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Airflow Reduction to Improve Building Comfort and Reduce Building Energy Consumption—A Case Study

Mingsheng Liu, Ph.D., P.E.
Member ASHRAE

Yeqiao Zhu, Ph.D.
Member ASHRAE

B.Y. Park, Ph.D., P.E.

David E. Claridge, Ph.D., P.E.
Member ASHRAE

Denis K. Feary

ABSTRACT

To remedy comfort problems in a 99,000 ft² (9,200 m²) office building, the total airflow rate was reduced by 35%, and the total outside airflow was reduced by 86% in four multi-zone air-handling units that serve the office building. After the airflow reduction, the peak room relative humidity level was reduced from 70% to 55%, and cold and hot deck reset schedules were implemented. These improved operating practices reduced building energy consumption by 27%.

INTRODUCTION

Indoor comfort conditions were improved in a Texas office building after the airflow rate was reduced and the cold and hot deck temperatures were reset. The case study building, located in Austin, Texas, consists of one three-story section and one six-story section with a total floor area of 99,000 ft² (9,200 m²). The three-story section was built around the turn of the century as a bakery. The six-story section was built as a separate bank in 1946. The two buildings were connected and renovated to form an office building in 1963.

In 1982, a replacement HVAC system was installed. However, the newly installed system caused a series of indoor air quality problems. Although a number of retrofits were performed, neither the indoor air quality problems were fixed nor was the anticipated energy efficiency obtained until 1995 when the authors recommended changes in operating practices for the building. This paper presents the processes involved and the measured results.

BUILDING AND HVAC SYSTEMS

In 1982, two 175-ton hermetic centrifugal chillers were installed in the basement to provide cooling. Two 2.4 MMBtu/h gas-fired boilers were also installed on the sixth floor parking garage ramp to provide heating. Four multi-zone air-handling units (AHUs) were installed to deliver the heating and cooling air (see Figure 1 for the schematic

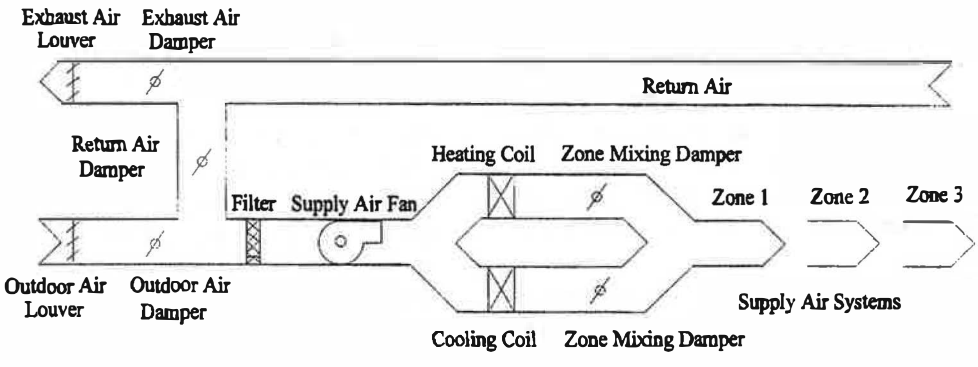

Figure 1 *Schematic diagram of the air-handling units.*

Mingsheng Liu is assistant director and **Yeqiao Zhu** is assistant research scientist in the Energy Systems Laboratory at Texas A&M University, College Station. **B.Y. Park** is director of Education, Korea Institute of Construction, Inchon. **David E. Claridge** is a professor in the Mechanical Engineering Department at Texas A&M. **Denis K. Feary** is energy manager in the State of Texas General Services Commission, Austin.

TABLE 1
Summary of Design Information for Each AHU

AHU			Area Supplied on Each Floor (ft²)				Total
Name	Hp	cfm	1st	2nd	3rd	4th	
A	30	36,800	10,100	10,100	10,100		30,300
B	40	44,300		20,600	20,200		40,800
C	20	21,125	21,100				21,100
F	7.5	10,000			2,900	3,900	6,800
Total	97.5	112,225	31,200	30,700	33,200	3,900	99,000

diagram). The outside air intake was designed to range from 8% to 15% of total airflow rate. Three of the four AHUs were equipped with economizers. No return air fan was installed for any of the units. Table 1 summarizes the basic AHU design information. AHUs A and B were located on the garage ramp. Since neither AHU had a return air fan, the pressure in both mixing chambers and the ambient portion of the return air ducts was negative. Consequently, when either exhaust air damper was open, polluted garage air would be drawn in through the open damper(s). Indoor air quality consultants were hired in the following years to solve the problem.

The consultants suggested installation of return air fans and ducting the return air from each room. These suggestions were rejected by the owner due to high costs and lack of confidence in the suggested measures. The consultant then suggested an increase in the outside air intake and sealing the exhaust outlet to prevent garage air "back flow." This solution successfully prevented entry of polluted garage air; however, the following problems soon appeared: (1) the HVAC system lacked the necessary capacity to cool the building in the summer and to heat the building in the winter; (2) the room relative humidity increased to 70% during summer months; (3) energy consumption increased significantly; (4) the security door could not be closed automatically due to overpressurization of the building; and (5) AHUs blew dirt into the rooms.

To remedy the overpressurization problem, in 1990 the building operator cut a 4 ft by 4 ft (1.2 m by 1.2 m) opening in one of the exterior walls to reduce the building pressure and let the security doors close automatically. However, the other problems were still present.

Installation of an energy management and control system (EMCS), along with numerous control valves and dampers, was completed during October 1994. The EMCS had the following capabilities: (1) nighttime and weekend shutdown; (2) cold and hot deck reset based on either outside air temperature or positions of zone dampers; (3) room condition monitoring; and (4) optimized chiller operation. The EMCS provided improved monitoring of room conditions, but due to the existing physical problems, capabilities 1, 2, and 4 could not be implemented. There was no noticeable improvement in the indoor comfort conditions.

In 1995, the authors corrected the airflow problems by using existing balance and control dampers for each AHU. After the airflow corrections, improved cold and hot deck reset schedules were implemented by using the EMCS system. As a result, the annual energy use index was reduced 33% from 150,800 Btu/ft² a year (1.71 GJ/m² a year) to 101,000 Btu/ft² a year (1.15 GJ/m² a year). The peak room relative humidity level was reduced from 70% to less than 55%.

AS-FOUND BUILDING AND HVAC SYSTEM PERFORMANCE

On June 8, 1995, the AHUs and the building thermal conditions were inspected. The measurement results are summarized in Table 2.

Airflow Rate. The measured total supply airflow rate was 140,700 cfm (66,400 L/s), which was 25% higher than the design value of 112,225 cfm (52,960 L/s) for the entire building. For units A, C, and F, the measured airflow rates were 99%, 32%, and 28% higher than the design values, while the airflow in AHU B was 39% less than the design value.

Outside Air Intake. The measured total outside air intake was 73,000 cfm (34,450 L/s), or 0.74 cfm/ft² (5.9 L/s·m²), which was seven times higher than the required value of 0.10 cfm/ft² (0.80 L/s·m²) or 20 cfm per person (9.6 L per person).

Deck Setpoint and Supply Air Temperature. The measured results showed that the cold deck temperature varied from 53.0°F (12°C) to 55.8°F (13°C) with an average value of 53.9°F (12°C). The heating coils were turned off. The measured air temperature leaving the diffusers varied from 59°F (15°C) to 65°F (18°C), which indicated significant mixing of hot deck air and cold deck air at the zone mixing dampers (see Figure 1).

Room Conditions. The room temperatures and relative humidity levels were measured at 18 locations from 2:00 p.m. to 4:00 p.m. on June 8, 1995, when the ambient temperature was 88°F (31°C) and the ambient relative humidity was 60%. The room temperatures varied from 67.3°F (20°C) to 74.5°F (24°C). The room relative humidity levels varied from 58% to 69%.

Building Positive Pressure and Air Infiltration. As mentioned previously, there was a 4 ft by 4 ft (1.2 m by 1.2 m) opening in the east wall on the first floor. Approximately

TABLE 2
Summary of AHU Measurement Results

AHU	A	B	C	F	Total/Average
Floor Area (ft^2)	30,300	40,800	21,100	6,800	99,000
Supply Air (cfm)	73,200	26,940	27,770	12,800	140,700
Supply Air (cfm/ft^2)	2.42	0.66	1.32	1.88	1.58
O. A. (cfm)	43,500	12,750	10,320	6,455	73,025
O. A. Fraction (%)	59%	47%	37%	50%	52%
Cold Deck Temp. (°F)	55.8	53.3	53.5	53.0	53.9
Ambient Temp. (°F)	88.8	88.2		86.0	87.7
Return Air Temp. (°F)	78.5	76.7	74.8	74.0	76.0
Static Pressure (in H_2O)	2.3	1.6		1.4	1.8
Room Air Temp. (°F)	72.8	73.2	73.6	72.0	72.9
Supply Air Temp. (°F)		65.6		59.3	64.5
Room RH (%)	68.8	64.2	59.5	62.7	63.4

8,000 cfm (3,780 L/s) of air flowed out of the building through this opening. The positive pressure was measured as 0.10 in. H_2O (26 Pa). When the opening was covered, the positive pressure increased to 0.15 in. H_2O (38 Pa).

IMPROVED OPERATING PROCEDURES

During the site visit, the following problems were identified: (1) room relative humidity levels were as high as 69%; (2) room temperatures could not be maintained at comfortable levels during peak summer and winter weather (according to operators and office workers); (3) both cold and hot "spots" coexisted in a number of rooms; (4) AHUs blew dust into the rooms; and (5) the security door could not be closed automatically. It appeared that all of these problems originated from the high total airflow and high outside airflow. Consequently, the following improved operating procedures were proposed.

Suggestion 1: reduce the total air supply rate from 1.42 cfm/ft^2 (7.2 L/s·m^2) to 0.88 cfm/ft^2 (4.5 L/s·m^2), reduce the outside air intake from 0.74 cfm/ft^2 (3.8 L/s·m^2) to 0.10 cfm/ft^2 (0.5 L/s·m^2) (see Table 3 for details), and correct zone airflow rates based on the zone loads.

Suggestion 2: optimize the cold and hot deck reset schedules.

Table 4 compares the as-found and recommended cold and hot deck reset schedules. The recommended schedules were developed using in-house air-side simulation software. The detailed calibration and optimization procedures are presented by Liu and Claridge (1995). The projected energy impacts of the recommended operating procedures were determined using the same simulation program.

Suggestion 3: do not implement suggestion 2 until suggestion 1 is implemented.

Analysis of the Improved Operating Procedures

The reduced total airflow and outside airflow rates will significantly improve the room relative humidity conditions and reduce the cooling and heating energy consumption as shown in the following example. The impacts of the reduced airflow rates may be seen in Figures 2 and 3.

Assume that before reducing the supply air and outside air flow rates, the zone supply air temperature is 65°F (18°C). Further assume the following conditions: (1) outside air condi-

TABLE 3
Summary of Airflow Management

AHU		A	B	C	F	Total
Total cfm	Current	73,200	26,940	27,770	12,800	140,700
	Suggested	25,700	37,700	17,900	5,800	87,100
	Reduction	47,500	−10,760	9,870	7,000	53,610
O. A. cfm	Current	43,500	12,750	10,320	6,460	73,000
	Suggested	3,000	4,000	2,000	600	9,600
	Reduction	40,500	8,750	8,320	5,860	63,400

TABLE 4
Summary of Airflow Management

AHU		A	B	C	F	Total
Total Air (cfm)	As-Found	73,200	26,940	27,770	12,800	140,700
	Recommended	25,700	37,700	17,900	5,800	87,100
	Reduction	47,500	−10,760	9,870	7,000	53,610
Outside Air (cfm)	As-Found	43,500	12,750	10,320	6,460	73,000
	Recommended	3,000	4,000	2,000	600	9,600
	Reduction	40,500	8,750	8,320	5,860	63,400

tion: 90°F (32°C) and 70% RH; (2) cold air: 55°F (13°C) and 90% RH; (3) hot deck air: mixed air; (4) outside airflow fraction: 50%; (5) room temperature: 73°F (23°C); (6) return air temperature: 75°F (24°C); and (7) relative humidity level increase due to moisture production in zone: 3%. Imposing these conditions on the AHU, we conclude that (1) the hot deck airflow fraction is 0.36 and (2) the room relative humidity is 70%. The AHU psychrometric process is shown in Figure 2.

By reducing the total airflow rate by 38%, the zone supply air temperature will be reduced from 65°F (18°C) to 59°F (15°C). When the outside air intake is reduced to 11%, the calculations show that (1) the hot deck airflow fraction is 19% and (2) the room relative humidity is 53%. This AHU psychrometric process is shown in Figure 3. The reduced total and outside airflow rates will also decrease the cooling energy consumption significantly. Under the weather conditions of this example, the potential cooling energy savings are 63%.

When the total airflow is too high, a significant amount of mixed air bypasses the cold deck to maintain suitable supply air temperature and, thus, the desired room temperature. Since no moisture can be removed by the heating coil, the room relative humidity cannot be controlled at a suitable level. When the total airflow is correct at design cooling conditions, more than 90% of the air should flow through the cold deck. If the cold deck temperature is controlled at a temperature lower than 57°F (14°C), the room relative humidity should be maintained at 55% or lower. Of course, the amount of outside air intake greatly influences the moisture content of the mixed air. Under these conditions, excessive total airflow can cause a high room relative humidity level. Correcting the total airflow will improve the room relative humidity conditions.

Figure 4 shows the predicted heating and cooling energy consumption as a function of outside air temperature before and after airflow reduction. It appears that the improved operating procedures will reduce the peak demand by 40%. Therefore, the AHUs should be able to maintain the cold deck temperature at 55°F (13°C) after the airflow reduction. The potential annual energy sa ings are summarized in Table 5. The boiler efficiency is assumed to be 70%. The chiller kW per ton is assumed to be 1.2, which includes the cooling tower and associated pump power as well. The potential fan power savings are not included.

After the outside air is reduced by 86%, the building pressure should be maintained at a normal level. After the tota airflow is reduced by 38%, the duct vibration should be reduced significantly. Consequently, less dirt should be blown into rooms.

Implementation of Improved Operating Procedures

From November 1 to November 3, the airflows were reduced by closing both return and outside air dampers. The relief air outlets were kept blocked. The economizers were

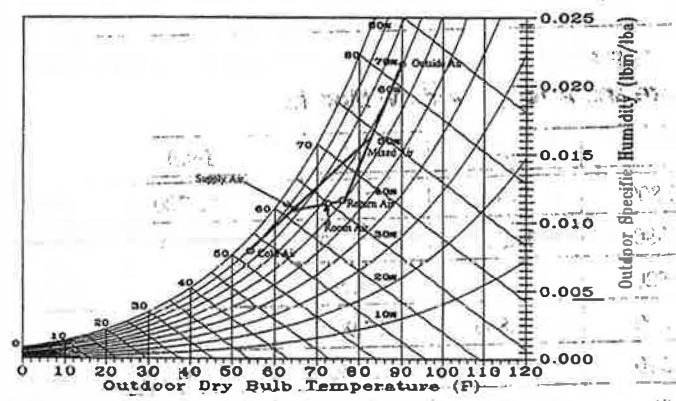

Figure 2 Psychrometric process under as-found conditions within typical Austin weather.

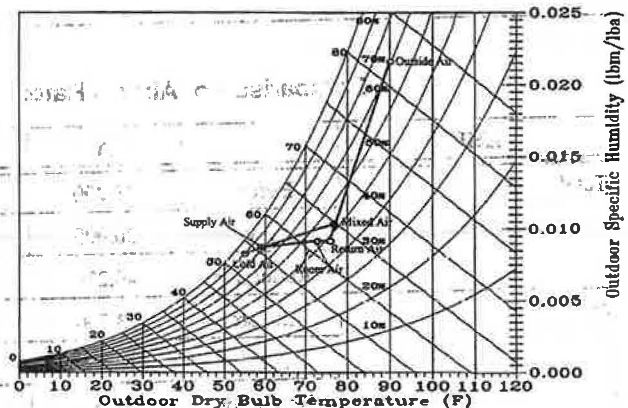

Figure 3 Psychrometric process after airflow reductions with typical Austin weather.

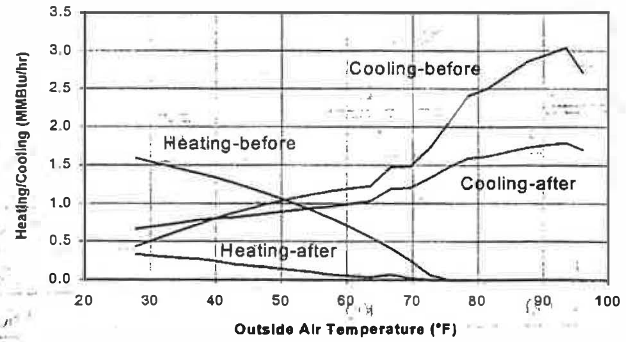

Figure 4 *Comparison of predicted chilled water and hot water consumption before and after the airflow reduction.*

kept disabled. Table 6 presents the results. The total air supply was reduced from 140,700 cfm (66,400 L/s) to 92,130 cfm (43,480 L/s), 0.90 cfm/ft^2 (4.6 L/s·m^2). The outside airflow was reduced from 79,950 cfm (37,730 L/s) to 10,840 cfm (5,120 L/s), 0.11 cfm/ft^2 (0.6 L/s·m^2). The airflow w s increased in AHU B. A relatively high airflow rate (1.3 cfm/ft^2, 6.6 L/s·m^2) was continued in AHU F since it serves a major conference room. The modified whole building airflow rate is 18% lower than the original design value.

D ing November 1995, the recommended cold and hot deck reset schedules were also fully implemented.

MEASURED RESULTS

On June 8, 1995 (before airflow reduction), we measured temperatures, relative humidities, and CO_2 levels in 18 preselected rooms to represent whole building conditions. The pressure across the security door (building pressure) was also measured. On June 14, 1996 (after airflow reduction), we repeated the measurements. Table 7 presents the results.

During the site visit on June 14, 1996, all the office workers said that they were satisfied with the room conditions except in two rooms that were served by AHU F. It was discovered that one of the hot air dampers was malfunctioning. All the office workers who were in the building during the summer of 1995 noticed the significant improvement in comfort conditions.

The measured energy savings agree closely with the predicted energy savings. To compare with the predicted energy savings, the monthly average hourly energy consumption was calculated from utility bill data. The monthly average hourly electricity consumption was determined as the ratio of the billed total electricity consumption and the oper ting hours in the billing period. The monthly average hourly gas consumption was determined as the ratio of total gas consumption and the number of hours in the billing period. The monthly average hourly heating consumption was determined using measured gas consumption with an assumed boiler efficiency of 70%. The monthly average temperature was determined as the average of the measured hourly temperature values in Austin.

The base period was taken from August 1994 to July 1995. The post period was taken from October 1995 to May 1996.

Figure 5 compares the measured monthly average hourly electricity consumption before (◆) and after (■) the implementation of the improved operating procedures. The simple

TABLE 5
Comparison of the As-Found and Recommended Outside Air Reset Schedules

Outside Air Temperature	Cold Deck Reset	Hot Deck Reset
As-Found	54.5°F	$100 - 0.7(T_{o.a} - 40)$°F
Recommended	$T_c = \begin{cases} 64 & T_{o.a} < 60 \\ 57 & T_{o,a} > 60 \end{cases}$	$T_h = \begin{cases} 85 & T_{o.a} < 40 \\ 85 - 0.29(T_{o.a} - 40) & T_{o,a} > 40 \end{cases}$

TABLE 6
Comparison of Airflow Rates Before and After the Airflow Modifications

AHU		A	F	B	C	Total	%
Total cfm	Before	73,200	12,800	26,900	27,800	140,700	35%
	After	26,990	9,980	33,510	21,650	92,130	
	Reduction	46,210	2,820	-6,610	6,150	48,570	
O. A. cfm	Before	43,500	6,500	12,800	17,900	79,950	87%
	After	2,740	2,510	3,940	1,650	10,840	
	Reduction	40,760	3,990	8,860	16,250	69,860	
Design	cfm	36,800	10,000	44,300	21,125	112,225	
Area		30,300	6,800	40,800	21,100	99,000	

TABLE 7

Comparison of Room Comfort Parameters Before and After Airflow Reduction

Item	Before	After
CO_2	400 ppm - 500 ppm	650 ppm - 800 ppm
Room Temperature	67.0°F - 74.5°F	72.0°F - 75.0°F
Room Relative Humidity	58% - 69%	30% - 55%
Building Positive Pressure	0.1 in. H_2O	0.02 in. H_2O

Note: On June 8, 1995, the ambient air temperature was 88°F (31°C) and the relative humidity was 60% (w_{oa} = 0.018) duri g the measurement. On June 14, 1996, the correspondi g conditions were 99°F (37°C) and 50% (w_{oa} = 0.021) during the measurement.

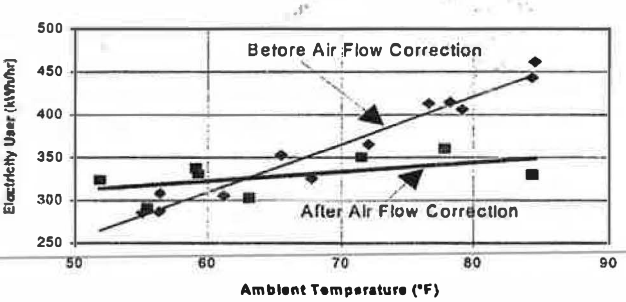

Figure 5 *Measured monthly average hourly whole building electricity consumption vs. the monthly average ambient temperature.*

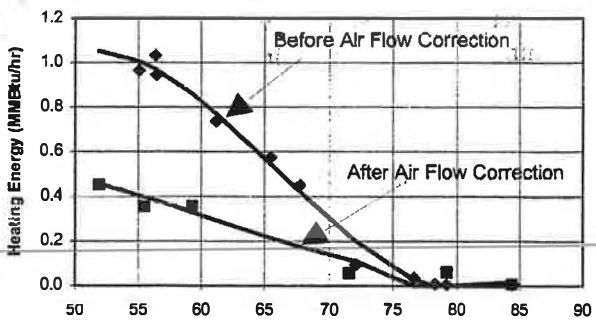

Figure 6 *Measured heating energy consumption vs. the monthly average hourly ambient temperature.*

linear regression models shown in the figure are based on the measured data. When the ambient temperature is low, the electricity consumption is higher than the baseline because less outside air is used. When the ambient temperature is high, the electricity consumption is significantly lower than the baseline. These observations agree with the model prediction (Figure 4). However, the crossover point is different (40°F in Figure 4 and 64°F [18°C] in Figure 5). Use of monthly average values of electricity use and temperature has increased the crossover point since the cooling energy consumption is not linearly dependent on the ambient temperature, as shown in Figure 4. The measured electricity savings are 90 kW when the ambient temperature is 85°F (29°C). Assuming this reduction is due to reduced chilled water consumption, it corresponds to 0.9 MMBtu/h at 1.2 kW/ton. This value is consistent with the predicted value shown in Figure 4. The electricity consumption has increased by 50 kWh/h (0.17 MMBtu/h) when the monthly average hourly temperature is 52°F (11°C) due to the operating changes.

Figure 6 presents the measured monthly average hourly heating energy consumption vs. the monthly average ambient temperature. Simple polynomial regression fits to the data are also presented in the figure. The measured heating energy savings vary from 0.05 MMBtu/h (29 kW) to 0.60 MMBtu/h (205 kW) as the monthly average hourly temperature varies from 75°F (24°C) to 52°F (11°C). The measured gas savings vary from0.12 MMBtu/h (35 kW) to 0.88 MMBtu/h (258 kW) as the monthly average hourly temperature varies from 75°F (24°C) to 52°F (11°C). The measured savings are approximately 15% smaller than the predicted savings.

Figure 7 presents the measured peak electrical demand vs. the monthly average hourly ambient temperature. Simple linear regression models based on the data are also shown in the same figure. The measured peak demand reduction varies from 30 kW to 70 kW as the monthly average hourly ambient temperature varies from 80°F (27°C) to 52°F (11°C). The peak demand reduction indicates smooth cooling system operation.

Based on utility bills (August 94 to July 95 for the baseline and August 95 to February 96 for the post period), the gas energy use index was reduced 51%, from 42 kBtu/yr·ft^2 (0.44 GJ/m^2·yr) to 22 kBtu/yr·ft^2 (0.23 GJ/m^2·yr). The electricity use index was reduced by 21% from 53.5 kWh/yr·ft^2

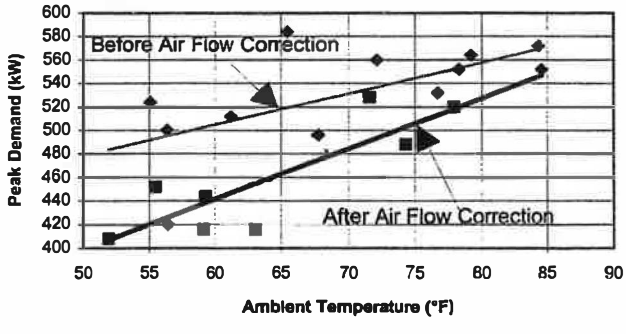

Figure 7 *Measured peak demand vs. the monthly average hourly ambient temperature.*

(576 kWh/m^2·yr) to 42.0 kWh/yr·ft^2 (452 kWh/m^2·yr). The building energy use index was reduced 27% from 225 kBtu/yr·ft^2 (2.34 GJ/m^2·yr) to 165 kBtu/yr·ft^2 (1.72 GJ/m^2·yr).

CONCLUSIONS

In the case study building, the airflow reduction decreased the peak room relative humidity from 70% to 55%. Reducing airflow rates and resetting cold and hot deck temperatures as functions of ambient temperature reduced building annual energy use by 27%. It appears that the excessive air bypass of the cold deck was one of the major reasons for the high indoor relative humidity in the building described. Correcting the total airflow and the outside airflow rates significantly improved the indoor comfort conditions.

ACKNOWLEDGMENT

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