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Supply Fan Control for Constant Air Volume Air Handling Units

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Abstract

Since terminal boxes do not have a modulation damper in constant volume (CV) air handling unit (AHU) systems, zone reheat coils have to be modulated to maintain the space temperature with constant supply airflow. This conventional control sequence causes a significant amount of reheat and constant fan power under partial load conditions. Variable Frequency Drives (VFDs) can be installed on these constant air volume systems. The fan speed can be modulated based on the maximum zone load. This paper presents the procedure to control the supply fan speed and analyzes the thermal performance and major fan energy and thermal energy savings without expensive VAV retrofit through the actual system operation.

Introduction

In constant volume (CV) systems, all delivered air is cooled to satisfy the maximum zone cooling load with constant speed fans. Air delivered to other zones is then reheated with heating coils in individual zone ducts. In ASHRAE Handbook, CV systems are generally limited to applications with fixed ventilation needs, such as hospitals and special processes or laboratories; nevertheless, CV systems are widely used in a lot of old office buildings because these systems can be easily controlled and have simple mechanical equipment.

However, there are issues in current CV systems. One issue is each zone has variable zone load. The CV system uses conditioned air from AHU generally at a fixed cold air temperature to meet the maximum cooling load. Reheat is added to the discharge air in each zone to match the cooling capacity to the current zone load. The result is very high thermal energy use. Another issue is the reheat energy can be reduced if the supply air temperature from the AHU can be varied. However, the supply air temperature has to be maintained at design value for the humidity control when the outside air humidity ratio is high. Significant reheat energy is consumed unnecessarily due to humidity control requirement. The other issue is normally the supply fan chooses bigger size than required design size when the HVAC systems design. This results in major fan power waste.

The following is the possible solution for how to solve these issues and control supply fan in the current CV systems. The best way to control supply fan is an installation of terminal box and modulate airflow with VFDs on AHUs. In this way, the system should be changed from CV system to VAV system. This method will be able to control adequate supply fan speed under partial load conditions. However, this method needs high installation and labor fees. Another way is an installation of the VFD on CV system and set the fan speed. In this method, the system will not change and does not need high installation and labor fees. The cost of using VFD is much higher than changing the pulley; however, it can allow accurate airflow adjustment. Consequently, using a VFD is a potential solution to control supply fan without system change and any retrofit fee. Fan energy and both the heating and cooling energy savings can potentially increase.

Few engineers recognize the advantage and requirement of VFD. Phillips (2004) tested the VFD energy usage. The data showed a 72 % energy savings and the payback on the VFD is 1.44 year and VFDs can help lower maintenance costs. Miller (2002) showed the minimum application requirement and said contemporary VFDs are more reliable, cost effective and more capabilities. Currently, in CV system, the supply fan controls with a fixed VFD speed by a rule of thumb. But, VFD speed can not adjust under partial load conditions due to lack of the capability of changing the airflow. Therefore, dynamic airflow adjustment using VFD on CV system should be studied.

The objective of this study is that existing CV system will not change VAV system and how to get the largest benefits from minimum investments by only installing VFD. This paper presents the issues of applied building with conventional CV systems, develops a procedure how to control supply fan using VFD on current CV system to solve these issues, and applies to the actual building. The energy consumption of conventional and improved supply fan controls were compared using measured data.

Building and Facility information

The applied building complex, located in Omaha, Nebraska, was built in the late 1960s and composed of building III, II and I. The 12-story building III, 8-story building II and 4-story building I are used as an office building complex with total area of 489,000 square feet, which is served by AHUs with VFD. Each system is a typical multi-zone CV system without terminal box. The typical office hour is from 8:00am to 5:00pm during the weekdays. Figure 1 shows a schematic diagram of the AHUs serving area layout. Two typical AHUs, located at the interior zone (AHU 7A) and exterior zone (AHU 8), were chosen in this study.

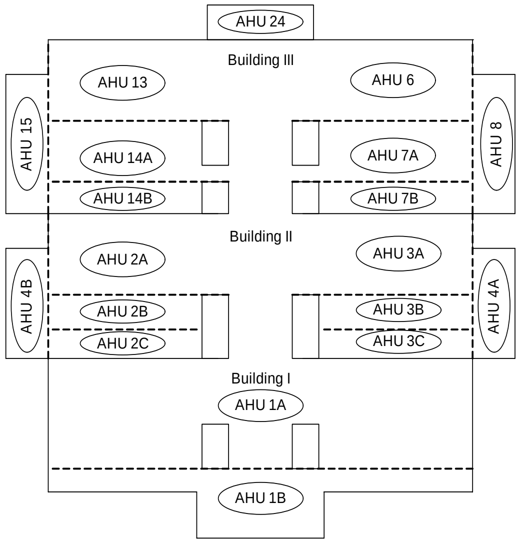

Figure 1. AHUs serving area layout

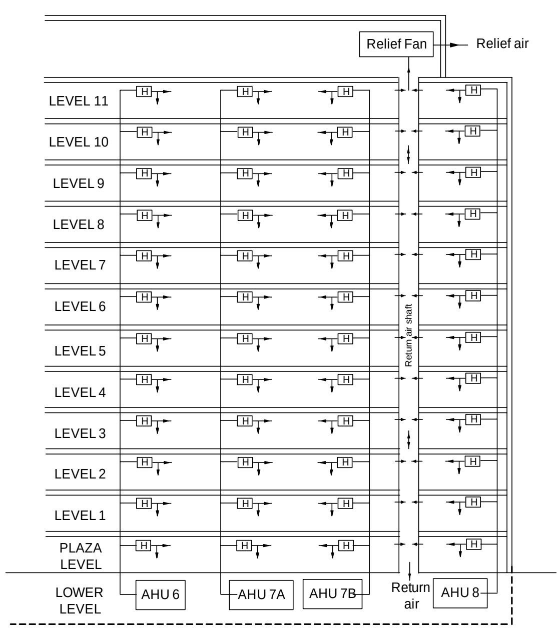

Figure 2. Section plan of the East of Building III

Figure 2 shows a schematic diagram for the section plan of Building III. Each AHU serves each floor. There is no terminal box and modulating damper. Each zone has reheat coils to maintain room air temperature for unequal loading. The AHU schematic diagram is shown in Figure 3.

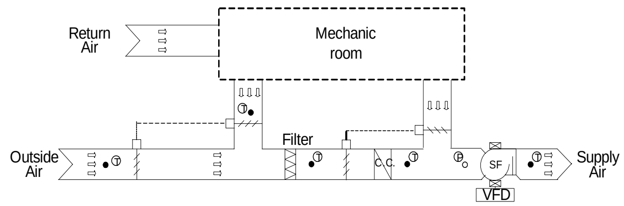

Figure 3. Schematic diagram of AHU

Procedure for supply fan control using VFD

Most of this energy waste can be avoided by simply installing a VFD on the fan without a major retrofit effort. The following is the procedure for the optimal supply fan speed.

Interior zone

Interior zone load profile in an office building is normally not too variable because there is no influence on the outside air conditions and similar heat gains. Cooling load for interior zones considers total sensible cooling load due to heat gains by people, lighting and office equipment conditioned spaces.

The supply fan speed for interior zone is typically optimized to be the following:

- *Identify the maximum load ratio for the AHU*

To identify the building load ratio for an AHU, room air temperature, supply air temperature and discharge air temperature should be measured daily. After measuring room conditions of each floor, choose the highest (zone B) and lowest load zone (zone A) of each zone as shown in Figure 4. The building load ratio can be calculated by the following equation:

$$\alpha = \frac{T_r - T_{DA}}{T_r - T_{SA}} \qquad (1)$$

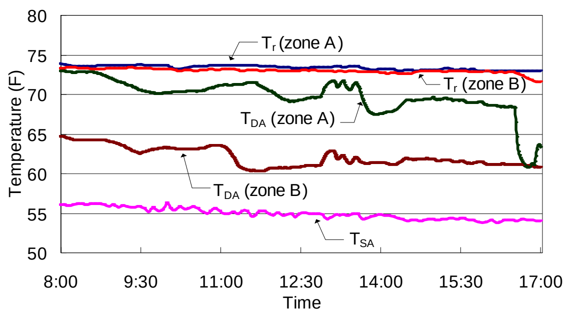

Figure 4. Measured data for building load ratio

Figure 5 shows the calculated load ratio graph by equation (1) to determine the supply fan speed. The graph shows zone A and B have different load ratios. Even though zone B has maximum load, it still wastes reheat energy because of high airflow ratio. The supply air temperature from central units provided to be maintained at design value (55 °F) for the humidity control, but discharge air temperature increased to maintain room conditions by reheat coil. Therefore, the reheat energy will be able to be reduced by increasing load ratio. There are two ways to increase the load ratio. One way is an increase of the supply air temperature (T_{SA}). However, this method may have a humidity issue. Another way is a decrease of the discharge air temperature (T_{DA}) by reducing airflow ratio.

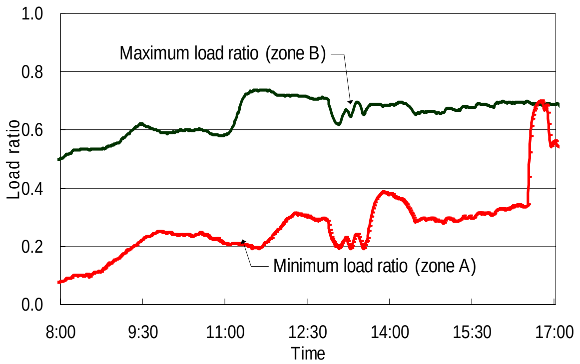

Figure 5. Calculated zone load ratio graph

- *Develop/implement the optimal fan speed*

Develop the supply fan speed set point based on the previous maximum load ratio and modulate the VFD to maintain the supply fan speed at its set point. Figure 6 shows the determined fan speed ratio based on the maximum load ratio. To verify this control, select critical zones like furthest zone and highest load zone from AHU and measure the room air condition.

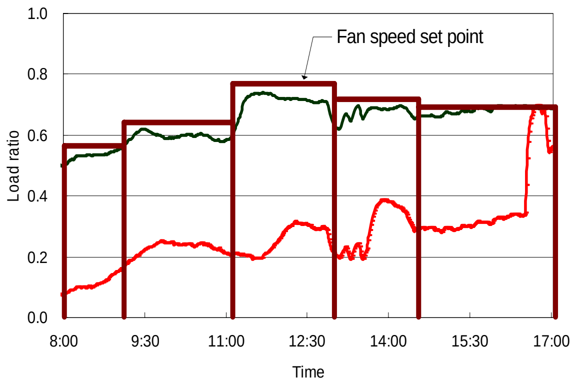

Figure 6. Determined fan speed ratio

Exterior zone

Exterior zone load profile in office building will be variable by the outside air conditions but internal heat gains are similar to interior zone loads.

The supply fan speed for the exterior zone is determined by the following:

- *Calculate design condition and determine the optimal fan speed*

Calculate the design airflow based on the real heat gains data such as U value of the wall and window, solar heat gain, lights, appliances and people. All referenced data are based on the ASHRAE Fundamentals Handbook.

- *Determine the optimal fan speed*

Based on the above calculation, the supply fan speed ratio was calculated by different outside air temperatures from 40°F to 100°F. During heating period, the fan speed has minimum fan speed like terminal box minimum airflow ratio.

If the system works continuously 24 hours and 7 days, the fan speed will have 3 different modes: occupied hours at weekday daytime, unoccupied hours at weekend daytime and unoccupied hours at weekday night time. VFD speed was modulated to maintain the supply fan speed at its set point. Figure 7 shows the determined supply fan speed ratio with 3 different modes.

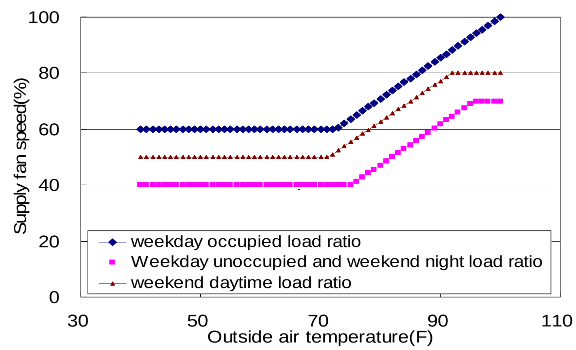

Figure 7. Supply fan speed ratio for dynamic VFD speed

- *Implement the optimal fan speed*

Implement the supply fan speed set point based on the pervious determined set point. Also, measure the room air condition to verify this control. Figure 8 shows the trending data for the supply fan speed and outside air temperature.

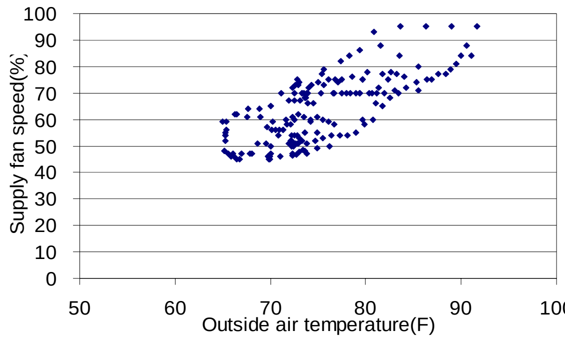

Figure 8. Supply fan speed and outside air temperature trend

Results and discussions

After applying the optimal supply fan control sequence in real buildings, the energy consumption and room conditions were measured and analyzed.

Energy consumption in the actual system

- *Interior zone*

Room, supply and discharge air temperature were measured with a data-logger to compare the thermal energy consumption. The results show room and supply air temperature are similar but discharge air temperature is different for each area served. Figures 9 and 10 show the discharge air temperature graphs of existing and optimal supply fan control. Two figures show each floor has a similar trend but different discharge temperature to maintain room air temperature by adjusting reheating coil. Each floor discharge temperature shows that there is unnecessary reheat energy from supply air condition to maintain room air temperature, even the 8th floor, and also wastes a fan power energy. After implementing a dynamic VFD speed, the discharge air temperature has a lower than fixed VFD speed as shown in Figure 9 and 10. The discharge air temperature is reduced and reheating energy should also be saved by dynamic VFD speed.

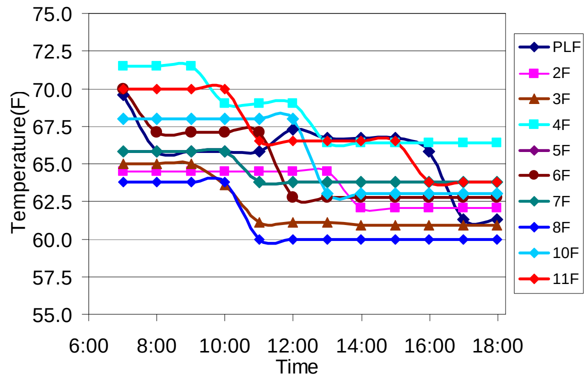

Figure 9. Discharge temperature graph of existing supply fan control

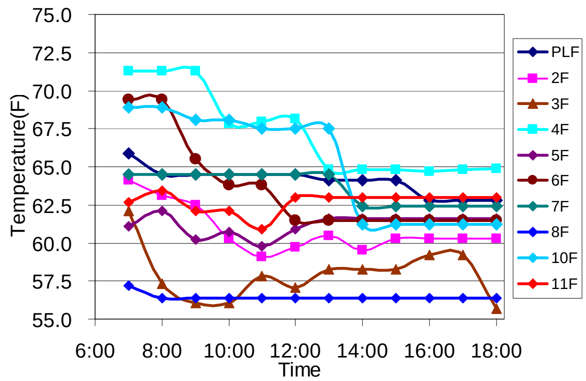

Figure 10. Discharge temperature graph of optimal supply fan control

The reheat energy savings can be considered as the reheat energy consumption required to heat the reduced airflow from the discharge air temperature to the room temperature. The reheating coil energy consumption can be estimated based on the equation (2).

$$\Delta E_{rh} = \dot{m} \cdot C_p \cdot (\alpha_{fixed} - \alpha_{dynamic}) \cdot (T_{DA} - T_r) \qquad (2)$$

From previous measured room condition data, the reheat energy consumption is 6,947,892 Btu/hr when there is fixed VFD speed, as shown in Table 1 and Figure 11. On the other hand, the energy consumption is 3,896,493 Btu/hr when there is dynamic VFD speed. The thermal energy consumption of dynamic VFD speed is less than that of the fixed VFD speed by 44 %. Therefore, when the supply fan speed control is optimized, thermal energy can be reduced.

Table 1. Comparison data of thermal energy consumption

Floor	Fixed VFD speed (Btu/hr)	Dynamic VFD speed (Btu/hr)	Energy saving (%)
PLF	784,891	502,611	36.0
2nd	680,100	350,993	48.4
3rd	527,329	163,780	68.9
4th	799,420	551,056	31.1
5th	693,341	344,032	50.4
6th	720,195	467,998	35.0
7th	624,738	429,354	31.3
8th	458,972	81,093	82.3
10th	822,660	599,766	27.1
11th	836,246	405,810	51.5
Total	6,947,892	3,896,493	43.9

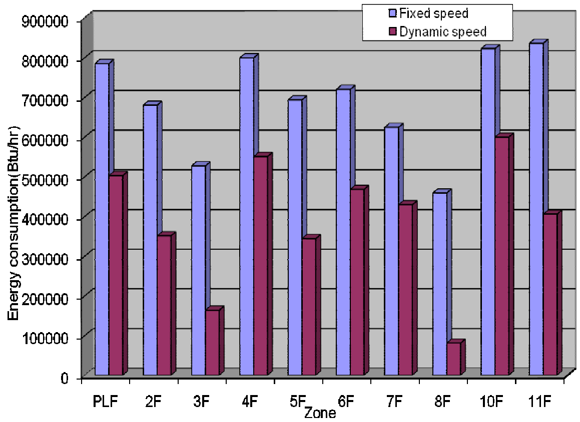

Figure 11. Comparison graph of the thermal energy consumption

The interior zone, before installing VFD, was a constant volume (CV) system. The supply fan power consumption was 2534.0 kWh per week. After installing VFD on the supply fan, the supply fan speed was set at a constant speed under normal operating conditions. The supply fan speed set point was 90% (25.7 kW) of the design speed. The supply fan power consumption was 1799.0 kWh per week. The operator can manually adjust the supply fan speed set point based on the requirement.

After implementing the dynamic speed reset, the fan speed is reset based on the maximum zone load. Figure 12 shows the trended data for the supply fan speed for the interior zone. The supply fan power was 1013.6 kWh per week. The supply fan speed was reduced significantly. The decrease of fan speed results in major fan power and zone reheating coil energy savings. The fan power savings is 29% after installing VFD and 60% after dynamic VFD speed. The system can be operated continuously without frequent speed adjustment by the building operator.

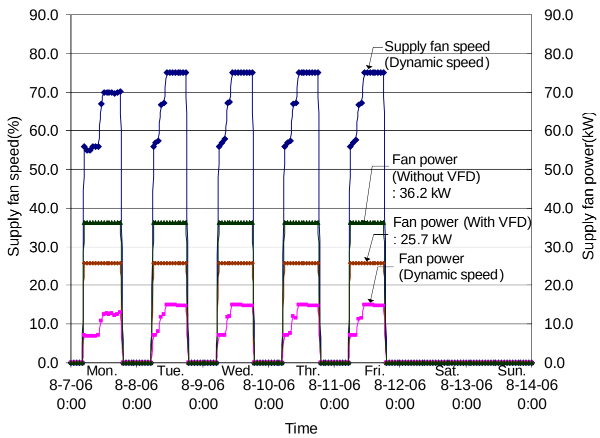

Figure 12. Supply fan speed and power trend graph for interior zone

- *Exterior zone*

In the exterior zone, before installing VFD, the supply fan power was 5796.0 kWh per week. After installing VFD on the supply fan, the supply fan speed set point was 90%(21.0 kW) of the design speed. The supply fan power was 3528.0 kWh per week. After implementing optimal speed, the supply fan power was 1426.1 kWh per week. Figure 13 shows the trend data for supply fan speed and fan power after implementation of the new control sequence for the exterior zone. The fan power savings is 39% after installing VFD and 75% after dynamic VFD speed. In addition, the discharge air temperature and airflow are reduced and then reheating energy should also be saved by dynamic VFD speed.

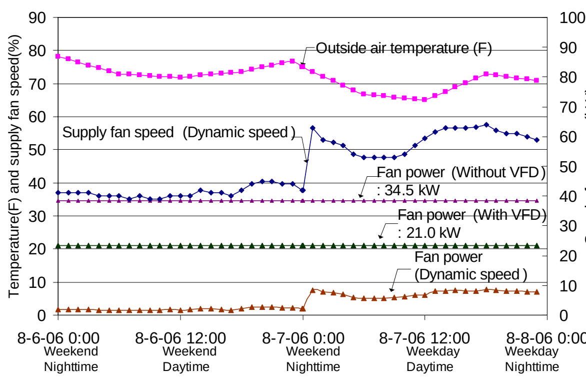

Figure 13. Supply fan speed and power trend graph for exterior zone

Evaluation of thermal performance

To verify the interior zone condition, select the furthest zone (11th floor) from AHU and highest load zone (8th floor). Exterior zone chooses the furthest zone (11th floor). The range of the room air temperatures is 72.4°F ~ 73.8°F during occupied hour, as shown in Figure 14. After implementing dynamic VFD speed, the room air temperature is able to maintain the set point and did not result in comfort issues. Therefore, it is assumed that there is proper and stable room air temperature control.

Measurement of CO_2 in occupied spaces has been widely used to evaluate the amount of outdoor air supplied to indoor spaces. The average CO_2 level on each floor was in the range of 400 ~ 500 ppm when the average outdoor air concentration was 320 ppm. According to ASHRAE Standard 62, comfort criteria are likely to be satisfied if the ventilation results in indoor CO_2 concentrations less than 700 ppm above the outdoor air concentration, which is representative of delivery rates of outside air. Therefore, it is judged that any IAQ problems due to reduction of the airflow will not happen.

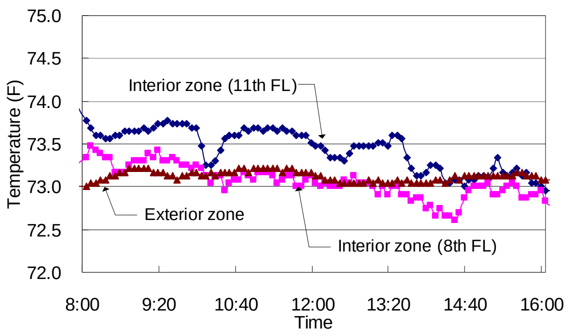

Figure 14. Room air temperature trend

This method is the possible solution for how to control supply fan in the current CV systems. This study shows energy savings without thermal comfort issues through the field test; however, it still has issues: (1) difficult to satisfy every zone in the multi-zone system, (2) verify the building load ratio during four seasons and (3) consider the supply air temperature reset when the outside air humidity ratio is low.

Conclusions

This study optimizes the supply fan speed using VFD with a CV system to reduce energy consumption. This paper shows a procedure to determine dynamic supply fan speed using VFD, and to apply the optimal supply fan control to an actual AHU system. Energy consumption was compared using measured data.

(1) The fan speed can be modulated based on the maximum zone load based on the measured data.

(2) The thermal energy consumption of dynamic VFD speed is 43% less than that of the fixed VFD speed.

(3) The fan power savings is 60% in the interior zone and 75% in the exterior zone after dynamic VFD speed.

NOMENCLATURE

c_p - specific heat capacity, Btu/lbm °F

$\dot{m}$ - mass flow rate, lbm/hr

T_r - room dry bulb temperature, °F

T_{SA} - supply dry bulb temperature, °F

T_{DA} - Discharge dry bulb temperature, °F

α - Building load ratio, %

α_{fixed} - fixed airflow

$\alpha_{dynamic}$ - dynamic airflow

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