A Comprehensive Continuous Commissioning Improve Building Comfort and Reduce Building Energy Consumption-A Case Study

M. Liu, B. Park, D. Claridge, D. Feary and J. Gains

A comprehensive continuous commissioning turned a sick and energy inefficient building into a comfort, efficient working environment. This case study shows that the reduced excessive supply air flow rate is one of the major measures to control indoor relative humidity level in the hot and humid climates when dual duct systems are used. The building and HVAC system information is presented first, followed by the measurement and analysis results. Finally, the measured room comfort conditions and energy consumption are compared before and after the Continuous Commissioning♦.

Introduction

The Austin, Texas building consists of one 3 story section and one 6 story section with a total floor area of 99,000 ft^2. The 3-story section was built at the beginning of the century as a bakery. The 6-story section was built in 1946 as a bank building. The two buildings were connected and renovated as 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 were the anticipated energy efficiency obtained until the building was recommissioned in 1995. This case study represents the Continuous Commissioning♦ activities and the measured impacts.

Building and HVAC Systems

In 1982, two 175-ton hermetic centrifugal chillers were installed in the basement of the building to handle the increased cooling load. Two 2.4 MMBtu/hr gas fired boilers were also installed in the parking garage ramp on the 6th floor to provide heating. Four multi-zone AHUs (See Figure 1 for the systematic diagram) were installed to deliver the heating and cooling air. The outside air intake was designed to be 8% to 15%. Three of the four AHUs were equipped with economizers. There was no return air fan installed for all the units. The basic AHU design information is summarized in Table 1. AHUs A & B were located on the garage ramp. Since there was no return air fan installed, the pressure at the mixing air chamber would be negative. Consequently, if the exhaust air damper was open, the outside air would flow back through the exhaust air damper.

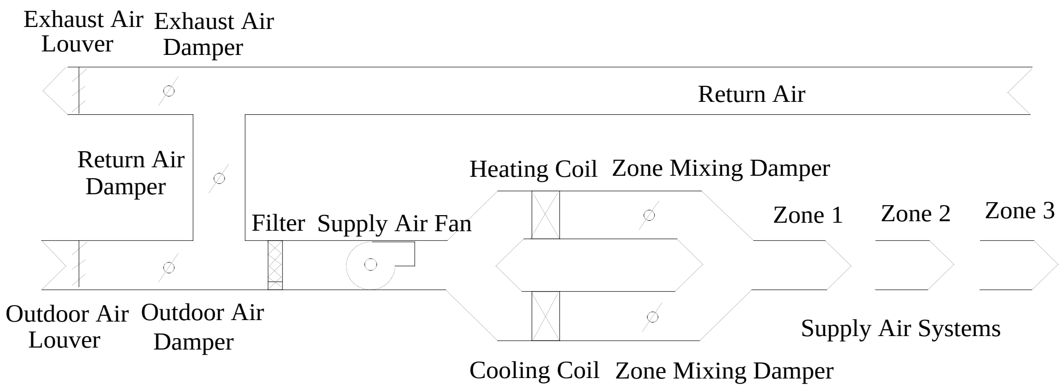

Figure 1: Systematic Diagram of the AHUs

When the AHUs were put into operation, the garage air was being pulled into the rooms through the exhaust ducts in AHUs A and B, which were located on the garage ramp. Indoor air quality consultants were hired in the following years to solve the problem.

It was suggested to install the return air fans and to duct the return air from each room. The suggestion was rejected by the owner due to high costs and lack confidence in the suggestion. It was then suggested to increase outside air intake and seal the exhaust outlet to prevent garage air “back flow”. It worked! However, the following problems soon appeared: (1) the HVAC system lacked capacity to cool the building in the summer and to heat the building in the winter; (2) room relative humidity went up to 70% during summer months; (3) the energy consumption increased significantly; (4) the security door could not be closed automatically due to over-pressurization of the building, and (5) AHUs blow down dirt to rooms.

To remedy the over-pressurization problem, the building operator cut a 4 foot by 4 foot hole in one of the exterior walls to reduce the building pressure to let the security doors close automatically in 1990. However, other problems still existed. The building's annual energy consumption index was 173,300 Btu/ft^2/yr.

Table 1: Summary of Design Information for each AHU

AHU			Area at Each Floor				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	98,900

In October 1994, an Energy Management and Control System (EMCS) was installed. The EMCS had the following capabilities: (1) nighttime and weekend shut down; (2) cold and hot deck reset; (3) room condition monitoring; and (4) optimized chiller operation. Due to the existing problems, most of the EMCS functions could not be implemented. The annual energy index was only reduced from 173,300 Btu/ft^2/yr to 150,800 Btu/ft^2/yr (13% reduction) in FY 1995 (09/01/94 to 08/31/95). There was no noticeable improvement on the indoor comfort conditions.

In 1995, the air flows for each AHU were corrected. After the air flow corrections, the optimized operation schedules were also implemented by using EMCS system. As a result, the annual energy index was reduced from 150,800 Btu/ft^2/yr to 101,000 Btu/ft^2/yr (33% reduction). The room relative humidity level was reduced from 70% to less than 55%.

Measured Building and HVAC System Performance Before Continuous Commissioning

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

Air Flow Rate: The measured total supply air flow rate was 140,700 CFM, which was 25% higher than the designed value of 112,225 CFM for the whole building. For unit A, the measured total air flow rate (73,200 CFM) was 99% higher than the designed value of 36,800. For unit B, the measured air flow (26,940 CFM) was 39% less than the designed value of 44,300 CFM. For unit C, the measured air flow rate (27,770 CFM) was 32% higher than the designed value (21,100 CFM). For unit F, the measured total supply air flow rate (12,800 CFM) was 28% higher than the designed value (10,000 CFM).

Table 2: Summary of AHU Measurement Results

AHU	A	B	C	F	Total/Average
Floor Area sq-ft	30,200	40,800	21,100	6,800	98,900
CFM	73,200	26,940	27,770	12,800	140,700
CFM/sq-ft	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%
Tc	55.8	53.3	53.5	53.0	53.9
Tout	88.8	88.2		86.0	87.7
Tret	78.5	76.7	74.8	74.0	76.0
Pstatic inH2O	2.3	1.6		1.4	1.8
Troom	72.8	73.2	73.6	72.0	72.9
Tsupply		65.6		59.3	64.5
RH room	68.8	64.2	59.5	62.7	63.4

Outside Air Intake: The measured total outside air intake was 73,000 CFM, or 0.74 CFM/sq-ft, which was seven times higher than the required value.

Deck Setpoint and Supply Air Temperature: The measured results showed that the cold deck temperature varied from 53.0°F to 55.8°F with an average value of 53.9°F. The hot decks were off. The measured air temperature leaving the diffusers was from 59°F to 65°F. Significant hot and cold air mixing existed in all AHUs.

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. The room temperature varied from 67.3°F to 74.5°F. The room relative humidity varied from 57.5% to 68.8%.

Building Positive Pressure and Air Infiltration: As mentioned previously, there was a 4 foot by 4 foot hole in the east wall on the first floor. About 8,000 CFM of air leaked out through this hole. The positive pressure was measured as 0.1 inH_2O when the hole was open. When the hole was covered, the positive pressure increased to 0.15 inH_2O.

Improved O&M Measures

During the site visit, the following problems were identified: (1) the cold air temperature could not be maintained at 55°F or lower; (2) room relative humidity were as high as 70%; (3) room temperature could not be maintained at comfort level during peak summer and cold winter; (4) both cold and hot "spots" coexist in number of rooms; (5) AHUs blow dusts to rooms; and (6) the security door could not be closed automatically. It appears that all of these problems originated from the high total air flow and high outside air intake. Consequently, the following improved O&M measures were proposed:

Suggestion 1: reduce the total air supply rate from 1.42 CFM/sq-ft to 0.88 CFM/sq-ft, reduce the outside air intake from 0.74 CFM/sq-ft to 0.1 CFM/sq-ft (See Table 3 for details), and correct zone air flow rate according to the zone load.

Table 3: Summary of Air Flow 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

Suggestion 2: optimize the cold and hot deck reset schedule. Table 5 compares the existing and suggested cold and hot deck reset schedules.

The optimized schedules were developed by using an in-house air side simulation software.

Table 5: Comparison of the Existing and Suggested Outside Air Reset Schedules

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

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

The zone supply air temperature is the indicator of the zone load when the air flow rate is correct. Therefore, when all zones are under the design or the similar load conditions, the zone supply air temperature should be similar. This principal was used to correct the air flow rate for each zone. In the air flow correction process, the air flow was adjusted until all zone supply air temperatures were within ±1°F.

Analysis of the Improved O&M Measures

The reduced total air flow and the outside air flow will significantly improve the room relative humidity conditions and reduce the cooling and heating energy consumption. The impacts of the reduced air flow rate are explained in Figures 2 & 3.

Before reducing the air flow rate, the zone supply air temperature was 65°F. If we assume the following conditions: (1) outside air condition: 90°F and 70%; (2) cold air: 55°F and 90%; (3) hot air: mixed air; (4) out side air flow fraction: 50%, (5) room temperature: 73°F; (6) return air temperature: 75°F; and (7) relative humidity level increase due to moisture production: 3%. To impose these conditions to the AHUs, we conclude: (1) hot air flow fraction: 0.36; and (2) room relative humidity: 68%. The AHUs working process is shown in Figure 2.

After reducing total air flow rate by 38%, the zone supply air temperature will be reduced from 65°F to 59°F. When the outside air intake is reduced to 11%, the calculation showed: (1) hot air flow fraction: 19%; and (2) room relative humidity: 53%. The AHUs working process is shown in Figure 3. The reduced total and outside air flow rates will also decrease the cooling energy consumption significantly. Under the weather condition mentioned above, the potential cooling energy savings were 63%.

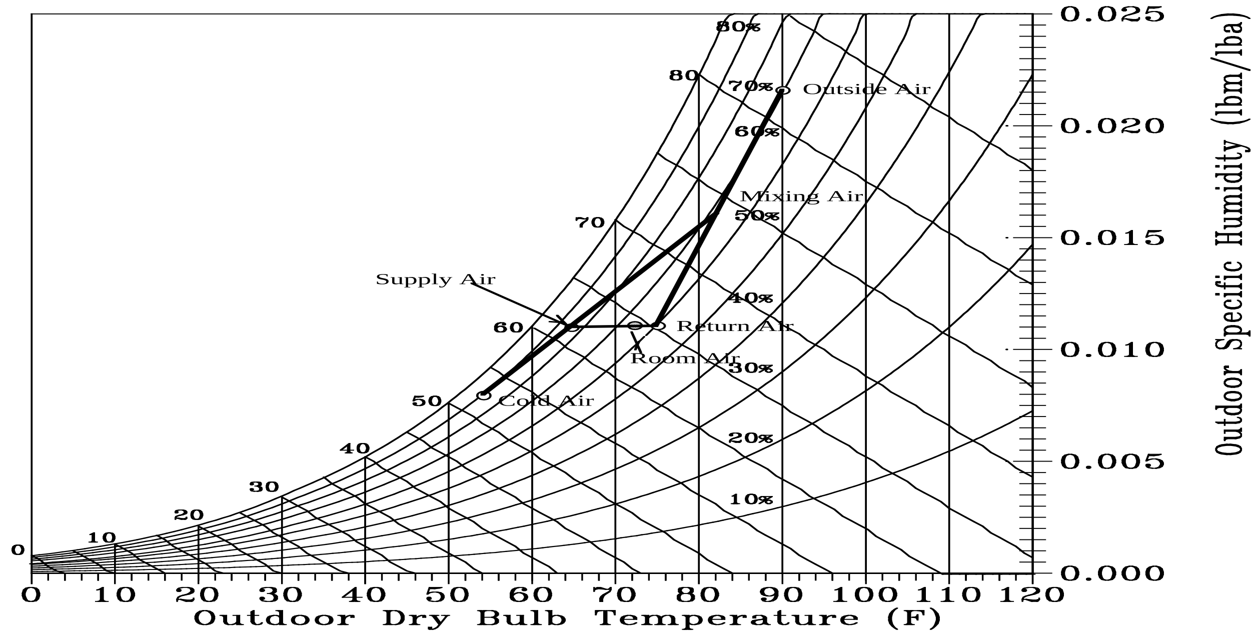

Figure 2: Air Conditioning Process Under Before Continuous Commissioning Under Typical Austin weather

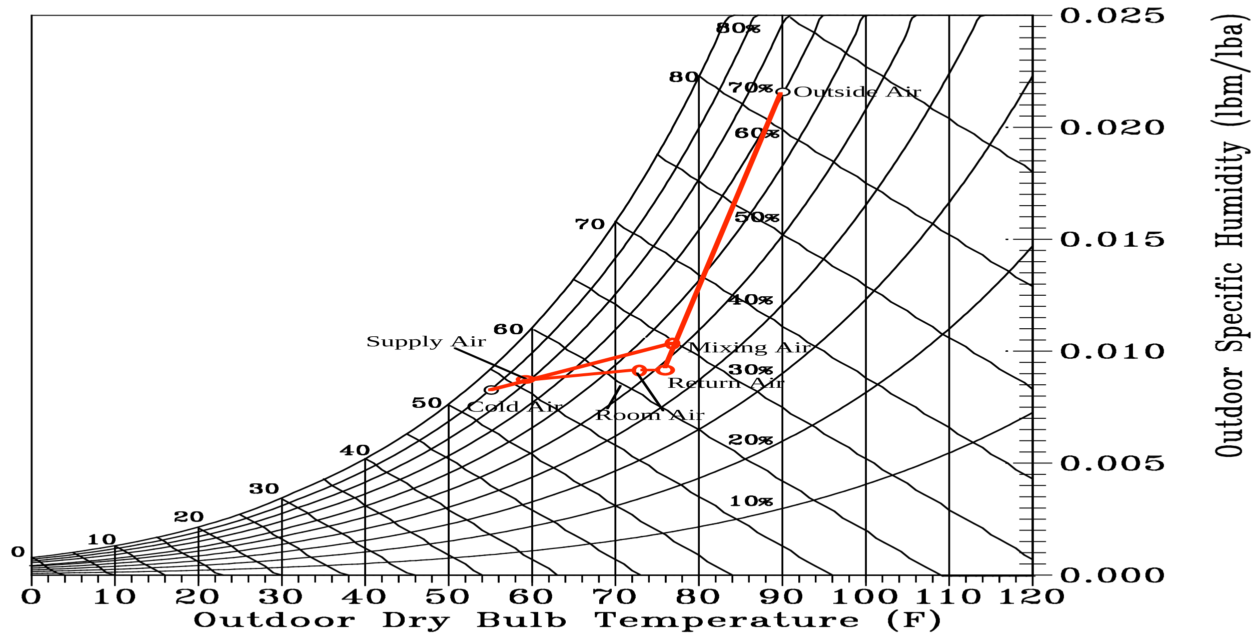

Figure 3: Air Conditioning Process After Continuous Commissioning Under Typical Austin Weather

When the total air flow is too high, significant amount of mixing air by-pass the hot deck in order to maintain suitable room temperature. Since no moisture can be removed by the heating coil, the room relative humidity cannot be controlled at the suitable level. When the total air flow is right, over 90% of the total air should flow through cold deck. If the cold deck temperature is controlled at lower than 57°F, the room relative humidity should be maintained at 55% or lower. Of curse, the amount of outside air intake greatly influence the moisture contents of the mixing air. It appears that excessive air flow can cause high room relative humidity problem. Correcting total air flow will improve the room relative humidity conditions.

The optimized operation schedules were developed by using an in-house air side simulation program. Detailed calibration and optimization procedures were provided. The energy impacts of the O& M measures were determined by the same program.

The predicted heating and cooling energy consumption versus the bin temperature is presented in Figure 4 for before and after continuous commissioning. It appeals that the improved O&M measures will reduce the peak demand by 40%. Therefore, the AHUs should be able to maintain the cold deck temperature at 55°F after the continuous commissioning. The potential annual energy savings were summarized in Table 4. The boiler efficiency was taken as 70%. The chiller kW/ton was taken 1.2, which include the cooling tower and associated pump power as well. The potential fan power savings were not included.

Table 4: Summary of Predicted Energy Consumption Before and After Continuous Commissioning

	Before CC	After CC	Savings	Savings %
Chiller Electricity (kWh/yr)	1,638,000	1,157,000	481,000	29
Hot Water (MCF/yr)	4897	681	4,216	86
Total				

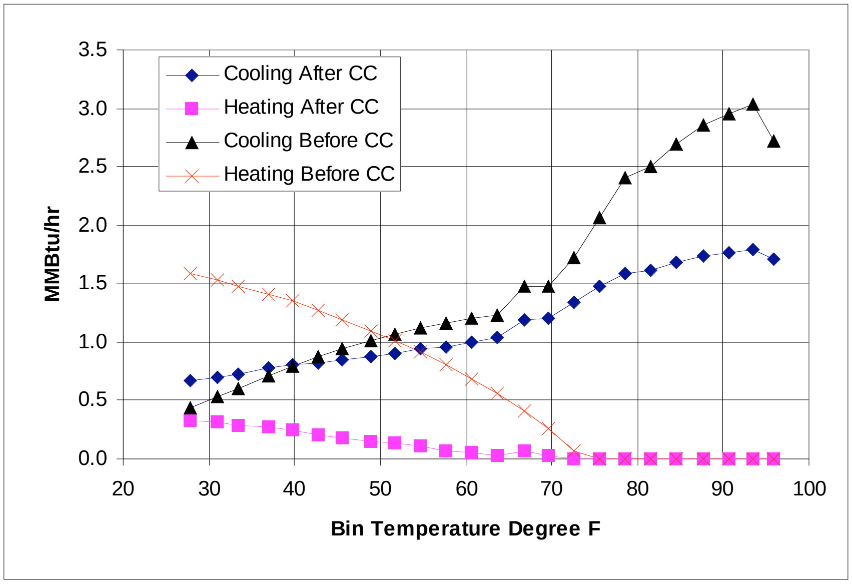

Figure 4: Comparison of Predicted Chilled Water and Hot Water Consumption Before and After the Continuous Commissioning

After the outside air was reduced by 86%, the building pressure should be maintained at normal level. After the total air flow is reduced by 38%, the duct vibration will be reduced significantly. Consequently, less dirt should be blown out through the AHUs.

Implementation of Improved O&M Measures

On July 20 and 21, 1995, the total supply air flow was reduced by 32% and the outside air flow was reduced by 62%. The 4 foot by 4 foot hole in the exterior wall was also blocked. In the beginning of September, the air flow rate was adjusted back to the condition before the continuous commissioning. On October 12, the air flow was readjusted. On November 1 to 3, the air flow was finally adjusted. The results are summarized in Table 6.

The total air supply was reduced from 140,700 CFM to 92,130 CFM (0.9 CFM/ft^2). The outside air flow was reduced from 79,950 CFM to 10,840 CFM (0.11 CFM/ft^2). The air flow was increased for AHU B. The relatively high air flow rate (1.3 CFM/ft^2) was used for AHU F since it served a major conference room. The corrected air flow rates are smaller than the designed air flow.

Table 6: Comparison of Air Flow Rate Before and After the Continuous Commissioning

AHU		A	F	B	C	Total	%
Total CFM	Before	73,200	12,800	26,900	27,800	140,700	
	After	26,990	9,980	33,510	21,650	92,130	
	Reduction	46,210	2,820	-6,610	6,150	48,570	35%
O.A. CFM	Before	43,500	6,500	12,800	17,900	79,950	
	After	2,740	2,510	3,940	1,650	10,840	
	Reduction	40,760	3,990	8,860	16,250	69,860	87%
Designed	CFM	36,800	10,000	44,300	21,125	112,225	
Area		30,300	6,800	40,800	21,100	98,900	

The implementation of the optimized operation schedule started on November 1, and finished on November 22.

Measured Impact of Continuous Commissioning♦ on the Building Comfort and Energy Consumption

Before the Continuous Commissioning♦, the room temperature, relative humidity, and CO_2 level were measured in 18 pre-selected room to represent whole building condition on June 8, 1995. The pressure cross the security door (building pressure) was also measured. After the continuous commissioning, the same measurements were repeated on June 14, 1996. The results are summarized in Table 7.

Table 7. Comparison of Room Comfort Parameters Before and After Continuous Commissioning

Item	**Before CC**	**After CC**
CO2	**400 ~ 500 ppm**	**650 ~ 800 ppm**
Room Temperature	**67 ~ 74.5** °F	**72 ~ 75** °F
Room Relative Humidity	**58% ~ 69%**	**30% to 55%**
Building Positive Pressure	**0.1 inH2O**	**0.02 inH2O**

Note: The ambient air temperature was 88°F on June 8, 1995 when the pre-CC test was performed. The ambient temperature was 99°F on June 14, 1996 when the post CC test was performed.

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

The measured energy savings agreed with the predicted potential energy savings. To compare with the predicted potential energy savings, the monthly average hourly energy consumption was calculated by using utility bill data. The monthly average hourly electricity consumption was determined as the ratio of the billed total electricity consumption and the operation 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 gas consumption combined with a boiler efficiency of 70%. The monthly average temperature was determined as the average value of the measured hourly temperature at Austin.

The pre-CC♦ period was from August 1994 to July 1995. The post O&M period was taken from October 1995 to May 1996.

Figure 4 compares the measured monthly average hourly electricity consumption before and after the implementation of the improved O&M measures. The simple linear regression model was also created based on the measured data. When the ambient temperature was low, the post electricity consumption was higher than that of pre-CC♦ because the reduced outside air flow. When the ambient temperature was high, the post CC♦ electricity consumption was significantly less than that of pre-CC♦. These observations agree with the model predicted (Figure 3). However, the change point was different (43°F in Figure 3, and 64°F in Figure 4). Since the monthly average value was used in Figure 4, this bias appears when the cooling energy consumption does not linearly dependent on the ambient temperature, which is the case here. The measured electricity savings was 90 kW when the ambient temperature was 85°F. If this savings was due to reduced chilled water consumption, it was converted to 0.9 MMBtu/hr with kW/ton value of 1.2. This value consists with the predicted value in Figure 3. The electricity consumption increased by 50 kWh/h (0.17 MMBtu/hr) when the monthly average hourly temperature was 52°F.

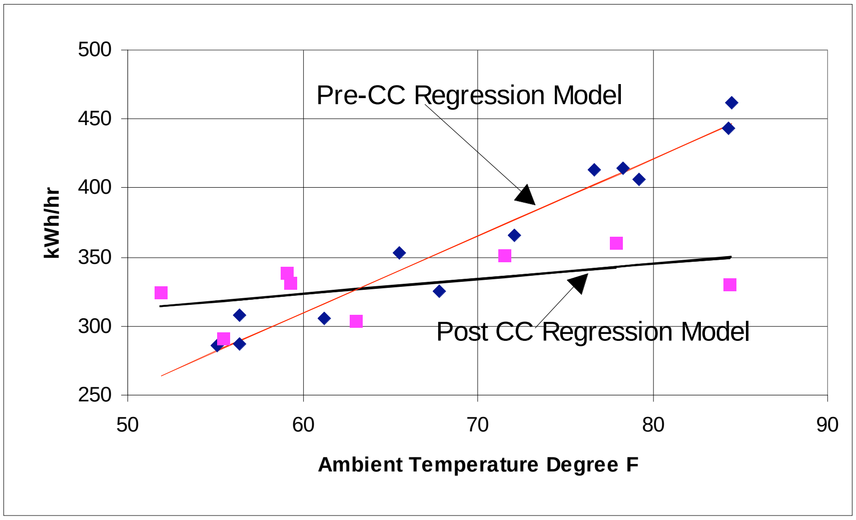

Figure 4: Measured Monthly Average Hourly Whole Building Electricity Consumption Versus the Monthly Average Ambient Temperature

Figure 5 presents the measured monthly average hourly heating energy consumption versus the monthly average ambient temperature. The regression models were also presented in the same chart. The measured heating energy savings varied from 0.1 MMBtu/hr to 0.7 MMBtu/hr when the monthly average hourly temperature varied from 75°F to 52°F. The measured gas savings varied from 0.12 MMBtu/hr to 0.88 MMBtu/hr when the monthly average hourly temperature varied from 75°F to 52°F. The measured savings were about 15% smaller than the predicted savings.

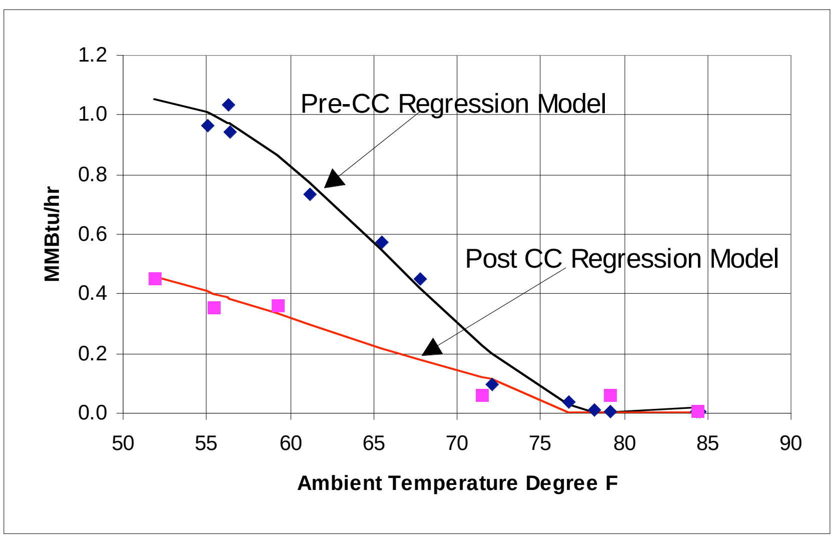

Figure 5: Measured Heating Energy Consumption Versus the Monthly Average Hourly Ambient Temperature

Figure 6 presents the measured peak electricity consumption versus the monthly average hourly ambient temperature. The simple linear regression models were also presented in the chart. The measured peak demand reduction varied from 30 kW to 70 kW when the monthly average hourly ambient temperature varied from 80°F to 52°F. The peak demand reeducation indicates a smoothly cooling system operation.

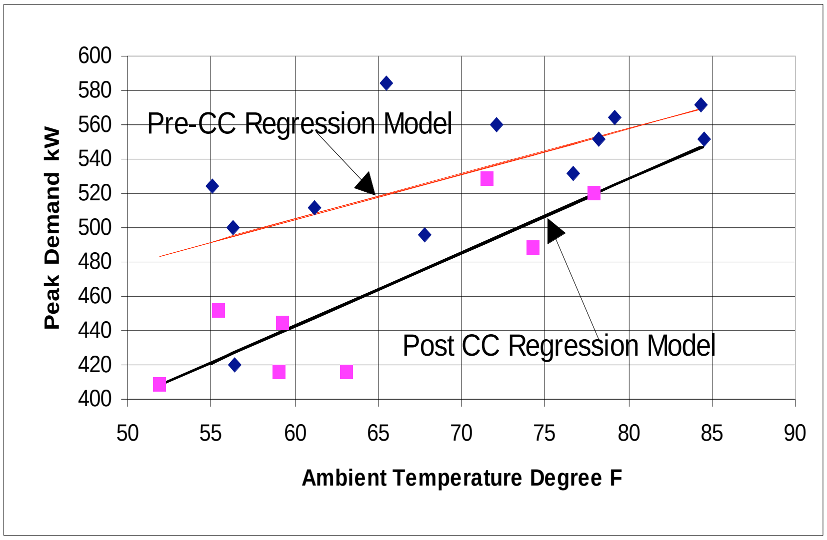

Figure 6: Measured Peak Demand Versus the Monthly Average Hourly Ambient Temperature

Based on the measured energy consumption, the annual energy savings can be determined by correcting predicted gas savings by a factor of 0.85. Then, the measured annual energy savings should be 4,940 MMBtu/yr, which includes 1,640 MMBtu/yr (481,000 kWh) for electricity, and 3,300 MMBtu/yr for gas. After the continuous commissioning, the annual energy index was reduced from 150,800 Btu/ft^2/yr to 101,000 Btu/ft^2/yr.

Conclusions

A comprehensive Continuous Commissioning♦ was performed on a 99,000 ft^2 old building. During the commissioning, both total and outside air flows were adjusted based on the existing building and occupancy conditions. The optimized cold and hot deck reset schedules were also implemented. After the building was recommissioned, the room relative humidity levels were reduced from 68% to lower than 55%. The high building pressure was also reduced to a range of 0.02 inH2O to 0.05 inH2O. The building annual energy index was reduced from 150,800 Btu/ft^2/yr to 101,000 Btu/ft^2/yr.

This case study showed that the excessive air by-pass cold deck was one of the major reasons for the high indoor relative humidity problem in the hot and humid climate. Correcting the total air flow and the outside air flow can significantly improve the indoor comfort conditions.