Impacts of Static Pressure Set Level on the HVAC Energy Consumption and Indoor Conditions

M. Liu, Y. Zhu, D. E. Claridge
Energy Systems Laboratory
Texas A&M University

Ed. White
Energy Management Operation
UTMB

ABSTRACT

Air static pressure must be maintained at a certain level leaving the air-handing unit (AHU) to force a suitable amount of air through the terminal boxes. However, an excessive static pressure level is often used due to (1) lack of a control device in a constant volume system (CV); (2) malfunctioning control device in a variable volume (VAV) system; and (3) fear of failure to maintain room temperature. High static pressure often develops excessive damper leakage in older mixing boxes. This results in an inappropriate mixing of hot and cold air and an excessive amount of air entering the space. Consequently, the actual fan power, heating and cooling energy consumption all become significantly higher than the design values. Even worse, the system may not be able to maintain room conditions due to unwanted simultaneous heating and cooling, and may be noisy due to the excessive static pressure. This paper proposes to control the hot duct pressure and the Variable Frequency Drives (VFD's) to control the fan static i.e. the cold duct pressure. Both a theoretical analysis and a case study results are presented in this paper.

NOMENCLATURE

cfm	total air flow rate (ft^3/min)
E	energy (MMBtu or kWh)
f	excessive air flow rate (% of designed flow rate)
h	enthalpy (Btu/lb)
N	number of hours in each temperature bin
p	pressure (inH2O)
T	temperature (°F)
v	specific volume (ft^3/lb)
α	excessive air flow coefficient (% per extra inH2O)
β	cooling load ratio or load fraction (0.0 ~ 1.0)
η	fan efficiency
γ	ratio of cold and hot air flow
δp	static pressure drop (inH2O/ft)

Subscripts

a	ambient
c	cold deck, cold air
f	fan
h	hot deck, hot air
m	mixed air
o	designed, exit of fan

INTRODUCTION

The air static pressure will be defined here as the air static pressure at 2/3 of the distance down the main air duct. The static pressure has to be maintained at a certain level, such as 1 inH2O, to overcome the air flow resistance in the remainder of the air distribution duct and that in the other flow components, such as the terminal boxes and diffusers. For VAV systems, the air static pressure is maintained either by adjusting the inlet guide vanes or by adjusting the motor speed. However, excessive air static pressure is often used due to: (1) a malfunctioning control device; and (2) fear of failure to maintain room temperature. For constant volume systems, there is no static air pressure control device for normal operation. The static air pressure is usually in a range of 2.5 inH2O to 6 inH2O for AHUs with 25 hp or larger motors [Liu et. al., 1995a and b].

The impact of static air pressure level on the fan power has been recognized [Warren and Norford, 1993]. The use of a static air pressure reset schedule has been investigated [Rose and Kopko, 1994]. The excessive static pressure can also cause excessive air flow through older dampers. An excessive static pressure at the hot air damper not only initiates a higher hot air flow but also increases cold air flow to compensate. Likewise, an excessive static pressure at the cold air damper also increases the hot air flow to compensate for the excessive cold air flow (this assumes the flow controller is also bad and will permit excessive flow).

The excessive air flow can cause a number of problems in building operations: (1) higher heating and cooling energy consumption, and higher fan and pump electricity consumption; (2) lack of capacity to maintain comfort condition during extreme hot or cold periods; (3) "hot" and "cold" complaints in some rooms; and (4) an unacceptable noise level in some rooms.

The static air pressure can be either controlled or influenced by the following measures: (1) installing a hot deck damper in the air handing unit to control the hot duct pressure at the minimum level (retrofit); (2) installing a VFD on the supply fan to control the static pressure in the cold duct at the minimum level (retrofit); and (3) decreasing the hot air temperature to increase hot air flow rate - thus lowering the pressure (O&M). Note that when the static pressures are controlled at the minimum level, the cold deck and hot deck temperature schedules can be optimized and the simultaneous heating and cooling energy consumption can be reduced substantially (O&M).

This paper investigates the impact of the proposed measures on dual-duct constant volume systems by using a model analysis. The measured energy impacts of adding hot deck dampers and VFDs are also presented for two buildings.

MODEL ANALYSIS

The HVAC system model was based on a medical building in Galveston, Texas. The building has five stories with the same floor plan(100 feet by 200 feet). One 120 hp AHU located on the roof supplies 100,000 CFM to the building with an outside air intake of 40%. The AHU has five branches with the longest branch extending 248 feet to the first floor boxes. This includes 100 feet from the fan exit to the branch on the fifth floor, 48 feet between 5th floor and 1st floor branches, and 100 feet of duct on the 1st floor.

The building and HVAC system were simplified as shown in Figure 1. Each branch was simplified to a single zone with a single terminal box. The pressure drop per foot of duct was taken as 0.025 inH2O under the ideal flow rate based on the measured results. This value is within ASHRAE and common design practice [ASHRAE, 1993, and Grimm and Rosaler 1990]. It was also assumed that both the hot and cold ducts are the same size at the same place to simplify the analysis and that all branches are 100% balanced under the design condition. To make the case more representative of a typical commercial building, the outside air intake fraction was assumed to be 20%. The building load and each system component are discussed next.

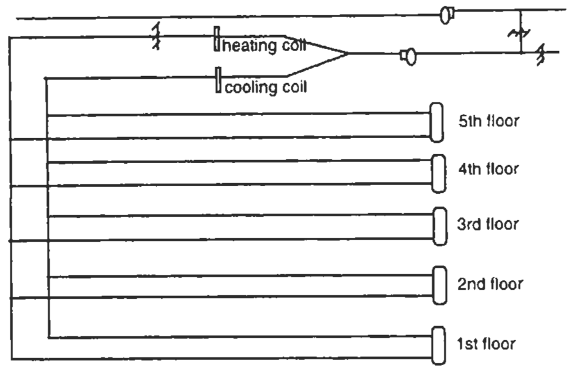

Figure 1: Schematic of the Simplified HVAC System Model

Heating/Cooling Loads: Based on building envelope and internal load conditions, the load profile was represented by equation 1 or Figure 2.

$$\beta = \begin{cases} 0.8 + 0.0167(T_a - 60) & T_a < 60 \\ 0.8 + 0.005(T_a - 60) & T_a > 60 \end{cases} \qquad (1)$$

The room conditions were assumed to be 74°F and 50% relative humidity.

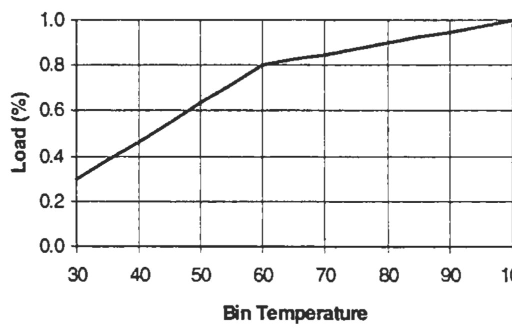

Figure 2: Cooling Load Profile Used in the Model Simulation

Terminal box: Under ideal operation, the terminal boxes maintain constant air flow to each room regardless of the load ratio. The room condition was maintained by adjusting the cold and hot air flow ratio according to Equations 2 and 3.

$$r_{c0} = (1-\beta)\frac{T_r - T_c}{T_h - T_c} \qquad (2)$$

$$r_{h0} = 1 - \gamma_{c0} \qquad (3)$$

Under real operating condition, the total air flow rates may be higher than the design flow due to excessive static pressure at either the hot or the cold air dampers of the terminal boxes. If the static pressure at the hot air damper is higher than that on the cold air damper, the excessive air flow rate was determined by Equation 4.

$$f_h = \alpha \times \max(0,(p_h - 3))$$
$$f_c = f_h \times (T_h - T_c)/(T_r - T_c) \quad (4)$$

If the static pressure at the hot air damper is lower than that on the cold air damper, the excessive air rate is determined by Equation 5.

$$f_c = \alpha \times \max(0,(p_c - 3))$$
$$f_h = f_c \times (T_r - T_c)/(T_h - T_c) \quad (5)$$

The actual hot and cold air flow rates were then determined by summing the ideal flow and the excessive air flow.

$$CFM_c = (r_{c0} + f_c)CFM_0$$
$$CFM_h = (r_{h0} + f_h)CFM_0 \quad (6)$$

It was assumed that the terminal box can maintain the designed air flow rate when the static pressure on the dampers was lower than 3 inH2O. Excessive air flow occurred when the static pressure on either cold or hot air damper was higher than 3 inH2O. The amount of the excessive flow also depends on the quality of the terminal box or the air leakage coefficient (α). The ideal terminal box has a leakage coefficient of 0 while normal terminal boxes may have leakage coefficients between 0.01 and 0.02.

Hot deck: The system design required the discharge air temperature to vary from 85 °F to 100 °F as the ambient temperature changed from 70 °F to 30 °F. When the ambient temperature was higher than 70 °F, the design discharge air temperature was the same as the mixed air temperature.

The hot air temperature could be reduced without degrading the room comfort conditions. The reduced hot air temperature schedule was called the improved operating schedule, and set the hot air temperature at 85 °F when the ambient temperature was lower than 70 °F. When the ambient temperature was higher than 70 °F, the heating coil was shut off. The hot deck operating schedules are shown in Figure 3

The heating energy consumption is calculated by Equation (7).

$$E_h = N \times CFM_h(h_h - h_m)60/v \quad (7)$$

Cold Deck: The discharge air temperature was designed to be 55°F. This low discharge air temperature was indeed required due to the excessive hot air damper leakage, humid outside air condition in the summer, and the internal load condition.

. When the excessive air leakage was eliminated, the cold deck discharge air temperature could be reset to vary from 60°F to 55°F as the ambient temperature varies from 30 °F to 100 °F. Figure 4 compares the designed and the improved cold deck reset schedules.

The cooling energy consumption is calculated by Equation (8).

$$E_c = N \times CFM_c(h_m - h_c)60/v \quad (8)$$

Hot deck damper in the AHU: The purpose of the hot deck damper was to reduce the static pressure at the terminal boxes so that the hot air damper would not allow excessive air flow. In the model simulation, the hot deck damper controlled the minimum hot duct air static pressure at 1.1 inH2O.

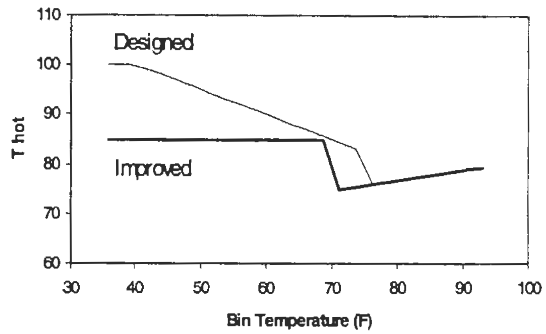

Figure 3: Hot Deck Discharge Air Temperature Schedules

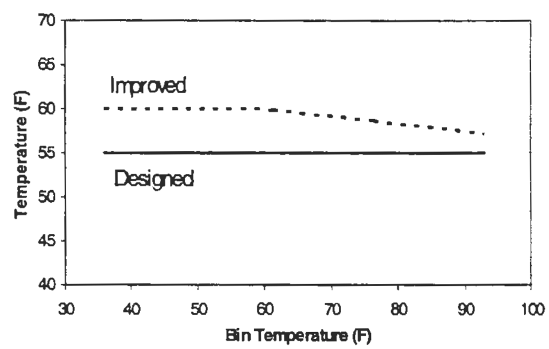

Figure 4: Comparison of the Designed and the Optimized Cold Deck Settings

Variable frequency drive on the supply air fan: A VFD is generally not used for the constant volume system in current engineering practice. It is suggested in this paper that the VFD be used to control the static pressure on the cold air duct to the minimum level. Consequently, the fan power energy consumption is reduced. It also reduces the possible excessive air leakage through the cold air damper. Otherwise the cold air leakage problem become as bad as the hot air leakage problem.

The fan power consumption is calculated by Equation 9.

$$E_f = 0.1174 CFM \times p_0 / \eta \qquad (9)$$

The static pressure is used to calculate the power consumption rather than the total pressure. This underestimates the fan power consumption slightly. However, the impact of this simplification on the fan power savings can be neglected.

The fan efficiency is taken as 0.8 in the model analysis.

Outside air and the return air dampers: These two dampers are used to control the constant outside air intake at 20% of the total air flow rate.

SYSTEM SIMULATION

The system performance is simulated by using Galveston bin weather data which includes ambient dry bulb and dew point temperatures.

First, the mixed air temperature, cold deck and hot deck temperatures, and the load ratio were calculated. Then, the ideal hot and cold air flow rates were determined according to Equations 2 and 3. Finally, the actual hot and cold air flow rates were iterated using the process described below and shown in Figure 5.

Step 1: Determine the static pressure drop per unit length under current hot and cold air flow rates.

$$\delta p_h = CFM_h^{\,2} \times \delta p_0$$
$$\delta p_c = CFM_c^{\,2} \times \delta p_0 \qquad (10)$$

Step 2: Determine the static pressure acting on the hot and cold air dampers.

$$p_h = p_0 - 248 \times \delta p_h$$
$$p_c = p_0 - 248 \times \delta p_c \qquad (11)$$

If the variable frequency drive is used, the static pressure on the cold air damper is assumed to be 1.1 inH20. If a hot deck damper is used, the static pressure at the hot air dampers is assumed to be 1.1 H2O.

Step 3: Determine the excessive air leakage.

If the cold air damper is subjected to a higher pressure, the air leakage rate is determined by Equation 4. If the hot air damper is subjected to a higher pressure, then the air leakage rate is determined by Equation 5.

Step 4: Determine the cold and hot air flow rates according to equation 6.

This procedure was repeated until the cold air flow rate converged to a limit of 0.001%.

The heating and cooling energy consumption, and the fan power consumption are finally determined by Equations 7, 8, and 9.

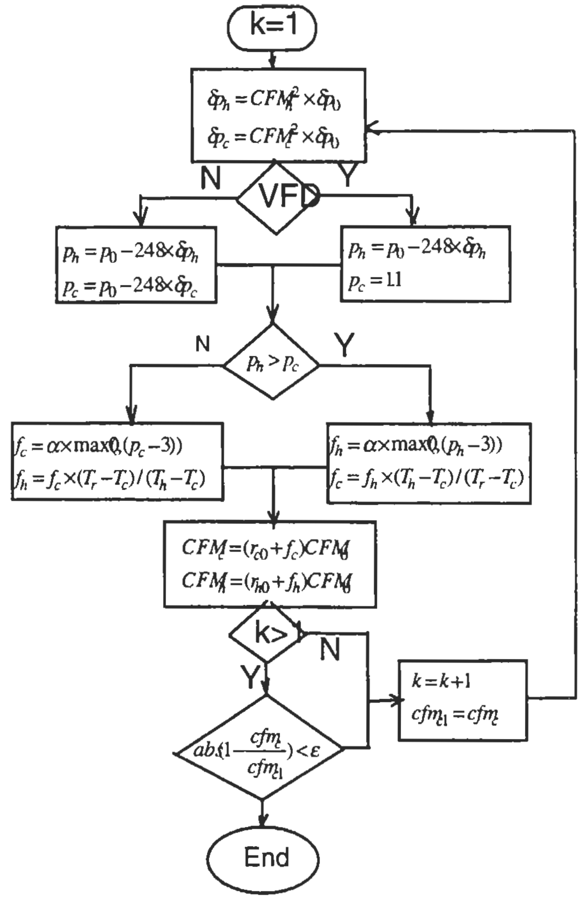

Figure 5: Block Diagram of Iteration Process to Determine the Air Flow Rates

SIMULATION CASES AND RESULTS

Seven cases were simulated to investigate the impacts of different measures on heating and cooling energy consumption. These seven cases are summarized below:

Case 1: Design case. There was no excessive air flow regardless of the static pressure on the dampers. There was no hot deck damper in the main AHU and no VFD on the supply air fan.

Case 2: Actual case. The excessive air flow coefficient was 0.015. Other conditions were the same as in the design case.

Case 3: Hot deck reset. The hot air temperature was 85 °F when the ambient temperature was lower than 70 °F. When the ambient temperature was higher than 70 °F, the hot deck was shut off. The excessive air flow coefficient was 0.015. No VFD nor hot deck damper was present.

Case 4: Install VFD. The VFD controlled the minimum static pressure at 1.1 in H2O. The excessive air flow coefficient was 0.015. No hot deck damper was present.

Case 5: Install hot deck damper. A damper in the AHU controlled the hot air static pressure at 1.1 inH2O. The excessive air flow coefficient was 0.015. No VFD was present.
Case 6: Combination case. Hot deck damper and VFD controlled both hot air and cold air static pressure at 1.1 inH2O. No excess air flow was assumed.

Case 7: Optimal case. Improved hot and cold deck reset schedules were implemented. Other conditions were the same as case 6.

Table 1 summarizes the simulated annual heating, cooling, and fan power consumption under different O&M and retrofit measures. The potential savings were also calculated. For all cases, the base line was taken as the actual case.

The simulation results show that the actual heating, cooling and fan power consumption were significantly higher than the designed annual consumption by 12%, 59% and 48%, respectively, due to lack of control of the static air pressure.

The hot deck reset could reduce the heating by about 26%, cooling by 2%, and fan power by 1%. Installing a VFD could reduce the cooling energy consumption by 7%, heating energy consumption by 22% and fan power by 25%. *Installing a hot deck damper could significantly reduce the heating (38%), cooling (11%) and fan power (16%).* When both a VFD and a hot deck damper were installed, the fan power could be reduced by 41%. When the improved reset deck schedules were implemented, the building energy consumption could be reduced by 39% for cooling, by 54% for heating, and by 42% for fan power.

The energy savings could be achieved because the measures, mentioned above, changed the static air pressure and reduced the excessive air leakage. This mechanism was explained by examining the detailed simulation results for the hot deck damper case.

Figure 6 presents the static air pressure for both cold and hot air. Under actual operation, the hot air static pressure was maintained in a very high range from 8 inH2O to 9 inH2O, while the cold air static pressure was maintained in a very low range from 1 inH2O to 3 inH2O. The hot air static pressure always initiated excessive air leakage. The air handling unit might not be able to maintain suitable room temperature because of the low cold air static pressure and high hot air leakage. The hot deck damper could control the static pressure at 1 inH2O. When the excessive hot air leakage was reduced, the cold air flow requirement was reduced. Consequently, the cold deck static pressure increased (See Figure 2) and the AHU was better able to maintain the room conditions.

Figure 7 shows the simulated air flow ratio defined as the air flow rate divided by the design air flow rate, versus the ambient temperature. Due to the high hot air static pressure, the actual total air flow rate is simulated to be 18% to 25% higher than the designed flow. When the hot air damper was installed, the total air flow rate should be controlled at the design level. To compare with the actual case, the hot deck damper may reduce the total air flow by 15% to 20%.

Figure 8 presents the energy impact of the hot deck damper. Due to the total air flow reduction, the fan power consumption was reduced from 1.6 W/CFM to 1.3 W/CFM with savings of 16%. The cooling energy consumption was reduced by 1.5 Btu/CFM. The heating consumption was also reduced when the ambient temperature was lower than 75°F. The hot deck damper should result in 218,700 kWh/yr (16%) fan power savings, and 3,390 MMBtu/yr (13%) thermal energy savings, which includes 2,570 MMBtu/yr (11%) for cooling, and 900 MMBtu/yr (38%) for heating.

Table 1: Summary of Simulated Annual Energy Consumption Under Different O&M and Retrofit Measures

	Cooling	Heating	Fan Power	Percent Savings		
	(MMBtu/yr)	(MMBtu/yr)	(kWh/yr)	Cooling	Heating	Fan power
Actual	23,000	2,400	1,332,800			
Design	20,400	1,500	899,600	11	38	33
Hot deck reset	22,500	1,780	1,313,000	2	26	1
VFD	21,400	1,870	996,000	7	22	25
Damper	20,500	1,500	1,114,000	11	38	16
Combination	20,400	1,500	788,000	11	38	41
Optimized	14,000	1,100	776,200	39	54	42

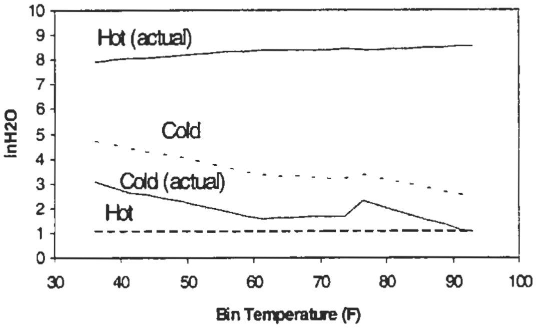

Figure 6: Simulated Static Air Pressure Versus the Ambient Dry Bulb Temperature Under Actual and the Hot Deck Damper Cases

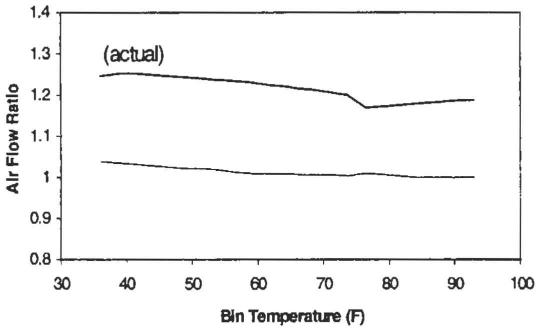

Figure 7: Simulated Air Flow Ratio Versus the Ambient Dry Bulb Temperature Under the Actual and the Hot Deck Damper Cases

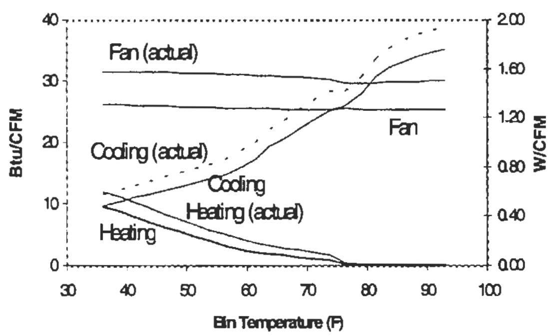

Figure 8: Simulated Fan Power, Heating, and Cooling Energy Consumption Versus the Ambient Dry Bulb Temperature

MEASURED IMPACT OF HOT DECK DAMPER

The hot deck damper was installed in the case building in September 1995. Before the installation of hot air damper, the hot air static pressure was measured as high as 5 inH_2O in a number of rooms in the first floor. The higher static pressure created three problems: (1) dust was blown in rooms when the hot air dampers opened; (2) noise level was as high as 78 db; and (3) room temperature could not be maintained.

After the installation of hot air damper, the hot air static pressure was controlled within a range of 1 inH2O to 4 inH2O according to outside air temperature. The reduced supply static pressure solved all the problems mentioned above. Moreover, significant energy savings were also measured. Figures 9, 10, and 11 compare the measured daily average hourly chilled water, hot water and electricity consumption between pre and post damper installation. The pre-installation data were measured from 09/25/94 to 09/24/95. The post period data were measured from 09/25/95 to 01/10/96.

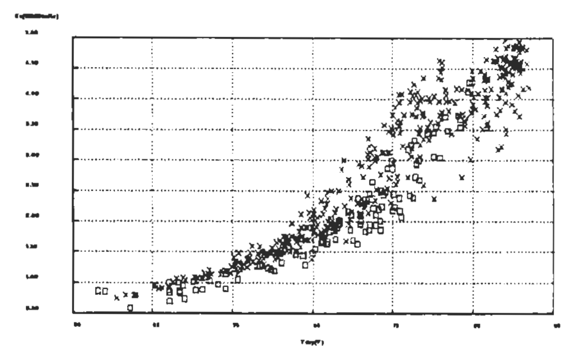

Figure 9: Measured Daily Average Hourly Chilled Water Consumption During Both Pre-damper and Post-Damper Periods in the Case Study Building

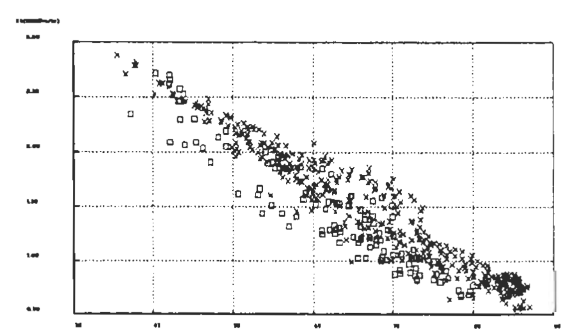

Figure 10: Measured Daily Average Hourly Hot Water Consumption During Both Pre-damper and Post-Damper Periods in the Case Study Building

In the post period (112 days), the measured chilled water savings were 552 MMBtu, or 12% of total consumption. The measured hot water savings were 980 MMBtu, or 15% of total consumption. The measured electricity savings were 49,670 kWh with a demand savings of 18.5 kW, which is about 21% of the supply air fan capacity. These energy savings converted to a total cost savings of $12,610 or $113/day with the following energy prices: $7.30/MMBtu for chilled water, $5.055/MMBtu for hot water, $8.07/kW for peak demand and $0.02659/kWh for electricity. The project cost was $4,040, which was paid back in the first month.

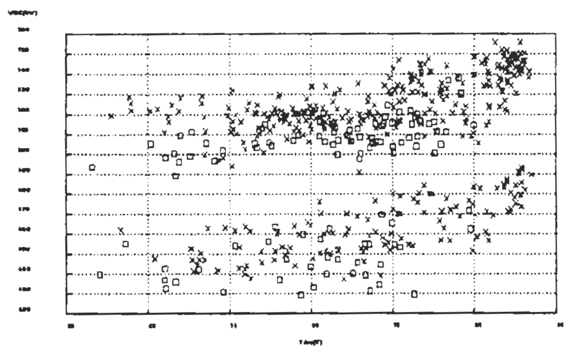

Figure 11: Measured Daily Whole Building Electricity Consumption Versus the Ambient Temperature in the Case Study Building

MEASURED IMPACT OF VFD

On February 24,1995, five Variable Frequency Drives (VFD) were installed on five major supply air fans in a medical building in Houston. The building had a total conditioned floor area of 276,000 ft^2 over seven floors. Five variable air volume (VAV 125 hp each) AHUs, located in the basement, had used inlet guide vanes to regulate air static pressure. The air static pressure had been controlled in a range of 3 inH$_2$O to 4 inH$_2$O. After the installation of VFDs, the air static pressure was gradually adjusted to 1 inH$_2$O. Consequently, significant energy savings were measured. Figures 12, 13 and 14 compare the measured whole building electricity, cooling, and heating energy consumption between pre-VFD and post-VFD periods.

After the supply air static pressure was reduced from 4 inH$_2$O to 1 inH$_2$O, the whole building electricity consumption was reduced by 80 kW, or 17% of the fans' capacity. The total electricity savings were measured as 270,000 kWh in 137 days with a peak demand reduction of 80 kW/month.

The cooling energy consumption was reduced by 1.3 MMBtu/hr or 23% of total consumption. The cooling energy reduction was due to (1) reduced fan power consumption (0.27 MMBtu/hr); and (2) reduced excessive air leakage (1.1 MMBtu/hr). The reduced air leakage simultaneously reduced heating and cooling and improved room conditions.

The heating energy consumption was reduced by 0.4 MMBtu/hr, or 30% as the ambient temperature varied from 55°F to 75°F. The higher post-VFD heating consumption at lower temperature was probably due to inappropriate reaction to the cold weather by the operators. The measured total energy savings were $50,770 in 137 days, or $370/day.

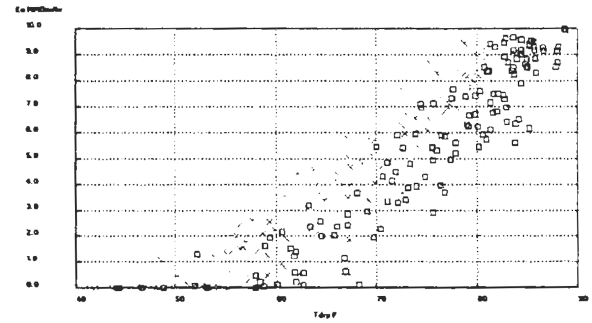

Figure 12: Measured Daily Average Hourly Chilled Water Consumption During Both Pre-VFD and Post-VFD Periods in the Case Study Building

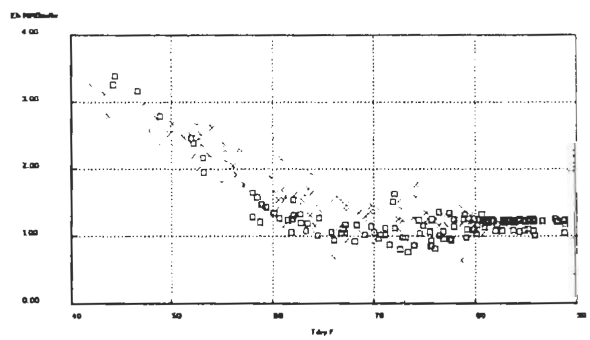

Figure 13: Measured Daily Average Hourly Hot Water Consumption During Both Pre-VFD and Post-VFD Periods in the Case Study Building

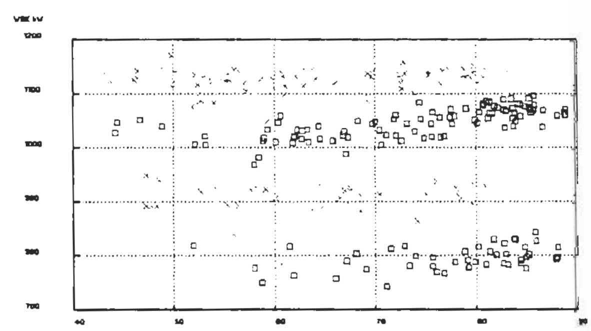

Figure 14: Measured Daily Whole Building Electricity Consumption Versus the Ambient Temperature in the Case Study Building

CONCLUSIONS

Excessive leakage from either the hot or cold duct dampers in mixing boxes of dual duct systems is a major factor in energy loss, poor room thermal conditions, high noise levels, and lack of capacity in some cases. Decreasing the duct static pressures can effectively eliminate these problems.

A hot deck damper was installed to reduce supply air static pressure from 4 inH_2O to 1 inH_2O in a case study building in Galveston, Texas. The lower static pressure eliminated dust from diffusers, reduced room noise levels, and improved indoor temperature control capability. Moreover, the measured energy savings was $113/day. The project was paid back in the first month after the hot deck damper was installed.

Five VFDs were used in another building to reduce the supply static pressure from 4 inH_2O to 1 inH_2O in dual duct VAV systems. The measured electricity use reduction was 18% of the fan capacity. The cooling energy was reduced by 23%. The heating consumption was reduced by 30% when the ambient temperature varied from 55°F to 75°F. The measured total energy savings were $50,770 in 137 days, or $370/day.

It should be pointed out that the model analysis was used to demonstrate the theory. The numerical results from the model analysis should be quoted with caution.

REFERENCES

ASHRAE, 1993. *ASHRAE Handbook: 1993 Fundamentals.* American Society of Heating, Refrigeration and Air-Conditioning Engineers, Atlanta, GA.

Liu M., Y. Zhu, and D. E. Claridge. 1995. *Potential O&M Savings by Eliminating Excessive Air Flow in the Clinical Science Building at UTMB.* Internal Report of the Energy Systems Laboratory, Texas A&M University, College Station, TX.

Liu M., J. A. Eggebrecht, D. E. Claridge. 1995. *Measured Energy Performance of Air Handling Units and Terminal Boxes in Clinical Science Building at UTMB.* Internal Report of the Energy Systems Laboratory, Texas A&M University, College Station, TX.

Liu M., 1995. *Measured Impact of VFD and Static Pressure Reduction on Energy Consumption.* Internal Report of the Energy Systems Laboratory, Texas A&M University, College Station, TX.

Liu M., A. Athar, Y. Zhu, and D. E. Claridge, 1995. Reduce Building Energy Consumption by Improving Supply Air Temperature Schedule and Recommissioning the Terminal Boxes. ESL-TR-95/01-04. Internal Report of the Energy Systems Laboratory, Texas A&M University, College Station, TX.

Nils R. Grimm and Robert C. Rosaler, 1990. *Handbook of HVAC Design.* Mc Graw-Hill Publishing Company, New York, NY.

Rose J. R., and W. L. Kopko. 1994. "A Novel Method for Resetting Duct Static Pressure for Variable Air Volume Systems". *Proceedings of ACEEE 1994 Summer Study on Energy Efficiency in Buildings. Vol. 5. pp. 219-223.*

Warren M., and L. K. Norford. 1993. "Integrating VAV Zone Requirements with Supply Fan Operation". *ASHRAE Journal*, Atlanta. GA. April, pp.43-46