Optimizing HVAC Control to Improve Building Comfort and Energy Performance

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Abstract

This paper demonstrates the benefits of optimal control in well-designed and operated buildings using a case study. The case study building was built in 2001. The HVAC and control systems have been installed with state-of-the-art equipment which include a terminal box temperature integrated minimum airflow reset. The building has been used and operated based on the design intents. This paper presents both the existing and the optimal control schedules, which include the VAV box operation schedule, AHUs optimal control, chiller and chilled water pump control, and boiler and hot water pump control. The measured hourly HVAC electricity consumption shows that annual savings of up to 40% can be achieved with an optimal control schedule.

Introduction

Within the last 20 years, more and more researchers have developed a number of new technologies to improve existing building energy and comfort performance [1~6]. As a result, Continuous Commissioning (CC) has been developed [7].

CC has been helping building owners achieve energy savings and thermal comfort improvement by retrofitting out-of-date facilities, detecting inefficient equipment, replacing malfunctioning components and updating incorrect operating sequences, and optimizing the system control sequences. Old buildings with no advanced equipment and control strategies definitely have energy savings potential. For example, C.R. Kjellman [8] successfully saved \$9,932/yr for a 34,173 square foot school building by converting a constant air handler to variable air volume, adding economizers, dampers and actuators, and replacing three-way valves with two-way valves. M. Liu [9] reduced the utility cost of a 123,000 ft^2 laboratory-office building up to \$369,000/yr by simply detecting and fixing a leaking pneumatic line on the cooling coil control valves and implementing a supply air temperature reset schedule.

It is generally believed that a well-designed new building with state-of-the-art technologies has little or no potential of reducing energy by retrofit or commissioning. This paper demonstrates the potential energy savings and performance improvement in buildings with state-of-the-art technologies through a case study. This paper will present the case study building information, existing and optimal control schedules, and measured building performance improvement.

Building information

The case study building is located in Omaha, Nebraska. It was built in 2001 with a total conditioned floor area of 247,000 square feet (see Figure 1). The major conditioned area is office space. The building is occupied from 8 a.m. to 5 p.m. Monday through Friday. The HVAC systems operate 24 hours per day, seven days a week. The HVAC systems operate in two modes: occupied and unoccupied. The HVAC occupied hours are defined from 3 a.m. to 10 p.m. Other times are defined as unoccupied hours. During system unoccupied hours, the terminal box minimum supply airflow is reduced to zero and the room temperature is set back to 85°F in cooling mode and 65°F in heating mode.

A total of 223 terminal boxes supply conditioned air to the space. The terminal boxes are equipped with flow stations. Wireless space temperature sensors are used for the open office area. Parallel fan-powered boxes with reheat coils serve exterior zones, series fan-powered boxes with reheat coils serve all conference rooms and labs, VAV boxes with reheat coils serve the interior zone on the fourth floor, and VAV boxes with no reheat coils serve the interior zones from the first floor to the third floor.

Figure 1~~1~~ Case Study Building

The building has two single-duct variable air volume air-handling units (MAHUs) for the office area and two auxiliary single-duct constant air volume AHUs (AAHUs) for the main lobby and entrance. Each MAHU has variable frequency drives installed for two supply fans (125hp/each) and three return air fans (40hp/each). Each AHU serves both interior and exterior zones. Each MAHU serves half of the building (south and north). The main supply air ducts of two MAHUs are interlinked in a so-called “donut” shape.

Two centrifugal chillers have been installed (450 ton/each). Each chiller has one dedicated constant-speed primary chilled water pump (15hp/each) and one dedicated constant-speed condensing water pump (25hp/each), respectively. A variable speed drive has been installed in the secondary chilled water pump (40hp/each).

Ten Gas-Fired Pulse Combustion boilers (PHW-1400 size: 1,400,000 Btu/hr/each) have been installed. A variable speed drive has been installed on the hot water pump (25hp).

Modern DDC control systems have been installed for AHUs, chillers, pumps, and 223 terminal boxes. The boiler has its own control panel, but it can receive global enable/disable commands from the EMCS. The HVAC hourly energy consumption is measured by dedicated meters.

Optimal Control Schedules

The control schedule optimization includes terminal boxes, AHUs, chillers, and boilers. In the following sub-section, the original control schedules are presented and analyzed first before the optimal schedules are presented.

VAV boxes

The integrated terminal box nighttime reset was implemented to reset the terminal minimum primary airflow to zero and heating and cooling space temperature set points to 65°F and 85°F, respectively, during unoccupied hours. The unoccupied hours were defined from 10 p.m. to 3 a.m., seven days a week. The original integrated nighttime reset has the following disadvantages: (1) negative building pressure when the supply air flow was lower than exhaust airflow, (2) high fan power during warm-up and cool-down periods (see Figure 3), and (3) comfort complaints from a few workers who worked during unoccupied hours.

To solve the problems stated above, the integrated terminal box reset schedule was switched to an airflow reset schedule [10], which reset the minimum airflow to a lower value during unoccupied hours while maintaining the space temperature at the occupied set point. The unoccupied hours were defined from 5:30 p.m. to 6:30 a.m. during weekdays and from 2:30 p.m. to 9 a.m. on the weekends. The number of unoccupied hours was extended from the existing 35 hours per week to 97 hours per week.

Air Handling Unit (MAHU):

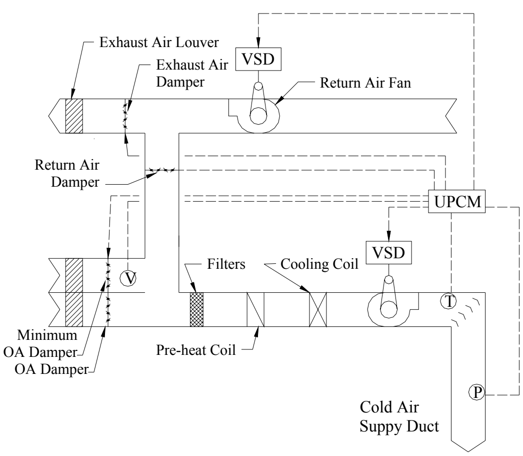

Figure 2: Schematic of MAHU system

Figure 2 presents the schematic diagram of the two main air-handling units. The supply air fan was controlled to maintain the duct static pressure set point between 1.5 inH_2O and 2.5 inH_2O depending on the maximum terminal box damper position. If the maximum damper position was less than 94% open, the static pressure set point would be decreased to open the damper more. If the static pressure reached the minimum value (1.5 inH_2O), the damper was closed more to maintain the room air temperature.

The return fan was controlled by relief damper position according to the following criteria:

a. When relief damper was less than 25% open, the relief chamber static pressure set point was controlled at -0.02 inH_2O.
b. When the relief damper was higher than 96% open, the relief chamber static pressure was controlled at 0.1 inH_2O.
c. When the relief damper was less than 50% and 75% open, the relief chamber static pressure was controlled at 0.0 and 0.01 respectively.
d. The relief damper position was controlled by the building static pressure sensor.

The supply air temperature was reset based on the outside air temperature. When the outside air temperature was lower than 50°F, the supply air temperature was maintained at 63°F. When the outside air temperature was higher than 75°, the supply air temperature was maintained at 53°F. The supply air temperature was reset linearly from 63°F to 53°F when the outside air temperature increased from 50°F to 75°F.

The enthalpy economizer was implemented. When the outside air enthalpy was lower than 17Btu/lbm, the economizer was activated. When economizer was disabled, the minimum outside air was measured using a flow station and controlled at the design value by adjusting the outside air damper during occupied hours. During unoccupied hours, the outside air damper was completely closed.

The original control schedules have the following disadvantages: (1) excessive minimum static pressure set point, (2) outside air backflow from the relief damper due to negative pressure set point at the mixing chamber, (3) high humidity during mild weather conditions due to relatively high supply air temperature set point, (4) excessive minimum outside air intake during occupied hours, (5) negative building pressure during unoccupied hours due to zero outside air intake, and (6) excessive mechanical cooling consumption due to an inappropriate enthalpy economizer set point.

Figure 3 shows the actual fan speed and static pressure set point for a typical day operation (May 2, 2003) under the original schedule. AHU2 had a static pressure set point of 1.5 inH_2O the entire time. AHU1 had a similar profile. Obviously, the minimum static pressure set point was too high. Extra fan power was being consumed and more noise was being produced.

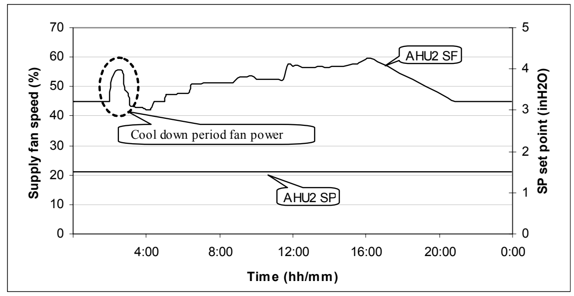

Figure 3: Supply Fan Speed and Static Pressure Set Point for AHU on May 2nd, 2003

In Omaha, Nebraska, there are about 1455 hours per year when the outside air enthalpy is between 17 Btu/lbm and 25 Btu/lbm. Due to the excessively conservative set point of the economizer, significantly more mechanical cooling was required.

Optimal control schedules were developed to solve the problems mentioned above and to improve the system energy efficiency. An optimal static pressure reset [11] was developed. Figure 4 shows the improved static pressure control schedule. The static pressure is reset within the range determined by the supply fan speed instead of constant values (1.5~2.5inH_2O), since fan speed is a good indicator of total airflow due to the static pressure reset strategy. The minimum static pressure set point was dynamically reset from 2.0 inH_2O to 0.5 inH_2O when the fan speed was reduced from 70% to 30%. The existing schedules are also presented in the same chart for comparison. The optimal static pressure set-point schedule allows the damper to be fully open to save fan power and reduce noise level.

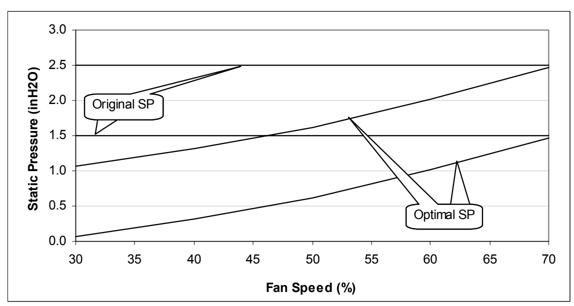

Figure 4: Existing and Optimal Static Pressure Reset Schedules

The return air fan speed is controlled by the indirect volume tracking method. Since the optimal static pressure reset is implemented, the supply air fan airflow can be determined using the fan speed. The return airflow can be determined by the return air fan speed and design airflow. The return air speed is controlled to maintain a constant difference between the supply airflow and the return airflow.

The economizer is enabled when the outside air enthalpy is less than the return air enthalpy. This increases the economizer operational time by 1450 hours per year and significantly reduces electricity energy consumption during these hours. The recommended economizer operation significantly improves the indoor air quality since more outside air is provided to the building 1450 hours per year.

The minimum outside air intake is determined based on typical day occupancy and exhaust fan operational schedules. The minimum outside air intake is reduced from the existing 20,000cfm to 8,000cfm (2000cfm/unoc). Newly occupied minimum outside air intake reduces cooling and heating consumption, and unoccupied minimum outside air intake keeps positive building pressure.

The optimal temperature reset maintains the supply air temperature at 55°F when the outside air temperature is higher than 55°F to eliminate the problems associated with the previous schedules. It also saves significant chiller and pump power during summer operations.

<u>Chiller and chilled water loop</u>

Figure 5 shows the chilled water system. The primary chilled water loop includes two chillers with a dedicated primary pump and a bypass pipe with a manual valve. The secondary chilled water loop includes the secondary chilled water pump with Variable Speed Drive (VSD) and cooling coils with control valves.

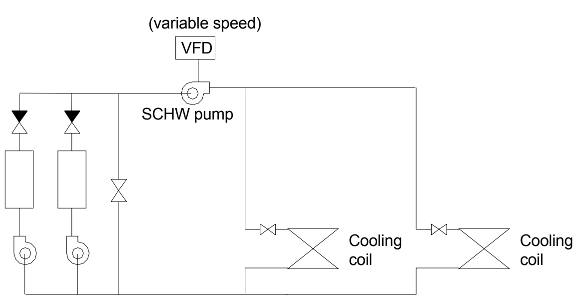

Figure 5: Schematic of the chilled water systems in the case study building

The original operation schedule turned the chiller and the primary pump on when the outside air enthalpy was higher than 17 Btu/lbm or when the economizer was disabled. Chilled water supply temperature was set at 40°F. When chilled water supply temperature was higher than 42.5°F, the second chiller was turned on. When the temperature difference between chilled water supply and return was less than 50% of the design values, the second chiller was turned off.

The secondary pump speed was controlled to maintain the differential pressure (DP) set point between 10psi to 15psi at the end of the loop depending on the maximum chilled water valve position. If the maximum valve position was less than 90% open, the differential pressure set point would be decreased to open the valve more. If the differential pressure reached the minimum value (10psi), the secondary pump maintained the differential at this value and the cooling coil valve was modulated to maintain the supply air temperature set point.

The original chiller and chilled water pump system controls have the following disadvantages: (1) excessive building by-pass flow, e.g., the temperature difference between chilled water supply and return can be as low as 2°F, (2) low load operation of both chillers, (3) excessive primary pump power and condensed water pump consumption, and (4) excessive secondary pump energy consumption due to inappropriate minimum differential pressure set points.

Figure 6 presents monitored maximum valve position and the loop differential set point for a typical day (May 3, 2003) operation. The minimum differential set point was maintained over half of the time. The maximum valve position was less than 60% open over 50% of the chiller operation time due to a higher than required minimum differential pressure set point.

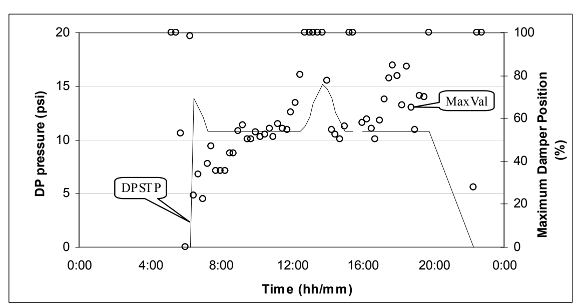

Figure 6: EMCS Monitored Maximum Valve Position and the Differential Pressure Set Point for May 3rd

To minimize chilled water pump power and chiller compressor power, the optimal operation control schedules were developed based on variable flow principles. A number of researchers and engineers have investigated the issues of variable flow system implementation and conversion from primary/secondary systems [12~16]. However, special control sequences have been developed to accommodate existing conditions. In this plant, there is no automatic isolation valve for each chiller and for the by-pass pipe.

To implement the variable water flow schedule, the chiller by-pass valve is shut off manually. Figure 7 shows a schematic of the chilled water system. A chiller is turned on when the outside air temperature is higher than 60°F or the mixed air temperature is higher than the supply air temperature set point plus the dead band. The chiller is operated according to building load which is divided into LOWLOAD mode, HIGHLOAD mode and SNDCHILLER mode.

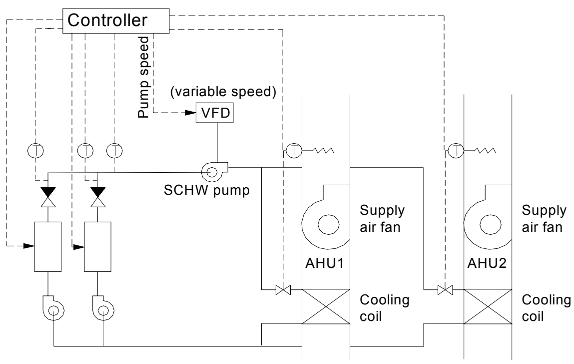

Figure7: Schematic of chilled water loop and control after CC

During LOWLOAD mode, only one chiller is on and one primary pump circulates the chilled water through the cooling coils. The chilled water supply temperature is reset to maintain cooling coil valve position at 80% ~ 100% open [17]. When the chilled water temperature reaches the minimum value and the maximum chilled water valve is 100% open, the chiller switches to HIGHLOAD mode. If the outside air temperature is below 57°F and the mixed air temperature is below the supply air temperature set point, the chiller and the primary pump are turned off.

During HIGHLOAD mode, the secondary chilled water pump is enabled to increase the chilled water flow rate through the coils with minimum chilled water supply temperature (45°F). The secondary chilled water pump speed is controlled by VSD to maintain the cooling coil valve at 80%~100% open. If the chilled water temperature is higher than the chilled water temperature set point and the maximum valve position is 100%, the chiller mode is switched to SNDCHILLER mode. The chiller mode switches to LOWLOAD mode automatically when the supply air temperature is less than the set point minus dead band.

Under SNDCHILLER mode, both chillers and primary pumps are on. When the temperature difference between the chilled water supply and return is lower than 5°F and the supply air temperature is lower than the set point minus dead band, the chiller mode switches back to the HIGHLOAD mode.

Boiler and hot water loop

The boiler and hot water loop consists of ten boilers, a hot water pump with VSD control and a bypass pipe with manual valve.

The original schedule turned the boilers on when the outside air temperature was lower than 80°F. The detailed hot water supply temperature is shown in Table 1. The hot water pump ran year-round to maintain the differential pressure (DP) of the remote reheat coil at 7.5psi even though the boilers were off.

The original boiler and hot water pump schedules have the following disadvantages: (1) excessive building bypass flow, e.g., the hot water supply temperature was the same as the return temperature most of the time under mild weather conditions, (2) excessive boiler operation when outside air temperature was high, and (3) excessive hot water pump energy consumption due to inappropriate DP set point and no disable command when boilers were off.

Table 1: Comparison of existing and improved hot water supply temperature schedules

OA Temp.	HWS Temp. before CC	HWS Temp. after CC
50ºF	140ºF	110ºF
40ºF	150ºF	120ºF
30ºF	160ºF	130ºF
20ºF	170ºF	140ºF
10ºF	180ºF	150ºF
0ºF	190ºF	160ºF

To reduce extra gas consumption and hot water pump operation during the hot summer months, the supply hot water temperature was rescheduled. Table 1 presents the hot water supply temperature comparison before and after CC. When the outside air temperature is higher than 60ºF, the boilers are turned off. The hot water pump speed is controlled to maintain the differential pressure of the remote reheat coil at 3.5psi. 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.

Results

This section presents the building energy system performance and energy savings after optimal control schedule implementation. The optimal VAV box operation schedule expands the building thermal comfort up to 24 hrs a day.

Figure 8 presents the measured AHU 2 supply fan speed and the static pressure set point on June 21, 2003. AHU1 has a similar profile. The duct static pressure can be as low as 0.2 during unoccupied hours and fan speed can be as low as 20%. Fan power and noise level are significantly reduced.

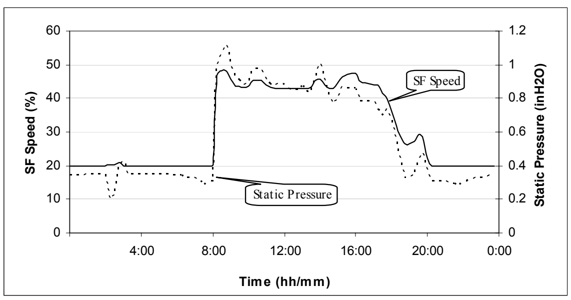

Figure 8: Measured supply fan speed and static pressure under the optimal schedules for AHU 2 on June 21st, 2003

Figure 9 presents the measured chilled water supply and return temperature, and the difference of the supply and return chilled water temperatures on July 2 under the optimal control schedule. The chiller operation stayed in LOWLOAD mode during unoccupied hours (midnight to 8 a.m. and 6 p.m. to 12 midnight). The chilled water temperature was reset to maintain minimum chilled water flow. When the outside air temperature increased and the building was occupied, the chiller operation was switched to HIGHLOAD mode. The chilled supply water temperature was maintained at the minimum set point (45ºF) and the secondary pump was enabled to circulate more chilled water flow through the cooling coils. At approximately 1:30 p.m., the chiller was not able to maintain the minimum supply water temperature due to increased chilled water flow. At 2:30 p.m., when the chilled water temperature was higher than the set point plus the dead band, the second chiller was turned on. The temperature difference between chilled water supply and return was maintained at about 10°F most of the time. The chiller and chilled water loop returned to LOWLOAD mode after 6 p.m. The chiller and chilled water loop now operate in response to the building load. Chiller efficiency and cooling coil heat transfer are improved significantly. The supply air temperature is still maintained at 55°F to satisfy the building humidity and comfort requirements.

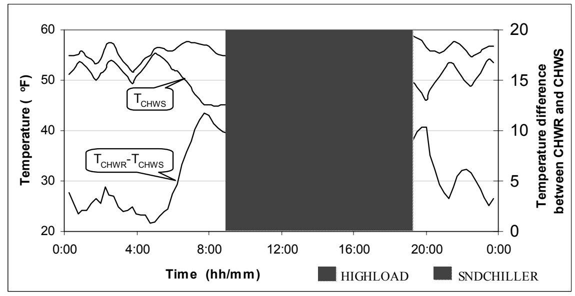

Figure 9: Measured chilled water supply and return temperatures, and their difference on July 2, 2003 (Remove the supply air temperature)

Figure 10 compares the daily HVAC electricity consumption under both original and optimal control schedules. The original data were measured from June 16 to June 30, 2002. The current data were measured on the same dates in 2003. The mean HAVC electricity consumption was generated for each hour. The HVAC electricity difference varied from 80 kW to 150 kW.

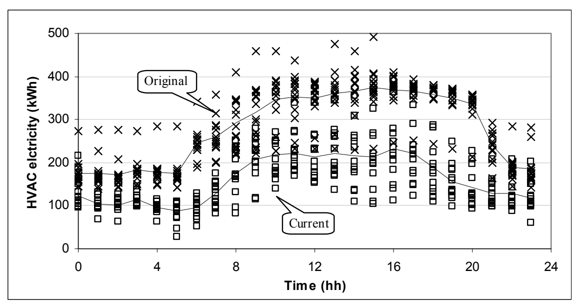

Figure 10: Comparison of daily profiles of HVAC electricity consumption

Figure 11 compares the HVAC electricity consumption under both previous and the optimal schedules against the ambient air temperature. With temperature corrections, it appears that the average hourly electricity savings is approximately 85 kW. Based on these measured results, the annual electricity energy savings was estimated as 744,600 kWh, which is 18% of the entire building electricity consumption and 40% of the annual HVAC electricity energy consumption. If the electricity price is $0.05/kWh, the annual electricity cost is $37,240/yr.

Gas savings was not able to be documented due to lack of utility data at the time the paper was written.

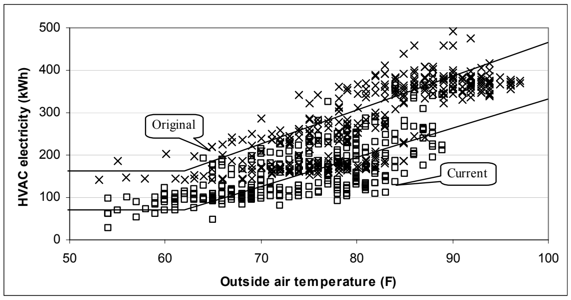

Figure 11: Comparison of correlations between HVAC electricity consumption and outdoor air temperature

Conclusions

Several new HVAC control technologies have been implemented in the case study building, such as nighttime airflow reset schedule, main duct static pressure reset schedule, volume tracking return fan control schedule, and a variable primary chilled water flow schedule. The case study shows annual HVAC electricity reduction of 40%.

Optimizing the existing control sequence in state-of-the-art buildings can significantly improve building comfort and reduce HVAC energy costs.

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References

1. Liu, M., Y. Zhu, and D. E. Claridge, “Use of EMCS Recorded Data to Identify Potential Savings Due to Improved HVAC Operations and Maintenance,” *ASHRAE Transactions-Research.* Volume 103, Part 2, 1997.
2. Liu, M., D. E. Claridge, and B. Y. Park, “An Advanced Economizer Controller for Dual Duct Air Handling Systems with a Case Study,” *ASHRAE Transactions-Research.* Volume 103, Part 2, 1997.
3. Liu, M., Y. Zhu, D. E. Claridge, and E. White, “Impacts of Static Pressure Set Level on the HVAC Energy Consumption and Indoor Conditions,” *ASHRAE Transactions-Research.* Volume 103, Part 2, 1997
4. Liu, M., Y. Zhu, B. Y. Park, D. E. Claridge, D. Feary, & J. Gain, “Air Flow Reduction To Improve Building Comfort and Reduce Building Energy Consumption.” *ASHRAE Transactions,* Part I, 1999.
5. Liu, M., and D. E. Claridge, “Converting Dual Duct Constant Volume Systems to Variable Volume Systems Without Retrofitting the Terminal Boxes.” *ASHRAE Transactions,* Part I, 1999.
6. Wei, G., M. Liu, and D. E. Claridge, 2001, “In-situ Calibration of Boiler Instrumentation Using Analytical Redundancy.*” International Journal of Energy Research*, Vol. 25, Issue 5, 2001.
7. Liu, M., D. E. Claridge, J. S. Haberl, and W. D. Turner, “Improving Building Energy Systems Performance by Continuous Commissioning.” *Energy Engineering*, May 1999.
8. Kjillman, C.R., 1997, “Commissioning as a Vital Component of Customer-Financed Performance Contracts”, Fifth National Conference on Building Commissioning, Huntington Beach, California.
9. Liu, M. and D.E. Claridge, 1998. “Use of Calibrated HVAC System Models to Optimize System Operation.” *ASME Journal of Solar Energy Engineering*, Vol. 120, pp. 131-138.
10. Liu M., Abbas M., Zhu Y., and Claridge D. E., 2002, “Terminal Box Airflow Reset: An Effective Operation and Control Strategy for Comfort Improvement and Energy Conservation,” Proceedings of Thirteenth Symposium on Improved Building Systems in Hot and Humid Climates, pp. 80-86, May 20-23, 2002, Houston, Texas.
11. Wei G., Claridge D. E., Sakuii Y., and Liu M., 2000, "Improved Air Volume Control Logic," Proceedings of The Twelfth Symposium on Improving Building Systems in Hot and Humid Climates, pp. 195-198, May 15-16, 2000, San Antonio, Texas.
12. Hartman, T.B., 1996, “Design Issues of Variable Chilled-Water Flow Through Chillers,” *ASHRAE Transactions*, Vol.102, Part 2.
13. Rishel, James B., “Control of Variable Speed Pumps for HVAC Water Systems.” *ASHRAE Winter meeting*, 2003.
14. Taylor, Steven T. , “Primary-Only vs. Primary-Secondary Variable Flow Systems” *ASHRAE Journal,* Vol.44, No.2, pp.25-29, Feb. 2002.
15. Avery, G., “Controlling Chillers in Variable Flow Systems” *ASHRAE Journal ,* Vol.40, No.2, pp. 42-45, Feb. 1998.
16. Liu, M., “Variable Water Flow Pumping for Central Chilled Water Systems”, *Transactions of ASME*, Vol.124, pp.300-304, Aug. 2002.
17. Kirsner, W, “Designing for 42degF Chilled Water Supply Temperature-does it save energy?” *ASHRAE Journal,* Vol.40, No.1, pp. 37-42, Jan. 1998.