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Building HVAC Control System Interaction Issues: Two Case Studies

Cory Toole, Chen Xu, Qiang Chen and Song Deng
Energy Systems Laboratory
Texas A&M University System

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ABSTRACT

Direct Digital Control (DDC) allows HVAC equipment to be controlled at an upper level (supervisory control) through commands from a central system, or at a lower-level (local-loop control) by local controllers. The various levels of equipment control can be allowed to interface through the building Energy Management and Control Systems. While implementing the Continuous Commissioning® (CC®)[1] process in two institutional buildings, a number of operational and control issues related to the interfacing of equipment control have been identified. These issues include improper zone damper control of multi-zone air handling units (AHU) and comfort complaints related to room temperature control in a dual duct application, among others. These issues not only resulted in comfort problems, but also compromised building energy efficiency. All of the issues to some extent were related to the building HVAC equipment selection and design, and would have been costly to correct by redesigning and retrofitting the existing systems. This paper presents the CC measures identified in two case studies to improve building comfort and energy efficiency with minimal hardware investment.

INTRODUCTION

In the course of implementing the CC process in two institutional buildings, some unique problems surfaced related to the interfacing of supervisory control (i.e. air handling units) with local-loop control (i.e. local terminal box controllers). These problems led to comfort issues in the buildings as well as unnecessary energy consumption. A thorough investigation of the building systems and their interaction allowed measures to be implemented that helped resolve the problems with very little cost. The problems, investigation, and solutions found in both of the buildings are presented in the two case studies that follow.

CASE STUDY – I

Facility Description

The first building evaluated for this case study was a four-story building (including basement) constructed in 1921. The building consisted primarily of offices with a few classrooms, and the total floor area was 39,887 square feet. The building underwent major renovation in 1962, and received an HVAC control system upgrade in 2004. The HVAC system consisted of eight constant-speed, multi-zone air handling units, each with one damper linkage for each zone that modulated between hot deck and cold deck flow to maintain the zone temperature at its set point. All equipment was controlled with a Direct Digital Control (DDC) system.

Pre-Investigation HVAC System Control

A diagram of the multi-zone AHU system in place in the building is shown in Figure 1. The individual zone control for each AHU zone was controlled by a local controller that was not designed for the type of system in place in the building. It is unclear whether a controller fitted to the system in place did not exist at the time or was not readily available, or if some other reason existed for the controller chosen. The controller utilized was designed for a single duct variable air volume pressure dependent cooling or heating terminal box. It was designed to determine if the supply air flow entering the box was cold or hot, and to switch to cooling or heating mode accordingly. Then, depending on the mode, either a cooling or heating loop would be run to open or close the damper according to the proximity of the zone temperature to its set point.

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[1] Continuous Commissioning and CC are registered trademarks of the Texas Engineering Experiment Station (TEES), the Texas A&M University System, College Station, Texas. To improve readability, the symbol "®" will sometimes be omitted.

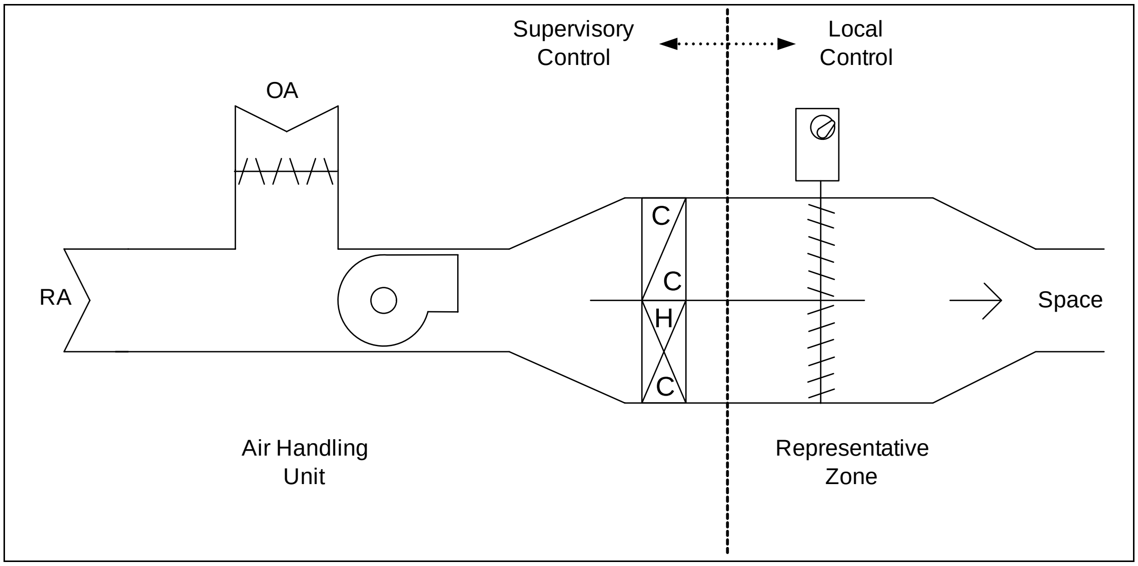

Figure 1. Multi-zone AHU system in place in building.

Obviously the design of this local controller was intended for equipment that was quite different from the multi-zone AHU system actually in place. To account for the differences, some overrides were set up within the controller. Since no supply air temperature sensor was actually available for measurement, a false supply air temperature was programmed into the controller that would maintain it constantly in cooling mode, and the damper was allowed to control accordingly. Because one actuator controlled the linkage between hot and cold decks for each zone, the zone always received full heating, full cooling, or a mixture of the two streams. This meant that a zone temperature dead band could not effectively be set at the local-loop control level, since the zone would be heated or cooled constantly to the zone cooling set point. Heating set points at the local-loop level were irrelevant, since all zones remained constantly in cooling mode.

The AHU cold deck and hot deck temperature set points were programmed to vary according to the maximum damper open or closed position for the dampers on each AHU. If the maximum zone damper was more than 80% open, the cold deck temperature set point would be decreased, whereas if the maximum damper was less than 70% open, the cold deck temperature set point would be increased. If the minimum zone damper position was less than 20% open, the hot deck temperature set point would be increased, whereas if the minimum damper was more than 30% open, the hot deck temperature set point would be decreased.

Problems Found

Constant volume multi-zone AHUs such as were in place in this building are inherently inefficient due to the large amount of simultaneous heating and cooling that occurs, and are not in compliance with the guidelines for new construction found in ASHRAE Standard 90.1-2004. These units were common, however, at the time that air conditioning was installed in this building. The inherent inefficiency of these units was compounded by the control strategies previously described that were being utilized in the building. Because of the control of AHU deck temperature resets already mentioned, it was not uncommon for the cold deck temperature to be at its minimum at the same time that the hot deck temperature was at its maximum for a given AHU, increasing the amount of simultaneous heating and cooling in the zones.

In addition to the inefficiency of the control, comfort complaints were also frequent throughout the building. The most significant of these came from occupants who were working in the building during night or weekend hours. Several complaints were received indicating that space temperatures during these times were as high as 90°F, even in the winter time. Data loggers left in the building over a period of time during the investigation confirmed these complaints, as shown in Figure 2. The temperatures trended represent three rooms served by a single zone, as well as outside air temperature.

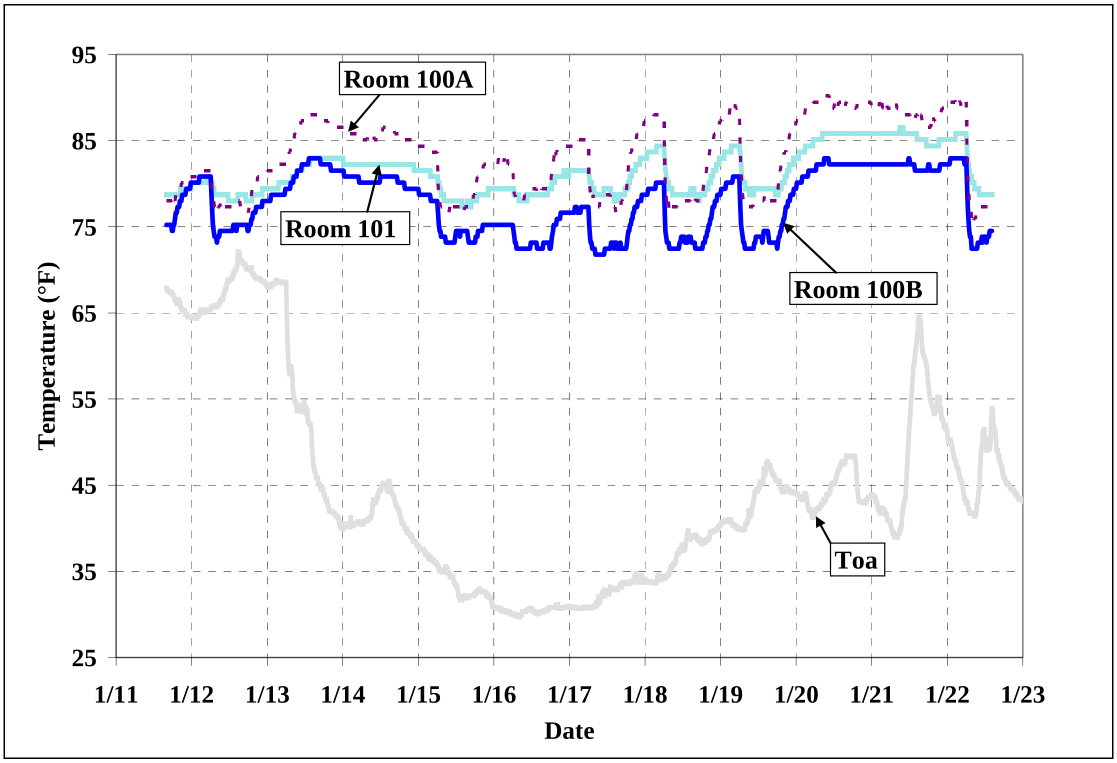

Figure 2. Trended temperatures for rooms 100A, 100B, and 101.

It was discovered during the course of the investigation that night and weekend setback and setup schedules were in place in the building. Specifically, the zone temperature cooling set points became 85°F at night. This was originally implemented at a time when AHUs were being shut down during unoccupied hours, in an attempt to conserve energy by not requiring the units to come on until zone temperatures exceeded this set point. However, at some time prior to the investigation, the shut down schedules on these units had been removed so that they would run continuously. Subsequently, due to the nature of the control of these units and of the local zones, when the cooling set point became 85°F, the zone dampers would close, causing full heat to go to each zone. Additionally, the hot deck temperature set point would rise to its maximum due to the low damper open position. This caused a rise in zone temperature up to 85°F, to which each AHU would control throughout the unoccupied period. Obviously this resulted in enormous energy waste, as well as very uncomfortable working conditions for anyone occupying the building during this time period.

Solutions

In order to fully resolve the problematic control issues identified, a retrofit of the HVAC system was recommended. It was suggested that individual control be given to hot deck and cold deck dampers for each AHU zone, and that a variable frequency drive be installed on each AHU. In this way, each zone could be made to operate similar to a dual duct variable air volume terminal box, and the effects of simultaneous heating and cooling could be dramatically reduced.

For the short term, however, in order to improve operation, measures were implemented that utilized the existing equipment to better control the spaces. In order to improve control of the AHUs and zone temperatures, as well as to improve comfort in the building, the supervisory control programming for the AHUs was rewritten. The local-loop zone control was overridden by the supervisory control programming in order to force the local controller into heating or cooling mode as needed with a ten minute switch delay. The heating temperature set point was set lower than the cooling temperature set point, creating a zone temperature dead band in each zone. The damper position for each zone was limited to operate from 50 to 100% open when in cooling mode, and from 0 to 50% open when in heating mode. Deck temperature set point resets were implemented based on weighted averages of zone damper positions, with the maximum and minimum dampers receiving greater weighting. Shut down schedules were once again implemented on all of the AHUs except one that conditioned a server room. In any case, when a zone temperature cooling set point was raised to 85°F, for example during an unoccupied period, if the AHU remained on, the zone damper would remain at 50% and supply essentially a neutral air flow to the space, allowing space temperature to slowly drift. Figure 2 is a flowchart demonstrating the logic behind the new control implemented.

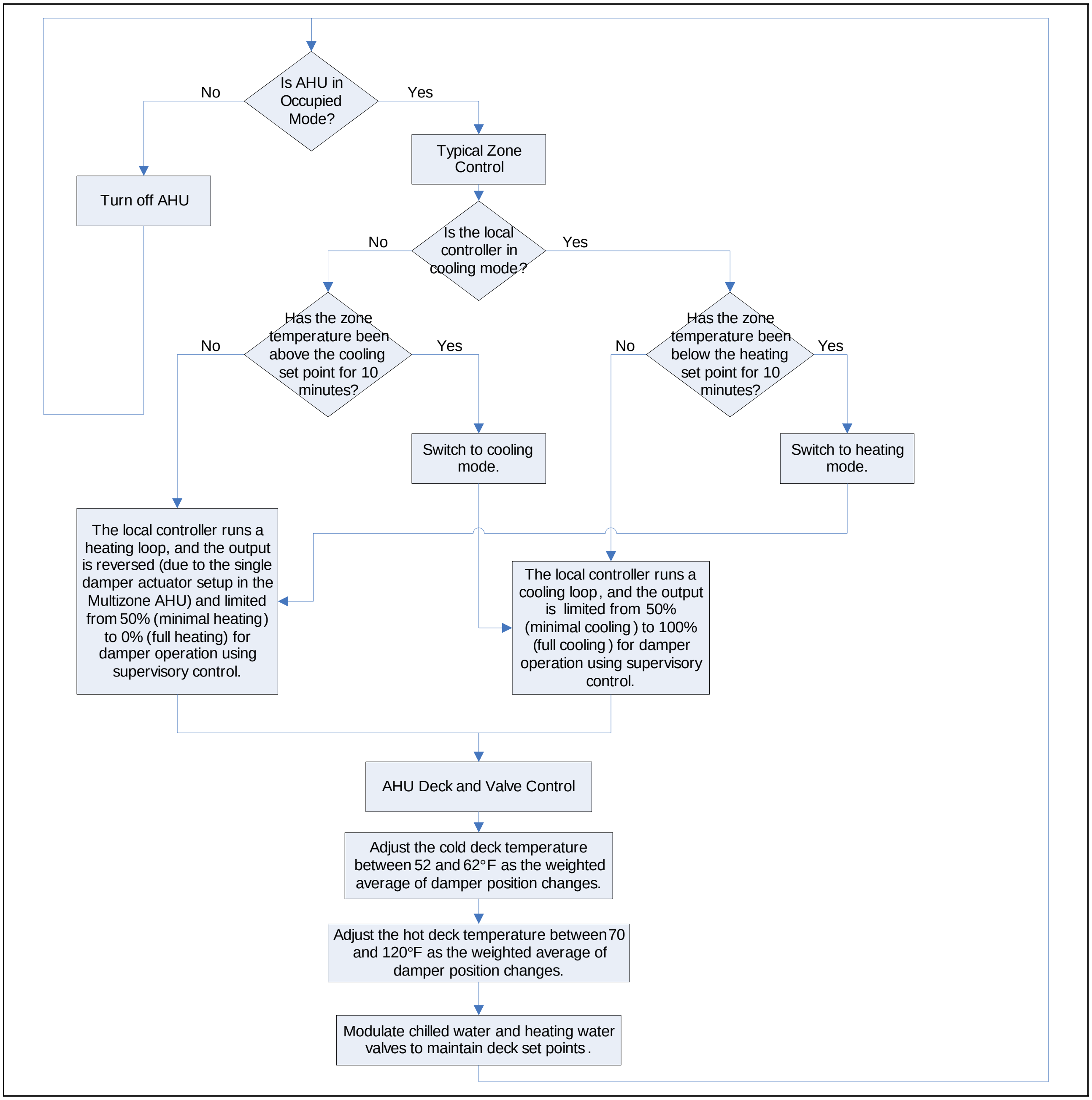

Figure 3. Flowchart demonstrating new control for multi-zone AHUs and local zone controllers.

Results

While the new control programming did not completely eliminate energy waste associated with simultaneous heating and cooling, it did serve to eliminate the severe comfort problems mentioned, and also resulted in better energy efficiency from the units. Because energy metering was not present in the building, energy savings achieved from these measures was not able to be determined from measured data. However, in order to provide a general idea of the effects on energy usage that might be expected from these measures, simulations of building energy consumption before and after CC were performed. Additionally, a simulation of the building with the proposed retrofit was also performed. These simulations were performed using a commercially available whole building simulation package. Tables 1, 2, and 3 give the major input parameters used in the simulation of each case. Figures 3, 4, and 5 show the simulated cooling energy usage, heating energy usage, and electricity usage for the pre-CC, post-CC, and post-retrofit conditions. The simulations showed that the implemented CC measures would result in savings somewhere in the range of 50% for cooling, 70% for heating, and 40% for electricity. If the proposed equipment retrofit were implemented, the simulation showed that these savings would increase to somewhere in the range of 75% for cooling, 90% for heating, and 60% for electricity. Because the simulations could not be calibrated to measured data, these figures cannot be assumed to accurately reflect building energy consumption or resultant savings. However, it is clear from the simulations that some level of savings can be expected from the measures implemented, and it would appear that these savings are significant. It also appears that the proposed retrofit would increase the expected level of savings.

Table 1. Major input parameters for the simulation of the pre-CC case.

Total Building Conditioned Area	40,000 ft^2
Number of Floors	4
System Type Simulated	Constant Volume Multi-zone AHU with HW Heat
Number of AHUs per Floor	1
Day Space Cooling Set Point	76°F
Day Space Heating Set Point	76°F
Night Space Cooling Set Point	85°F
Night Space Heating Set Point	85°F
Cold Deck Reset Control	Warmest Zone
Minimum Cold Deck Temperature	53°F
Maximum Cold Deck Temperature	65°F
Hot Deck Reset Control	Coldest Zone
Minimum Hot Deck Temperature	70°F
Maximum Hot Deck Temperature	120°F
Supply Air Flow (per AHU)	12,612 CFM
Outside Air Flow (per AHU)	1,450 CFM
Fan Schedule	On 24/7
Day Mode Times	M-F 8:00AM-5:00PM

Table 2. Major input parameters for the simulation of the post-CC case.

Total Building Conditioned Area	40,000 ft^2
Number of Floors	4
System Type Simulated	Constant Volume Multi-zone AHU with HW Heat
Number of AHUs per Floor	1
Day Space Cooling Set Point	75°F
Day Space Heating Set Point	70°F
Night Space Cooling Set Point	85°F
Night Space Heating Set Point	60°F
Cold Deck Reset Control	Warmest Zone
Minimum Cold Deck Temperature	52°F
Maximum Cold Deck Temperature	62°F
Hot Deck Reset Control	Coldest Zone
Minimum Hot Deck Temperature	70°F
Maximum Hot Deck Temperature	120°F
Supply Air Flow (per AHU)	12,612 CFM
Outside Air Flow (per AHU)	1,450 CFM
Fan Schedule	Off 10:00PM-6:00AM
Day Mode Times	M-F 8:00AM-5:00PM

Table 3. Major input parameters for the simulation of the post-retrofit case.

Total Building Conditioned Area	40,000 ft^2
Number of Floors	4
System Type Simulated	Variable Air Volume Dual Duct AHU with HW Heat
Number of AHUs per Floor	1
Day Space Cooling Set Point	75°F
Day Space Heating Set Point	70°F
Night Space Cooling Set Point	85°F
Night Space Heating Set Point	60°F
Cold Deck Reset Control	Warmest Zone
Minimum Cold Deck Temperature	52°F
Maximum Cold Deck Temperature	62°F
Hot Deck Reset Control	Coldest Zone
Minimum Hot Deck Temperature	70°F
Maximum Hot Deck Temperature	120°F
Supply Air Flow (per AHU)	12,612 CFM
VAV Minimum Flow	50%
Outside Air Flow (per AHU)	1,450 CFM
Fan Schedule	Off 10:00PM-6:00AM
Day Mode Times	M-F 8:00AM-5:00PM

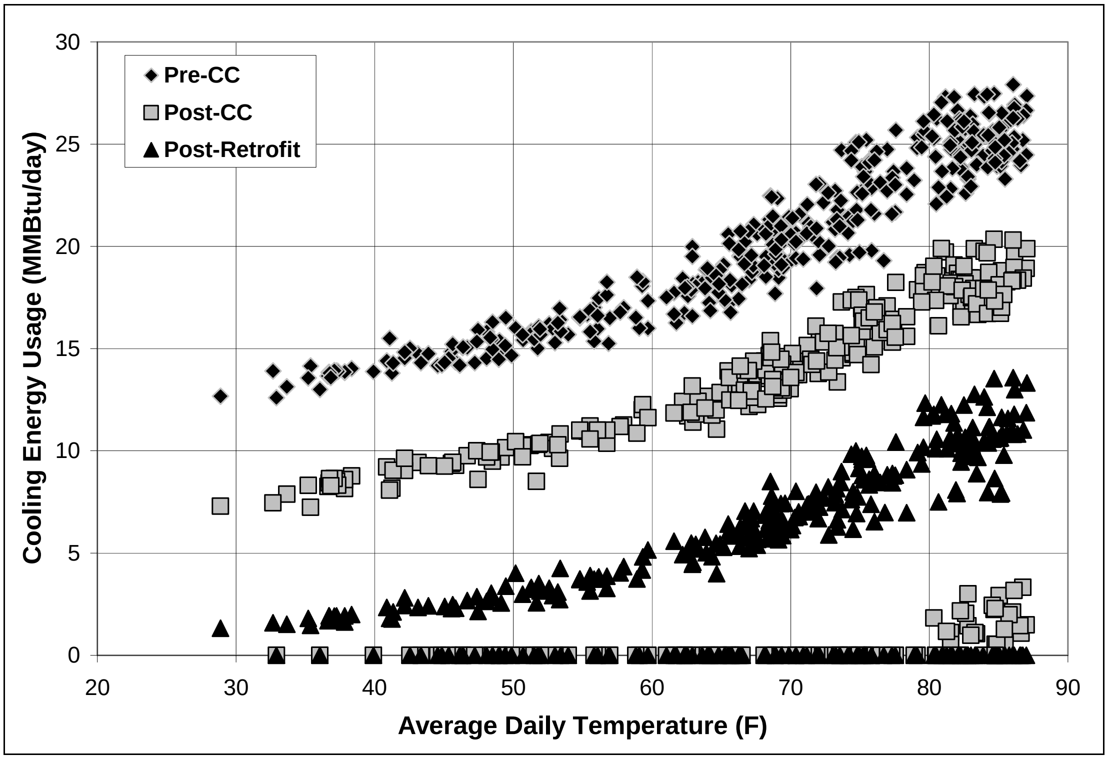

Figure 4. Simulated cooling energy usage for pre-CC, post-CC, and post-retrofit conditions.

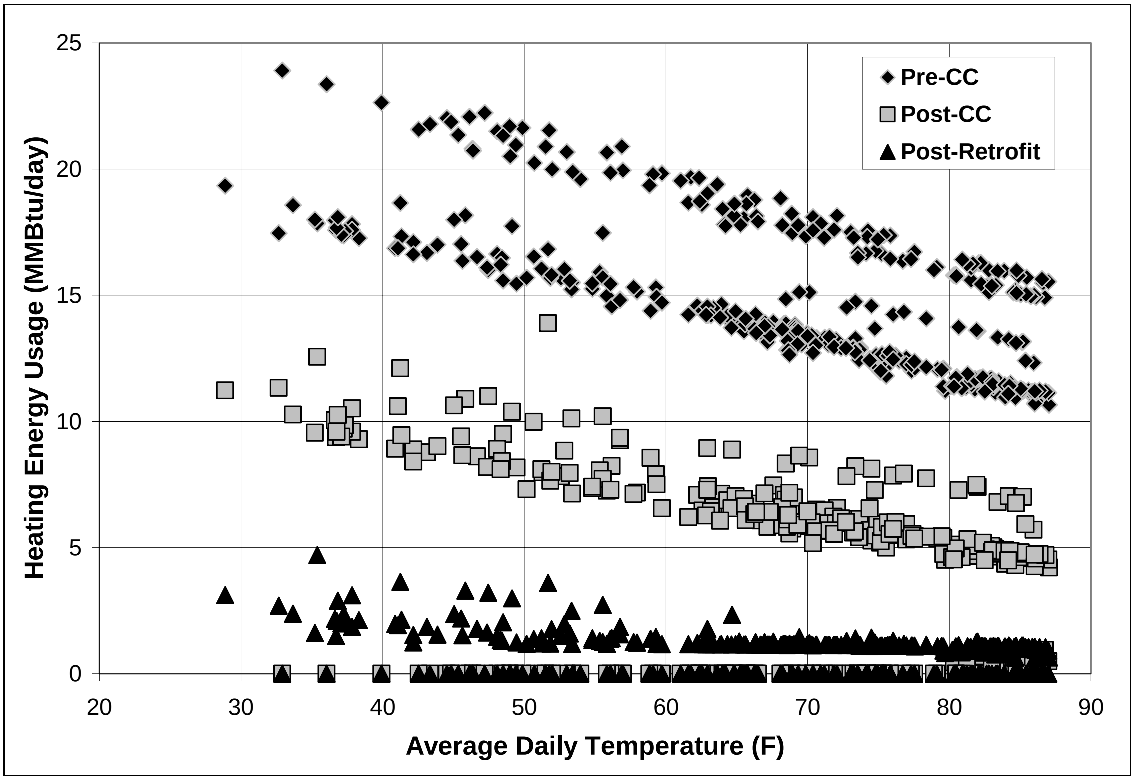

Figure 5. Simulated heating energy usage for pre-CC, post-CC, and post-retrofit conditions.

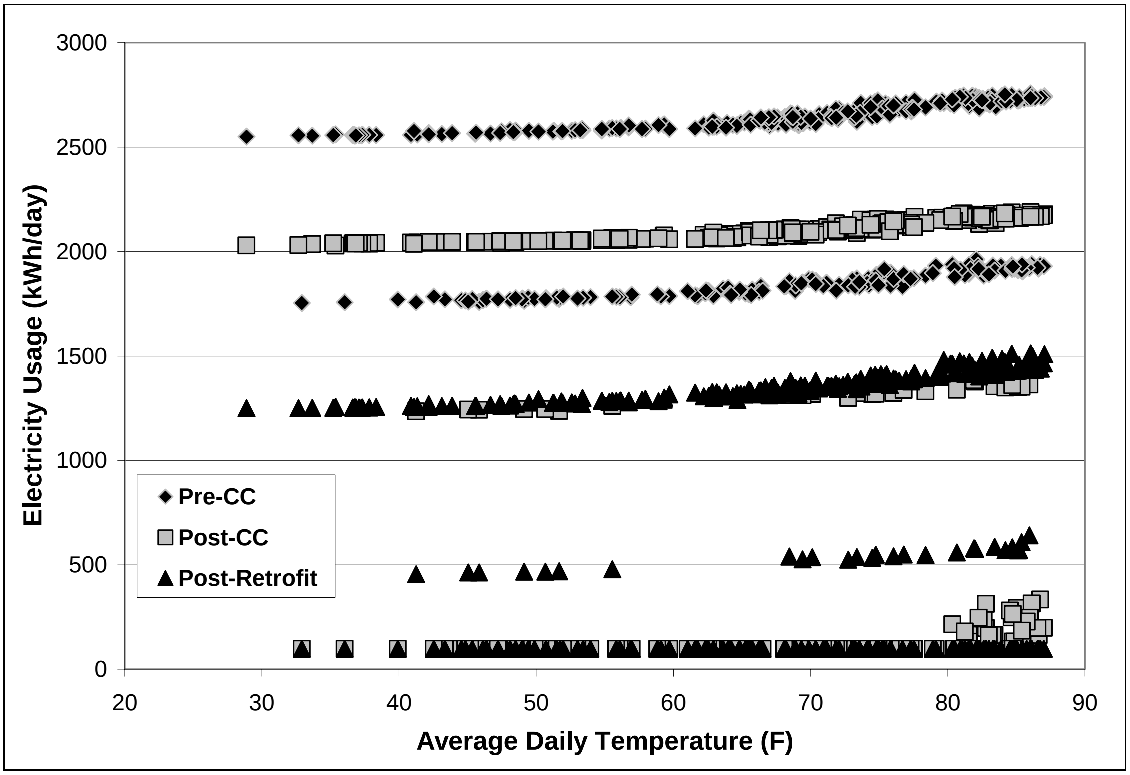

Figure 6. Simulated electricity usage for pre-CC, post-CC, and post-retrofit conditions.

CASE STUDY – 2

Facility Description

The facility evaluated for the second case study was a three-story building, approximately 10 years old, consisting of offices and classrooms, and totaling 133,326 square feet of floor area. The building HVAC system was dual duct variable air volume with pretreated outside air, and incorporated 10 air handling units, three outside air handling units, and nearly 200 terminal boxes. All equipment was controlled with a DDC system.

Pre-Investigation HVAC System Control

At the onset of commissioning, the air handling units were all set up with demand based cold deck and hot deck temperature set point reset schedules, with average terminal box heating and cooling demands being used as the reset parameters. During normal operation, the cold deck temperature set point was allowed to vary from 53°F to 58°F, and the hot deck temperature set point was allowed to vary from 72°F to 120°F. AHU static pressure set points were also allowed to vary based on average terminal box damper positions, and generally ranged from 0.3 in. W.C. to 1.5 in. W.C. during normal operation.

Unlike the previous case, this facility did not suffer from a mismatch in system and local zone controller. The local controller utilized was designed for dual duct variable air volume terminal boxes with two inlet sensors and optional reheat. The controller had a programmable minimum air flow from the cold deck, as well as a total minimum air flow from a combination of hot and cold decks. At the onset of commissioning, the average total minimum flow for all terminal boxes represented approximately 27% of the cooling maximum flow, and approximately 30% of the heating maximum flow. Several of the terminal boxes also had cooling minimum flow settings greater than 0 cfm. The heating space temperature set point for almost all of the boxes was 70°F. The cooling space temperature set point for almost all of the boxes was 76°F.

Problems Found

Numerous comfort complaints were received in the building. Investigation during the CC process uncovered a variety of reasons for these complaints, related to a number of building HVAC system issues. In particular, however, it was discovered that the way the terminal boxes were being controlled was a large factor in some of the comfort complaints received. The most significant of these was a complaint by many office occupants of uncomfortably cool temperatures in the offices during periods of mild to cool outdoor air temperatures. Further investigation of these complaints revealed that a large percentage of spaces in the building, including interior spaces, were being constantly maintained at 70°F, the heating set point, and the terminal boxes were remaining in heating mode, even with outdoor air temperatures higher than 70°F. This caused discomfort for some occupants, who were no longer dressed for winter conditions with the mild outdoor climate, and who often went for lengthy periods of very light activity while performing typical office work. It was observed more than once during the CC process that an occupant would hang a bag of ice on the thermostat in order to force the terminal box to provide more heat.

In addition to the discomfort brought on by these conditions, it was suspected that energy was unnecessarily being consumed to overcool the spaces. Further investigation confirmed this theory. It was discovered that the local terminal box control was designed such that when in cooling mode, the cold deck air flow was modulated to maintain the space temperature set point, and the hot deck air flow was modulated to ensure that the total flow from the terminal box was at least the total minimum flow set in the controller. When in heating mode, this was reversed, with the hot deck air flow being modulated to maintain the space temperature set point, and the cold deck air flow being modulated to ensure adequate total minimum flow from the terminal box. Typically when a majority of the terminal box controllers were in cooling mode, the hot deck temperature would drift down to return air temperature, essentially providing neutral air as makeup air to the space, a method that worked very well. However, as noted above, even if all terminal box controllers were in heating mode, the cold deck temperature was never allowed to drift above 58°F, meaning that any time the cold deck air flow was used as makeup air to provide proper total minimum flow, at least some cooling would occur. For offices with small interior heat loads, the minimum flow being made up by the cold deck air flow was very often sufficient to never allow the space to warm up above 70°F, also meaning that the terminal box controller would remain in heating mode, and cooling energy usage would occur unnecessarily at the AHU cooling coil.

Solutions

Ideally, each space would be allowed to naturally drift between the heating and cooling space temperature set points with little to no heating or cooling as heat loads in the space increased or decreased. This would minimize energy consumption and reduce discomfort associated with overcooling. In order to attempt to achieve this operation, two CC measures related to this issue were implemented in the building. The first measure was to set the cooling minimum flow point (different from the total minimum flow point, as noted earlier) to 0 cfm for all terminal boxes. The purpose of having a separate cooling minimum flow point is for ventilation if the AHU received fresh air only through the cold deck. However, in this case fresh air mixed with return air before being cycled through both decks, meaning that the minimum flow required for ventilation could come from either of the two decks. Therefore, having a value greater than 0 cfm for the cooling minimum flow was unnecessary, and further exacerbated the problem of simultaneous heating and cooling in some instances. The second CC measure related to this issue was to modify the upper limits of the cold deck temperature set points for all AHUs. Because the outside air brought into the building was pretreated by outside air handling units before mixing with return air at the AHUs, it was unnecessary to set the upper limit of the AHU cold deck temperature set point at a level that would dehumidify this air. Therefore, the limiting factor for the upper limits of the AHU cold decks was only the space cooling demand. The upper limit for the cold deck temperature set point of each AHU was set to 72°F during the CC process. The supervisory control programming for this reset schedule was written to frequently sample the controllers for all terminal boxes served to determine a weighted average of cooling needs in the spaces, with the maximum cooling need being weighted more heavily than other boxes in the average. This would allow the cold deck to essentially provide neutral makeup air when the controllers for the majority of terminal boxes served were in the heating mode, which would allow the space temperatures throughout the building to slowly drift up in conjunction with increasing heat loads until the terminal box controllers switched over to cooling mode.

Results

The CC measures described were implemented during the spring, with generally mild outdoor air temperatures. Shortly before implementation, it was observed that a majority of the terminal box controllers in the building remained constantly in heating mode with space temperatures of 70°F. Shortly after implementation, it was observed that nearly all of the terminal box controllers had switched over to cooling mode, and that space temperatures fell in the range of 70°F to 76°F (the cooling set point). Although only a couple of the

AHU cold deck temperatures had risen to 72°F, nearly all had risen some, mostly to a level in the range of 60°F to 70°F. Significant energy savings resulting from these measures and other CC measures identified in the building immediately began to occur following implementation of the measures. The other CC measures implemented included AHU shut downs, reducing unnecessary outside air intake, better pre-treat control of outside air, and improved AHU static pressure reset schedules, among others. Figures 6, 7, and 8 demonstrate the building chilled water, heating water, and electricity consumption before and after the CC period. The red line in each figure represents the regression model used to project the baseline data to the same weather conditions as the post-CC data, in accordance with ASHRAE Guideline 14-2002, Measurement of Energy and Demand Savings.

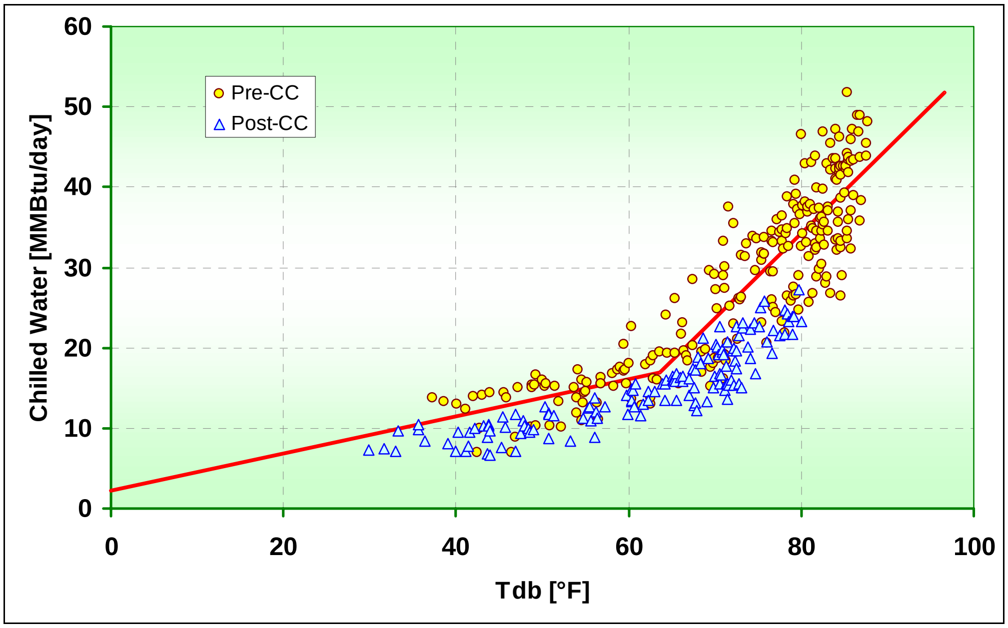

Figure 7. Measured building chilled water consumption for pre-CC and post-CC conditions.

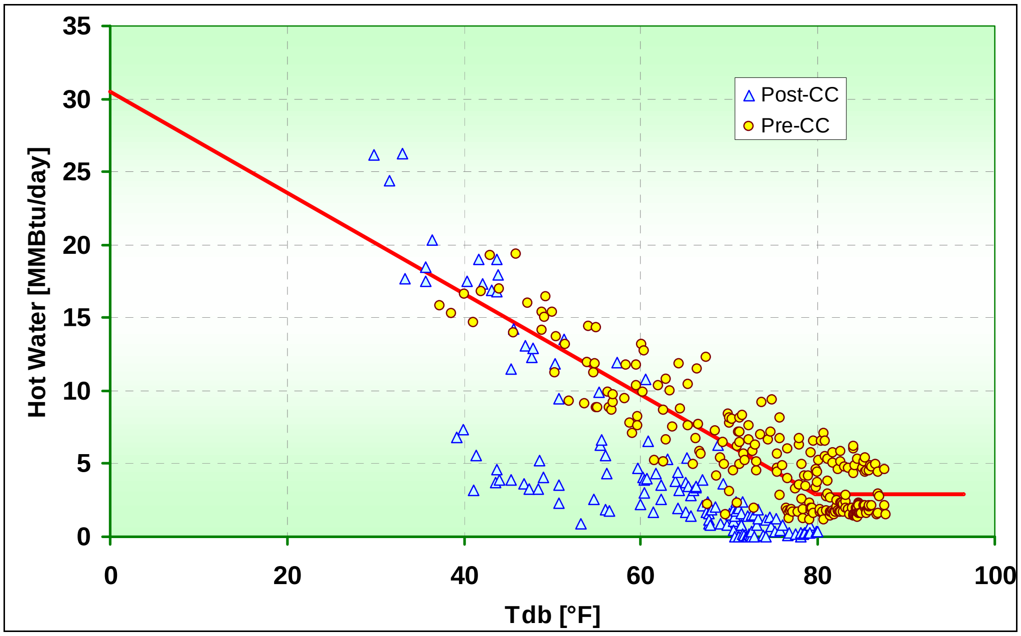

Figure 8. Measured building hot water consumption for pre-CC and post-CC conditions.

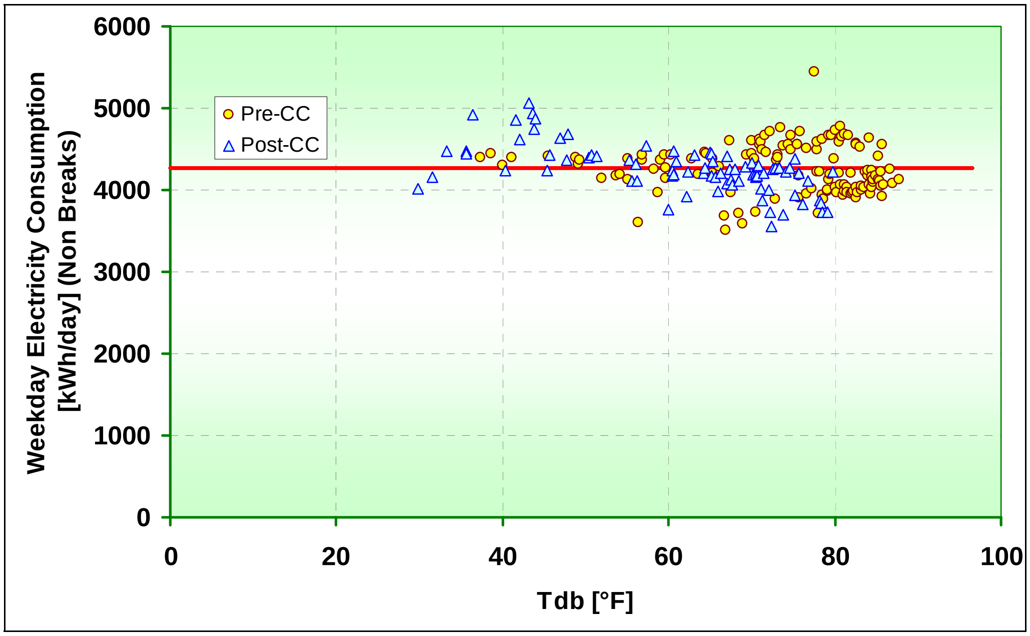

Figure 9. Measured building electricity consumption for pre-CC and post-CC conditions (normal weekday operation).

Based on whole building calibrated simulations of the building using a commercially available simulation package, it was predicted before implementation of the CC measures that the implementation of the two mentioned in this paper would save approximately $1,000 per year, and that all of the identified CC measures together would save approximately $20,000 per year. Updated predictions based on a whole building approach using the measured data obtained thus far estimate that $40,000 of total energy cost per year will actually be saved from all of the measures combined. Although the energy savings achieved from the two measures described for this case study were relatively small, the comfort related improvements from these measures were significant.

It should be noted that the solutions implemented in this case study could also have been implemented in a building without pretreated outside air, though to a lesser extent. However, the outside air dew point temperature would need to be monitored, and the upper limit of each cold deck temperature set point would need to change depending on outdoor air humidity levels. As long as outdoor air humidity was lower than desired for indoor air conditions, the same upper limits could be set as were done in this case study. During humid conditions, however, the upper limit on the cold deck could not be allowed to drift any higher than required for proper indoor humidity control.

It should also be noted that large differences in heat load among spaces served by an AHU could reduce the effectiveness of the strategies implemented in this study. For example, spaces with large, constant heat generation that always require cooling may not allow the cold deck temperature to even approach a neutral air temperature. For this case, it may be desirable to attempt to increase the maximum air flow possible to these high load areas, such that with more flow the temperature required for cooling would be reduced. However, in all cases a limit on the effectiveness of this process would occur. It would be helpful if local-loop control existed that could change which deck was used for makeup air depending on which deck temperature was closer to space temperature. Although this would add a level of complexity to the control, it could further reduce the effects of simultaneous heating and cooling, as well as overcooling of spaces.

CONCLUSIONS

Problems stemming from the interfacing of supervisory control with local-loop control were identified during the commissioning process for two institutional buildings. In one building, the problem resulted from the improper utilization of a local-loop controller designed for a different system that was installed on the multi-zone system during a controls upgrade. Using modifications in supervisory control programming to override much of the local-loop control, the comfort problems and much of the excessive energy usage resulting from this problem were largely resolved, while a system retrofit was also recommended for further improvement. Substantial energy savings are believed to have occurred from the measures implemented in this case, as well as comfort improvements. In the second building, the local-loop controller was properly matched to the equipment being controlled, but an issue with how minimum airflow was being achieved created comfort problems and wasted energy in the building. Two measures were implemented that helped improve comfort in the rooms and also provided some energy savings. For both buildings, the problems identified were able to be largely resolved with very little financial investment.

It can be concluded from these case studies that an in-depth understanding of the interaction of various components within an HVAC system, as well as good fundamental understanding of the impacts of control strategies on overall energy consumption are essential in allowing a building to operate effectively and efficiently. Specifically, local-loop control must be viewed in the broader perspective of whole building operation and energy consumption to ensure that local-loop control does not become counterproductive to the goals of building performance. Fortunately, even in buildings where this lack of understanding has led to problems such as the ones described in this paper, in many cases significant improvements can be made through creative control strategies, without necessarily requiring extensive retrofit or capital investment. This ability to positively affect building comfort and energy efficiency with minimal cost is the goal and the benefit of the Continuous Commissioning process, and it has been demonstrated here that local-loop control optimization can be a prime target for achieving these benefits in a facility.

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