

# Continuous Commissioning<sup>®</sup> of an Office Building

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## Abstract

*The case study building is a typical 10-story office building. The continuous commissioning (CC) started in May 2004 and completed in March 2005. Major commissioning measures include: improvement of the VAV box fan operation; modification of the supply fan and return fan control algorithm; correction of the outside air intake setting; modification of the chilled water pump control sequence. The implementation results are presented. After CC, the supply fan speed was reduced, economizer operation hours were increased, the chilled water system operation period was shorten, chilled water pump speed control was optimized and indoor comfort was improved. In a nine-month period, the electricity demand was reduced by 18%, and electricity consumption was decreased by 1,177,857 kWh.*

## Introduction

The 10-story office building was built in 1987. The total gross area is about 200,000 ft<sup>2</sup>. Most of the areas are occupied 40 hours per week. There are two floors occupied 24 hours on weekdays.

A total of five main air handling units (AHU) supply the conditioned air to most area of the building. Among them, AHU1, AHU2, AHU3 and AHU4 are variable air volume (VAV) systems with fan powered boxes on floors except the executive level. AHU5 is a single zone and single duct VAV system serving Atrium. Another single duct constant volume system, AHU6 only serves the executive level.

There are about 200 fan powered terminal boxes installed in the building. The VAV series fan powered boxes are installed in 10<sup>th</sup> floor, and the VAV parallel fan powered boxes are installed in other floors. The exterior zone boxes are equipped

with hot water reheat coils, and the interior zone boxes do not have reheat coils.

The central plant consists of one rotary chiller to produce chilled water to the chilled water secondary loop and two heat pumps to make ice for the ice storage system. They provide the required cooling to the entire building.

The capacity of the rotary chiller is 200 Ton. The system is a constant primary and variable secondary chilled water flow distribution system.

The capacity of each ice-making heat pump is 160 Ton. The total capacity of the ice-storage system is 2,850 ton-hours. Three identical tanks were installed in the basement mechanical room. The type of the ice-storage is external melt-on-coil. At the charging mode, the brine cooled by two heat pumps is fed to the coils submerged in tanks. At the discharging mode, the ice water in tanks is pumped through a frame-plate heat exchanger to cool the return chilled water from end users. The warm water returning from the heat exchanger is cooled by directly contacting with the melting ice.

Direct digital control (DDC) is used in the energy management and control systems.

The Continuous Commissioning<sup>®</sup> (CC) team from Energy Systems Laboratory at University of Nebraska-Lincoln and Omaha Public Power District initiated the CC project in May 2004 on the request of the owner. Most of the recommendations were implemented by December 2004.

The CC process was started by verifying the existing control sequences and taking measurements of selected parameters under the existing operating schedules. Based on the actual site conditions, CC recommendations were developed. The major CC measures included: optimizing terminal box minimum flow set point, modifying the interior zone fan control sequence of the parallel fan powered boxes, improving the VAV air handler supply fan and return fan control, changing one constant volume

system to VAV system, optimizing the chilled water system operation schedule, optimizing chillers, cooling tower and chilled water pump control .

### Main CC Measures

The main CC measures are summarized below.

#### Terminal boxes

The terminal box CC measures include resetting the minimum airflow set points on fan-powered boxes and modifying the fan control for the interior zone parallel fan power boxes.

#### Existing control

When there was a heating requirement, the terminal box fan was on. Both interior zone boxes and exterior zone boxes used the same control sequence.

For interior zones, the terminal box fans had never run since there is no heating requirement.

#### Improved control

For interior zone boxes, when the airflow is low at partial loads, because the parallel fans would not run, the throw from the diffuser is reduced. Reduced throw will result in poor air mixing. Cold air drops directly down to the occupants below the diffuser, which caused comfort problems. Turning on the fan when the terminal box flow is low can mix the cold air with the air in the ceiling better and provide more uniform air distribution. After CC, parallel fans will be on when box airflow is less than certain box flow low limit.

This measure provides better indoor air quality, but increases electricity consumption.

### Variable Volume Air Handling Systems

There are four variable air volume (VAV) systems, AHU1, AHU2, AHU3 and AHU4. These single duct (VAV) systems provide conditioned air to the fan powered boxes. In each unit, both the supply fan and return fan are installed with variable frequency drives. Figure 1 shows the air handling unit schematic. Figure 2 depicts the building main air conditioning system diagram. Commissioning of these units focused on supply fan control, return fan control and economizer control.

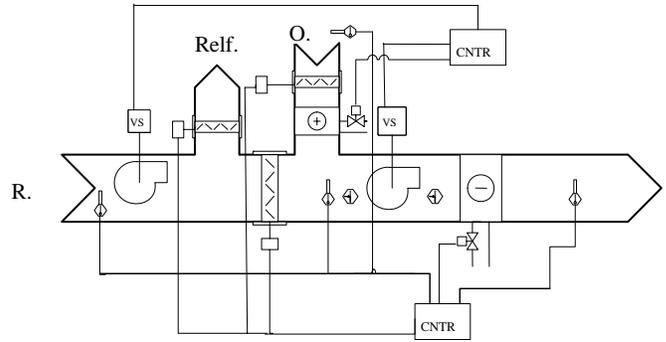


Figure 1: AHU1, AHU2, AHU3 and AHU4 schematic

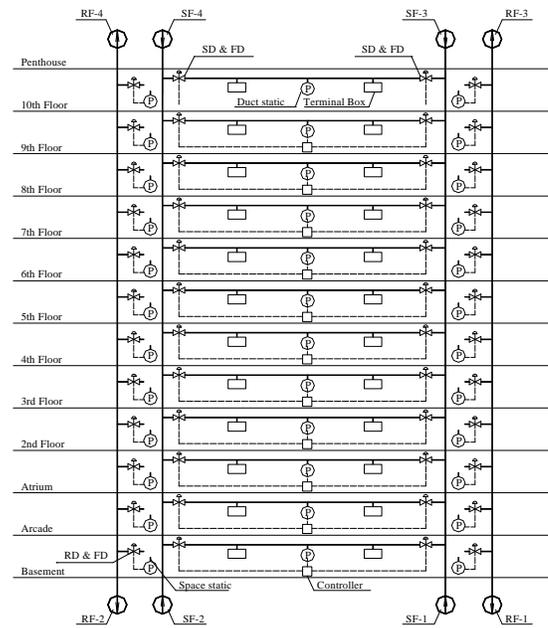
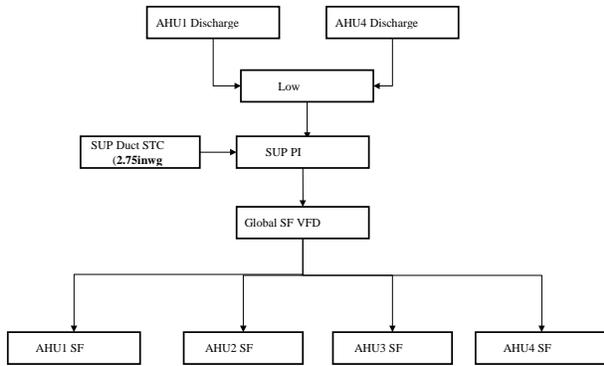


Figure 2: Building main air conditioning system diagram

### 1. Supply fan control

#### Existing control

The supply fan VFD speed was modulated to maintain the main duct static pressure set point, which was set at 2.75" water gauge. Figure 3 sketches the control logic. Under the existing control, most terminal box dampers were partially open at the full cooling condition.



**Figure 3:** supply fan VFD control diagram of AHU1, AHU2, AHU3 and AHU4

**Improved control**

In this building, the branch dampers on each floor also serve as fire dampers. To reduce the system resistance, fully open the supply and return duct branch dampers when there is no fire alarm.

After implementing the CC measures for branch dampers, the supply fan speed is modulated to maintain the most resistant duct branch static pressure set point instead of the main duct static pressure set point.

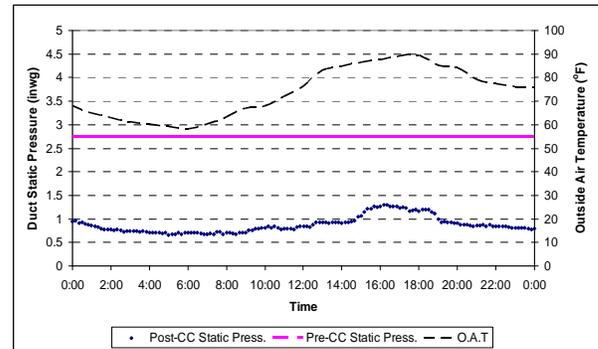
Prior to CC, high static pressure caused noise, overcooled the building and consumed more fan power. After implementing the new static pressure control algorithm, the main duct static pressure is significantly reduced while improving the indoor comfort. Figure 4a and Figure 4b show the main duct static pressure comparison pre-CC and post-CC on May 21, 2005 (Saturday) and May 24, 2005 (Tuesday).

The VFD speed on two similar days is compared before and after CC. Both days were Wednesday. Figure 5 shows the outside air temperature on the two days. Figure 6 shows the fan speed ratio as the actual speed to the full speed (60Hz). As shown in Figure 6, after CC, the reduction of main duct static pressure results in the remarkable decrease of fan speed. Based on fan affinity laws, there is a cubic relationship between fan speed and fan power, therefore, the reduction of fan speed provides considerable fan electricity savings.

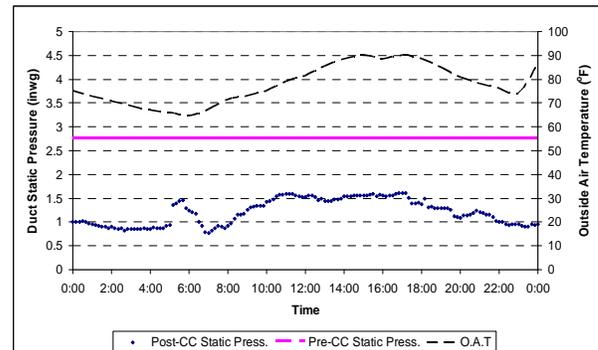
**2. Return fan control**

**Existing control**

Adjust return fan speed to maintain the mixed air chamber static pressure set point for AHU1 and AHU2. Keep constant speed for AHU3 and AHU4.

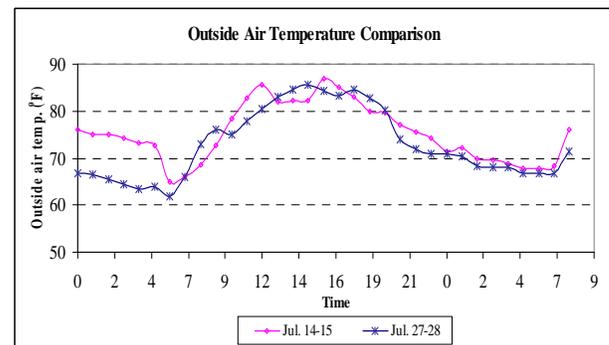


4a: main duct static pressure on May 21, 2005 (Saturday)

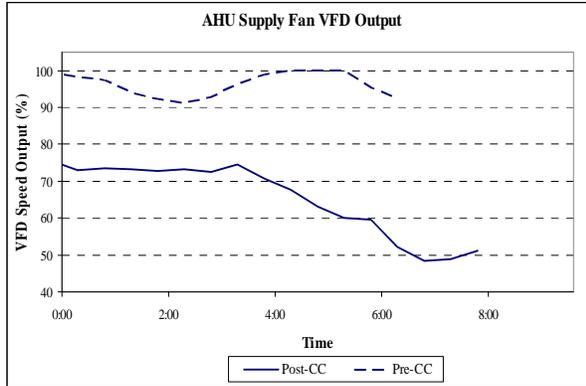


4b: main duct static pressure on May 24, 2005 (Tuesday)

**Figure 4:** Main duct static pressure comparison



**Figure 5:** Outdoor air temperature comparison (in year of 2004)



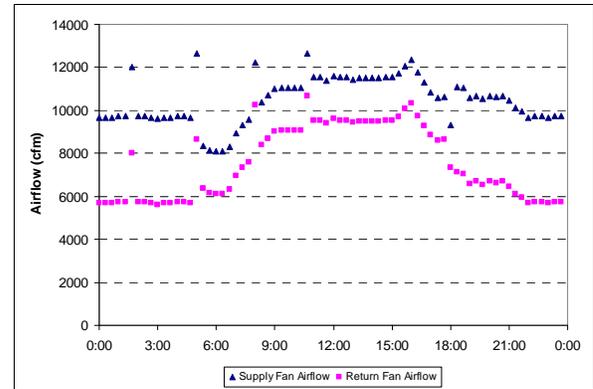
**Figure 6:** supply fan VFD speed output comparison

**Improved control**

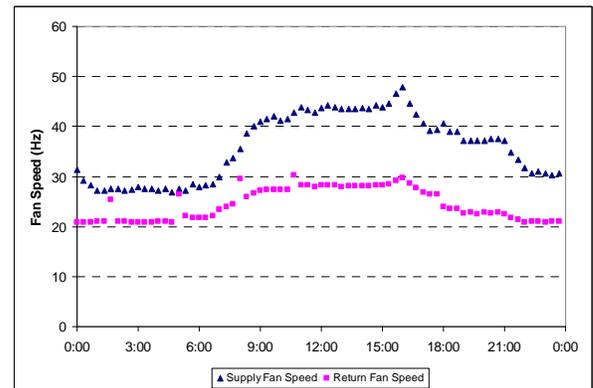
Fan air flow stations were installed in the units. The return fans are modulated to maintain the return fan speed set points. The return fan speed set points are calculated through the fan airflow stations based on the difference between supply airflow and building exhaust airflow [1-3]. The return fan airflow set point is the desired airflow to maintain building static pressure. Building static pressure depends on the airflow difference between supply air and return air. According to the rationale of a fan airflow station, the desired return fan airflow rate can be obtained through the fan airflow station [4]. This control strategy tightly links the return fan control with building static pressure and makes the return fan speed vary with the supply fan speed.

Figure 7a and 7b show the airflow and speed profiles for both the supply fan and return fan of AHU3 on January 27, 2005. In Figure 7a and 7b, at occupied hours, the airflow difference between supply fan and return fan was two times of that at unoccupied hours because all four handlers were on during occupied hours and only two air handlers were in operation during unoccupied hours.

Table 1 and Table 2 give the building pressure data on one day. The data indicate that control building static pressure using fan airflow stations achieves satisfactory results.



7a: hourly fan airflow



7b: hourly fan speed

**Figure 7:** Hourly fan airflow and speed on January 27, 2005

**Table 1:** Average pressure of each floor in inch of water gauge

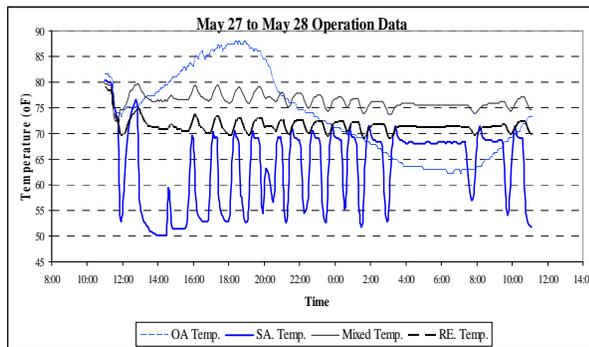
	Arcade	2nd	3rd	4th	5th	6th	7th	8th	9th
Pre-CC	0.016	0.058	0.027	0.027	0.025	0.016	0.022	0.022	0.016
After-CC	0.020	0.059	0.028	0.035	0.034	0.024	0.032	0.031	0.031

**Table 2:** Each floor pressure (inwg) distribution with the 90% confidence interval

	Arcade	2nd	3rd	4th	5th	6th	7th	8th	9th
Sample mean	0.020	0.059	0.028	0.035	0.034	0.024	0.032	0.031	0.031
Standard Deviation	0.006	0.013	0.015	0.019	0.019	0.020	0.020	0.020	0.020
Confidence lower limit	0.019	0.057	0.026	0.032	0.030	0.020	0.029	0.028	0.028
Confidence upper limit	0.021	0.061	0.031	0.038	0.037	0.027	0.036	0.035	0.035

## Constant Volume System, AHU5

Originally, AHU5 was a single duct single zone constant volume system serving the atrium space. Its operation period is 5:00 am to 1:30 pm on weekdays. Since most of the time, the original system worked at partial loads and supplied near constant air flow, the supply air temperature had great hunting. Figure 8 shows the operation data before CC on May 27 and May 28, 2004. As shown in the figure, the average supply air temperature was around 63°F. The higher supply air temperature resulted in the humidity problem, and the higher airflow rate consumed more fan power. After CC leading retrofit, it became a single duct, single zone variable volume system. Both the supply fan and the return fan were installed with VFDs.



**Figure 8:** May 27 to May 28, 2004 operation data (prior to CC)

### 1. Supply fan VFD control

#### *Existing control*

The constant speed fan was used.

#### *Improved control*

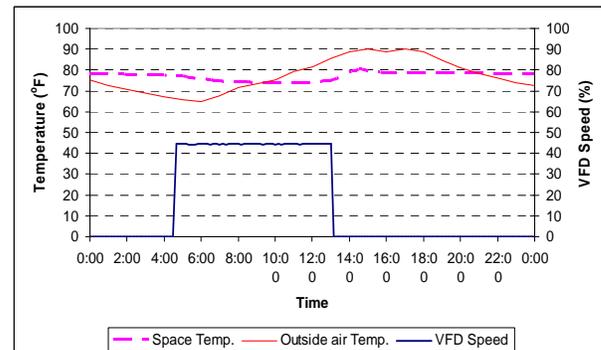
There are three kinds of mode to control the supply fan VFD speed, heating fan mode, cooling fan mode and normal fan mode.

In normal fan mode, the supply fan works at its adjustable minimum speed.

When supply air temperature is lower than its low limit and supply fan is running at minimum speed for ten minutes, fan mode switches from normal mode to cooling fan mode. In cooling fan mode, adjust supply fan VFD speed between maximum speed and minimum speed to maintain the space cooling temperature set point.

When supply air temperature is higher than 130°F and fan runs at its minimum speed, the fan mode switches from normal mode to heating fan mode. In heating fan mode, adjust the supply fan VFD speed between its maximum and minimum speed to maintain the space heating temperature set point.

After the installation of VFD, there is significant reduction of the fan speed to maintain the same space temperature. Figure 9 shows the normal fan mode speed and the space temperature.



**Figure 9:** space temperature and supply fan VFD speed on May 24, 2005

## 2. Supply air temperature control

#### *Existing control*

The supply air temperature is controlled to maintain the space temperature set point by modulating the heating valve, cooling valve or economizer dampers.

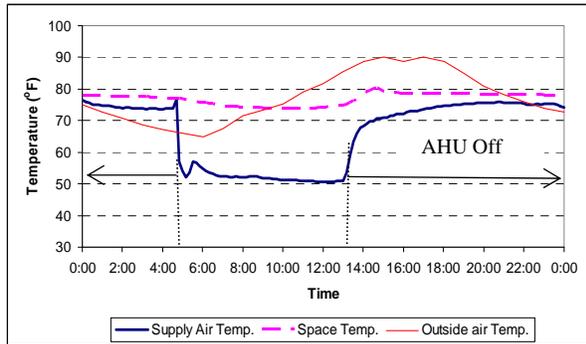
#### *Improved control*

The supply air temperature set point is 55°F in cooling fan mode. The supply air temperature set point is 130°F in heating fan mode. The supply air temperature set point is reset to maintain the space temperature set point in normal fan mode.

In cooling fan mode, modulate the cooling control valve to maintain the supply air temperature set point.

In normal fan mode, modulate the heating control valve, economizer dampers and the cooling control valve in sequence to maintain space temperature set point. In heating fan mode, modulate the heating control valve to maintain the supply air temperature set point.

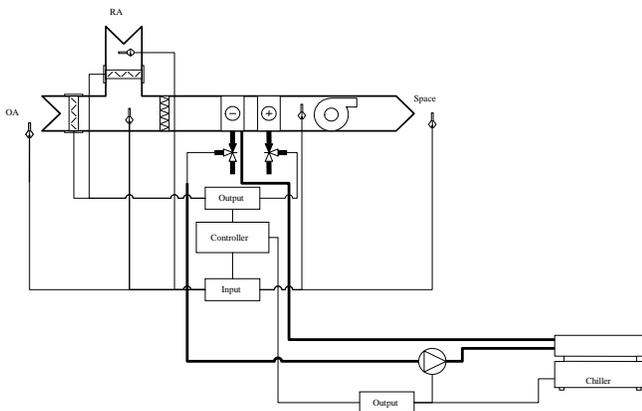
After the system is retrofitted to a VAV system, the supply air temperature oscillation is reduced. Figure 10 shows the supply air temperature profile on May 24, 2005. In the figure, the space temperature set point is 74°F. It can be seen the set point is maintained well when the system was running from 5:00 to 13:00, and supply air temperature is almost no hunting.



**Figure 10:** AHU5 supply air temperature on May 24, 2005

### Constant Volume System, AHU6

AHU6 system serves west building executive level interior zone offices. It is a single duct single zone constant volume system. This system has its own chilled water system to provide cooling to the handler. Figure 11 shows its schematic. The CC measure presented is the improvement of its economizer control.

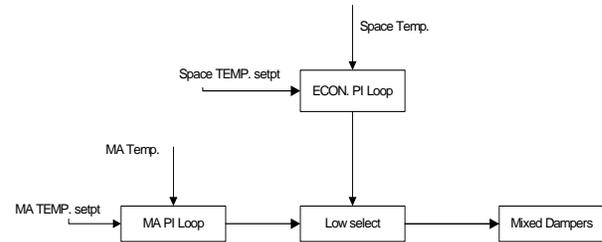


**Figure 11:** AHU6 system schematic

#### Existing control

Figure 12 shows the existing control logic. With the logic, the economizer cannot maximize free cooling if the mixed air set point input through EMS does not match the supply air temperature. For example, the

existing mixed air temperature set point is 70°F, the desired supply air temperature is around 60°F, and outside air temperature is 60°F; to maximize free cooling, the outside air damper should be fully opened based on the control signal from the economizer function loop; however, the control signal from the mixed air loop may be 40%. Eventually, the control signal to the mixed air dampers will be 40% after lower selecting, which leads to the outside air damper only 40% open, and mechanical cooling is required.



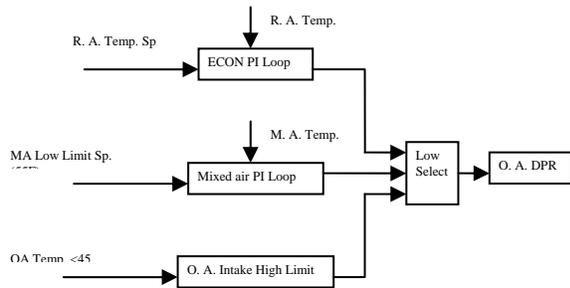
**Figure 12:** pre-cc mixed air damper control logic

The return air inlet is on the upper part of the air handler mixed air chamber, and the outside air inlet is on the side of the mixed air chamber. The mixed air chamber is too small to mix the return air and outside air well. An ordinary RTD sensor instead of an average temperature sensor is used to measure the mixed air temperature. The above factors resulted in (1) the outside air and return air can not mixed well before going through coils; (2) the mixed air temperature sensor can not reflect the actual mixed air temperature. Under these conditions, when outside air temperature is lower than the freezestat low limit, the air handler will trip very easily at early morning when occupants increase in the early winter morning if the outside air damper widely open due to large economizer signal. To avoid tripping the air handler, the existing mixed air temperature was set to 70°F by the operator which prevented the use of free cooling.

#### Improved control

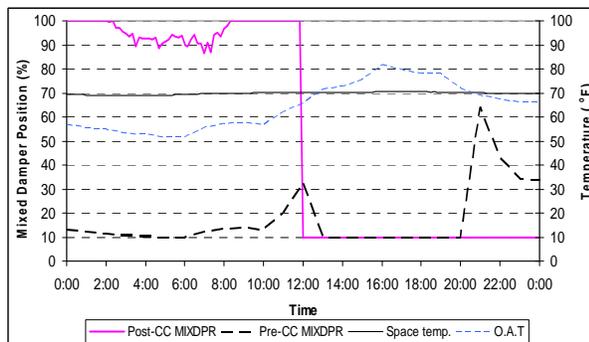
Figure 13 depicts the improved control logic. In the new control logic, when outside air temperature is lower than 45°F, based on site test, an outside air damper position high limit is added to the original control. The limit is reset based on outside air temperature from 0°F to 45°F. The economizer control logic is summarized below. Modulate the economizer dampers between its minimum and fully open position to maintain the space temperature set point. The outside air damper position should satisfy both the following conditions:

- The damper position should satisfy that mixed air temperature is not below the lowest mixed air temperature set point.
- The damper position should not exceed the damper position high limit when outside air temperature is below 45°F.



**Figure 13:** post-cc mixed air damper control logic

This CC measure reduces operator’s labor and air handler trip possibility; meanwhile, avoids wasting mechanical cooling. Figure 14 shows the outside air damper position comparison. After CC, the economizer working range is greatly increased. The increase of the economizer working range decreases the amount of mechanical cooling.



**Figure 14:** Outside air damper position comparison between pre-CC and post-CC

## Chilled Water System

### 1. Chillers Start and Stop

Prior to CC, the building cooling energy was provided by both the two ice making chillers and the rotary chiller. The total capacity of the ice-making chillers is 320 Ton. The capacity of the rotary chiller is 200 Ton.

Three ice tanks with a total capacity of 2850 Ton-hours were installed. The ice pumps circulate ice

water between the ice-storage tanks and the heat exchanger. The warm water from air handler coils is cooled down in the heat exchanger.

#### Existing control

The ice-storage system provides cooling all year long; in addition, under the following conditions, the rotary chiller is also on.

- When outside air temperature is higher than 70°F;
- When the average ice-level is lower than 15%;
- If chilled water supply temperature (the water temperature after heat exchanger to end users) exceeds 47°F;
- If the ice-storage unit is not at discharging cycle;
- Ice water temperature exceeds 42°F.

Based on observation and trending data, in summer, the rotary chiller ran at most of time while the ice-pump was running.

#### Improved control

Data indicate rotary chiller itself had enough capacity to serve the building when outside air temperature was below 95°F. After CC, the rotary chiller works as the primary chiller and ice system works as the back-up.

The rotary chiller is enabled based on outside air temperature and occupancy schedules. The ice-making chillers will be on when all of the following are true:

- Chilled water system is enabled;
- Ice-tank average level is lower than its low limit;
- At scheduled off-peak hours ( from 6:00 pm to 12:00 am)

This measure simplifies the operation and save electricity energy.

### 2. Secondary Chilled Water Pump Control

The chilled water system is a primary-secondary system. The secondary loop is a variable volume system. A variable frequency drive was installed on the lead pump.

#### Existing control

The secondary pump speed was modulated to maintain the constant secondary loop differential pressure set point (8 psi).

#### Improved control

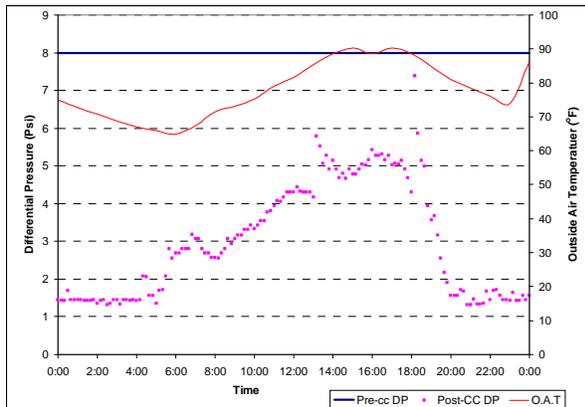
For this system, since the pump runs at partial loads most time, keeping constant differential pressure is not energy efficient. After CC, the secondary pump is modulated to maintain the most resistant air handler supply air temperature set point when the air handler is on. To minimize the pump energy consumption, the cooling valves will automatically open as wide as possible.

Figure 15 shows the water loop differential pressure at the most resistant loop. As shown in the figure, at most of the time, the loop differential pressure is lower than 8 psi. The pressure difference between pre-cc and post-cc implies the pump power savings. Modulating VFD speed to maintain the supply air temperature of the most resistant water loop and making the valve in the loop fully open can reduce pump power at partial loads; consequently, save electrical energy.

### Energy savings

Figure 16 and Figure 17 give the entire building electricity consumption comparison. Since there is no dedicated power meter for HVAC systems, the electricity consumption data included lighting, equipment and HVAC systems consumption.

After CC in a nine-month period, the electricity demand was decreased by 18.3%; and the total electricity consumption savings was 1,177,857 kwh



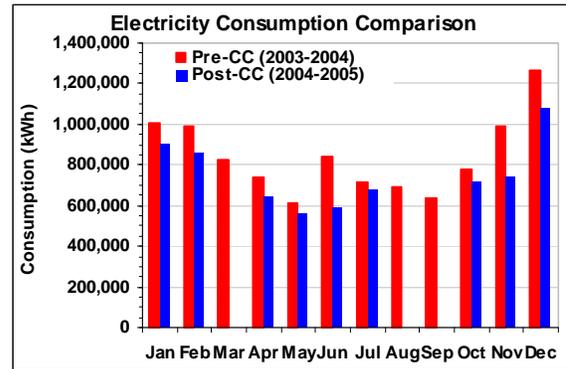
**Figure 15:** May 24, 2005 (Tuesday) the most resistant water loop differential pressure

### Conclusions

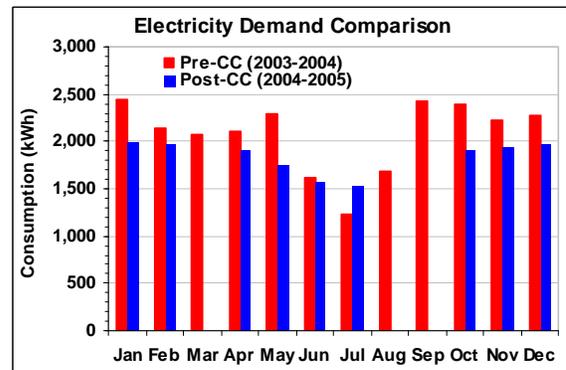
The implementation of CC measures positively influenced this office building on several aspects. The building indoor comfort was improved. Fan airflow station as an alternative building pressure control method provided satisfactory results to track

supply fan airflow. The fan and pump speed was reduced. The free cooling operation range was maximized.

The electricity demand was decreased by 18%. The electricity consumption was reduced by 1,177,857 kWh in nine months.



**Figure 16:** Electricity consumption comparison



**Figure 17:** Electricity demand comparison

### References

- [1] Liu, M, 2003, "Variable Speed Drive Volumetric Tracking (VSDVT) for Airflow Control in Variable Air Volume Systems," Journal of Solar Energy Engineering, Vol.125, pp.318-323.
- [2] Yuill, P.D., Redmann, N.K and Liu. M, 2003, "Development of A Fan Airflow Station for Airflow Control in VAV Systems," Proceedings of ISEC 2003.
- [3] Liu, M, G. Liu, Joo, I, Song, L. and Wang, G. 2004, "Development of In-situ Fan Curve Measurement for VAV AHU System," Proceedings of ISEC 2004.
- [4] Zheng, B., Pang, X. and Liu, M., 2005, "Using a Fan Airflow Station to Control Building Static Pressure," Proceedings of ISEC 2005.