, the maximum CO$_2$ levels of cases 1-2, 1-3 and 2 were below the comfort and health criteria for ventilation.

The CO$_2$ level of the improved minimum airflow can maintain the ventilation requirement but case 1-1 showed a maximum CO$_2$ level of more than 1,000 ppm. Levels lower than design minimum airflow setpoint (case 1-1) may cause indoor air quality issues but levels higher than the design minimum airflow setpoint (case 1-2) showed a low CO$_2$ level with high-energy consumption.
3.3 Air Circulation for Uniform Air Distribution

Twenty-five data loggers were positioned to measure the vertical room air temperature difference at 1-minute intervals. Five vertical lines have 5 data loggers with an 80 cm difference. To analyze the effects of air circulation, the measurements were performed using the vertical differences of the room air temperature.

Fig.8. shows the trending data for the vertical room air temperature difference profile for each case. The X (horizontal) axis was room air temperature and the Y (vertical) axis was the height. Fig.8. and Table 4. show the vertical room air temperature difference profile at 5 different heights. The maximum temperature difference between the ceiling and bottom of cases 1-1, 1-2, 1-3, and 2 was 11.5°C, 2.1°C, 6.2°C, and 1.8°C, respectively. The vertical air temperature difference of 5 points of cases 1-1, 1-2, 1-3, and 2 ranged from 19.8 ~ 31.3°C, 20.8 ~ 22.9°C, 21.3 ~ 27.5°C, and 20.9 ~ 22.7°C, respectively. Around the ceiling area, the temperature of cases 1-1, 1-2, 1-3 and 2 was more than 32.2°C, 23.9°C, 28.3°C, and ~23.4°C, respectively.

In case 1-1, the high discharge air temperature and low airflow reduced the level of air circulation, and the airflow directly returned to the ceiling duct without good air mixing for the occupancy area. Under high heating load conditions, the room temperature cannot maintain its setpoint, and might cause thermal discomfort. To solve this, the airflow should be increased. In case 1-2, the high airflow increased the air circulation with good air mixing. The room air can be mixed well and have a uniform air distribution and maintain thermal comfort. On the other hand, the high airflow may increase the energy consumption. In case 1-3, the room has good air mixing and an acceptable air distribution when the minimum airflow is the required value but this case can reduce the airflow during most of the heating season. In case 2, the room has good air mixing and an acceptable air distribution. The local discomfort due to the vertical difference in room air temperature is not believed to be significant. The vertical distribution is lower than the value proposed by the study on comfort because the vertical temperature difference is less than 3°C between the head and ankles (1.1m and 0.1 m above the floor). Local discomfort might not occur due to the high vertical difference in the room air temperature. The upper level temperature of case 1-1 was much higher than the lower level temperature. The range of air temperatures between the head and ankles of case 1-1 was below 2.2°C, whereas the temperature around the ceiling area was approximately 31.1°C because of the high air temperature and low airflow. This can increase the level of high reheating energy consumption.

3.4 Discharge Air Temperature and Room Reheating Outputs

To measure the room reheating outputs, the airflow, primary air temperature, discharge air temperature and room air temperature were measured. Data loggers were positioned to measure the temperature at 1-minute intervals.

Fig.9. shows the trending data for the correlation between the discharge air temperature and airflow for each case. The X (horizontal) axis was airflow and the Y (vertical) axis was the discharge air temperature. As shown in Fig.9. and Table 4., the measured discharge air temperature of case 1-1 was 19.6°C ~ 57.2°C and the calculated room reheating outputs ranged from 0 to 1648 W. In cases 1-2, 1-3 and 2 the corresponding values were 18.0°C ~ 27.9°C & 0 ~ 1087 W, 18.0°C ~ 37.9°C & 0 ~ 1586 W, and 18.8°C ~ 50.9°C & 0 ~ 1666 W.
In case 1-1, when the minimum airflow is lower than required, the room reheating outputs and fan power can be minimized but it cannot guarantee room comfort and IAQ. In case 1-2, the high minimum airflow can provide room comfort and IAQ but can waste reheating energy and fan power. When the minimum airflow was at the required value, the room reheating outputs and fan power usage were still high. The minimum airflow should be reduced further during most of the heating season to save fan power and reheating energy. The minimum airflow increased followed by an increase in the heating load to maintain the room air temperature. This can save fan power and room reheating outputs during most of the heating season. The high airflow of case 1-2 has a low discharge air temperature. The variable airflow of case 2 has variable discharge air temperature to maintain the room air temperature. A lower airflow can save fan power and room reheating output.

Fig. 10. compares the trend data for the room reheating outputs of each case. The X (horizontal) axis and Y (vertical) axis were the room load and reheating energy, respectively. When the room load was 293 W, the room reheating output was 664 W when there was constant minimum airflow (case 1-3). On the other hand, the room reheating output was 293 W when there was a variable minimum airflow (case 2). The room reheating output using the variable minimum airflow was less than that when using the conventional constant minimum airflow.

4. Energy Analysis
4.1 Simulation Modeling
To evaluate the energy consumption of 4 cases with constant and variable minimum airflow, a dynamic building simulation model was developed using EnergyPlus software. EnergyPlus is a building performance simulation program that combines the best capabilities and features of BLAST and DOE-2. The following information provides a brief description of the simulation model. Table 5. lists the input data for the simulation.

| Item                   | Input data                        |
|------------------------|----------------------------------|
| Simulation tool        | EnergyPlus                        |
| Simulation model       | Location: Omaha, Nebraska         |
| Size                   | - Floor area: 17.2 m²             |
|                        | - Volume: 57.7 m³                 |
| Window                 | - Double glazing                 |
|                        | - Area: 14 m²                    |
| Weather condition      | TMY2 (Typical meteorological years 2) |
| People load            | Sensible: 70 W/person Latent: 45 W/person Number: 4 Persons |
| Lighting load          | 14 W/m²                           |
| Equipment load         | 8 W/m²                            |
| Operation condition    | Control: Continuous heating and cooling control Schedule: Occupied hour (8:00 a.m.–5:00 p.m.) Room Set point: 22 °C |

The zone envelope used as a simulation model is the same as the previous experiment test room. The outdoor conditions were assumed to follow the standard Omaha, Nebraska weather conditions in Typical Meteorological Years 2 (TMY2). According to the ASHRAE Handbook of Fundamentals, the occupant based heat gains are 115W/person. The ASHRAE Standard and ASHRAE Handbook of Fundamentals further instruct that the heat gain caused by lighting has a maximum of 14W/m², whereas the equipment-based heat gain has a maximum of 8W/m². The typical occupied hours are from 8:00 a.m. to 5:00 p.m. during weekdays.

Energy analysis of 4 cases was conducted to evaluate the impact on the energy consumption of minimum airflow and discharge air temperature. The simulation items were used to compare the conventional constant minimum airflow and the variable minimum airflow. Cases 1-1, 1-2 and 1-3 were conducted to determine the problems with the conventional constant minimum airflow. Case 2 evaluated the variable minimum airflow. The building simulation zone load calculations were performed year round at 15-minute intervals. Over that time period, the rate of energy consumption in the constant minimum airflow and variable minimum airflow were analyzed and compared. The supply air temperature of the AHU was 13°C, and the room temperature was 22°C.

4.2 Simulation Results
The cooling energy equals the difference in enthalpy between the mixed and supply air. The savings can be estimated. When the impact of the fan efficiency decrease is neglected, potential power savings can be estimated. When the supply fan airflow is reduced, the supply fan speed and total fan power should also be lowered. Therefore, significant fan power can be saved. Reheat energy savings can be considered as the reduced airflow from the supply air temperature to the room temperature.
Fig. 11. compares the annual cooling, reheating energy and fan power of 4 cases. The results show that the energy consumption of case 2 can be minimized compared to the other cases. In addition, the minimum airflow of 1-2 was higher than the required value. This can provide thermal comfort and meet the IAQ requirements for the occupant. Nevertheless, it can waste enormous cooling, reheating energy and fan power. Case 1-3 was determined to be the minimum airflow using the principle method for a conventional constant minimum airflow. Therefore, this case (case 1-3) can be used as a base case to compare with the improved case (case 2).

Fig. 12. and Table 6. compare the monthly cooling, reheating energy and fan power between the base case (constant minimum airflow, case 1-3) and improved (variable minimum airflow, case 2). During the winter season, there was no mechanical cooling and the economizer was not in operation. Therefore, the energy consumption data for this time period is not available. A larger amount of reheating energy was saved in the improved case because of the unnecessarily high constant minimum airflow and simultaneous heating and cooling in the summer months.

Fig. 13. compares the cooling, reheating energy and fan power per year between the base case (constant minimum airflow, case 1-3) and improved case (variable minimum airflow, case 2). The cooling energy consumption of the base case was 372 MJ/m² per year, and that of the improved case was 345 MJ/m² per year when an economizer was used. The cooling energy savings were 27 MJ/m² per year. The cooling energy consumption per year of a variable minimum airflow was 7% less than that of the conventional minimum airflow. The reheating energy consumption of the base case was 651 MJ/m² per year, and that of the improved case was 472 MJ/m² per year. The reheating energy savings were 179 MJ/m² per year. The reheating energy consumption per year of the variable minimum airflow was 27% less than that of the conventional minimum airflow. The fan power consumption of the base case was 78 MJ/m² per year, and that of the improved case was 72 MJ/m² per year. The fan power savings were 6 MJ/m² per year. The fan power savings per year with the variable minimum airflow was 8% less than that of the conventional minimum airflow.

Table 6. Comparison Data of the Annual Cooling, Reheating Energy and Fan Power of the 4 Cases

| Case     | Cooling energy (MJ/m²) | Reheating energy (MJ/m²) | Fan power (MJ/m²) | Total (MJ/m²) |
|----------|------------------------|--------------------------|------------------|---------------|
| Case 1-1 | 341                    | 518                      | 72               | 932           |
| Case 1-2 | 381                    | 686                      | 80               | 1147          |
| Case 1-3 | 372                    | 651                      | 78               | 1101          |
| Case 2   | 345                    | 472                      | 72               | 889           |

Fig. 12. Comparison Chart of the Monthly Cooling, Reheating Energy and Fan Power of the 4 Cases

(a) Monthly cooling energy consumption.

(b) Monthly reheating energy consumption.

(c) Monthly fan power.

Fig. 13. Comparison Chart of the Annual Cooling, Reheating Energy and Fan Power Between the Base Case (Case 1-3) and Variable Minimum Airflow Case (Case 2)
The annual energy consumption of the base case and improved case was 1101 MJ/m² per year and 889 MJ/m² per year, respectively. The annual energy savings totaled 212 MJ/m². Overall, the improved case showed a calculated 20% annual energy saving.

5. Conclusion

In this study, a dynamic building simulation and field experiment were conducted to evaluate the impact of the minimum airflow on the energy consumption of single duct VAV terminal boxes. The results were as follows.

(1) The improved control algorithms have a variable minimum airflow and discharge air temperature setpoint. These algorithms can automatically identify the minimum heating airflow rate and discharge air temperature under actual working conditions. The discharge air temperature has the function of the minimum airflow setpoint to maintain acceptable air circulation and minimum energy usage. This can also meet the ventilation requirement using a CO₂ sensor.

(2) The experiment for the improved control algorithms with a variable minimum airflow shows that the room air temperature can be maintained stably and the vertical difference in the room air temperature is lower than the comfort value when the minimum airflow setpoint is reduced. Measurements of the CO₂ levels also show there was no indoor air quality problem.

(3) In the energy-measured results, the reheating energy consumption with a variable minimum airflow was less than that with the conventional constant minimum airflow.

(4) Building simulations were performed to evaluate the improved control algorithms with a variable minimum airflow of energy consumption. The simulation data revealed that the variable minimum airflow setpoint exhibited an annual energy savings of 20% compared to the conventional constant minimum airflow setpoint.

Acknowledgment

This work was supported by Mid-career Researcher Program through NRF grant funded by the MSIP (No.2011-0028990).

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