

Single-duct constant volume system optimization

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ABSTRACT

This paper demonstrates the implementation of new innovative technologies to improve building operations and reduce energy costs. The buildings are used as an office building with a total area of 45,429 m². An energy audit was conducted to evaluate the building's energy performance and identify potential cost-effective energy saving opportunities. This paper presents procedures for supply fan speed control, implementations and detailed descriptions of control sequence, and operation comparisons. The results show electricity savings of 23% and gas savings of 19% over a six-month period.

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1. Introduction

An ASHRAE committee proposed a building commissioning in 1988 to ensure that the system performance met the design specifications. Continuous Commissioning[®] technology was developed and implemented in 1992. Continuous Commissioning[®] is an ongoing process to resolve operating problems, improve comfort, optimize energy use and identify retrofits for existing commercial and institutional buildings and central plant facilities [1–4].

In constant volume 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 the ASHRAE Handbook, constant volume systems are generally limited to applications with fixed ventilation needs, such as hospitals and special process buildings or laboratories; nevertheless, constant volume systems are still used in the old office buildings because these systems can be easily controlled and have simple mechanical equipment [5].

However, there are issues in current constant volume (CV) systems. One of these is that each zone has a variable zone load. The constant volume system uses conditioned air from air-handling unit (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 supply air temperature has to be maintained at design value for the humidity control requirement when the outside air humidity ratio is high. Significant reheat energy is consumed unnecessarily.

And the supply fan was chosen in the bigger size than required design size when the heating, ventilating and air conditioning (HVAC) systems were designed. This results in major fan power waste.

The following is the possible solution for these issues and to control the supply fan in the current constant volume systems. The best way to control the supply fan is to install the variable air volume (VAV) terminal boxes and modulate the airflow with variable frequency drive (VFD) on air-handling units. In this way, the system should be changed from a constant volume system to a variable air volume system. This method will be able to control adequate supply fan speed under partial load conditions. However, this method requires high installation and labor fees. Another way is to install the variable frequency drive on the constant volume system and to set the fan speed. In this method, the system will not change and does not require high installation and labor fees. The cost of using variable frequency drive is much higher than changing the pulley; however, it can allow accurate airflow adjustment. Consequently, using a variable frequency drive is a potential solution to control the supply fan without a system change and any retrofit fee. Fan power and both the heating and cooling savings can potentially increase [8–10].

Liu et al. [7] presented a method to convert dual-duct constant volume systems without retrofitting the terminal boxes to a variable air volume system. A damper is installed in the main hot duct. During summer, the hot air is shut off. This method reduces both fan and thermal energy. For single-duct constant volume systems, Liu et al. [1] suggested installing variable frequency drive on both supply and return air fans to reduce airflow at night and on weekends. This practice can be extended to normal operation hours with the use of a proper control sequence for the constant volume system.

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Nomenclature

c_p	specific heat capacity (kJ/kg K)
$\dot{m}$	mass flow rate (kg/h)
T_r	room dry bulb temperature (°C)
T_{SA}	supply dry bulb temperature (°C)
T_{DA}	discharge dry bulb temperature (°C)

Greek symbols

α	building load ratio (%)
α_{fixed}	fixed airflow
$\alpha_{dynamic}$	dynamic airflow

The objective of this study is to develop a supply fan control method by only installing variable frequency drive on a current constant volume system. This paper presents the facility information of the applied building, develops a procedure for the supply fan control, and describes the application of this method to the actual building. The heat, electricity and gas consumption of conventional and improved supply fan controls were compared using measured data.

2. 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 a total area of 45,429 m², which is served by air-handling unit with variable frequency drive. Each air-handling unit is a typical single-duct constant volume system with terminal boxes. The typical office hours are from 8:00 a.m. to 5:00 p.m. during the weekdays. Fig. 1 shows a schematic diagram of the air-handling units serving area layout. Table 1 shows the air-handling unit information, and the air-handling unit schematic diagram is shown in Fig. 2.

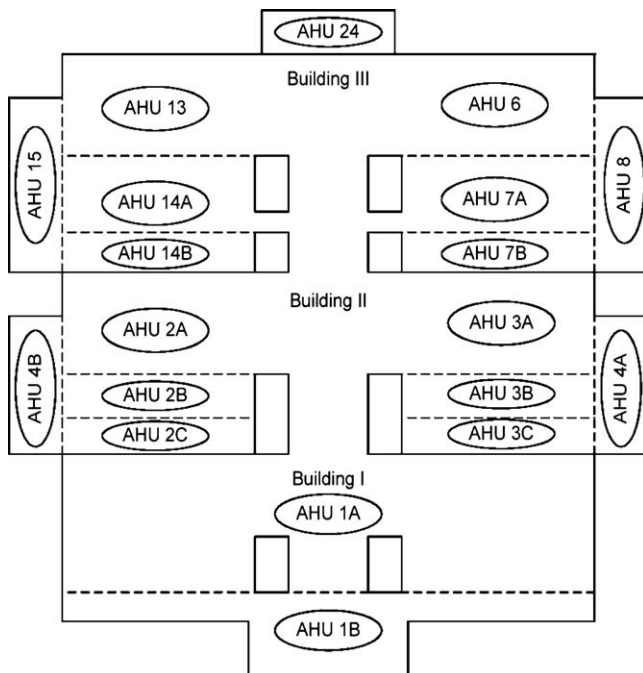


Fig. 1. Air-handling units serving area layout.

Table 1

Air-handling units information.

	Flow rate (L/s)	Fan power (kW)	Serving area	
AHU 6	35,488	45	East interior zone	Building III
AHU 8	13,781	19	East exterior zone	
AHU 13	26,387	30	West interior zone	
AHU 15	16,410	22	West exterior zone	
AHU 7A	26,775	37	East interior zone	
AHU 7B	3,704	7	South exterior zone	
AHU 14A	23,064	22	West interior zone	
AHU 14B	7,646	11	South exterior zone	Building II
AHU 2A	19,178	22	West interior zone	
AHU 2B	12,156	11		
AHU 2C	5,753	6	South exterior zone	
AHU 3A	17,697	22	East interior zone	
AHU 3B	16,618	15		
AHU 4A	7,137	11	East exterior zone	
AHU 4B	7,899	11	West exterior zone	Building I
AHU 4C	5,879	6	South exterior zone	
AHU 1A	26,017	30	Interior zone	
AHU 1B	17,039	15	Exterior zone	
AHU 21	13,266	15	North exterior zone	

Fig. 3 shows a schematic diagram for the section plan of Building III. Each zone has reheat coils to maintain room air temperature for unequal loading.

3. Supply fan control method

Most of this energy waste can be avoided by simply installing a variable frequency drive on the fan without a major retrofit effort. The following is the procedure for the optimal supply fan speed [11].

3.1. Interior zone

The 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. The supply fan speed for an interior zone is typically determined to be the following.

3.1.1. Identify the maximum load ratio

To identify the building load ratio for an air-handling unit, room air temperature, supply air temperature and discharge air temperature should be measured daily as shown in Figs. 4(a) and 5(a). After measuring the room conditions of each floor, choose the highest (zone B) and lowest load zone (zone A) of each zone. The building load ratio can be calculated by the following equation:

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

Figs. 4(b) and 5(b) show the calculated load ratio using Eq. (1). The graph shows zone A and B have different load ratios.

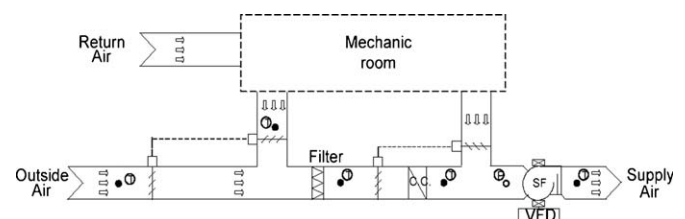


Fig. 2. Schematic diagram of air-handling unit.

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