Importance of Monitoring Air Handler Performance

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Introduction

Detailed monitoring of energy systems has multiple benefits, including (i) improved energy accounting, (ii) system diagnostics, and (iii) performance monitoring. In the Texas LoanSTAR program (Claridge et al. 1991), over fifty buildings and selected individual systems in buildings are being monitored to study specific energy conservation measures. This paper describes the value of monitoring air handlers in diagnosing problems which impair system efficiency and effectiveness.

Description of the Building

A large engineering center (EC) in Central Texas with a conditioned area of 240,000 ft^2 (324,000 gross ft^2) was instrumented extensively to monitor the hourly energy end use (beginning May 1989) (Katipamula and Claridge 1992). The EC had twelve identical large air handlers and one was instrumented as shown in Figure 1. In addition, the whole building data collected included electricity use, air handler electricity, chilled water load (Btu), hot water load (Btu), and hot water and chilled water pump run times. A weather station on the roof of the EC collected outdoor dry-bulb temperature, relative humidity, horizontal solar radiation and wind velocity data.

Monitoring of Air Handler Performance

The monitored air handler (AHU-10) is a dual-duct constant volume (DDCV) system with a flow rate of 20,000 cfm including 2,000 cfm of outside air when the outside air damper is fully open. It has a motor rated at 40 hp (29.8 kW). The heating coil is a two-row fin and tube heat exchanger with a surface area of 34.66 ft^2 while the cooling coil has eight fin and tube rows with a surface area of 62.16 ft^2.

Fan power consumption as a function of air flow is shown in Figure 2. The power consumption of the constant volume fan motor over a period of seven months varied by only three percent while the flow rate varied by ten percent. The pressure drop across the fan remained constant at about 6.5 in. of water throughout this period.

A cross plot of outdoor air temperature and mixed air temperature is shown in Figure 3. The mixed air temperature is strongly correlated with the outdoor air temperature, showing a slope of about 0.1°F per°F, consistent with 10% outdoor air. The scatter in mixed air temperature at fixed outdoor air temperature is probably due to changes in return air temperature or imperfect mixing. Time series graphs of outdoor air temperature and mixed air temperature for a two-week period in December 1990, are shown in Figure 4. On December 20, 1990, the outdoor temperature dropped from 75°F to 15°F, causing the mixed air temperature to drop from 75°F to 69°F, again consistent with 10% outside air. The operators normally close the outdoor air dampers when freezing weather is eminent and this would have reduced the heating load by approximately 2 MMBtu/h.

The cold deck supply temperature also shows a strong correlation to the outdoor air temperature (Figure 5). The range in the cold deck temperature is the same as the mixed air range (14°F) and the scatter increases with outdoor temperature as well.

The design cold deck supply temperature was 53°F. However, the cold deck supply temperature never reached the design temperature during the monitoring period. Since the central plant did not have a strainer to filter dirt on its chilled water lines, the cooling coils were severely fouled, so parts of the cooling coil no longer cooled supply air. Therefore, they were effectively operating with a partial bypass. When the outdoor dry-bulb temperature was above 60°F, the variation in the cold deck temperature at a given outdoor temperature increased, probably due to changes in latent load and/or return air temperature.

The hot deck supply temperature was designed to be controlled by the outdoor air reset schedule shown as the solid line in Figure 6. The data points shown in the figure are the measured hot deck supply temperatures. The hot water pumps are supposed to be staged on when the pressure from the central plant is insufficient to meet the heating load. One of the two hot water pumps was on continuously, leading to hot deck supply temperatures above 100°F, except between August 12 and 14 when both pumps were off (Figure 7) and the hot deck supply temperature dropped to 80°F. At low outdoor temperatures (below 40°F) the hot deck supply temperatures were lower than the schedule. If the second pump had been on during cold weather, the actual hot deck supply temperatures might have followed the schedule. However, the scatter in the hot deck supply temperature below 70°F is still large, indicating improper control.

Outdoor, mixed air, and cold deck specific humidities for November 8 to December 20, 1990 are shown in Figure 8. The mixed air and cold deck specific humidity profiles have the same trend. The average chilled water supply and return temperatures are 45°F and 55°F, respectively. The average coil surface temperature would be approximately 50°F. At 50°F dew point the specific humidity of the air is about 0.008 lbw/lba. Therefore, below a mixed air specific humidity of 0.008 lbw/lba, the specific humidity of the cold deck is approximately the same as that of the mixed air.

Without monitoring the air handler continuously the malfunction of the hot and cold deck temperature controls would have been difficult to detect. In buildings where the air handlers are not monitored continuously the air handlers should be periodically checked. However, most of the physical plant personnel spend their time attending to day-to-day crisis. Unless the building has an energy management control system (EMCS) or some type of continuous monitoring, these problems tend to go undetected for long periods of time.

Summary

Electricity energy used by the air handlers varied by 3% while the air flow varied by 10% over the period of a typical year.

The monitoring effort identified three problems in air handler operation:

(i) the cold deck temperatures showed a strong relationship to the outdoor air temperature (due to fouled coils), although the controls were intended to maintain a constant supply temperature

(ii) the outdoor air dampers were left open during a hard freeze, when operating practice called for them to be closed, causing increased heating consumption

(iii) the hot deck supply temperature was controlled based on an outdoor air reset schedule; however, the measured hot deck temperatures varied significantly from the reset schedule, due to a control malfunction.

References

Katipamula, S., and Claridge, D. E. 1992. "Use of Simplified System Model to Measure Retrofit Savings," ASME International Solar Energy Conference April 5-9, 1992, Maui, Hawaii.

Claridge, D. E., Haberl, J. S., Turner, W. D., O'Neal, D. L., Heffington, W. M., Tombari, C., Roberts, M., and Jaeger, S., 1991. "Improving Energy Conservation Retrofits with Measured Savings, ASHRAE Journal, Vol. 33 No. 10 pp 14-22.

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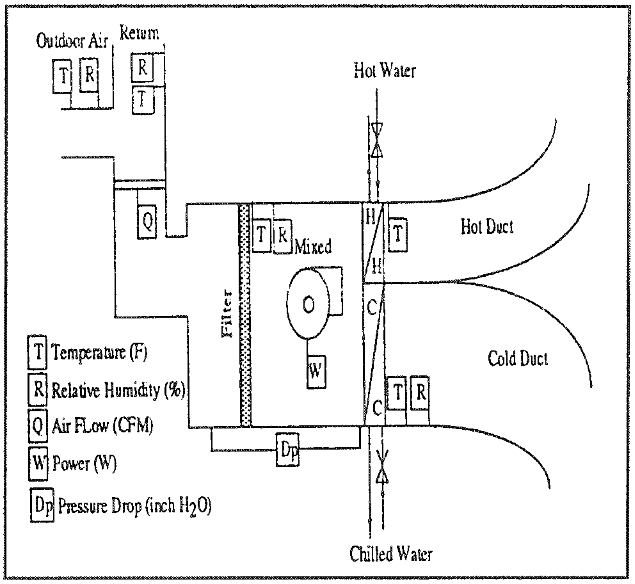

Figure 1. Air Handling Monitoring Schematic

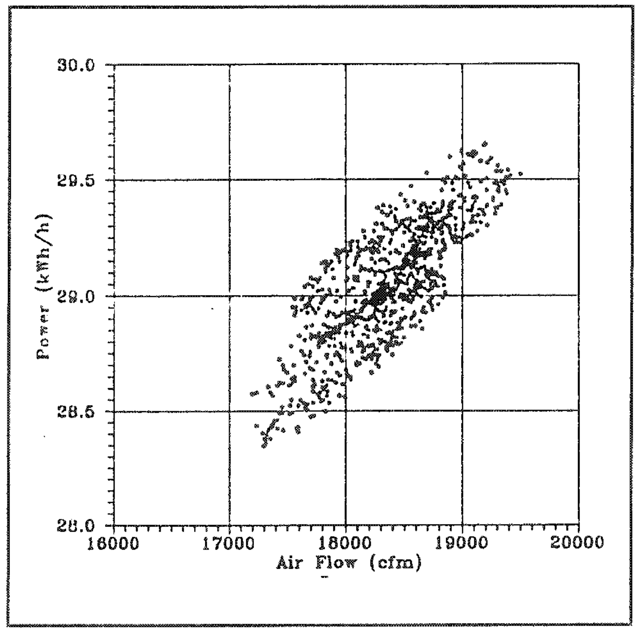

Figure 2. Fan Motor Power Consumption as a Function of Air Flow Rate

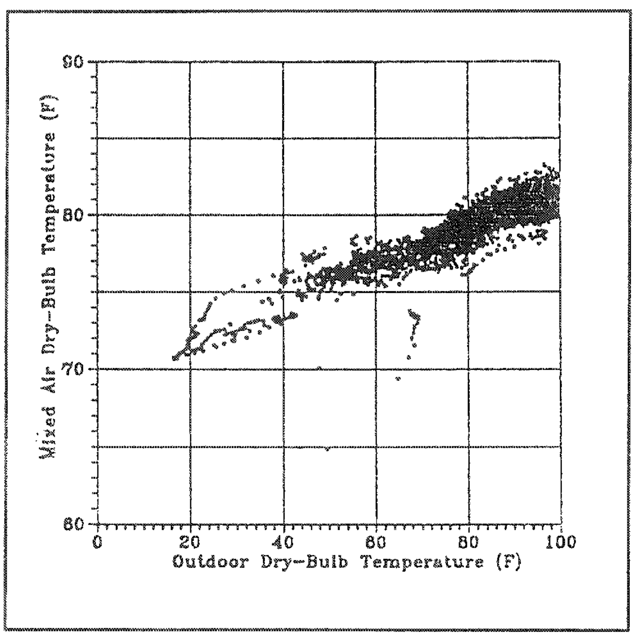

Figure 3. Mixed Air Dry-Bulb Temperature vs Outdoor Dry-Bulb Temperature

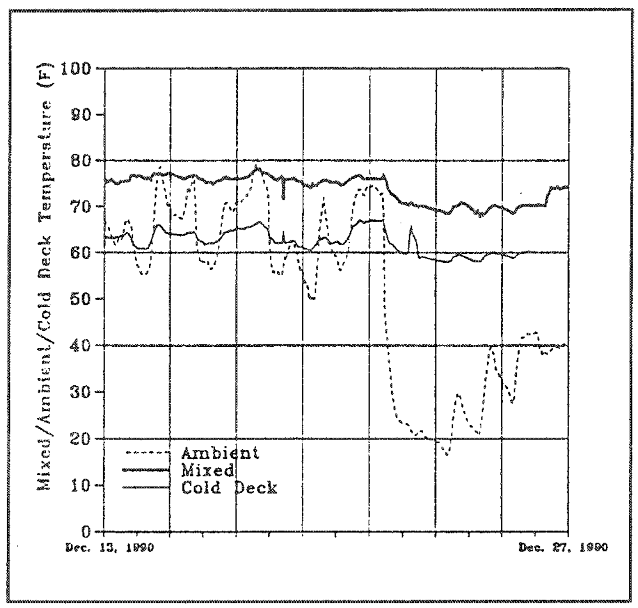

Figure 4. Mixed, Cold Deck and Outdoor Dry-Bulb Temperature

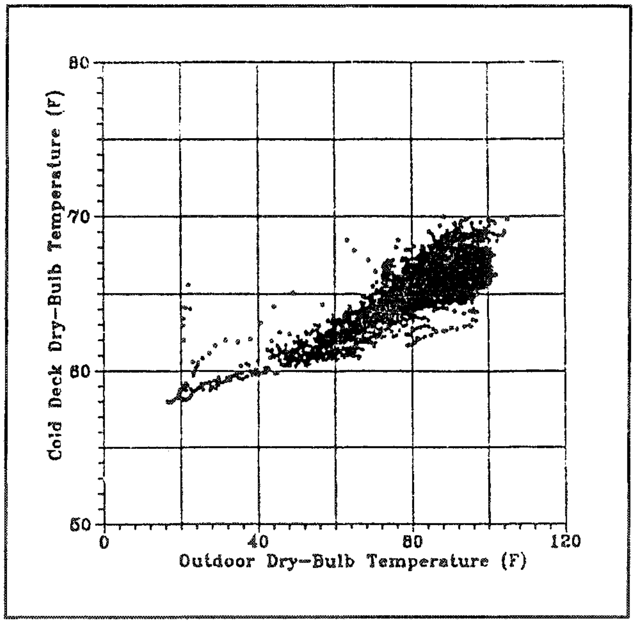

Figure 5. Cold Deck Supply Dry-Bulb Temperature vs Outdoor Dry-Bulb Temperature

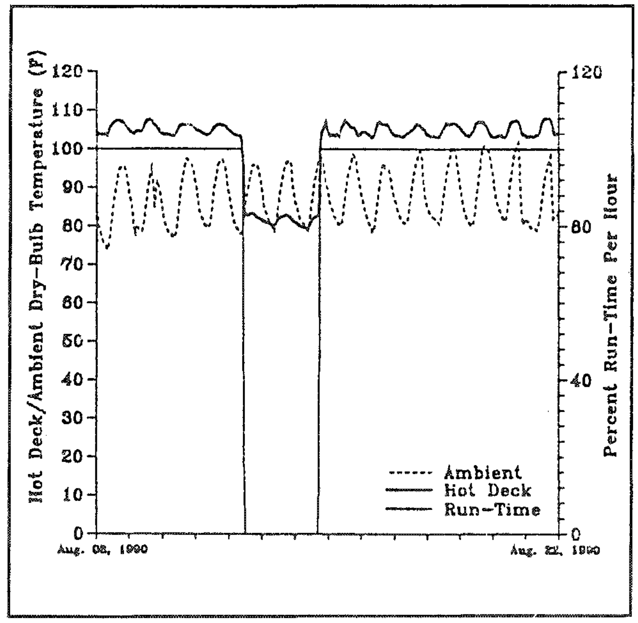

Figure 6. Hot Deck Supply Temperature vs Outdoor Temperature

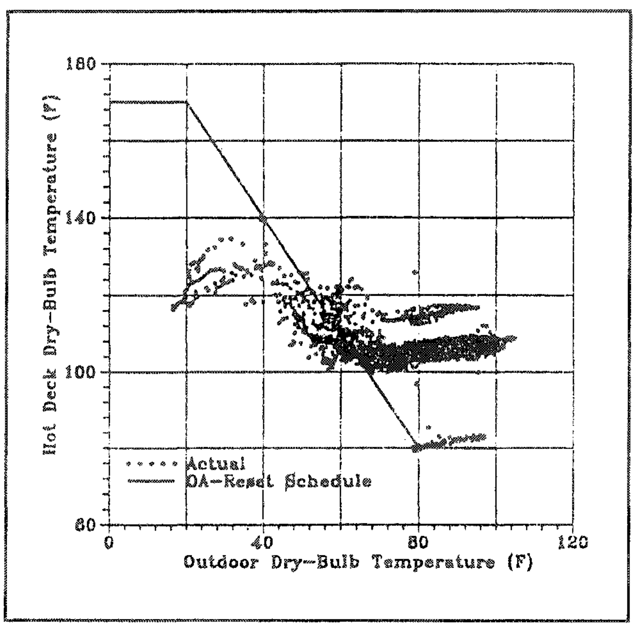

Figure 7. Hot Deck Supply and Outdoor Dry-Bulb Temperature

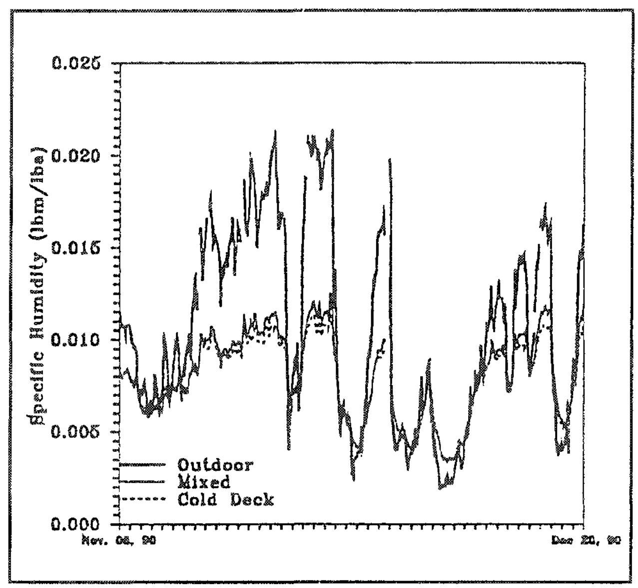

Figure 8. Mixed Air, Cold Deck and Outdoor Dry-Bulb Specific Humidities