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Case Study of Continuous Commissioning® in an Office Building

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Abstract: The case study building is a two-story office facility with total floor area of 31000 square feet, of which most is open space office. The HVAC system operates 24 hours a day, 7 days a week. An energy audit was conducted to evaluate the building's energy performance and identify potential cost-effective energy saving opportunities. Specific Continuous Commissioning® measures were implemented in the building. Indoor comfort was improved and the energy report shows a reduction in electricity consumption by 33.2% and gas consumption by 51.2%. This paper introduces the Continuous Commissioning® process and presents detailed information on the Continuous Commissioning® implementation in the case study building.

Introduction

"Continuous Commissioning® (CC®) 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]. HVAC researchers and engineers are motivated by an ultimate goal to achieve the best building performance with minimum energy. Some exciting results have been achieved with CC® [2-4].

In 1988, a group of ASHRAE members proposed building commissioning to ensure that system performance met design specifications. In 1992, the Energy Systems Laboratory at Texas A&M University examined system operations in a number of newly retrofitted buildings and found that optimizing the system can double energy savings and improve building comfort [Liu et al.]. In 1996, the CC® process was first developed. While most commissioning processes focus on bringing building operation performance to the original design intent, CC® is different in that it focuses on optimizing HVAC system operation and control for existing building conditions. CC® has produced typical savings of 20% with payback fewer than three years (often 1-2 years) in more than 130 large buildings [1].

The CC® process typically consists of two phases. The first phase involves conducting an energy audit, which evaluates the building's energy performance and identifies potential cost-effective energy saving opportunities in order to develop the project scope. The second phase is to implement CC® measures and verify the building energy and comfort performance.

The case study building is a two-story office facility with a floor area of 31000 square feet, most of which is an open space office area. The system operates 24 hours a day, 7 days a week. DDC pressure-independent VAV terminal units with reheat coil are used in this building to provide variable air volume and reheat capabilities to condition the space. One air handling unit with inlet guide vanes at the 40 HP supply fan is installed to serve the entire office space. The chilled water system is equipped with one 140 ton air-cooled screw chiller and one constant speed chilled water pump. One 1678 MBTH gas boiler provides hot water to the AHU heating coil and the reheat coils of the terminal boxes. The heating system is equipped with a constant speed hot water pump, and there are three-way control valves in a hot water loop. The automatic control systems use DDC control, with EMS software installed in the central computer.

This paper presents the results of the second phase of this project. In this phase, current CC® technologies were introduced, and energy consumption before and after CC® implementation was compared.

CC® Measures and Implementation

Major CC® measures included the following: 1) calibration and optimization of the terminal box minimum air flow to reduce the cooling and heating energy consumption; 2) development and implementation of a supply air temperature reset schedule for the AHU to reduce heating energy consumption; 3) setting up the fan air flow station to control the relief fan so that the building can be pressurized better; 4) Improvement of the economizer control to minimize the energy consumption meanwhile ensure the indoor air quality; and 5) implementation of a static pressure reset schedule to reduce the supply fan power. The remainder of this paper provides a general summary of the CC® measures and presents some preliminary results.

Terminal Boxes

Figure 1 shows the schematic diagram of a typical terminal box used in this building. It is a pressure-independent DDC single-duct VAV terminal box. A flow station is installed at the inlet of the terminal box. As the room temperature deviates from its set point, the DDC controller will calculate a needed air flow set point and modulate the zone damper to maintain the air flow between the minimum air flow and the maximum air flow. After it reaches the minimum air flow, the reheat coil will be modulated to maintain the room temperature set point.

Existing operation: Among all the terminal boxes, the ratios of the minimum air flow to the maximum air flow vary greatly from 5% to 50%, even to 100%; the average ratio is about 30%. Because minimum air flow is high in some terminal boxes in summer, air flow through these boxes needs to be reheated, which leads to extra heating and cooling.

CC® Improvement: For the interior zone, set minimum air flow to zero independent on the occupancy, and disable the heating valves. For the exterior zone, apply a minimum air flow factor of 0.2 CFM/ft^2 to most of the boxes, and 0.25CFM/ft^2 to the worse-case boxes.

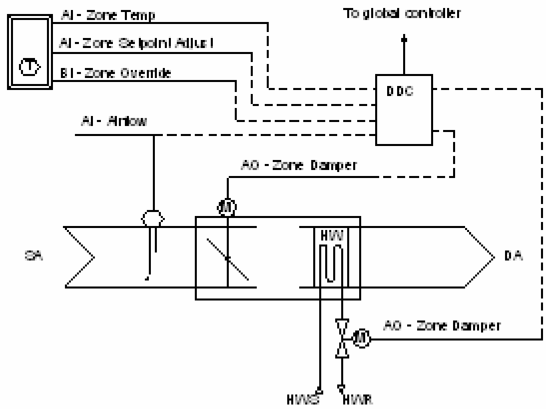


Figure 1 Schematic diagram of terminal box

Results: After CC® implementation, the cooling, reheat energy and fan power were reduced, while the indoor air quality was maintained at a satisfactory level. Before CC®, the measured average CO_2 level was 500PPM. After CC®, the measured CO_2 level was below 600PPM under the same time period and building occupancy conditions.

Air Handling Unit

One air handling unit with inlet guide vanes at supply fan is installed to serve the entire office space, as shown in Figure 2. It works 24 hours a day, 7 days a week. The inlet guide vanes are modulated to maintain a constant duct static pressure set point. The building pressure was maintained by modulating the relief air damper. The air handling unit has temperature economizer control. The economizer locks out at an outdoor air temperature of 35^{o}F. When the outside air temperature is lower than the set point, the air handling unit takes in the minimum outdoor air flow. In this building, the minimum outside air damper position is 25%. The supply air temperature set point is 53^{o}F when outdoor air temperature is higher than 55^{o}F and 58^{o}F when the outdoor air temperature is below 55^{o}F. The AHU has freezestat safety control to shut down the supply fan when low mixed air temperature is detected. The CC® measures and trouble-shooting applied to the AHU are presented in the subsections below.

Static Pressure Control

Existing control: A year-round constant main duct static pressure set point was used. The inlet guide vanes were modulated to maintain the set point

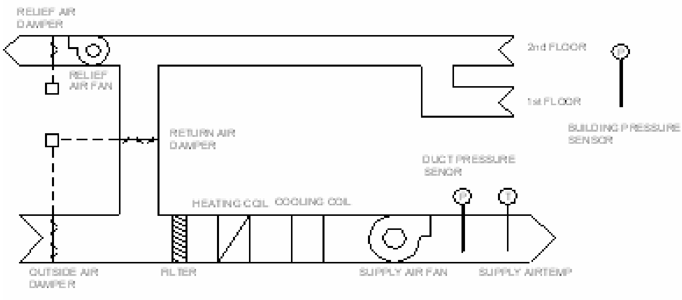


Figure 2 Schematic diagram of AHU

CC® Improvement: Set the inlet guide vanes to be fully open, and install a VFD for the supply air fan. Apply a supply duct static pressure reset schedule. The dynamic static pressure set point is determined by the critical zone terminal box cooling signal. When the worse-case terminal box cooling signal is lower than 85%, decrease the current static pressure set point by 0.03 in.w.g. When the worse-case terminal box cooling signal is higher than 95%, increase the static pressure by 0.03 in.w.g. If the cooling signal is between 85% and 95%, keep current static pressure set point. This should be repeated every 20 minutes. The initial static pressure set point is 0.5 in.w.g.

Results: The fan power was reduced. Figure 3 shows a 24-hour-period static pressure reset profile, and Figure 4 compares a 24-hour-period supply fan operation before and after CC®. As shown in Figure 4, the fan speed after CC® is reduced significantly.

Relief fan control / Building pressure control

Existing control: The relief air damper is modulated to maintain the building pressure set point. The inherent draw back of this control method results in some building pressure control problems in this building. During economizer period, when outside air intake is large, especially 100% outside air intake, some exterior doors can not fully close by themselves, although the building pressure sensor reading is around 0.02 in.w.g.

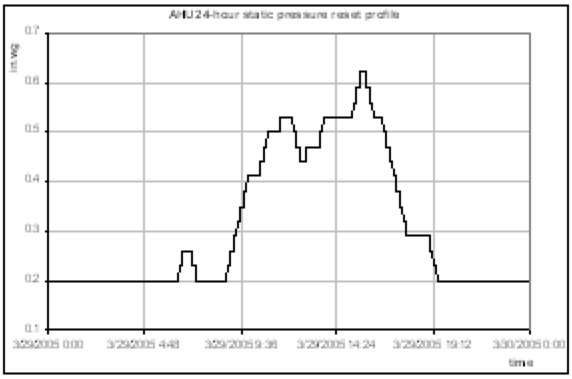


Figure 3 AHU static pressure reset profile

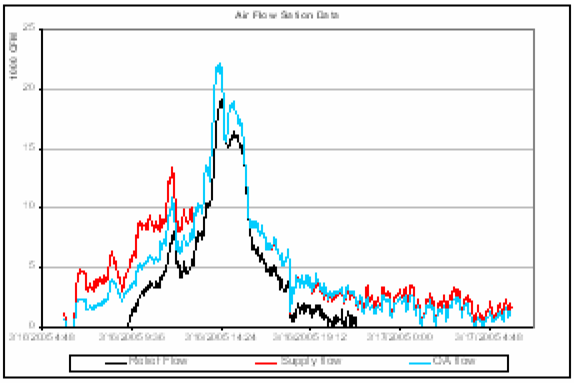


Figure 5 Fan power air flow station operation data

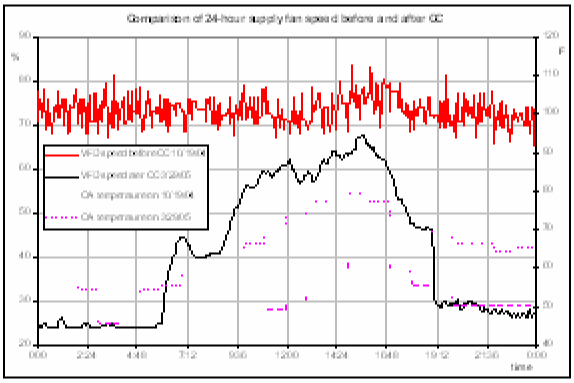


Figure 4 Comparison of supply fan speed

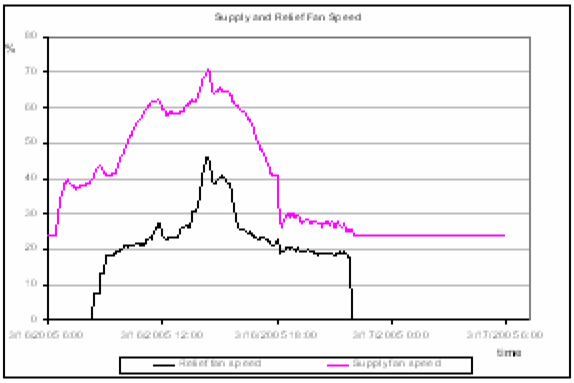


Figure 6 Supply and relief fan speed after CC®

CC® Improvement: Convert the relief air damper to On/Off control in response to the relief fan start/stop command. Install VFD for the relief fan and modulate the relief fan to maintain the relief air flow set point, which is calculated by:

$$Q_{r,sp} = Q_{OA} - 2300 \qquad [1]$$

where $Q_{r,sp}$ = Relief air flow set point, CFM,

Q_{OA} = Outside air flow, CFM.

The fan air flow station gives the supply and relief fan air flow. For detailed information about the relief fan control and fan air flow station, refer to [5~8].

Results: Figures 5 and 6 provide 24-hour AHU operation data for the fan power air flow station. As can be seen in these two figures, the relief fan can track the supply fan trend well. Investigation of the building pressure tells that the building pressure is acceptable.

Supply air temperature control

Existing control: The supply air temperature set point is 53°F when outdoor air temperature is higher than 55°F. When the outdoor air temperature is below 55°F, it is raised to 58°F. The heating coil, economizer and cooling coil are controlled in

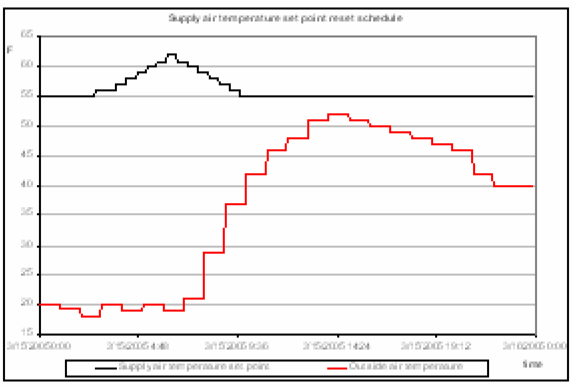


Figure 7 Supply air temperature set point reset schedule

Results: Both fan power and exterior zone boxes reheat can be saved, while satisfying the interior zone temperature. The supply air

sequence to maintain the supply air temperature set point.

CC® Improvement: Apply a supply air temperature set point reset schedule. It should be activated based on the supply fan air flow. The basic philosophy is that a percentage threshold value (15% is used here) of the design fan air flow is applied to determine whether the building is cooling-dominated. If it is cooling dominated, the supply air temperature set point will be maintained at the low limit of 55˚F. if not, the supply air temperature will be increased to maintain the fan air flow not less than 10% of the design air flow until the high limit of 68˚F is reached, and at the same time, the interior zone space temperature set point is maintained. The detailed algorithm is presented below.

With the priority of ensuring the temperature of the space serving by the interior zone terminal boxes, the dynamic supply air temperature set point is determined by maintaining the second floor worst-case terminal box heating signal at zero. When this box heating signal is not zero, increase the supply air temperature set point by 1˚F every 15 minutes until the box heating demand is zero. Before increasing the supply air temperature, check the interior zone space temperature; if space temperature is 2˚F higher than its set point, decrease the supply air temperature set point by 1˚F until the space temperature is not 2˚F higher than the set point.

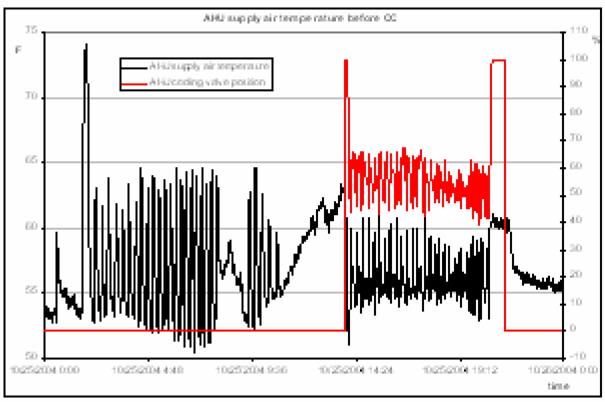


Figure 8 Supply air temperature before CC®

temperature set point responds to changes in the building load mode. As can be seen in Figure 7, the supply air temperature increases in the early morning in response to the low outside air temperature. It decreases when outside air temperature increases.

Trouble-Shooting of supply air temperature hunting

Existing problem: Figure 8 shows the supply air temperature hunting before CC®. After further investigation of the existing control sequences for the AHU, it was found that the cooling valve, economizer and heating valve were controlled by the same PI controller module, as shown in Figure 9. Although this algorithm can interlink these three components and effectively prevent them from simultaneously heating and cooling, it results in hunting of the supply air temperature because these three components have totally different dynamic characteristics. Another fact arising from this control algorithm is that the economizer locks out at an outdoor air temperature of 35°F. If the economizer operates below 35°F, because of the poor control of the economizer dampers, the freezstat will send out alarms and shut down the unit frequently, and will also put the cooling coil at the risk of freezing.

CC® Improvement: The control loop shown in Figure 9 is separated into three loops, i.e. each component has a dedicated PI control module. These three PI control modules can be tuned individually based on their own dynamic characteristics. They are also interlinked to prevent simultaneously heating and cooling.

Results: Stable supply air temperature control was achieved as can be seen in Figure 10. The economizer operating period is extended, and the following subsection gives the details.

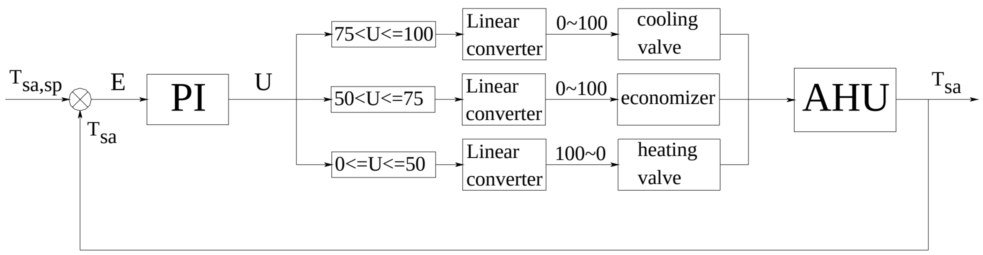


Figure 9 AHU control loop before CC®

Economizer

Existing control: The economizer locks out at an outdoor air temperature of 35°F. When the outside air temperature is lower than the set point, the outside air damper will be in the minimum position of 25%.

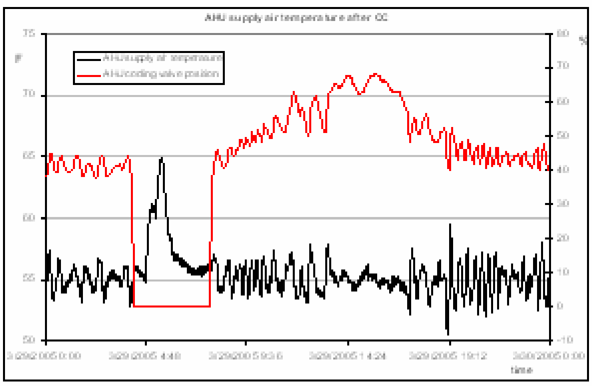


Figure 10 Supply air temperature after CC®

CC® Improvement: When outside air temperature is lower than 65˚F, the economizer is enabled. When economizer is disabled, at occupied hours, the outside air damper minimum position is 10%; at unoccupied hours, the minimum outdoor air damper position is zero. Table 1 gives the outside air damper control sequence when the AHU is under economizer mode.

Table 1 OA damper under economizer mode

Outside air temperature	Outside air damper operation
55˚F<T_{OA}≤65˚F	Fully open
T_{OA}≤55˚F	Modulate to maintain supply air temperature set point determined by the supply air temperature control within 10% ~100% position.

Results: The improved schedule greatly reduces the cooling energy consumption by the chiller.

Chilled water system

The chilled water system is equipped with one air-cooled screw chiller and one constant speed chilled water pump. The chiller capacity is 140 tons, and the nominal supply and return temperatures are 45ºF and 57ºF, respectively. Figure 11 depicts a schematic diagram of chilled water system. Existing control: The chiller operates at constant water flow and constant chilled water supply temperature. The chiller-enabled outside air temperature is 40ºF.

CC® Improvement: Enable the chiller when outside air temperature is beyond 55ºF, corresponding to the economizer operation.

Results: The CC® improvement saves the chiller electricity consumption and extends the life-span of the chiller.

Conclusions

The implemented control sequences can significantly improve building comfort and reduce HVAC energy cost. As can be seen in Figure 12, both the electricity and gas consumption are substantially reduced. The electricity consumption is reduced by 33.2% and gas consumption is reduced by 51.2%, based on 6 months (5 months for gas) of utility data since project completion.

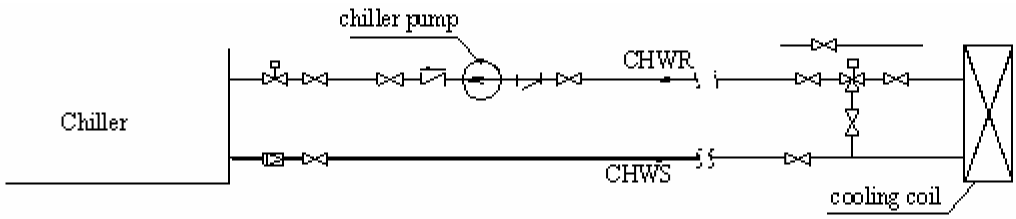
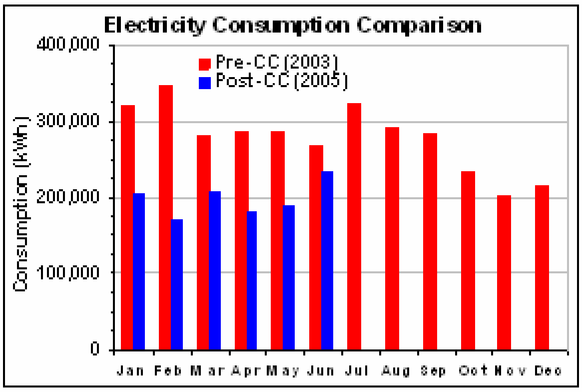


Figure 11 Schematic diagram of chilled water system



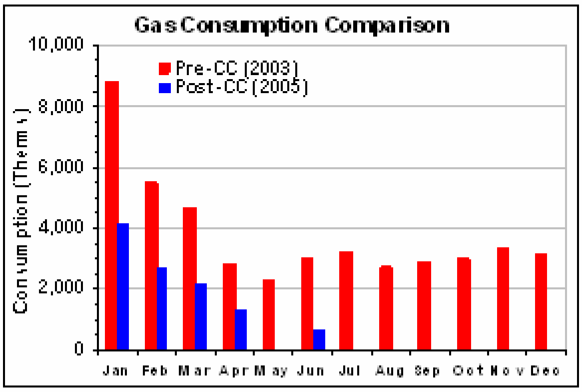


Figure 12 Energy consumption before and after CC®

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