Continuous Commissioning® of a Single Fan Dual Duct System in an Office Building

D. Dong
Energy Systems Laboratory
University of Nebraska−Lincoln
243 1110 South 67th Street
Omaha, NE 68182
ddong@unlnotes.unl.edu

M. Liu, Ph.D., P.E
Energy Systems Laboratory
University of Nebraska−Lincoln
206C 1110 South 67th Street
Omaha, NE 68182
mliu@unlnotes.unl.edu

Jinrong Wang, P.E
Omaha Public Power
District

ABSTRACT

This paper discussed the Continuous Commissioning® process of an office building at Nebraska. The building was built in 1970, serving by a single fan dual duct constant air volume system. The terminal boxes are independent boxes, which are operated in CAV modes. The building continuous commissioning was started on June 2004. It is being commissioned following the approaches used by the Energy Systems Laboratory at Nebraska University, which in this case include installing VFD on the supply fan and return fan, updating all terminal boxes from CAV modes into VAV modes, resetting duct static pressure and supply air temperature, optimizing outside air intake and installing VFDs on chiller and boiler pumps, and optimizing the control of chillers, boilers and pumps.

This paper presents procedures, implementations and detailed descriptions of control schedules and operation comparisons before CC® and after CC® for AHU, as well as the chillers and boilers.

INTRODUCTION

More and more studies have been investigated to improve building HVAC system operations and reduce energy costs. Researchers have also developed a number of new technologies to improve the existing building energy and comfort performance [1~2], especially for dual duct AHU system operation [3~15]. These technologies can be applied to actual HVAC systems during a Continuous Commissioning® process [16]. The applications of recent technologies are introduced in this paper. The technologies involve dual duct static pressure reset control, outside air intake in the morning, variable chilled water system control, and variable hot water system control.

CC® has been achieving superior energy and comfort performance by identifying potential energy retrofits, detecting declined efficiency equipment, updating mal-functional components, and optimizing the system control sequences. It is believed that new cost-effective system control technologies and system trouble shooting can be applied to improve comfort issue and enhance the building energy performance. This paper demonstrates the energy savings and performance improvements of optimizing HVAC system control sequences in buildings through a case study. This paper presents the case study facility information, existing and optimal control schedules, and measured building performance improvement and energy savings before CC® and after CC®.

FACILITY INFORMATION

The case study building, located at Omaha, Nebraska, was built in 1970. The 5-story office building has a floor area of about 71,000 square feet, which is shown in Figure 1. The typical office hours are from 7:00am to 6:00pm during the weekdays. The AHU system is located in the penthouse and serves the entire building. The system operates from 5:00am to 7:50pm during the weekdays, from 6:00am to 5:00pm on Saturday and from 7:00am to 2:00pm on Sunday.

Figure 1: Case Study Building

The building has a total of 216 dual duct independent terminal boxes with air flow stations. The boxes are used for both interior and exterior zones. Pneumatic controllers and actuators are installed.

A dual-duct single fan AHU serves the building. The AHU is equipped with two constant speed supply fans (125hp each) and two return air fans (100hp each).

The central plant consists of chilled water and hot water systems. The chilled water system includes three air cooled chillers (110 ton each), two constant speed chilled water pumps (50 hp each) and ten air-cooled cooling tower fans (1 hp each). The hot water system consists of three hot water boilers (2,010 MBH each), two constant speed hot water pumps (20 hp each) and two constant speed perimeter pumps (1.5 hp each). One hot water loop supplies hot water to the heat coil of AHU. The other hot water loop supplies hot water to the perimeter heating system, which is driven by perimeter pumps if outdoor air temperature is below the setting.

OPTIMAL CONTROL SCHEDULES

The control schedule optimization includes terminal boxes, AHU, chillers, and boilers. All the pneumatic controls for the AHU system, except box controllers, are updated to DDC controls before the optimal control sequences are implemented. In the following sub-section, the existing control schedules are presented and analyzed before the optimal schedules are introduced.

VAV boxes

There are a total of 216 terminal boxes with pneumatic controllers in the building. Figure 2 shows the schematic of the dual duct mixing box. The space thermostat resets the control point of hot duct velocity controller inversely relative to space temperature. The hot duct velocity controller positions the hot duct damper to maintain hot duct airflow at its set point. The discharge air velocity controller positions the cold duct damper to maintain discharge airflow at its set point. The box was originally operating in a constant air volume mode, which caused simultaneous heating and cooling consumption when the building load was less than the design load. However, the variable air volume box system can reduce fan power, heating and cooling consumption significantly under partial load conditions.

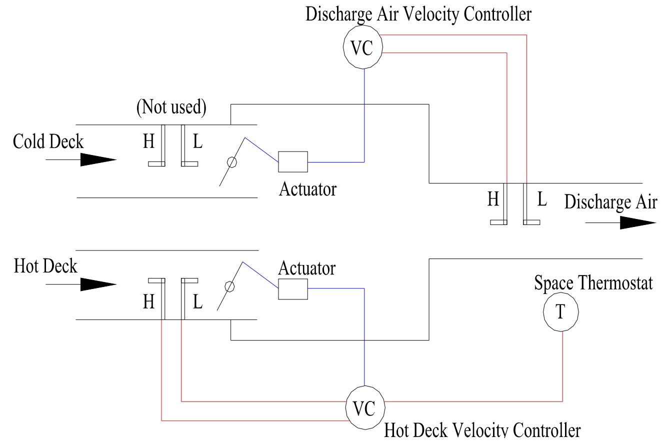

Figure 2: Dual Duct Terminal Box

The optimal control sequence takes advantage of the existing box configuration, which converts box performance from a CAV mode into a VAV mode without retrofitting. The two velocity controllers were reset in the same operation range of 7psi — 13psi in the existing control sequence. The controllers are reset respectively in the different action ranges in the optimal control sequence. The hot deck velocity controller is reset to response at a range of 3psi — 8psi when receiving a pneumatic pressure signal from the thermostat. The discharge air velocity controller is reset at an operation range of 9psi — 14psi when the hot deck damper is fully closed. The previous settings and modifications for velocity controllers are shown in Figure 2. The minimum airflow setting for exterior zone is reset to 0.3 cfm/ft^2 (cold air) and reset to 0.0 cfm/ft^2 for interior zone.

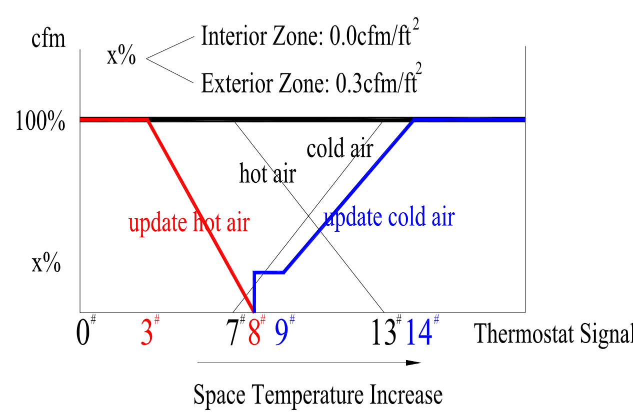

Figure 3: Terminal Box Performance

Air Handling Unit (AHU)

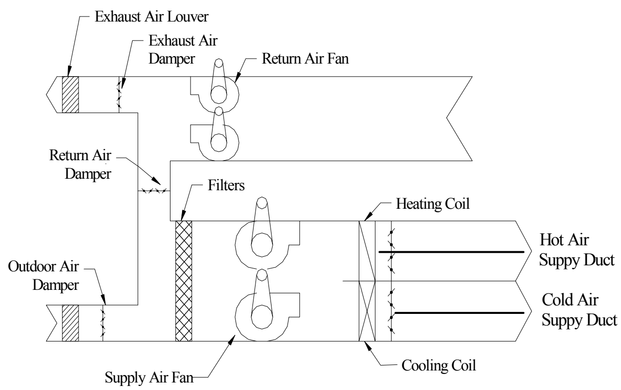

Figure 4: Schematic of AHU system

Figure 4 presents the schematic diagram of the air handling unit, which is a single fan dual duct constant volume system with two supply fans and two return fans serving the building. The supply fans were running at constant speeds. The discharge air static pressure was maintained as 3 inH_2O using inlet guide vanes (IGV). The cold and hot duct pressure sensors positioned the duct dampers to prevent duct pressure above their respective settings. The return fans were also running at constant speeds. The building pressure (0.025 inH_2O) was maintained by return fan IGV.

The hot duct air temperature was reset inversely to outside air temperature. The cold duct temperature was maintained at a constant set point (55°F).

The enthalpy economizer was implemented. When the outside air enthalpy was lower than 27Btu/lbm and outside temperature was below the setting (68°F), the economizer was activated. When economizer was disabled, the outside air damper was maintained at a minimum position.

The existing performance and control schedules have the following problems: (1) measured excessive airflow (2 cfm/ft^2) causes more energy consumption (2) noise problem and air leakage through box dampers due to high duct static pressure (3) excessive heating when constant cold deck temperature setting is maintained (4) simultaneous heating and cooling occurs when the hot deck is energized.

The optimal control schedules are developed to solve the above problems and improve the system energy efficiency. The dual duct single fan constant airflow system is converted to a VAV system by installing VFD on both supply and return fans. The duct dampers are modulated to maintain the duct static pressure at their set points and the supply fan VFD is also modulated to maintain at least one duct static pressure at its set point. If the cold deck static pressures are maintained by supply fan VFD, the cold deck dampers are fully open and hot duct static pressures are controlled by hot duct dampers, and vice versa.

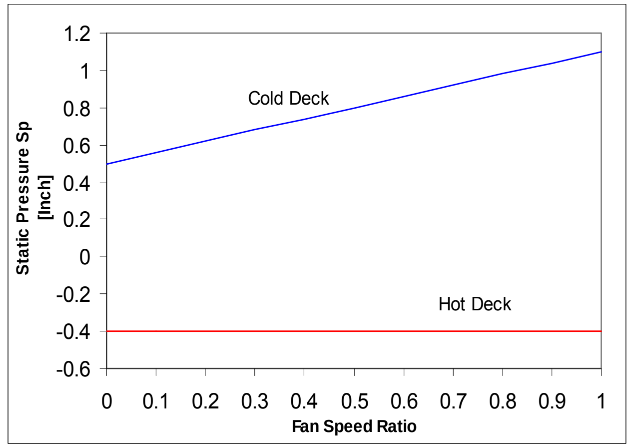

Figure 5: Cold and Hot deck static pressure set point when Toa ≥ 70 °F

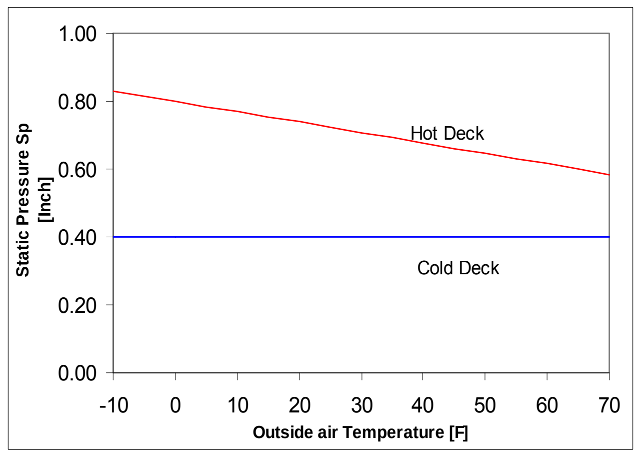

Figure 6: Cold and Hot deck static pressure set point when Toa < 70 °F

The optimal static pressure reset is developed. Figures 5 & 6 show the improved static pressure control schedule. The cold and hot duct static pressures are reset based on supply fan speed when outside air temperature is higher than 70°F (See Fig. 5). The cold and hot duct static pressures are reset based on outside air temperature when outside air temperature is lower than 70°F (See Fig. 6). The hot duct dampers are shut down during the summer period to reduce hot air leakage through terminal boxes. In order to warm up the building during the morning, the hot duct pressure is reset to 0.65 inH_2O when outside air temperature is below 66°F until 9:00 AM. Meanwhile, the hot duct pressure is reset to 0.80 inH_2O when outside air temperature is below 38°F until 10:00 AM. The optimal static pressure set-point schedule allows the damper to be fully open to save fan power and reduce noise level.

The return fan VFD is controlled to maintain building pressure, which measures the building pressure with reference to outdoor pressure. If the building is under-pressurized, the return fan slows down, the less return air can be drawn from the space, and less air is supplied to relief duct, and vice versa.

The hot duct air temperature is reset based on the outside air temperature and hot duct static pressure. The cold duct air temperature is kept constant (55°F) during summer and mild weather conditions, and the cold duct air temperature is reset to a higher setting (60°F) in winter (See Fig. 7). It saves heating consumption in the winter without sacrificing the space humidity level. It also saves chiller and pump power during the transition period operations.

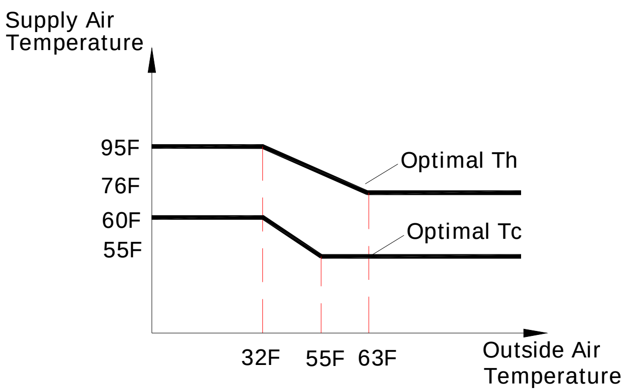

Figure 7: Cold and hot duct supply air temperature control schedule

The outside air intake is optimized based on typical day occupancy. Outside air is reset to zero during the early morning (unoccupied hours) before 8:00 AM. When outside air temperature is lower than 52ºF or higher than 74ºF, outside air damper is fully closed. When outside air enthalpy is higher than 27.2 Btu/lbm and outside air temperature is higher than 55 ºF, outside air damper is also fully closed. When outside air temperature is between 50ºF and 72ºF, the economizer control will be enabled. The economizer control is also enabled after 8:00 AM. This method can save unnecessary mechanical cooling and heating since the building needs to be cooled down in the summer morning and warmed up in the winter morning.

Chiller and chilled water loop

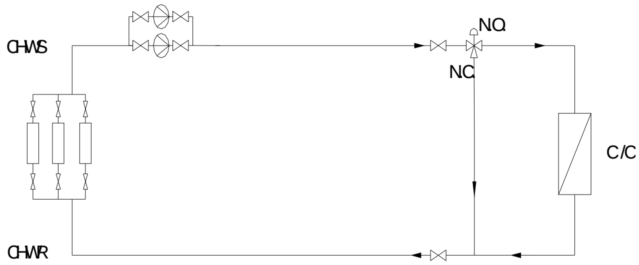

Figure 8: Schematic of the chilled water system

Figure 8 shows the chilled water system. The chilled water loop includes three rotary chillers with two water pumps, one of which is for back up use. The three way control valve through the cooling coil maintains chilled supply water temperature. There is no automatic isolation valve for each chiller. In the existing control schedule, all three chillers were enabled when the outside air temperature was higher than 55°F and disabled during unoccupied hours. The chilled water supply temperature set point was maintained as 44°F. The chilled water pump was running at a constant speed.

The existing chiller and chilled water pump system controls have the following problems: (1) excessive building by-pass flow, for example, the temperature difference between chilled water supply and return can be as low as 2°F, (2) low load operation of three chillers, and (3) excessive pump power consumption during a low cooling load.

To improve the existing chiller plant operation, the automatic isolation valves are installed for each chiller and optimal operation control schedules are developed. During occupied hours, the lead chiller is enabled when outside air temperature is higher than 57ºF and also mixed air temperature is higher than 59ºF. The chillers are operated according to the building load and divided into **1stChiller** mode, **2ndChiller** mode and **3rdChiller** mode. Each mode includes **LowLoad** module and **HighLoad** module. There are different minimum pump speed settings in the **LowLoad** modules of chillers.

When the lead chiller is enabled, the isolation control valve is open and the pump speed is maintained at a minimum speed (35%). The three way control valve is modulated to maintain cold deck air temperature. When the valve is fully open to the cooling coil and cold air temperature is higher than the set point plus the dead band, the chiller operation switches to a **LowLoad** module. If supply water temperature reaches maximum chilled water set point and cold air temperature is lower than the set point minus the dead band during a certain period, the three way control valve is activated to maintain cold air temperature. If outside air temperature is lower than 55ºF and mixed air temperature is lower than 57ºF, the lead chiller is disabled.

During a **LowLoad** module, the pump speed is maintained at a minimum speed. Chilled water temperature is modulated to maintain the cold deck air temperature. If supply water temperature reaches minimum chilled water set point (reset based on outside air temperature) and cold air temperature is higher than the set point plus the dead band during a certain period, the chiller operation switches to a **HighLoad** module.

During a **HighLoad** module, chilled water temperature is maintained at minimum chilled water temperature set point and pump VFD speed is controlled to maintain cold deck air temperature. If the VFD speed is less than minimum pump speed during a certain period, the chiller operation switches back to a **LowLoad** module.

If the VFD speed is higher than a certain value and cold air temperature is higher than the set point plus the dead band during a certain period, the chiller operation switches to a **2ndChiller** mode.

In the **2ndChiller** mode, isolation control valves are open and two chillers are on, performing the **LowLoad** and **HighLoad** module automatically based on cooling load conditions. The minimum pump speed is reset to a higher value (55%) when two chillers are on. If the VFD speed is higher than a certain setting and chilled air temperature is higher than the set point plus dead band during a certain period, each chiller operation switches to a **3rdChiller** mode. If chilled water temperature reaches maximum chilled water set point during a certain period, the second chiller is disabled and chiller operation switches back to a **HighLoad** module.

Under the **3rdChiller** mode, all isolation control valves are open and three chillers are on, performing the **LowLoad** and **HighLoad** module automatically based on cooling load conditions. The minimum pump speed is reset to a higher value (75%) when three chillers are on. If chilled water temperature reaches maximum chilled water set point during a certain period, the third chiller is disabled and each chiller operation switches back to a **HighLoad** module.

Boiler and hot water loop

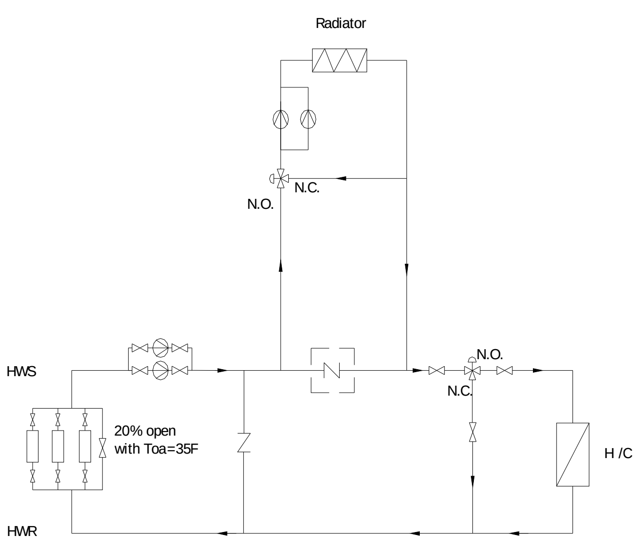

Figure 9: Hot water system

The boiler and hot water loop consists of three boilers, two hot water pumps for the primary loop and two pumps for the radiation perimeter hot water system. The three way control valve through the heating coil maintains hot deck air temperature. The three way control valve through the heating radiator maintains perimeter hot water supply temperature. The by-pass valve for boilers is manually opened and closed based on outside air temperature. The existing schedule turned boilers on when the outside air temperature was lower than 70°F. The hot water pump was running at a constant speed. The perimeter heat system was enabled when outside air was below 50°F and the perimeter supply water temperature is manually reset between 140°F and 150°F.

The existing boiler and hot water pump schedules have the following problems: (1) excessive building bypass flow, for example, the hot water supply temperature was the same as the return temperature most of the time under mild weather, (2) excessive boiler operation when outside air temperature is high, (3) excessive hot water pump energy consumption during a low heating load.

To reduce extra gas consumption and hot water pump operation, the hot water system is turned on when the outside air temperature is lower than 63ºF and the hot water system turned off when outside air temperature is higher than 66ºF. The by-pass valve for the boilers is shut down. The isolation valve is installed on each boiler and is always open for the lead boiler. The boilers are operated according to the building load and divided into **1stBoiler** mode, **2ndBoiler** mode and **3rdBoiler** mode. Each mode includes **LOWLOAD** module and **HIGHLOAD** module. If the hot water system is enabled, the lead boiler is enabled when hot deck temperature is lower than hot deck temperature set point minus the dead band.

When the lead boiler is enabled, the pump speed is maintained at a minimum speed (40%) and hot water temperature is maintained at 130°F. The three way control valve is modulated to maintain hot deck air temperature. When the valve is fully open to the heating coil and hot air temperature is lower than the set point minus the dead band, the boiler operation switches to a **LOWLOAD** module. If supply water temperature reaches minimum hot water set point and hot air temperature is higher than the set point minus the dead band during a certain period, the three way control valve is activated to maintain the hot air temperature. The lead boiler is disabled when hot water system is disabled or when hot deck temperature is higher than hot deck set point plus the dead band if the hot water supply temperature set point is at the minimum value (130ºF) for a certain period with fully open three-way valve.

During a **LOWLOAD** module, the pump speed is maintained at a minimum speed. The hot water temperature is modulated to maintain the hot deck air temperature. If supply water temperature reaches maximum hot water set point (170°F) and hot air temperature is lower than the set point plus the dead band during a certain period, the boiler operation switches to a **HIGHLOAD** module.

During a **HIGHLOAD** module, hot water temperature is maintained at maximum hot water temperature set point (170°F) and pump VFD speed is controlled to maintain hot deck air temperature. If the VFD speed is less than minimum pump speed during a certain period, the boiler operation switches back to a **LOWLOAD** module.

If the VFD speed is higher than a certain value and hot air temperature is lower than the set point minus the dead band during a certain period, the boiler operation switches to a **2ndBoiler** mode.

In the **2ndBoiler** mode, both boilers are on and perform the **LOWLOAD** and **HIGHLOAD** module automatically based on heating load conditions. The minimum pump speed is reset to a higher value (60%) when two boilers are on. If the VFD speed is higher than a certain setting and hot air temperature is lower than the set point plus the dead band during a certain period, each boiler operation switches to a **3rdBoiler** mode. If hot water temperature reaches minimum hot water set point during a certain period, the second boiler is disabled and boiler operation switches back to a **HIGHLOAD** module.

Under a **3rdBoiler** mode, three boilers are on and perform the **LOWLOAD** and **HIGHLOAD** module automatically based on heating load conditions. The minimum pump speed is reset to a higher value (80%) when three boilers are on. If hot water temperature reaches minimum hot water set point during a certain period, the third boiler is disabled and each boiler operation switches back to a **HIGHLOAD** module.

If the pump speed is higher than a certain value (95%) for a certain period, the pump with VFD is switched to the constant speed pump. If the hot air temperature is higher than the set point plus the dead band for a certain period, the constant speed pump switches back to the pump with VFD. The hot water pump is shut off when boilers are disabled, and the pump is enabled once a month during the boiler disable period for maintenance purposes.

For the perimeter heating system, the control schedule is also improved. During the unoccupied hours, the perimeter system is enabled when outside air temperature is below 45°F. During occupied hours, the perimeter system is enabled when outside air temperature is less than 35ºF and disabled when outside air temperature is higher than 45ºF. Perimeter supply water temperature is reset based on outside air temperature: when outside air temperature is 40°F, water temperature is set at 120°F and at 150°F when the outside air temperature is 10°F.

RESULTS

The operation comparisons are provided before CC ® and after CC ® for AHU, chillers and boilers. The building energy system performance and energy savings after optimal control schedule implementation are also present.

The data is compared using trending data in the Trane summit® operation system every 5 minute except measured fan power data.

Figure 10 presents the trended AHU supply fan speed ratio associated with outside air temperature. The speed ratio is defined as the ratio of an actual supply fan speed to a design supply fan speed. When outside air temperature is lower than 20°F or higher than 80°F, the speed ratio reaches the maximum value, which is only half of the supply fan speed ratio before CC ®. Compared to the constant fan speed ratio before CC ® and measured fan power at different speed ratios in Figure 11, fan power is reduced significantly.

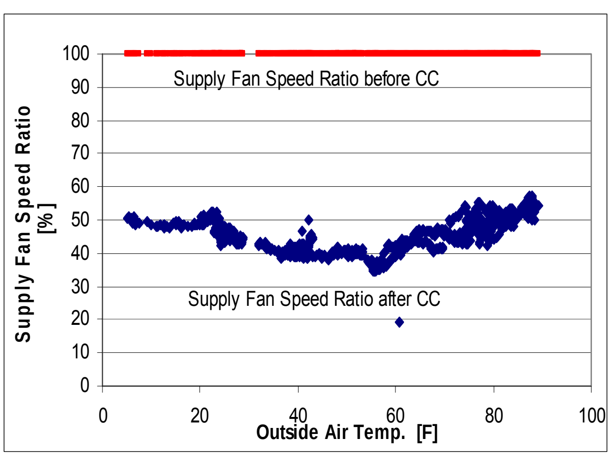

Figure 10: Supply fan speed ratio comparison before CC ® and after CC ®

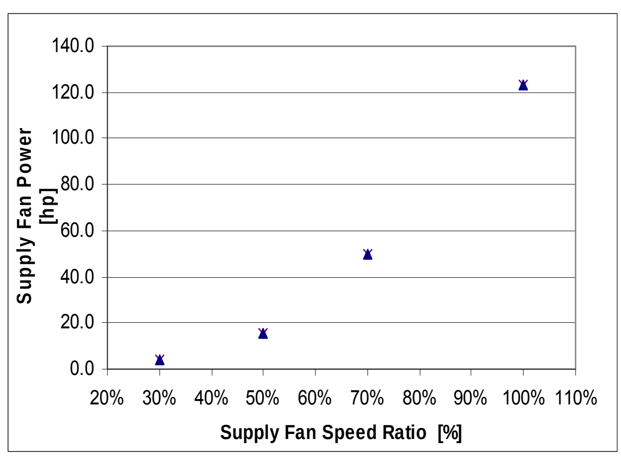

Figure 11: Measured supply fan power at

different fan speed ratios

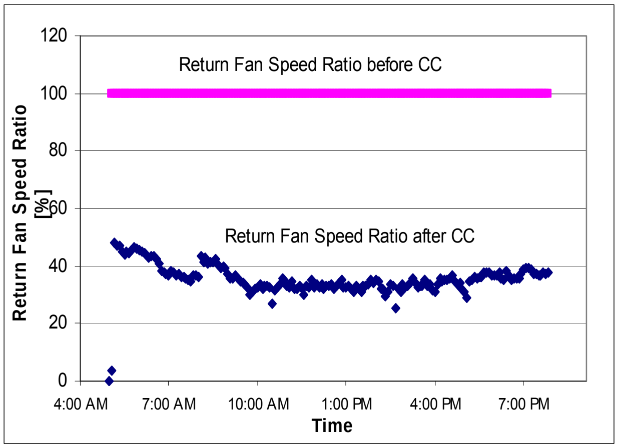

Figure 12: Return fan speed ratio comparison before CC® and after CC®

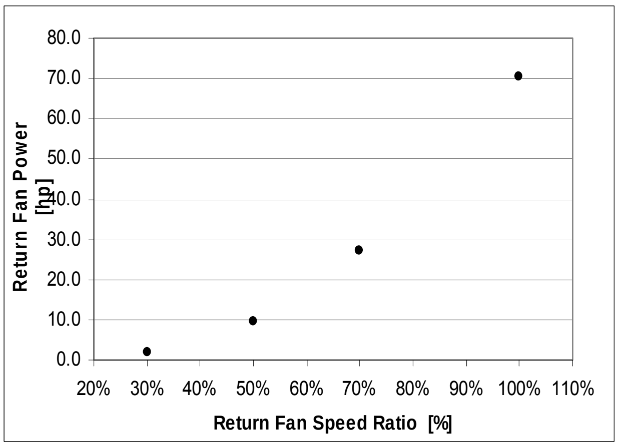

Figure 13: Measured return fan power at different fan speed ratios

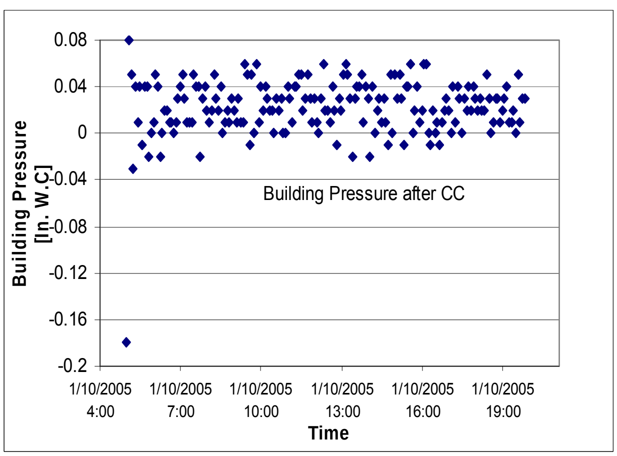

Figure 14: Building pressure control after CC®

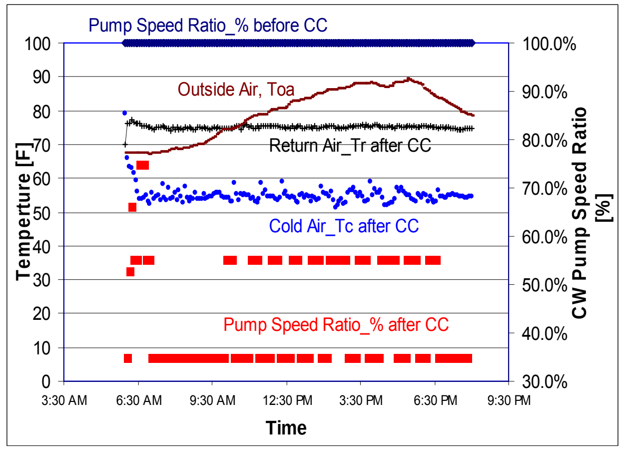

Figure 15: Chilled Water Pump speed ratio before CC® and after CC®

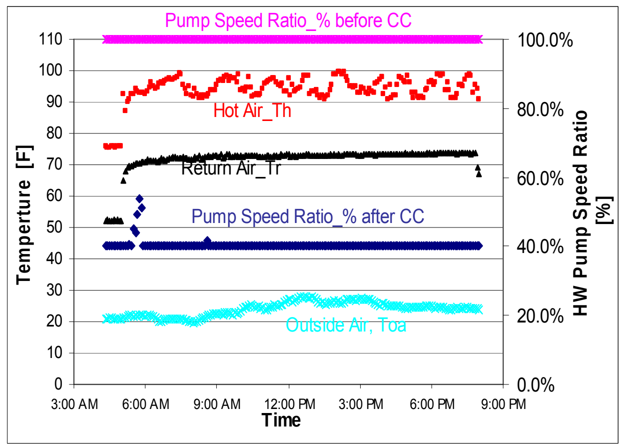

Figure 16: Hot Water Pump speed ratio before CC® and after CC®

Figure 12 presents the trended AHU return fan speed based on outside air temperature. Figure 13 presents measured return fan power at different speed ratios. Figure 14 shows the building pressure control by return fan in one day. The building pressure can be maintained at the setting with the substantial fan power savings that accompany the reduced speeds shown in Figure 12.

Figure 15 compares the daily chilled water pump operation in both existing and optimal control schedule. The cold air temperature can be maintained at about 55°F with reduced pump speed ratios.

Figure 16 compares the daily hot water pump operation under both existing and optimal control schedule. The hot water pump power is significantly saved, compared to the pump speed before CC®

Figures 17 & 18 compare the electricity consumption and gas consumption in 5 months before CC® and after CC®. The total cost savings of electricity and gas is $37,800 during the first five months after CC® completion. The electricity utility cost is reduced by 39.7% and gas utility cost is reduced by 80.6% based on the last five months of utility data since project completion (August, 2004)

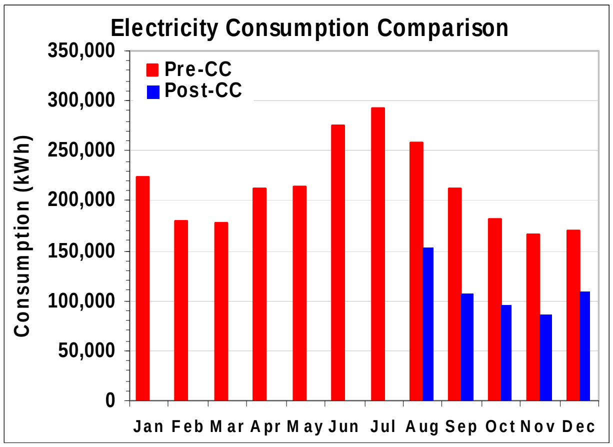

Figure 17: Electricity consumption comparison before CC® and after CC®

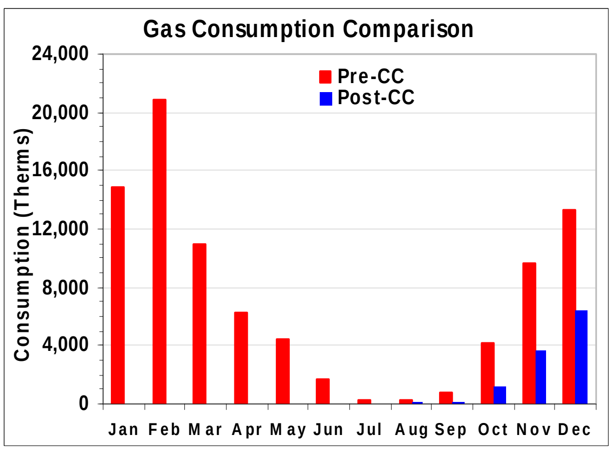

Figure 18: Gas consumption comparison before CC® and after CC®

CONCLUSIONS

All HVAC control technologies have been implemented in the case study building. The building performance is improved by reduced comfort complaints and improved system performance. The case study shows that electricity utility costs were reduced by 39.7% and gas utility costs by 80.6%.

The optimization of AHU control sequences can significantly improve building comfort and reduce HVAC energy cost.

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