Application of Innovative Technologies during Continuous Commissioning

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Abstract

This paper demonstrates the implementation of new innovative technologies during continuous commissioning practices to improve building operations and reduce energy costs. A 30-year-old typical commercial building with a floor area of about 49,436 square feet was used as a case study building. The new technologies are a variable speed drive volumetric tracking method for building pressure control, a recently developed fan airflow measurement method for duct static pressure reset, and a new operational strategy based on the current variable chilled water flow technology. The results showed that these technologies improve building operation and maintenance and significantly reduce energy costs. The building energy consumption has been reduced by 48% based on monthly utility bills, of which 32% are contributed to the Continuous Commissioning savings.

Introduction

Buildings older than 20 years generally face degradation of energy systems. Most of these systems use out-of-date equipment and pneumatic controls. Energy improvement opportunities exist in many buildings due to the inefficient operation of energy systems. These inefficient systems not only waste energy but also increase maintenance costs and decrease building comfort. HVAC system retrofits and upgrades improve operating efficiency and maintenance, but the payback can be tedious. Even with retrofits and upgrades, continuous commissioning practices can create opportunities for energy efficiency, maintenance and comfort.

Continuous Commissioning (CC) has been one of the most prominent energy conservation processes for over a decade. CC has been developed to help building owners achieve energy savings, improve thermal comfort and reduce maintenance costs [1]. During the CC process, the system operations and controls can be further optimized by applying new technologies.

This paper presents the applications of three new technologies in a case study building. The new technologies are a variable speed drive volumetric tracking method for the building pressure control, a duct static pressure reset using newly developed fan airflow stations, and a new operational strategy based on the current variable chilled water flow technology.

Technical background

Building Pressure Control

In a VAV system, return fan speed control is critical in maintaining appropriate building pressure. In most VAV systems, direct fan tracking control, direct building pressure control or volumetric tracking by direct airflow measurement are commonly used for the return fan control. Liu [2] has demonstrated that these methods have difficulty in maintaining proper building pressure.

In the same paper, Liu introduced a building pressure control method named variable speed drive volumetric tracking (VSDVT). The return airflow is controlled at an amount less than the supply airflow to maintain appropriate positive building pressure. The difference between supply and return airflows corresponds to the building exhaust airflow and the envelope tightness. In this method, fan airflow stations are used to obtain an accurate airflow for the supply and return fans. The fan airflow station measures fan speed and fan head, and utilizes the fan curve to calculate the airflow. The fan curves can be expressed by using a second order polynomial equation which works well in the fan's operational range. Under full speed, the fan curve equation is:

$$H = a_0 + a_1 \cdot Q + a_2 \cdot Q^2 \tag{1}$$

where H is a fan head [inch H_2O, Pa] under full fan speed.
Q is an airflow rate [ft^3/min, m^3/s] under full fan speed.
a_0, a_1 and a_2 are fan curve coefficients.

If the fan runs under partial speed by the VFD, the fan head is correlated with the fan speed. Equation 2 is derived by using the fan laws:

$$H = a_0 \cdot \overline{\varpi}^2 + a_1 \cdot \overline{\varpi} \cdot Q' + a_2 \cdot Q'^2 \tag{2}$$

where: $\overline{\varpi}$ is a current fan speed over 100% fan speed.
Q' is a calculated airflow rate under partial speed of the fan.

The airflow rate at any measured fan head and fan speed can be obtained by solving Equation 2. The actual airflow rate was calculated by Equation 3.

$$Q' = \frac{-a_1 \cdot \overline{N} - \sqrt{a_1^2 \cdot \overline{N}^2 - 4 \cdot a_2 \cdot \left(a_0 \cdot \overline{N}^2 - H\right)}}{2 \cdot a_2} \tag{3}$$

The theoretical model of the fan airflow station has been experimentally tested, and it was found that the model closely agreed with the experimental values [3]. This method is a very effective way to measure the airflow accurately under all weather and working conditions of air-handling units (AHUs).

Duct Static Pressure Reset

The fan airflow station technology was used not only for the variable speed drive volumetric tracking, but also for the duct static pressure reset control. The supply air fan is controlled to maintain the duct static pressure at a set point. When the cooling load is low, the supply airflow rate is reduced. Consequently, the pressure loss through the duct will be reduced. Duct pressure will be maintained even though the fan speed is lowered.

For constant static pressure control, extra fan head is consumed to maintain the constant static pressure set point when terminal box dampers downstream of the sensor are closed to consume the fan head. The duct static pressure reset can be applied if the pressure sensor is located at the upstream or the middle of the main duct rather than at the end location. The reset lowers the duct static pressure to open terminal box dampers relatively more. Therefore, the amount of supply airflow to the space for the reset will be the same as for the constant static pressure control during the same load condition.

The static pressure required through ductwork and boxes is proportional to the square of the supply airflow ratio. The supply airflow ratio denotes the calculated supply airflow rate from the fan airflow station over the supply airflow rate under design conditions.

$$P' = P_1 \cdot \left(\frac{Q'}{Q_1}\right)^2 + \alpha \tag{4}$$

where: P' is a calculated static pressure [inch H_2O or Pa].
P_1 is a measured static pressure when all the boxes are open.
Q' is calculated supply airflow rate [ft^3/min, m^3/s]. (See Equation 3)
Q_1 is a measured supply airflow when all the boxes are open.
α is a additional static pressure

In reality, it is difficult to obtain measured data under the design conditions. Therefore, P_1 and Q_1 in Equation 4 can be obtained by measuring the actual static pressure and supply airflow when most terminal boxes are fully open. An additional pressure (α) is added in the calculation due to different load profiles in different zones. The static pressure set point is reset depending on the total supply airflow rate, which theoretically implies that the load condition of each zone be the same. Therefore, the actual static pressure set point should be a little higher than the calculated reset value. This will ensure that zones which have higher cooling loads can be supplied with enough airflow. The additional value can be different and should be adjustable based on the system characteristics.

Variable Chilled Water Control

In addition to the air side, an innovative control method was implemented for the chilled water system. Recently, new chillers have been improved and are able to run at a variable water flow. This technology has been in use for more than a decade [1, 4, 5, 6 and 7]. Figure 1 shows a schematic diagram of a new single-chiller and single-cooling coil variable chilled water system.

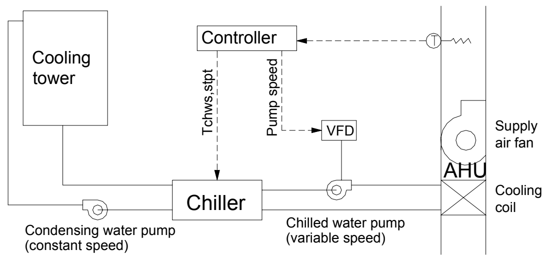

Figure 1: Schematic diagram of the new chilled water system

The new system is equipped with a variable speed drive (VFD) at the chilled water pump instead of the three-way valve for supply air temperature control. The VFD controls pump speed to maintain the supply air temperature at its set point after the chiller is enabled. Generally, the chiller specifications limit the minimum pump speed due to concerns about accumulation of dirt or scales inside the tubes at low water speed. When the pump speed reaches the minimum, the supply chilled water temperature set point will be reset to a higher temperature. The purpose of the reset is to control the supply air temperature when the cooling load is low and the pump speed is at the minimum.

Facility

The case study building was built in Omaha, Nebraska in 1972. This three-story building is used as a typical rental office building with a floor area of 49,436 square feet. The building is served by a single-duct cooling-only variable air volume air-handling unit and perimeter hot water heating. The HVAC systems are located in the penthouse and serve the entire building. The AHU consists of a supply fan of 100 HP and a return fan of 50 HP. There are a total of 52 original pneumatic VAV boxes (no reheat coils) which were installed in 1972.

The original 30-year-old chiller was replaced by a 155-ton chiller, the chilled water valve was removed and a VFD was installed on the chilled water pump as a retrofit process before commissioning. The original pneumatic controller system was upgraded to an electronic energy management and control system. Along with the controller upgrade, the newly developed fan airflow stations were installed to the supply and return fans. The fan airflow station comprises a differential pressure transducer to measure the fan head, a digital tachometer to measure the fan speed, and the design fan curve to calculate the airflow rate. In the actual system, the fan curve coefficients (a_0, a_1 and a_2) are 6.417, 0.0002284 and -0.000000003239 for the supply fan and 6.119, 0.0001931 and -0.000000004112 for the return fan, respectively.

These retrofits and upgrades allow the application of recently developed innovative technologies described in the previous section. Figure 2 shows a schematic diagram of the AHU fan airflow station system control.

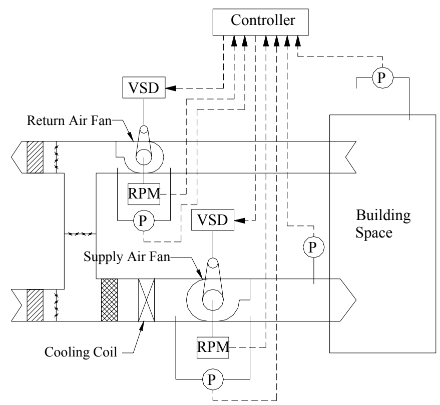

Figure 2: Schematic diagram of the AHU fan system control

Results and Discussion

Building Pressure Control

The return air fan is controlled to maintain the calculated return airflow rate at a set point of 3,000 ft^3/min less than the supply airflow rate. The set point was determined by an experiment in which the value from the building pressure sensor was maintained at a constant positive value of 0.03 inch H_2O in a very calm day.

Figure 3 presents the results of the building pressure control by the VSDVT on a typical calm day. The data were collected at five minute intervals. The return airflow rates were approximately 3,000 ft^3/min less than the supply airflow rate, as intended. The measured data from the building pressure sensor were maintained within 0~0.05 inch H_2O most of the day.

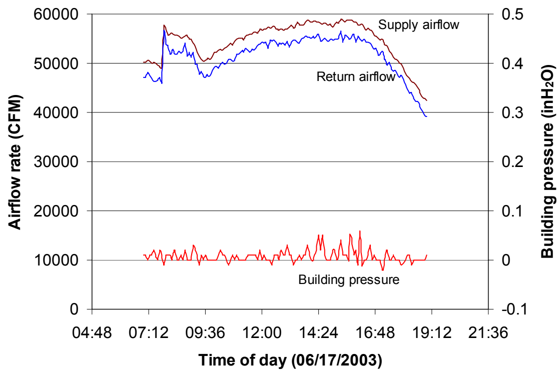

Figure 3: An example of building pressure control on a typical calm day

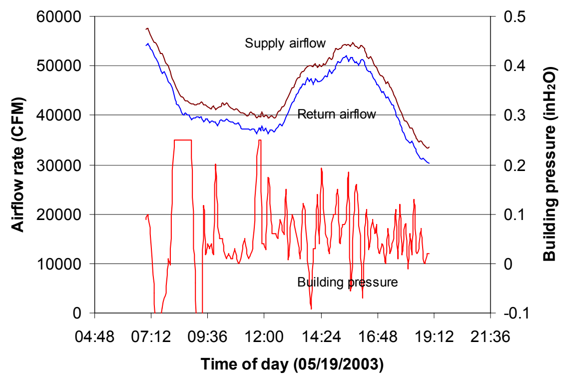

Figure 4: An example of building pressure control on a windy day

Figure 4 presents the results of the building pressure control by the VSDVT on a windy day. Even though the differences between the supply airflow rate and the return airflow rate were maintained at 3,000 ft^3/min, the values from the building pressure sensor were agitated. The accuracy of the building pressure sensor was significantly impaired by wind. Figures 3 and 4 prove that the volumetric return fan tracking control works better than the building pressure control by actual building pressure differential measurement.

Figure 5 presents the relationship between the supply and return airflow rate difference and the measured building pressure. The coordinate represents the difference between the supply airflow and the return airflow rates, and the abscissa represents the measured value from the building pressure sensor. The data were collected from May 17, 2003 to June 30, 2003. Most measured airflow differences remained around 3,000 ft^3/min, but the measured data from the building pressure sensor were scattered. Area B can be well represented as typical measurement errors of the building pressure sensor affected by wind. Point B in Figure 6 describes one of the points of Area B in Figure 5. Even though the airflow difference was about 3,000 ft^3/min, the building pressure was 0.1 inch H_2O at Point B in Figure 5.

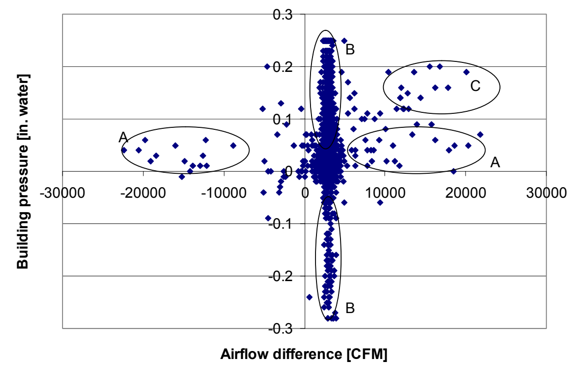

Figure 5: Airflow difference versus building pressure

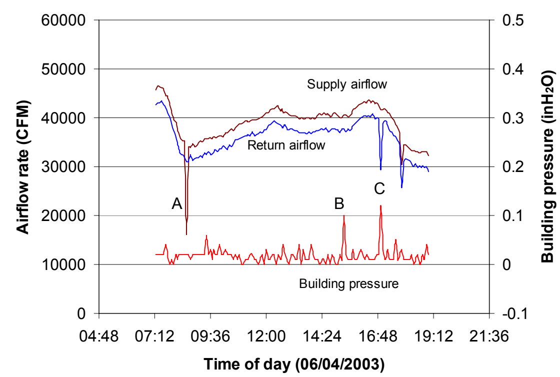

Figure 6: An example of several different disturbances on a typical calm day

In Figure 5, several points in area A and C were out of boundary near 3,000 ft^3/min. These two areas are also depicted as the momentary points A and C in Figure 6, respectively. The supply air fan was running as a normal condition at point A according to the collected data, but the value from the tachometer was very low. This means that the signal from the tachometer to the controller might be disturbed. Normally, this abnormal signal disturbance does not affect the return fan operation as shown by point A. At point C, however, the return air fan was actually running slow according to the return fan speed output. The calculated airflow difference was 12,286 ft^3/min, while the value from the building pressure sensor was 0.12 inch H_2O. The miscalculated supply airflow rate caused by the signal disturbance may affect the return fan operation. The difference between the A and C cases is that the actual system was affected by the signal disturbance in case C. The case like point C occurred within less than 0.5% of the collected data. Therefore, it can be asserted that the VSDVT method is free of weather conditions and its operation is very stable.

Duct Static Pressure Reset

The duct static pressure reset method can be customized for the AHU system by utilizing the airflow measurement of the fan airflow station. In this system, the static pressure sensor is located almost at the end of the main duct. Figure 7 shows the calculated and current duct static pressure set point depending on the supply airflow ratio, compared with the constant set point at 1 inch H_2O. The current set point is 0.2 inch H_2O higher than the calculated value, and the low limit is set at 0.5 inch H_2O to ensure that all terminal boxes supply enough airflow.

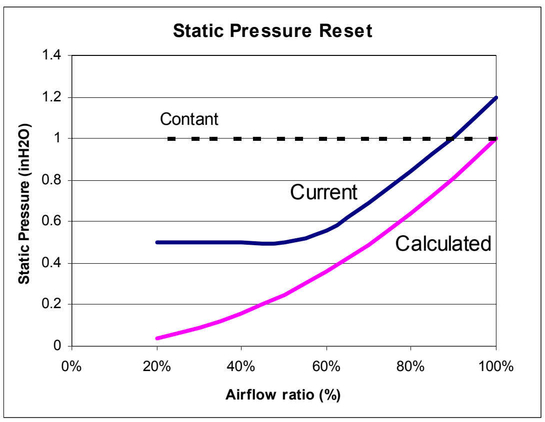

Figure 7: Static pressure reset schedules

Currently, the static pressure reset control is overridden by the constant static pressure control because a remote terminal box serves interior zone offices as well as a conference room. If the conference room is occupied, one diffuser is not able to satisfy the space cooling load. The override will be released after a new arrangement can be made.

Variable Chilled Water System Control

In the new single-chiller and single-cooling coil variable chilled water system, the supply air temperature of the air-handling unit is controlled by either the pump speed under high cooling load conditions or the supply chilled water temperature reset under low cooling load conditions due to the limitation of minimum water flow by the chiller specifications. The minimum pump speed was limited to 62% in this system. The low and high limits of supply chilled water temperature set points are 45ºF and 55ºF, respectively.

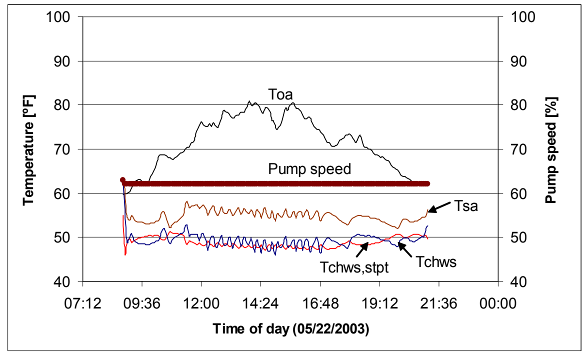

Figure 8: Chilled water system operation on a typical spring day.

Figure 8 presents an example of chilled water system operation on a typical spring day. The pump was operated at the minimum speed and the supply chilled water temperature (Tchws) was reset to maintain the supply air temperature (Tsa) at the set point under partial load conditions. The supply chilled water temperature was controlled to match its set point (Tchws,stpt) by an independent chiller controller. The supply air temperature was maintained within the dead-band (4ºF) of its set point (55ºF). Eventually, this variable chilled water system will reduce pump power and achieve stable system operation.

The pump power consumption was measured and regressed as a function of pump speed. When the pump speed was 100%, the pump power was 4.2 kW. When the pump speed was 62%, the pump power was about 1.2 kW. Therefore, the power consumption can be obtained through the collected pump speed data. The data were collected from May 17, 2003 to June 30, 2003. The chilled water system was operating for 457.5 hours during the measurement period. Figure 9 shows the hour fraction of different pump speed ranges. The pump ran at the minimum during about 67 % of the operating hours. The pump power consumption was under 50% (2.1 kW) during 87% of the operating hours. The pump power savings in May and June were 69% and 62%, respectively. Therefore, substantial pump power savings can be achieved by using the variable chilled water system.

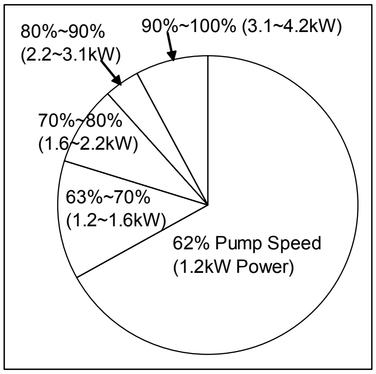

Figure 9: Hour fraction of different pump speed ranges

Energy Savings

Building electricity consumptions has been reported from a local power distribution company as shown in Figure 10. In the chart, pre-commissioning columns are monthly electricity consumptions of year 2002, and post-commissioning columns are monthly electricity consumptions during or after the commissioning process in 2003.

After the commissioning practices, average 48 percent of electricity savings are achieved. Among this savings, 32 percent accounts for the Continuous Commissioning that includes the application of innovative technologies, chiller replacement and implementation of improved operating schedules, and 16 percent accounts for the lighting retrofit.

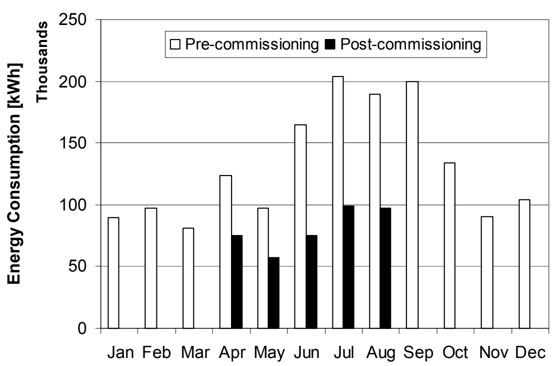

Figure 10: Comparison of pre-commissioning and post-commissioning electricity consumptions

Conclusion

This paper presents three innovative technologies and their applications in a commercial office building as part of the Continuous Commissioning process. The new technologies include the building pressure control, the duct static pressure reset and the variable chilled water control. The results show that the building pressure was accurately controlled by the variable speed drive volumetric control that is equipped with the fan airflow stations on the supply and return fans. By utilizing the airflow measurement on the supply fan, the static pressure set point can be adequately reset, which also leads to fan power savings. Significant pump power savings were obtained by an innovative variable water flow method in a single-chiller, single-cooling coil chilled water system. The application of these newly developed technologies improved building operation and reduced energy costs. The Continuous Commissioning saved average 32 percent of electric power for the building.

Acknowledgement

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