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# Simulated Energy Savings Comparison Between Two Continuous Commissioning® Methods Applied to a Retrofitted Office Building

Texas A&M Campus Building CC® Team
Energy Systems Laboratory
Texas A&M University System

## KEYWORDS

Continuous Commissioning®, Outside Air Based Reset, Demand Based Reset, Electric Reheat, Simulation, EnergyPlus

## ABSTRACT

Continuous Commissioning® (CC®)[1] was performed on a 24,446 square foot institutional building used primarily to house offices and conference rooms. The building was constructed in 1950, and then had a complete Heating, Ventilating, and Air Conditioning (HVAC) retrofit in 2001 to include new equipment and Direct Digital Control (DDC). Following the retrofit, CC was carried out in this building, mainly implementing outside air temperature (OAT) based temperature and static pressure reset strategies. In 2008, a second round of CC was performed (and is still in progress), which has focused on incorporating demand based reset strategies in addition to the reset strategies already in place. This paper examines the advantages of demand based reset strategies used in conjunction with outside air based reset strategies, with this building serving as a case study. In a more general sense, the building also serves as a case study of the benefits of performing multiple rounds of CC in a facility over time. Predicted energy savings from the second round of CC over the first round were reported. Since the second round of CC was still in progress as of completion of this paper, savings were predicted using simulation models developed in Energy Plus, a commercial simulation software package. Actual savings achieved from CC will be determined and presented in a 2nd paper when sufficient post-CC data have been obtained. This paper also mentions two specific challenges that were encountered and addressed during the second round of CC, including: 1) control of terminal boxes with inline electric reheat, and 2) control of a constant speed multi-zone air handling unit with zone temperature control. Recommendations were made to optimize control relative to these and other issues, in order to improve comfort and energy efficiency in the facility.

[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.

## INTRODUCTION

A 24,446 square foot institutional building was constructed in 1950, and underwent a complete Heating, Ventilating, and Air Conditioning (HVAC) system retrofit in 2001. This renovation included all new equipment and modern Direct Digital Control (DDC). Just after this retrofit occurred, Continuous Commissioning® (CC®) was performed in the building. Several years later, in 2008, another round of CC was begun. This paper will describe the two CC processes performed, including differences between the first and second rounds of commissioning. A major focus will be to observe the effects of two different methods of implementing reset strategies in the HVAC equipment, but the building will also serve as a case study for performing multiple rounds of CC in a facility over time. Since the second round of CC is still in progress, energy consumption comparisons presented will be based on simulated models.

## FACILITY DESCRIPTION

The building under evaluation was a three-story facility encompassing 24,446 square feet, and consisting primarily of offices and conference rooms. After the 2000 HVAC system retrofit occurred, the equipment serving the building included a constant speed, multi-zone air handling unit (AHU), three single-duct variable air volume (VAV) AHUs, a constant speed outside air pre-treat unit, and a chilled water pump and hot water pump with variable speed drives. The single-duct VAV AHUs supplied air to a total of 19 terminal boxes with inline electric reheat.

## ORIGINAL (POST-RETROFIT) CONTROL

When the HVAC system retrofit was completed in the summer of 2001, the control system was set up with a prescribed sequence of operation for each piece of equipment.

The multi-zone AHU was started and stopped by an associated DDC panel. Once energized, the outside air damper would open fully and the cold and hot decks would begin to control. Cold and hot deck temperature sensors would control their respective control valves to maintain deck temperatures at their set points. DDC damper controllers modulated each zone mixing damper to maintain zone temperatures at their set points. Deck temperature set points and zone temperature set points were constant, fixed values.

The three VAV AHUs were also started and stopped by associated DDC panels. Once energized, the outside air damper of each unit would open fully and the chilled water coil would begin to control. Temperature sensors controlled their respective control valves to maintain discharge temperature at its set point. Static pressure sensors controlled fan speed to maintain duct static pressure at its set point. Discharge air temperature set points and static pressure set points were constant, fixed values for all three VAV units.

For the chilled water and hot water systems, with the system pump off, the system return valve was controlled to maintain building differential pressure at its set point. If the control valve was fully open and the building differential pressure was less than its set point after a time delay, the system pump was energized. Once the system pump was energized, the speed of the pump was controlled to maintain building differential pressure at its set point. The system return valve was controlled to limit flow to the building to the maximum design flow for the building. The chilled and hot water systems were energized any time an AHU was energized.

## FIRST CC INVESTIGATION

The first round of CC took place between September and October 2001, just after completion of the HVAC system retrofit. Major activities that occurred are summarized in the subsections below.

*AHU Deck Temperatures*

As noted, pre-CC operation of the three VAV AHUs maintained the discharge air temperature set point at a fixed, constant value (55°F). During the first round of commissioning, a reset schedule was developed for the discharge air temperature set point of each of these AHUs based on outside air temperature. When the outside air temperature was less than 45°F, the discharge air temperature set point was set at 60°F. When the outside air temperature was 65°F the discharge air temperature set point was 57°F. For outside air temperatures above 75°F the discharge air temperature set point was 55°F. This reset schedule is shown below in Figure 1.

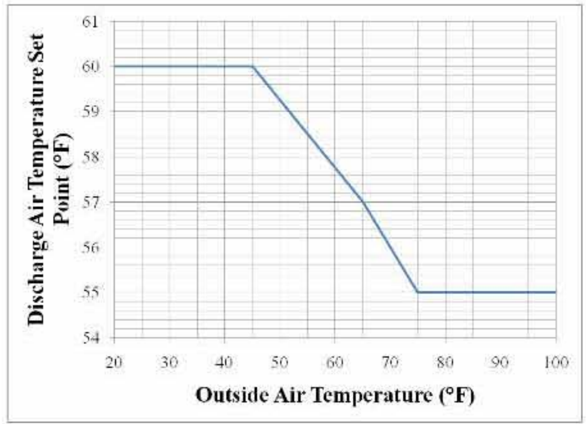


**Figure 1. Outside air based discharge air temperature reset schedule for VAV AHUs.**

The multi-zone AHU had fixed, constant set points for cold and hot deck temperatures. These were replaced during this commissioning with a reset schedule for each of the decks that determined the maximum and minimum zone temperatures and reset the deck temperature set points accordingly. A maximum zone temperature of 68°F would result in a cold deck discharge temperature set point of 58°F, and a maximum zone temperature of 74°F would result in a cold deck discharge temperature set point of 55°F. A minimum zone temperature of 65°F would result in a hot deck discharge temperature set point of 110°F, and a minimum zone temperature of 74°F would result in a hot deck discharge temperature set point of 85°F. Figures 2 and 3 illustrate these reset strategies.

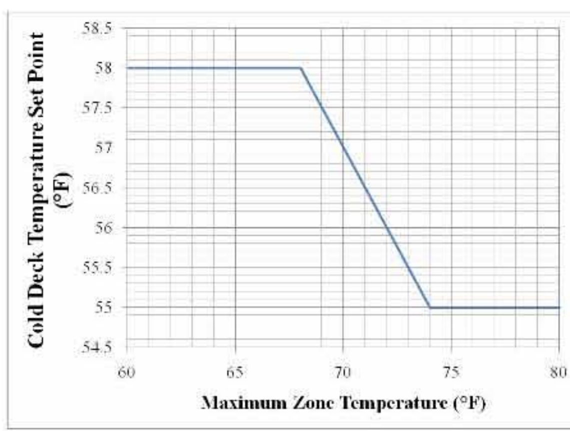


**Figure 2. Zone temperature based cold deck temperature reset schedule for multi-zone AHU.**

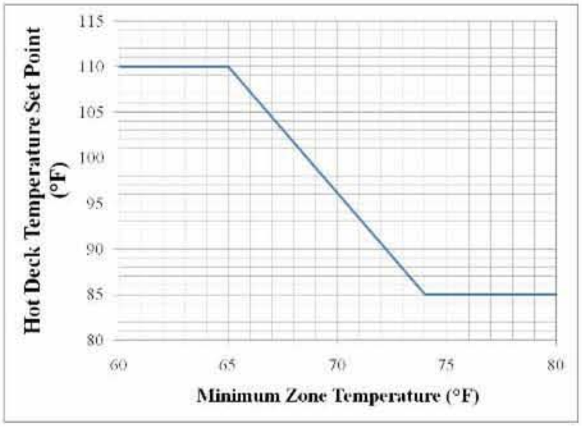


**Figure 3. Zone temperature based hot deck temperature reset schedule for multi-zone AHU.**

*AHU Static Pressure Set Points*

As noted, pre-CC operation of the three VAV AHUs maintained the static pressure set point for each unit at a fixed, constant value. During the first round of commissioning, a reset schedule was developed for the static pressure set point of each of these AHUs based on outside air temperature during occupied periods. Another outside air temperature based reset schedule was implemented for unoccupied periods. For occupied periods, when the outside air temperature was less than 45°F, the static pressure set point was set at 1.0 inch of water. When the outside air temperature was 65°F the static pressure set point was 1.2 inches of water. For outside air temperatures above 75°F the static pressure set point was 1.5 inches of water. This reset schedule is shown below in Figure 4.

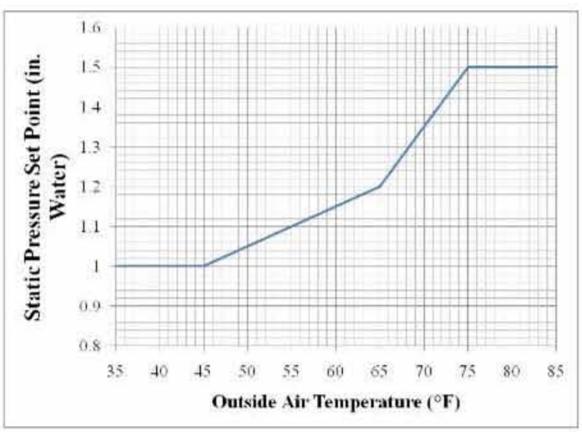


**Figure 4. Outside air based static pressure reset schedule for VAV AHUs during occupied periods.**

For unoccupied periods, when the outside air temperature was less than 45°F, the static pressure set point was set at 0.8 inches of water. When the outside air temperature was 65°F the static pressure set point was 1.0 inch of water. For outside air temperatures above 75°F the static pressure set point was 1.2 inches of water. This reset schedule is shown below in Figure 5.

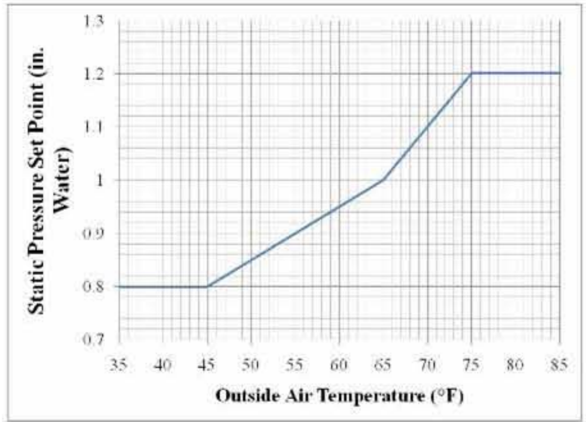


**Figure 5. Outside air based static pressure reset schedule for VAV AHUs during unoccupied periods.**

*Water Loop Differential Pressures*

As discussed, the loop differential pressure set points and flow set points were constant before CC took place. During the first round of commissioning, reset schedules based on outside air temperature were developed for both the hot water and chilled water loops. For the hot water loop, the differential pressure set point was 10 psi when the outside air temperature was less than 50°F. The set point was 9 psi when the outside air temperature was 65 °F. The set point was 7 psi when the outside air temperature is higher than 75°F. This reset schedule is shown in Figure 6 below.

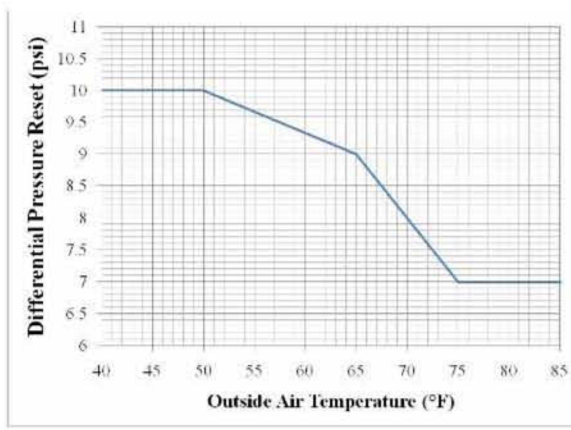


**Figure 6. Outside air based differential pressure reset schedule for hot water loop.**

The hot water flow set point was 50 GPM when the outside air temperature was lower than 50°F, 45 GPM when the outside air temperature was 65 °F, and 40 GPM when the outside air temperature was higher than 75°F. This reset schedule is shown in Figure 7.

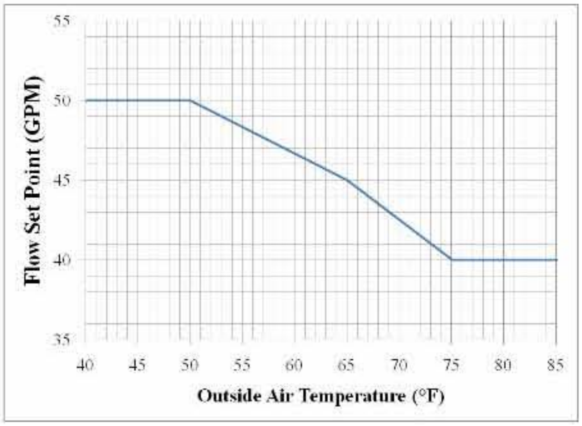


**Figure 7. Outside air based flow set point reset schedule for hot water loop.**

For the chilled water loop, the differential pressure set point was reset to 7 psi when the outside air temperature was less than 50°F, 8.5 psi when the outside air temperature was 65 °F, and 10 psi when the outside air temperature is higher than 75°F. This reset schedule is shown in Figure 8.

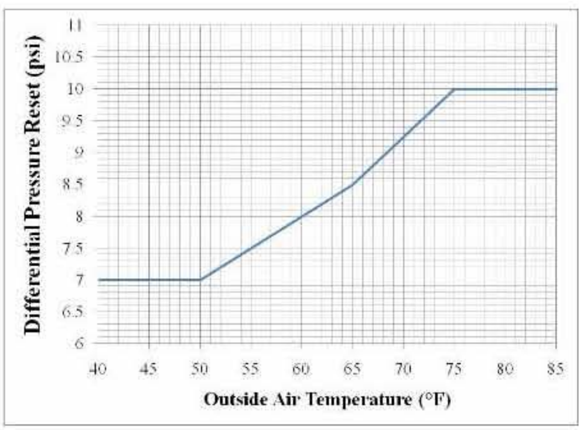


**Figure 8. Outside air based differential pressure reset schedule for chilled water loop.**

The chilled water flow set point was 150 GPM when the outside air temperature was lower than 50°F, 170 GPM when the outside air temperature was 65 °F, and 200 GPM when the outside air temperature was higher than 75°F. This reset schedule is shown in Figure 9.

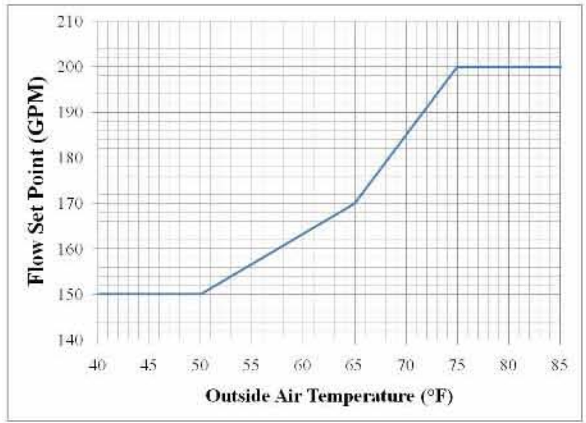


**Figure 9. Outside air based flow set point reset schedule for chilled water loop.**

## SECOND CC INVESTIGATION

Several years after the first round of CC occurred the building began to be considered again as a candidate for CC. Despite the energy efficiency measures implemented during the first round of CC, metered energy consumption data showed the cost per square foot to operate the building was $11.90 per year, an enormous cost for a building of its type. Comfort complaints also began to be received from selected locations throughout the building. It would be discovered later that a metering issue had grossly overestimated building energy consumption, and that the true cost of operation was closer to $2.10 per square foot per year. Nonetheless, the building was selected to undergo another round of CC, and during this second process significant additional energy saving initiatives were identified. The second round of CC began in January 2008, and is still ongoing as of completion of this paper. Major findings and recommendations from this second round of CC are summarized in the subsections below.

*Equipment Schedules*

The second round of CC found all equipment operating continuously. Recommendations were made to shut down all AHUs at night and on weekends, with overrides of the shutdown for each AHU possible through a button at each thermostat.

*Multi-zone AHU*

Recommendations were made to improve operation of the multi-zone AHU beyond the reset schedule implemented during the first round of CC. It was recommended that zone temperatures and damper positions be sampled at regular intervals (e.g. 5 minutes) to determine the maximum damper position and minimum temperature. If the maximum zone damper position was greater than 90% open, the cold deck temperature set point would be lowered by 0.5 degrees, not to fall below 55°F. If the maximum zone damper position was less than 80% open, the cold deck temperature set point would be raised by 0.25 degrees, not to exceed 70°F. If the minimum zone temperature was less than 70°F, the hot deck temperature set point would be raised by 2.0 degrees, not to exceed 90°F. If the minimum zone temperature was greater than 70.5°F, the hot deck temperature set point would be lowered by 1.0 degrees, not to fall below 70°F.

*Electric Terminal Reheat*

As noted, the VAV AHUs served terminal boxes with electric reheat. The reheat was done completely against supply flow from the AHU, with no mixture of plenum air, and no terminal fan. It was observed that the minimum airflow values for the terminal boxes during heating mode were much higher than in cooling mode, due to the flow requirements of the heat strips. Each heat strip had an airflow requirement as a safety precaution, so that the strip would not operate when it sensed an airflow lower than required. The heat strips also ran intermittently as needed in order to maintain the space temperature at its set point.

This system design and operation caused significant simultaneous heating and cooling to occur. It was also the cause of a number of comfort complaints during the heating season. Rooms were cooled to the heating set point unnecessarily due to the relatively high amount of cold airflow that would enter during heating mode when the heat strips were not activated.

Terminal box control for this building was local loop control, but was available for access by the supervisory control through the DDC system. It was recommended that the supervisory control program be utilized to sample the terminal box operation continuously. When a box was in heat mode but with the heat strip off, it was recommended that the minimum airflow requirement be overridden to the minimum value required during cooling mode. When a heat strip was commanded on, it was recommended that the minimum airflow value be released in order to store the heating minimum value, allowing the heat strip to operate.

*VAV AHU Deck Temperatures*

Recommendations were made during the second CC process to utilize a demand based reset schedule for AHU discharge air temperature set points, rather than an outside air temperature reset as instated during the first round of CC. It was recommended that the terminal box controllers be sampled periodically by the supervisory control program, and that the terminal box with the maximum cooling demand be used as the basis for raising or lowering the discharge air temperature set point. For the VAV AHU served by an outside air pretreat unit, it was recommended that the upper limit for the discharge air temperature set point be 65°F, and the lower limit 55°F. For the two VAV AHUs with untreated outside air, it was recommended that the discharge air temperature set point not exceed 57°F when the outside air dew point temperature was higher than 55°F. For all other conditions, the same limits were recommended as for the unit with pretreated outside air.

*VAV AHU Static Pressure Set Points*

Recommendations were made during the second CC process to improve upon the outside air based static pressure reset schedule implemented during the first round of CC by incorporating a demand based reset schedule as well. It was observed that the outside air based reset seemed to work well in satisfying the spaces served. However, at times when building usage was low, the static pressure set point was still higher than necessary to meet building demand. Therefore, it was recommended that the outside air based reset be left in the supervisory control programming, but also that the damper position of each terminal box be sampled periodically to identify the maximum open position among the terminal boxes served. The static pressure set point would then be raised or lowered as needed to maintain the maximum open damper position between 85% and 95% open. However, the static pressure set point would not be allowed to exceed the value calculated by the outside air based reset.

*Outside Air Handling Unit*

Control of the outside air handling unit discharge air temperature prior to the second round of CC reset the cooling set point from 60°F to 55°F as outside air temperature varied from 45°F to 75°F, and maintained the preheat set point at 40°F. It was recommended during the second round of CC that the preheat set point continue at 40°F, but that the cooling set point be altered somewhat so that when outside air dew point temperature was above 55°F, the cooling set point would be 57°F, but when outside air dew point temperature was 55°F or below, the cooling set point would be 75°F.

*Water Loop Control*

Recommendations were made during the second CC process to improve upon the outside air based loop differential pressure reset schedules for hot water and chilled water loops implemented during the first round of CC by incorporating demand based reset schedules as well. It was observed that the outside air based reset schedules seemed to work well in satisfying the hot water and chilled water demand of the building. However, at times when building usage was low, the differential pressure set points were still higher than necessary to meet building demand. Therefore, it was recommended that the outside air based reset schedules be left in place, but also that the valve position of each AHU valve be sampled periodically to identify the maximum open position. The differential pressure set point would then be raised or lowered as needed to maintain the maximum open damper position between 80% and 90% open. However, the differential pressure set point would not be allowed to exceed the value calculated by the outside air based reset. This type of reset was recommended for both hot and chilled water loops.

It was also recommended that the control valve for each loop control to maintain the differential pressure set point instead of a flow set point, and that the pumps be allowed to shut off when primary pressure is enough to satisfy building needs. Additionally, it was recommended that a bypass line in one of the hot water coils be closed once the hot water pump was allowed to shut off. Previously some consumption had occurred because of this flow through the bypass line even when both hot water coils in the building were closed.

## DIFFERENCES IN CC RECOMMENDATIONS

From the discussion thus far, it is obvious that there were some significant differences between the two rounds of CC that occurred in the facility. The emphasis in the first round of CC was on outside air based reset schedules. These were implemented on AHU deck and discharge air temperature set points, static pressure set points, water loop differential pressure set points, and water flow set points.

The emphasis of the second round of CC was on improving the previous reset schedules through incorporating demand based resets. It was desired to know how much more savings could be achieved by utilizing demand based schedules over simple outside air based resets.

Additionally, due to comfort complaints in the building, the second round of CC had to focus more on issues relating to the terminal boxes. An innovative solution to the minimum flow problems encountered was presented as a recommendation. It was felt that this solution would aid in alleviating comfort complaints while also saving energy.

Table 1 is a summary of the differences in what was implemented during the first round of CC (or what was already in place and left unchanged) and what was recommended during the second round of CC.

**Table 1. Summary of differences between first and second rounds of CC.**

| Category | 1st CC Result | 2nd CC Recommendation |
|---|---|---|
| AHU Discharge Air Temperatures | Reset linearly based on outside air temperature. | Reset between maximum and minimum limits based on maximum zone cooling demand. |
| AHU Static Pressure Set Points | Reset linearly based on outside air temperature. | Reset between maximum and minimum limits based on maximum terminal box damper position. Maximum limit is value set by 1st CC linear outside air based reset. |
| Water Loop Differential Pressures | Reset linearly based on outside air temperature. | Reset between maximum and minimum limits based on maximum AHU chilled water valve position. Maximum limit is value set by 1st CC linear outside air based reset. |
| Multi-zone AHU Deck Temperatures | Reset linearly based on maximum and minimum zone temperatures. | Cold deck reset based on maximum zone damper position. Hot deck modulated to maintain minimum zone temperature at 70 degrees F. |
| Electric Terminal Reheat | Minimum flow during heating is value needed for heat strip to operate. | Minimum flow during heating is the same as during cooling when the heat strip not commanded on. When the strip is commanded on, the minimum flow is increased to the required value. |
| Outside Air Handling Unit | Cooling temperature reset linearly based on outside air dry bulb temperature. | Cooling temperature set to dehumidify when needed based on outside air dew point temperature, otherwise set at 75 degrees F. |

## SAVINGS ESTIMATION

Reliable metered energy consumption data for this building were available only after June 1, 2007. Therefore, no savings comparison between the initial equipment set up and the first round of CC was performed. Since the second round of CC is still in the implementation phase, a comparison of measured data before and after commissioning for savings analysis purposes also was not yet possible. Therefore, in order to estimate savings from implementation of the CC measures, calibrated simulation was performed, in accordance with ASHRAE Guideline 14.

The simulation software chosen was EnergyPlus. Metered data from June 1, 2007 through December 1, 2007 were chosen to represent the baseline or pre-CC period (referring to pre-2nd round of CC). Using known building inputs, the building was simulated, and this simulation was then calibrated to the pre-CC measured data. Figure 10 is a snapshot of the building as modeled in EnergyPlus. Figure 11 is an exploded view of the model showing the zones. Table 2 gives some detail as to the inputs used in the simulation for the calibrated pre-CC model.

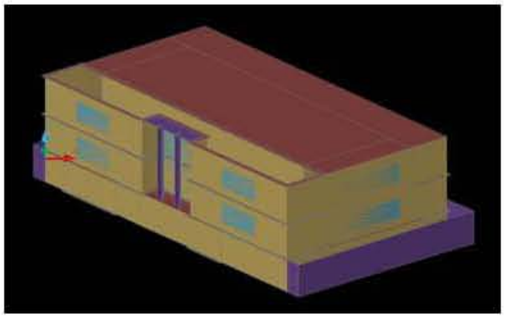

**Figure 10. Screenshot of building model in EnergyPlus.**

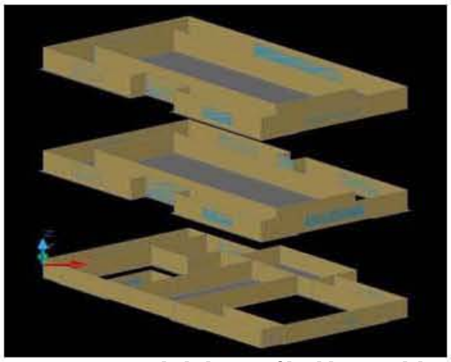

**Figure 11. Exploded view of building model.**

**Table 2. Summary of major input parameters used in pre-2nd CC simulation.**

| Zone | Floor Area (ft$^2$) | Peak Occupancy | Peak Electrical + Lighting Load (W) | Cooling Setpoint (°F) | Heating Setpoint (°F) | HVAC System | Cold Deck Setpoint (°F) | Hot Deck Setpoint (°F) | Pre-treat | Reheat |
|---|---|---|---|---|---|---|---|---|---|---|
| Server Room | 240 | 0 | 3000 | 70 | 70 | Four Pipe FCU | N/A | N/A | - | - |
| Basement-1 | 653 | 5 | 2000 | 75 | 75 | Multizone (Modeled as a Dual Duct Constant Volume) | 55-58 (based on warmest zone) | 85-110 (based on coldest zone) | Preheat (HW) and Precool (CHW) Coils[1] | - |
| Basement-2 | 1507 | 6 | 4376 | 75 | 75 | | | | | |
| Basement-3 | 580 | 2 | 1472 | 75 | 75 | | | | | |
| Basement-4 | 1383 | 0 | 2640 | 75 | 75 | | | | | |
| Basement-5 | 2326 | 10 | 5960 | 75 | 75 | | | | | |
| Basement-6 | 330 | 2 | 1112 | 75 | 75 | | | | | |
| Basement-7 | 400 | 2 | 1232 | 75 | 75 | | | | | |
| First Floor Interior | 3533 | 16 | 9100 | 75 | 71 | Single Duct VAV | 60-55 as OAT varied from 45-75 | N/A | Preheat (HW) and Precool (CHW) Coils[2] | Electric |
| First Floor Exterior | 4085 | 19 | 10900 | 75 | 71 | | | | | |
| Second Floor Interior | 3533 | 11 | 8300 | 75 | 71 | Single Duct VAV | 60-55 as OAT varied from 45-75 | N/A | Electric Preheat[3] | Electric |
| Second Floor Exterior | 4138 | 14 | 10500 | 75 | 71 | | | | | |

[1,2,3] The preheating coil set point is 40°F for all preheating coils. The pre-cooling set point is reset between 55°F and 60°F as the outside air temperature changes from 75°F to 45°F.

Figures 12, 13, and 14 show a comparison of simulated versus measured daily chilled water, hot water, and electricity consumption, respectively, as a function of daily average outdoor air temperature.

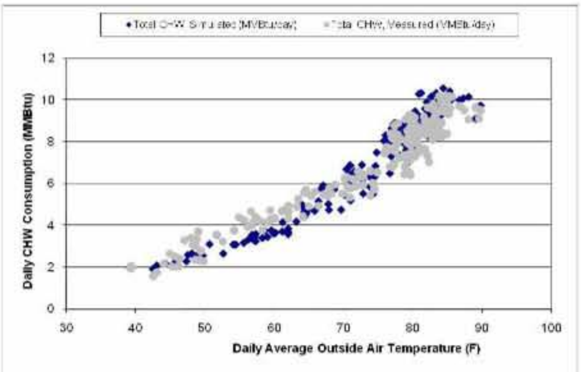


**Figure 12. Daily measured and simulated chilled water consumption versus average daily outdoor air temperature.**

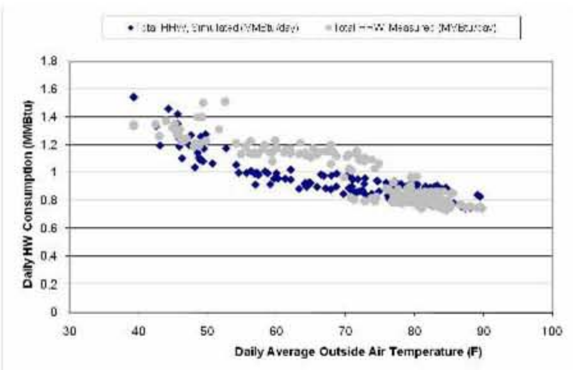


**Figure 13. Daily measured and simulated hot water consumption versus average daily outdoor air temperature.**

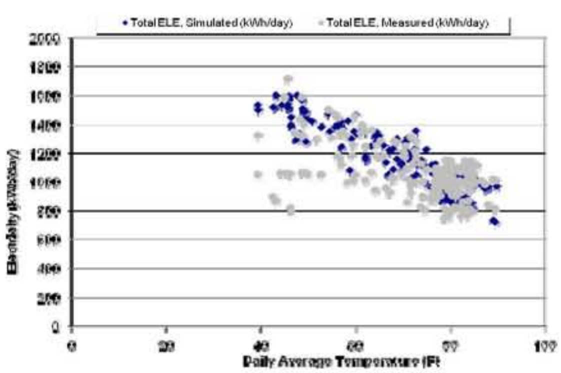


**Figure 14. Daily measured and simulated electricity consumption versus average daily outdoor air temperature.**

For this simulation, the cooling root mean square error (RMSE) was 10.4% of the average measured value, and the heating RMSE was 12.9% of the average measured value. The chilled water model was a very good fit to the measured data. The hot water model underestimated consumption for the middle range of outside air temperatures. This is likely due to a failed hot water valve on the multi-zone AHU, because after repairing the valve during CC the consumption dropped to levels closer to what is predicted by the calibrated model. The electricity model overestimated consumption somewhat at colder temperatures. However, based on the RMSE values this model was chosen as the calibrated model.

Once this calibrated simulation was obtained, those recommended measures that could be simulated were simulated, in order to estimate the savings that could be expected from implementation. A comparison of simulated chilled water, hot water, and electricity consumption before and after the second round of CC is shown in Figures 15, 16, and 17 that follow.

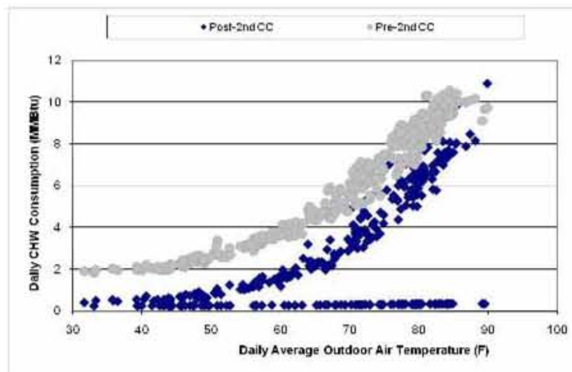


**Figure 15. Daily simulated chilled water consumption versus average daily outdoor air temperature before and after 2nd round of CC.**

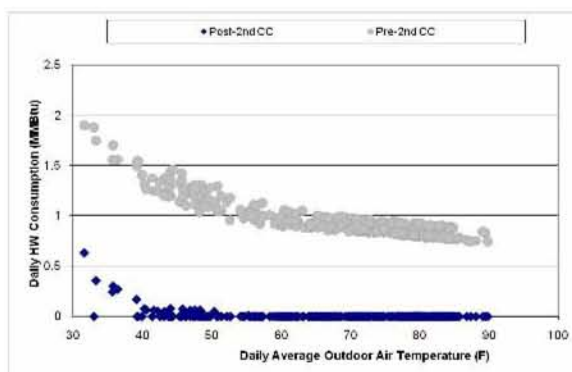


**Figure 16. Daily simulated hot water consumption versus average daily outdoor air temperature before and after 2nd round of CC.**

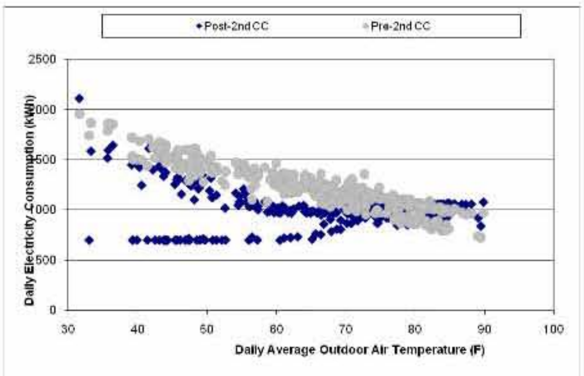


**Figure 17. Daily simulated electricity consumption versus average daily outdoor air temperature before and after 2nd round of CC.**

The estimated savings resulting from the measures simulated were 51% cooling, 99% heating, and 15% electricity. At energy prices of $7.347/MMBtu for chilled water, $9.735/MMBtu for hot water, and $0.079/kWh for electricity, this comes to a total of around $16,500 saved over the consumption levels before the second round of CC. These estimates do not include the recommendations for AHU static pressure resets and water loop differential pressure resets, nor do they include the recommendations dealing with terminal box minimum flow adjustment during strip heat operation. It was estimated that these measures would save an additional $3,500 per year, bringing total expected savings to $20,000 for all measures. This would decrease the building energy cost index from $2.11 per square foot per year to around $1.29 per square foot per year.

These savings estimates are rather large relative to the total energy consumption of the building, and they show a large potential for efficiency improvement. The hot water usage in the building can essentially be eliminated at outdoor temperatures higher than around 60 degrees F. Chilled water consumption can be significantly reduced in all periods, and electricity consumption can be reduced during colder temperatures. The measure with the largest effect on predicted energy savings was the AHU shut down schedule. The temperature deadband implementation for AHU 1 followed closely behind, followed by the improved minimum flow operation of terminal boxes in heating mode. Discharge air temperature resets on the other AHUs were also significant. Other measures had only slight impacts on savings.

A future paper is planned to address the savings actually achieved from these measures after the measures have been implemented, and when sufficient time has past to be able to compare measured energy consumption before and after the second round of CC.

## CONCLUSIONS

From the savings analysis performed, it is apparent that the second round of CC was beneficial in further reducing energy consumption in the building, beyond the reductions which occurred during the first round of CC. This was partly due to an emphasis in the second round on implementing demand based reset strategies in the building, as opposed to reset strategies solely based on outside air temperatures. More aggressive equipment shutdown schedules were also a large factor in the predicted savings from the second round.

Improvements in technology in recent years make demand based reset strategies much easier to implement and more effective. The ability of a supervisory control system to sample demands in spaces and coils allows the program to effectively control to minimize energy consumption at all times. Combining this ability with a traditional outside air based reset strategy is an effective way to reduce energy consumption even further, since it helps mitigate the effects of rogue dampers (Wei, 2004).

This investigation reaffirmed the inefficiency inherent in systems with electric terminal reheat in the supply airstream. Without a complete retrofit of the system, recommendations were made to reduce the impact of simultaneous heating and cooling at these boxes as much as possible.

It can be concluded that additional rounds of CC in a facility can be useful over time, even if an increase in energy consumption has not necessarily been detected. This is because continual improvements in technology over time sometimes allow improved CC measures to be implemented, particularly with regard to demand based reset strategies. Additionally, for those buildings that utilize terminal boxes with electric reheat in the supply air stream, a strategy similar to the one employed in this study can be implemented for improved energy efficiency and comfort.

## REFERENCES

ASHRAE Guideline 14

Wei, G., Liu, M., and Claridge, D. "Integrated Damper and Pressure Reset for VAV Supply Air Fan Control." ASHRAE Transactions: Research, 2004.