Impacts of Optimized Cold & Hot Deck Reset Schedules On Dual Duct VAV System - Application and Results

Mingsheng Liu, Ph.D., P.E.
Energy Systems Laboratory
Texas A&M University
College Station, Texas

Mustafa Abbas
Energy Systems Laboratory
Texas A&M University
College Station, Texas

Bryan Veteto
Energy Systems Laboratory
Texas A&M University
College Station, Texas

David E. Claridge, Ph.D., P.E.
Energy Systems Laboratory
Texas A&M University
College Station, Texas

Homer Bruner
Physical Plant
Texas A&M University
College Station, Texas

Abstract

In the theory and modeling part, the principles of VAV box optimization are investigated. The simulations show that optimizing cold and hot deck reset schedules can significantly reduce the amount of energy consumption.

During a Continuous Commissioning (CC) process, the optimal cold and hot deck reset schedules were implemented in 12 dual duct variable air volume systems which serve a 324,000 ft^2 university building. The improved cold and hot deck reset schedules combined with other CC measures reduced the chilled water energy consumption by 7,571 MMBtu/yr (18%) and 303 MWh/yr (18%) for the fan power. The estimated heating energy savings are 2,327 MMBtu/yr. The estimated annual cost savings are $75,040/yr. This paper presents the building and AHU system information, optimal reset schedule, and measured results.

Introduction

Zachry Engineering Center is a major engineering research and teaching building at the Texas A&M Campus. The building was built in 1973 with a total floor area of 324,000 ft^2. Initially, the building was equipped with 12 major dual duct constant volume air systems. In 1991, the constant volume systems were converted into VAV system by installing variable frequency drives and retrofiting terminal boxes. During the retrofit, a modern DDC system was installed to control AHUs, pumps, and terminal boxes. The building was commissioned from November 1996 to April 1997 by the authors. During the commissioning, a number of mechanical problems were corrected and the optimal hot and cold deck reset schedules were implemented. These activities reduced the building energy consumption significantly. This paper describes the building information, VAV conversion, commissioning, and compares the measured energy consumption.

Building Information

The Zachry Engineering Center was built in 1973 on the North side of the Texas A&M University, College Station, Texas. The three and one half story structure (plus additional ground floor level), has 324,400 square feet of gross floor area.

The rectangular shape building with cement block exterior walls has approximately 40% glass exposure. The glass is a single pane with a shading coefficient of 0.5 and built in place vertical blinds. The building is oriented such that the long axis is along a Northeast to Southwest direction. There is a large open area on the East side of the building that is open from first level to the ceiling of the third level with approximately 7,000 square feet of area on the first floor. On the third floor, there is about 3,000 square feet open area as well. The ground floor has the same floor area as the first floor except the center portion is filled with small offices.

The main functions of the building are for classrooms, laboratories, computer facilities, and offices. There are also some clean rooms for solid state electronics studies. The building is open 24 hours per day. All AHUs operate 24 hours to satisfy fume hoods, late-night studies, research activities, and computer facility operations. However, the late-night occupancy is much less than that during daytime.

Chilled water, hot water, and electricity are being provided to the facility by the Main Campus Central plant via an underground tunnel. There are two chilled water pumps (2-30 hp) and two hot water pumps (2-20 hp) in the building. The central plant supplies 42°F chilled water to the building year around. The hot water temperature varies from 150°F to 200°F according to the ambient temperature.

There are 12 dual-duct air-handing units (AHU) with single supply air fan (12-40 hp) installed in the basement to serve about 90% of the total building floor area. These 12 AHUs are spaced uniformly around the exterior wall. Each AHU has two risers from the basement to the third floor. Each AHU serves about the same amount of floor area on each floor. There are 10 more small AHUs (32 hp in total) to serve conference rooms (rooms 052 and 202), third floor open area, clean rooms, and penthouse. These units are either multi-zone or single duct units. The computer facility is conditioned by Liebert units on the ground floor.

VAV Conversion

A comprehensive energy audit was performed in 1986. By March, 1991, VFDs were installed in 12 AHUs. All the terminal boxes were retrofitted into VAV boxes. A modern DDC system was installed to control all AHUs, terminal boxes, hot water and chilled water pumps, and automatic control valves in all heating and cooling coils.

After the retrofit, the minimum air static pressure was controlled by modulating the fan speed. The static pressure set-point was reduced from 4 inH_2O (before retrofit) to a range of from 2.5 inH_2O to 3.5 inH_2O. The fan speed varied from 60% to 80% with a 45% fan power savings.

The cold deck and hot deck temperatures were controlled through the EMCS system. The EMCS system senses the cold and hot deck discharge air temperature and compares the measured values with the set-points. The bias was sent to a PI loop to control an automatic valve in each coil. The cold deck temperature was set in a range of 52°F to 55°F for different AHUs. The hot deck set-point varied from 110°F to 80°F when the outside air temperature varied from 40°F to 65°F. When the outside air temperature was lower than 40°F, the hot deck set-point was 110°F. When the outside air temperature was higher than 65°F, the hot deck set-point remained 80°F.

Each VAV box has a DDC controller, a hot air control damper, a cold air control damper, and a total flow sensor. The systematic diagram is shown in Figure 1. The DDC controller communicates with the central control system and commands hot and cold air dampers to maintain room temperature.

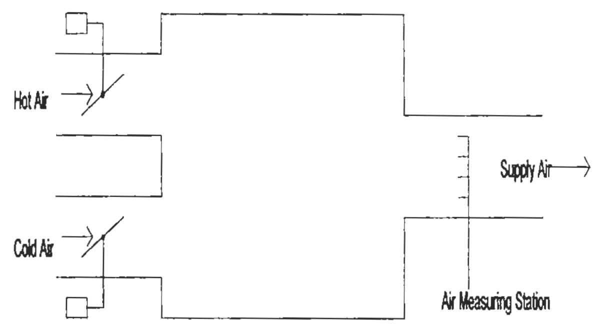

Figure 1: Systematic Diagram of a DD VAV Terminal Box

The DDC controller requires the following inputs: heating temperature set-point, cooling temperature set-point, switch temperature band, switch time band and room temperature set-point. The DDC controller switches between three operation modes: heating, light cooling and cooling.

In day heating mode, the cooling damper is set to the minimum. In night heating mode, the cooling damper is completely shut.

In the light cooling mode, the amount of cooling air needed to satisfy the cooling load is not enough to satisfy the minimum amount needed for ventilation and circulation. Both hot and cold air dampers are modulated to satisfy both cooling load and the minimum total air flow.

In heavy cooling mode, the cold air flow is more than the minimum total air flow rate. The cold air damper is modulated to satisfy the cooling load and the room temperature. The hot air damper is totally shut.

The heating/cooling switch-over determines whether the controller is in heating or cooling mode by monitoring the room temperature and the demand for heating and cooling.

If the following conditions are met for the length of switch time band (10 minutes), then the controller switches from heating to cooling mode:

(1) The heating damper command is less than 5.2% open.
(2) The room temperature is above the room temperature set-point by at least the value of switch temperature band.
(3) The room temperature is greater than the appropriate cooling temperature set-point minus switch time band.

If the following conditions are met for the length of switch time band (10 minutes), then the controller switches from cooling to heating mode:

(1) The cooling damper command is less than 5.2% open.
(2) The room temperature is below the room temperature set-point by at least the value of switch temperature band.
(3) The room temperature is less than the appropriate heating temperature set-point plus switch time band.

In day mode, the cooling temperature set-point is 74°F, the heating temperature set-point is 70°F. In night mode, the cooling temperature set-point is increased to 78°F, and the heating temperature set-point is decreased to 65°F. In this application, the room temperature set-point equals the cooling temperature set-point when the terminal box is in cooling mode. The room temperature set-point equals the heating temperature set-point when in heating mode. Significant amount of energy can be saved at night since the AHUs maintains the room temperature at 78°F for cooling and 65°F for heating. Due to the different requests, a total of eleven nighttime schedules were used in the building (See Table 1 for detail).

Table 1: Summary of Terminal Box Schedules

Schedule	No. of Boxes	Min. CFM	Max. CFM	Day Mode Hours at Weekday	Day Mode Hours at Weekend
e	1	251	659	6a~8p	10a~6p
f	11	2007	4931	6a~11p	9a~11p
g	1	135	280	6a~12p	10a~12p
l	6	1077	3652	8a~10p	11a~10p
m	2	616	1684	6a~10p	8a~8p
n	3	514	2187	6a~12p	8a~12p
q	3	670	1630	6a~7p	10a~5p
r	2	778	1896	6a~9p	7a~6p
t	98	24201	54390	6a~7p	off
u	7	1399	4284	6a~7p	7a~5p
v	39	10225	36843	6a~10p	off
w	96	19130	54632	24	24
x	40	10156	27218	6a~10p	6a~10p
y	58	13962	39014	6a~10p	8a~10p
z	17	3296	7489	6a~8p	off

Table 2: Summary of Terminal Box Information

Schedule	No. of Boxes	Sum of Min CFM	Sum of Max CFM	Min CFM Ratio
Nighttime Reset	134	31,565	87,435	0.36
Weekend Off	154	37,722	98,722	0.38
24 Hours Operation	96	19,130	54,632	0.35
Total	384	88,417	240,789	0.37

The schedules can be separated into three major groups: (1) nighttime reset; (2) weekend shut off; and (3) 24 hour-operations. Table 2 summarizes these three types of boxes.

There are a total of 384 boxes in the building. The nighttime reset was applied in 134 (35%) boxes; 154 (40%) for shut off during weekends; and 96 (25%) for 24 hours operation.

The total maximum air flow was determined as 240,789 cfm, or 0.82 cfm per gross floor area. If we assume that the net usable floor area is 80% of the grass floor area, the air flow rate is approximately 1.00 cfm per net usable floor area.

The DDC system was also installed in other small units. Optimal control schedules were also implemented by the control system contractor.

VFDs were also installed for the chilled water pumps. The chilled water flow, supply and return temperatures and pressures were measured. The VFD modulates the chilled water pump speed to maintain the differential pressure set point (10 psi). Neither chilled water flow rate, nor the chilled water supply and return temperature were used for the pump control.

No VFD is installed in the hot water pump. The minimum differential pressure was maintained by a automatic valve in the return water line. If the differential pressure is lower than the set-point, the valve will open to allow more water flow. When the differential pressure is higher than the set-point, the automatic valve will close more. The differential pressure was set at 10 psi regardless of the actual heating load.

Before VAV conversion, extremely low room temperature set-points (55°F to 65°F) were found during the audit, which indicated insufficient cooling capacity. After VAV conversion, most rooms can maintain suitable room temperature level. However, "lack of fresh air and air circulation" and "too hot" were consistently reported in a number of rooms. Indoor air quality was evaluated three times in 1996 at the request of occupants.

Continuous Commissioning

Continuous Commissioning is a process to solve the existing building operational problems and optimize the building operation [Liu et all.].

The Continuous Commissioning was initiated on November 1996 and completed on April 1997.

The cold deck and hot deck reset schedules were first optimized. After a careful investigation, it was decided to reset both hot and cold deck temperature according to outside air temperature. The cold deck temperature varies from 60°F to 54°F when the ambient temperature changes from 55°F to 90°F. When the outside air temperature is lower than 55°F, the cold deck temperature is set at 60°F. When the outside air temperature is higher than 90°F, the cold deck temperature remains 54°F. The hot deck set-point varies from 90°F to 70°F when the ambient temperature changes from 55°F to 70°F. When the ambient temperature is lower than 55°F, the hot deck set-point is 90°F. When the ambient temperature is higher than 70°F, the hot deck temperature set-point is 70°F which will force the hot water valve closed.

During the continuous commissioning process, both maximum air flow and the minimum air flow were adjusted according to the actual load conditions. The operation schedules were also adjusted according to occupant need. The cooling temperature set-point was reduced to 73°F, and heating temperature set-point was increased to 68°F to satisfy occupant comfort. The temperature switch band was set at 2°F and switch time band was set at 10 minutes. Table 3 summarizes the three schedules used after Continuous Commissioning.

The total maximum air flow was increased from 240,789 cfm (0.82 cfm/ft^2gross floor area) to 271,424 cfm (0.93 cfm/ft^2gross floor area). At the request of occupants, the number of 24 hour operation boxes was increased from 93 to 162. The unoccupied period was limited to 7:00 p.m. to 6:00 a.m. for 74 boxes, and 11:00 p.m. to 6:00 a.m. for 148 boxes. The weekends and weekdays used the same schedules to provide a comfortable environment for occupants who may work during weekends.

The indoor air quality problems were carefully investigated. The IAQ problems investigated during the CC process had one or more of the following causes:

(1) There was no return air path, which makes room pressure significantly higher than other areas.
(2) The flex duct was kinked, which makes the static pressure too low before the control damper in the terminal boxes.
(3) The damper shaft was glued to the box shell by paint, which makes it impossible to move the damper.
(4) The damper shaft was not adjusted right. When the controller commands to close, the damper was actually partially open.
(5) Nighttime reset schedules were used while the room was used 24 hours a day.
(6) Minimum and maximum air flow rates were not correct.

These problems were solved first during the commissioning process by using appropriate methods. The IAQ problems and complaints were solved. The hot and cold calls were reduced significantly.

After solving existing comfort and IAQ problems, the static pressure was reduced from a range of 2.5 inH_2O to 3.5 inH_2O to a range of 1.5 inH_2O to 2.0 inH_2O. When the ambient temperature is lower than 55°F, the static pressure set-point is 1.5 inH_2O. When the ambient temperature is higher than 90°F, the static pressure set point is 2.0 inH_2O. When the temperature varies from 55°F to 90°F, the static pressure set-point changes from 1.5 inH_2O to 2.0 inH_2O.

During the CC process, all the PI loops were fine tuned. Figure 2 compares a typical actual discharge air temperature before and after commissioning. Before commissioning, the cold deck discharge air temperature varied from 55 °F to 60 °F with maximum bias of 3 °F. After fine tune-up of the PI gains, the discharge air temperature were maintained at the set-point with a band of 0.5 °F.

Table 3: Summary of Terminal Boxes Schedules

Schedule		No. of Boxes	After CC		
Day Mode at Weekday	Day Mode at Weekend		Sum. Of Min CFM	Sum of Max CFM	Min CFM Ratio
6am - 6pm	6am - 6pm	74	16,325	51,984	0.29
6am - 10pm	6am - 10pm	148	30,123	101,184	0.30
24 hours	24 hours	162	33,764	118,256	0.29
Total		384	80,212	271,424	0.30

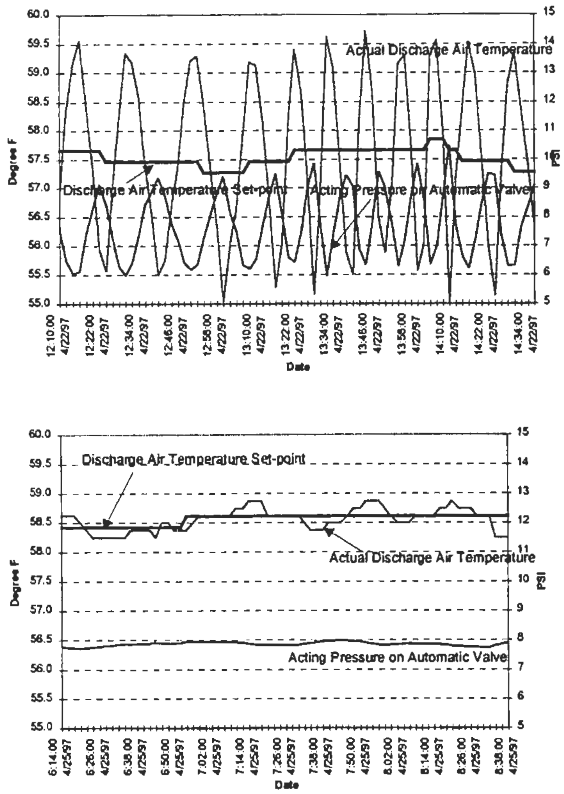

Figure 2: Comparison of Discharge Air Temperature Before and After PI Fine Tune Up

Measured Energy Savings

Under the LoanSTAR program [Verdict, et al, 1990], the building chilled water consumption, whole building electricity consumption, fan power and weather data have been measured since 1989. The energy savings of VAV conversion and CC are documented by using these measured data.

Figures 3 and 4 compares the measured fan power, and chilled water consumption, before retrofit, after retrofit, and after CC. The data were measured from 01/01/90 to 02/28/91 for the before retrofit period. The data were measured from 01/01/96 to 11/14/96 for the post retrofit but before CC. The data was from 04/01/97 to 12/02/97 for the post CC period.

The 12 fans have a total capacity of 480 hp or 358 kW. The measured daily average fan electricity power varied from 300 kWh to 358 kWh. Before retrofit, the fans were running at full load most of the time. After VAV conversion, the measured daily average fan power varied from 175 kW to 235 kW with an average of 180 kW.

The optimal static pressure reset schedule not only reduced the fan lift, but also reduced total air flow due to reduced simultaneous heating and cooling as well as excessive air leakage through air dampers in the box [Liu et al, 1995]. Consequently, the fan power consumption is reduced. The measured fan power varied from 90 kW to 200 kW.

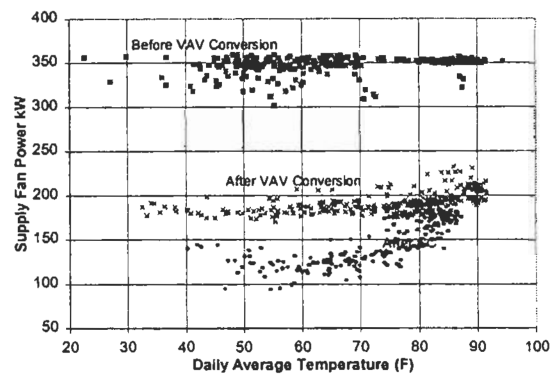

Figure 3: Comparison of Measured Daily Average Fan Power Before Retrofit, After Retrofit, and After Continuous Commissioning

Figure 4 compares the measured daily average chilled water energy consumption. Before VAV conversion, the hourly daily average chilled water consumption varied from 3 MMBtu/hr to 7.5 MMBtu/hr when the daily average ambient temperature varied from 20°F to 90°F. The higher chilled water consumption at the low ambient temperature is apparently due to simultaneously heating and cooling. After VAV conversion, the measured hourly daily average chilled water consumption varied from 2.0 MMBtu/hr to 7.5 MMBtu/hr. The simultaneous heating and cooling is reduced significantly when the cooling or heating load is low. The chilled water consumption was the same when the ambient temperature is above 90°F. After continuous commissioning, the measured hourly daily average chilled water energy consumption varied from 0.6 MMBtu/hr to 6.0 MMBtu/hr when the ambient temperature varied from 20°F to 90°F.

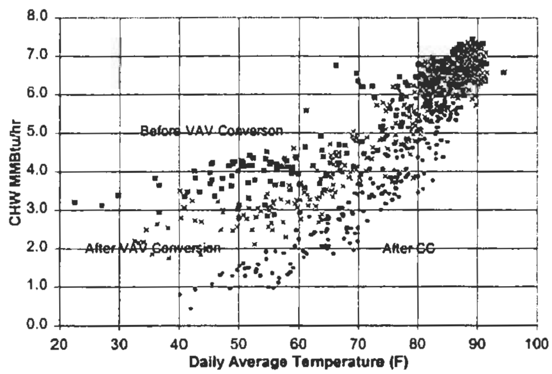

Figure 4: Comparison of Measured Daily Average Chilled water Consumption Before VAV Retrofit, After VAV Retrofit, and Post CC

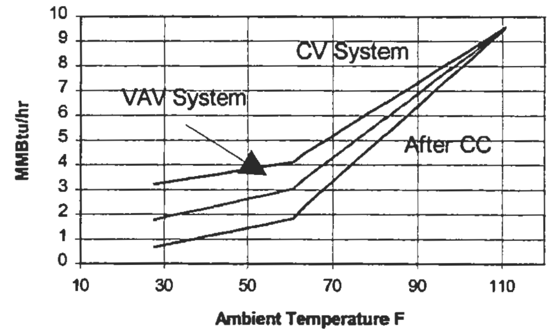

Figure 5: Chilled Water Consumption Regression Models for Before VAV Retrofit, After VAV Retrofit, and Post CC

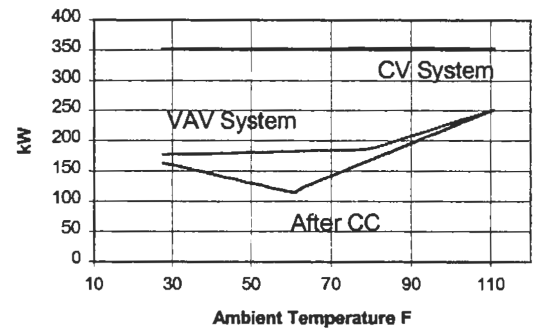

Figure 6: Fan Power Regression Models for Before VAV Retrofit, After VAV Retrofit, and Post CC

To evaluate the energy savings, the cooling energy consumption and fan power models were developed by regressing the measured hourly energy consumption versus the measured ambient temperature. The regression models are shown in Figures 5 and 6. The energy savings were determined by using measured ambient temperature in 1995 and these models. The results are summarized in Table 4. The heating energy consumption was estimated as the difference between the cooling energy consumption and the fan power savings since no heating energy consumption data was available.

Table 4: Summary of Measured Energy Consumption and Savings

	CHW (MMBtu/yr)			Fan Power (MWh/yr)			HW (MMBtu/yr)	Cost ($/yr)
	Base	Savings	%	Base	Savings	%	Savings*	Savings**
VAV	48,651	7,174	15	3,092	1,420	45	2,327	$84,084
CC	41,476	7,571	18	1,671	303	18	6,535	$75,040
Total		14,745	30		1,723	56	8,862	$159,124

*The heating savings was determined as the difference between the chilled water savings and fan power savings.

** The energy prices are following: $0.0278/kWh for fan and pump power, $4.67/MMBtu for chilled water, and $4.78/MMBtu for hot water.

The VAV retrofit decreased the chilled water energy consumption by 7,174 MMBtu/yr and 1,420 MWh/yr for the fan power. The estimated heating energy savings are 2,327 MMBtu/yr. The retrofit cost savings are $84,084/yr. The improved cold and hot deck reset schedules combined with other CC measures reduced the chilled water energy consumption by 7,571 MMBtu/yr or 18% and 303 MWh/yr for the fan power. The estimated heating energy savings are 2,327 MMBtu/yr. The annual cost savings are $75,040/yr.

It appears that the optimized cold and hot deck reset schedules can significantly reduce the building energy consumption in VAV systems.

Conclusions

The energy impact of the optimal cold and hot deck reset schedules were measured in a 320,000 square feet office building located in College Station, Texas. The building has 12 VAV dual-duct systems with the average minimum air at 40%. The optimal cold deck and hot deck reset reduced cooling energy consumption by 7,571 MMBtu/yr (18%), 6,535 MMBtu/yr for heating energy consumption, and 303 MWh/yr (18%) for fan power. The measured annual cost savings are $75,040. It appears that the optimized cold and hot deck reset schedules can significantly reduce the building energy consumption in VAV systems.

The measured energy savings are compared with VAV retrofit savings in the same building. The VAV conversion and modern EMCS system upgrade reduced the building heating energy consumption by 2,327 MMBtu/yr, 7,174 MMBtu/yr (15%) for cooling energy consumption, and 1,420 MWh/yr for fan power. The measured annual cost savings are $84,084/

Acknowledgment

This work was supported by the Texas State Energy Conservation Office through the LoanSTAR program.

Reference

Verdict, M., Haberl, J., Claridge, D., O'Neal, D., Heffington, W. and Turner, W. D., "Monitoring $98 Million in Energy Efficiency Retrofits: The Texas LoanSTAR Program," Proceedings of the ACEEE 1990 Summer Study on Energy Efficiency in Buildings, American Council for an Energy Efficient Economy, Washington, D.C., pp. 7.261-7.271, August, 1990.

Liu M., Y. Zhu, D. E. Claridge, Ed. White. 1997. "Impact of Static Pressure Set level on the HVAC Energy Consumption and Indoor Conditions". ASHRAE Transactions, 1997