Demand-based Optimal Control to Save Energy: A Case-Study in a Medical Center

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ABSTRACT

Continuous Commissioning®[1] (CC®) strategies include reducing simultaneous heating and cooling, scheduling the facility's occupancy needs, utilizing free cooling, and minimizing excessive supply air and outside air. Most significantly, this demand-based control energy conservation strategy can facilitate mechanical system performance at near optimal conditions through the gradual advancement of control systems and the ability of upstream systems reading the status of downstream systems.

This paper demonstrates demand-based temperature, pressure and economizer control by the mathematical optimization methodology illustrated by a case-study, implemented with actual systems in a 1.2 million square foot medical center. Based on the optimization results, the facility saved over 5% total building electricity and over 10% gas consumption in a period of one year while improving thermal comfort and maintenance operations drastically.

INTRODUCTION

In a large medical facility, various mechanical systems are coordinated to support thermal comfort and indoor air quality. This achieved through the management of temperature, pressure, airflow, and economizer control. In general, Continuous Commissioning® (CC®) strategies include reducing simultaneous heating and cooling, scheduling the occupancy needs, utilizing free cooling, minimizing excessive supply air and outside air, and many others. Among these practices, a demand-based control energy conservation strategy can facilitate mechanical system performance at near optimal conditions through the gradual advancement of control systems, and the ability of upstream systems to read the status of downstream systems. The application of this strategy depends on how it is implemented within the actual systems, in which many limitations and technical difficulties remain. Still, many facilities have not adopted this demand-based control strategy due to lack of knowledge, fear of change by maintenance staff, and other reasons.

Typically, control schedules are programmed according to commonly accepted practices by maintenance staff depending on their field experiences or by using "canned" software programs provided to the control technicians. These schedules are likely to be set beyond necessary ranges due to concerns that some building spaces may suffer from thermal discomfort. However, demand-based control can achieve optimal efficiency without compromising thermal comfort or indoor air quality when applied correctly.

The objectives are to optimize the temperature control, pressure control, and economizer switch-over, needed to minimize the total energy cost and/or consumption. This paper presents demand-based temperature, pressure and economizer control in part by using optimization methodology developed by Joo's dissertation [Joo 2004]. The control strategies were implemented with actual systems in a 1.2 million square foot medical center based on the optimization results.

DEMAND-BASED CONTROL TO OPTIMIZE

The concept of demand-based control is not very new and last for almost a decade. [Hartman 2001; Seidl 2001] Current advancement of DDC and network systems in HVAC industry has made this control strategy possible. With conventional control methods, the control loop uses a fixed set point or a set point reset based on simple parameters, which are unrelated to the actual demand. Cooling and heating required in an actual system's operation can be a demand, but those may not be readily available through a control system. The position of dampers and valves, room temperature, static pressure, or fan and pump speed are good indicators of demand for they can be read through the control system. For example, a VAV box position can be an indicator of its zone load condition. A cooling valve position can be a representation of its unit's cooling load condition. A fan's speed can be a representation of the amount of airflow and pressure required for the

[1] Continuous Commissioning® or CC® is a registered trademark of the Texas Engineering Experiment Station at Texas A&M University.

unit. Therefore, there are many ways to read the demand of equipment and buildings. The demand-based control is achieved by reading the demand through the network of different controllers.

The demand-based control can be applied to various systems in different control such as temperature control in all kinds of air-handling units and pressure control in variable speed fans or pumps. The true optimization can be accomplished by the demand-based control as all of the systems are operated at near optimal condition. In theory, the demand represents the edge point or optimal point. If the demand can be monitored, the system can be optimized.

In a large-scale medical facility, there are various kinds of energy systems, and each system is exposed to different objective functions with many constraints for the optimization. Therefore, this paper will first introduce brief overview of the medical facility and its systems, explain each system's optimization theory, describe application to the case, and present results.

FACILITY INFORMATION

The new Madigan Army Medical Center (MAMC) was built in 1992 with a gross floor area of about 1.2 million square feet. The central plant consists of four (4) upgraded 635 ton chillers and two (2) 345 ton absorption chillers with a primary/secondary chilled water loop configuration. The condenser water is cooled by five (5) wells, and two (2) booster pumps serve absorption chillers. 100 pound per square inch (PSIG) [689476 pascals (Pa)] high pressure steam is supplied by a separate boiler plant building. The high-pressure steam is reduced to 60 PSIG [413685 Pa] medium-pressure steam for sterilization and to 15 PSIG [103421 Pa] low-pressure steam for heating. The steam is converted to hot water for the heating.

There are 111 operational air-handling units serving this building. The types of existing air-handling units include dual-duct constant volume systems (DDCV), single-duct variable air volume reheat systems (VAV Reheat), single-duct constant volume terminal reheat systems (SDCVTR), single-zone constant volume units (SZCV) and dedicated computer room units.

OPTIMIZATION THEORY

In the HVAC application, an objective function is defined as energy costs, consumptions or savings. Eventually an optimal point is expressed as a minimum value of the energy cost or consumption, or a maximum value of energy savings within constraints. The constraints are defined as the physical boundaries that a system can reach or a system should be operated. Joo's dissertation illustrates optimization in dual-duct systems as an example [Joo 2004].

Optimization in some HVAC systems can be very complex. However, simple optimization methods could be applied to the systems in the case study facility because the objective function in control of most parameters only move to one direction, either high or low in order to achieve the optimal point. Joo's dissertation explains the reason in optimal temperature control in a dual-duct constant volume system with constant fan speed. In the dual-duct constant volume system, for instance, the optimal point can be achieved with the lowest possible hot deck temperature and the highest possible cold deck temperature because the optimization results in reducing simultaneous heating and cooling. In a VAV system, likewise, the optimal fan speed occurs when the system runs with the lowest possible duct pressure. This 'one-directional optimization' makes the demand-based control easy to achieve with simple changes in control systems.

APPLICATION

The demand-based control was applied to the whole facility. In some systems, the demand-based control was not feasible. The feasibility depends on the capability of network communication between controllers and physical constraints which allows optimization. The facility had well-established control system and network as well as very knowledgeable HVAC and control engineers. Therefore, their systems were running very well in terms of not only maintenance point of view but control strategies. Some of their control sequences already adopted demand-based control, in which cases we redefined some of the constraints in order to maximize the savings output. The control schemes should be illustrated system-by-system in order to understand the optimization efforts.

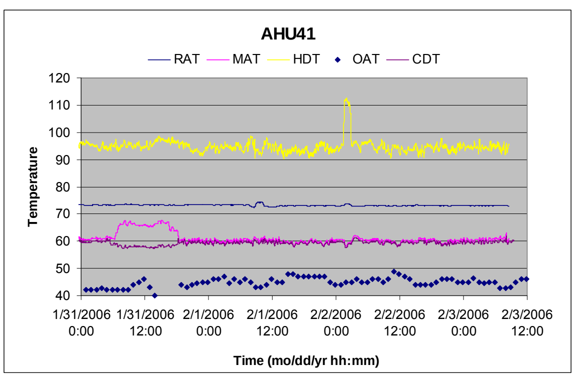
Dual-Duct Constant Air Volume (DDCV) System

A dual-duct air-handling unit system is an apparatus that supplies both cold and warm air to spaces where cooling or heating is required. The DDCV system comprises a supply air fan, a return air fan, mixing/relief dampers, a pre-heating coil, a cooling coil, a heating coil, and zone terminal boxes. The cold air is cooled by cooling coils in a cold deck, and hot air is heated by heating coils in a hot deck. A fan delivers the conditioned air through two parallel air ducts. Terminal boxes modulate either the hot airflow or the cold airflow, or both, to maintain room air temperature.

Typical characteristics of this DDCV system are 1) simultaneous heating and cooling by mixing heated air and cooled air at the terminal boxes wasting both heating and cooling when each zone is exposed to partial load, and 2) economizer penalties.

<u>Hot and Cold Deck Temperature Reset</u>

In the original control, the cold deck temperature set point was reset to maintain the hottest room at a maximum room temperature. The hot deck temperature set point was reset to maintain the coldest room at a minimum room temperature. A kind of the demand-based control was in place. The actual deck set point boundaries are set as constraints using a low limit and a high limit. The actual cold deck and hot deck temperature set points are determined by reset calculated from selected hottest room and coldest room within the reset bands: the difference between the low limit and high limit. The cold deck low limit and high limit were generally set at 55 ºF [12.8 ºC] and 65 ºF [18.3 ºC]. The hot deck low limit set points ranged from 85 ºF [29.4 ºC] to 95 ºF [35 ºC], while the hot deck high limit was generally set at 110 ºF [43.3 ºC].



RAT: return air temperature
MAT: mixed air temperature
HDT: hot deck air temperature
OAT: outside air temperature
CDT: cold deck air temperature

Figure 1. 4-day trended temperature data in a sample unit before implementation of demand-based control

As shown in Figure 1, the cold and hot deck temperatures were maintained constant through out a few days of measurement. The existing demand-based control was not working correctly because of 1) lack of controllability in objective parameter settings and 2) wrong setting of constraints (reset bands). The reason for 1) is that an actual one room can be set or maintained at the minimum room temperature set point which is a control parameter. Therefore, the actual temperature was constant. The reason for 2) is that the low limit of hot deck temperature was set too high, and vice versa for the cold deck temperature.

Alternative control parameters and constrains were implemented in order to make the system control as true demand-based control:

1) The hot and cold deck temperatures are reset depending on actual room conditions of all the occupied areas. The building automation system (BAS) reads room temperatures and set points from all the boxes. Then it calculated the maximum value of the difference of those two parameters. The hot and cold deck temperature set points are reset to maintain the difference at minimum levels (0.8ºF [0.44 ºC] in the actual programs).

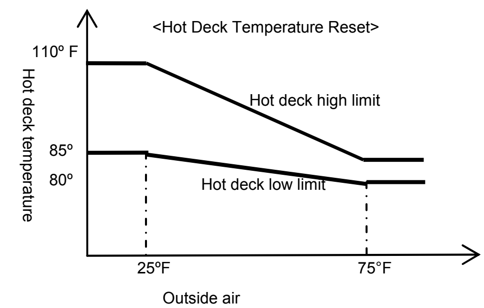


Figure 2. Hot deck temperature reset boundaries in a DDCV system

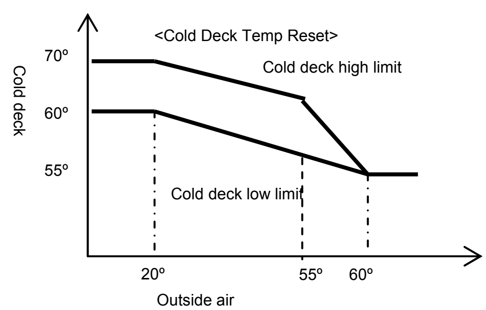


Figure 2. Cold deck temperature reset boundaries in a DDCV system

2) The boundaries of low limits and high limits are set for the reset schedules based on the outside air temperature for all DDCV units except critical units serving the ER and Dialysis areas. The actual deck set point boundaries are set at the programming level. The set point remains within this boundary even if a reset value determined by the temperature difference becomes higher or lower than the boundary. The

purpose of the boundary reset is to ensure that those temperature resets are performed regardless of communication loss from boxes and not affected by false temperature readings. If the set points stay constantly at the boundary, it is recommended to inspect for fault detection of sensors, dampers or valves' operation. *Figures 2* and *3* describe the boundaries resets for hot and cold deck temperature resets, respectively, based on outside air temperature.

<u>Optimal Economizer Control</u>

Originally, the economizer was enabled when outside air temperature is lower than 68ºF. The mixed air temperature was controlled at cold deck temperature set point when the economizer was enabled. If more zones are calling for heating, however, the economizer may yield more heating penalty than cooling savings in DDCV systems. [Liu et. al. 1997; Joo 2004]

The optimal economizer control sequence is implemented for the DDCV units except units serving several critical areas per facility staff's request. The diagram of the control program implemented is shown in *Figure 4*. The program adds actual heating and cooling airflow rates separately from all DDCV terminal boxes in a unit, compares actual heating consumptions and cooling consumptions, and select a lower cost to operate between economizer and non-economizer. In the control diagram the result of lower cost is represented as I1 (or Input 1), and the determination of economizer-enable signal is represented as I2 (or Input 2). If I1 and I2 are satisfied, then the system will turn on the economizer. Otherwise, it will turn off the economizer. The costs of electricity and steam need to be updated.

Single-Duct Variable Air Volume (SDVAVRH) Reheat System

A single-duct VAV reheat air-handling unit system (see *Figure 1*) supplies conditioned air through a single duct route to spaces. Each zone has a terminal box which controls the amount of air flowing to the zone. In a SDVAV reheat system, the terminal box houses a heating coil which reheats discharge air to accommodate the zone load.

Typical characteristics of this SDVAV system are 1) simultaneous heating and cooling by reheating discharge air and 2) duct pressurization to accommodate all the zones.

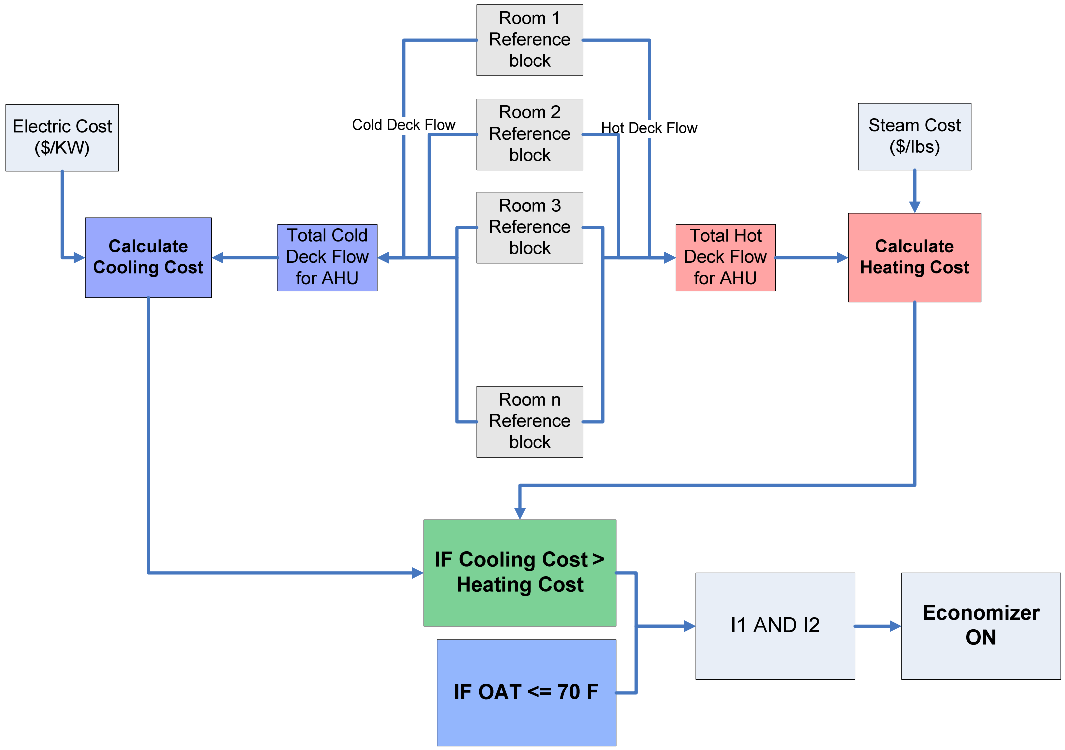


Figure 4. A Flowchart of Economizer Control for a Dual Duct Constant Volume Unit

Discharge Air Temperature Reset

The control sequences optimization of supply air temperature reset in the SDVAV is similar to the cold deck temperature reset described in the DDCV above.

Static Pressure Reset

For the exiting operation, the SDVAV units were typically controlled to provide a constant 1 inch H_2O static pressure set point at two (2) sensor locations. Some AHU's used a higher set point such as 1.2 or 1.5 inch H_2O [300 or 374 pascals (Pa)] due to the system or room requirement.

A new demand-control strategy was implemented to reset the static pressure. Now the static pressure is controlled based on remote and critical zones' damper positions. For most AHUs the static pressure is reset within a range of 0.5 to 1.2 inH2O [125 to 300 Pa] as boundaries. If any of the remote dampers is above 95%, the control loop will raise the static pressure set point within the depicted boundary as shown in *Figure 5*. Therefore, the supply air fan speeds up to accommodate the maximum zone demand. If any of the remote dampers is below 95%, the control loop will decrease the static pressure set point. Within selected boxes one box will maintain 95% of its damper position as a maximum, and all others' damper positions will be lower than 95%. This control eventually open all the boxes as much as possible without compromising thermal comfort of any zone, and thus maintains the lowest possible static pressure set point.

The boundaries are determined by the fan speed and the relation between the flow and pressure using Equation 1. The minimum was set at 0.5 inch H_2O [125 Pa]. The purpose of the boundary reset is to ensure that the pressure reset by damper positions should be performed regardless of communication loss from boxes and not affected by false damper position readings.

$$P_{\max} = P_{\min} + (P_{\max} - P_{\min}) \times \overline{N}_{fan} \pm \alpha \qquad \text{Eq (1)}$$

$P_{\max}$: Maximum Static Pressure, normally design set poin

$P_{\min}$: Minimum Static Pressure

$\overline{N}_{fan}$: Fan speed ratio

α : boundary ranges

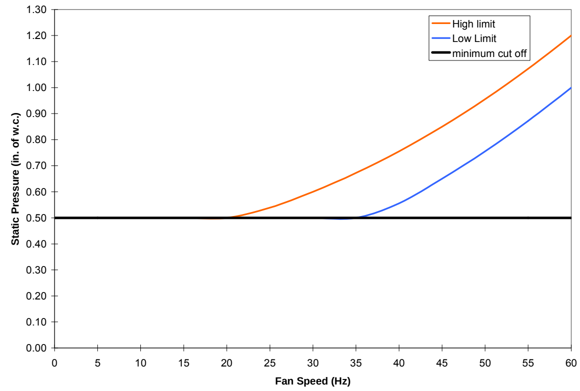


Figure 5. Variation of Static Pressure Reset based on Fan Speed

Minimum Supply Airflow Reset

The minimum supply air flows were previously set to a constant value for both exterior and interior zones. The minimum flow set point varied from 50% to 100% of maximum design flow.

The minimum supply flow rates were reset for both interior and exterior zones. For interior zones, the Minimum Heating flows were reduced to 0 since there would be no heating load for interior zones. The minimum cooling airflow was set to 10% of the design airflow for most boxes. For a few boxes in which the damper locked up due to the low minimum flow rates the minimum cooling flow rate was set to 30% of the design flow rate. For exterior zones, minimum cooling flow rates were set to 10% of the design flow rate. The minimum heating flow rates were set to vary between 15~30% of the design flow when the Outside Air temperature varied between 20~60 °F [-6.7~15.6 ºC]. Specialized areas such as exam rooms and materials are excluded due to the air circulation rate requirement.

Chilled Water System

The chilled water system has a primary-secondary loop. Each chiller has a dedicated primary chilled water pump. Those pumps seem balanced to maintain design water flow through the chillers. Three sets of secondary pumps (two identical pumps per each set) supply chilled water to Hospital, Clinic and Tower. There is no valve in the decoupling pipe.

Secondary Chilled Water Pump Pressure Reset

The chilled water pumps were previously controlled to maintain a differential pressure (ΔP) of 15 pounds per square inch (PSI) [103421 Pa] across the pump. A control logic was created to modulate the chilled water pump pressure between 12.5 and 22 PSI [86184 and 151685 Pa] based on the remote eight (8) air handlers' demand. The logic monitors remote air handlers' cooling valve positions and compare them against a value of 95%. Should any of the valves be above 95% the loop raises the ΔP set point to supply higher loop pressure, and vise versa.

RESULTS

The results in this paper are shown as an aspect of whole facility energy savings. First, the energy baseline is determined by using monthly electricity consumption by utility bills and hourly steam consumption which can be measured because the facility purchases steam from a nearby plant. Then, weather-dependent models are simulated by using EModel's program. [Kissock. et. al. 1993]

Whole Facility Energy Baseline

The baseline models of electric and steam energy consumptions was derived from monthly electricity utility bills for the years 2004 and 2005 (excluding two unusual and missing data) and available hourly steam data also in 2004 and 2005 (total 9,354 data), which were normalized by outside air temperature. The impact of internal heat load variation was ignored because there was no significant change in heat load over the implementation process according to the facility management.

Figure 6 shows the results of a weather-dependent baseline model for daily electricity usage. The model uses regression of daily average consumptions (from monthly utility bills) verses monthly average outside air temperature. The savings will be calculated by comparing the monthly baseline consumption (kWh/day * number of days per month) to actual electricity bills. *Figure* 7 shows the result of a weather-dependent baseline model for hourly steam usage, simulated by EModel's program. The savings are calculated by comparing the hourly baseline consumption (kLbs [1 kLb = 4.536 kg]/hour) to actual hourly steam data. The savings are summed into monthly savings in the following section.

Whole Facility Energy Savings

The implementation activities began in January 2006, but the major implementation started in June 2006 through January 2007. Therefore, the savings calculation starts from July 2006. The utility data were continuously collected until September 2007. The facility saved electric and gas consumption for the period of 15 months by 2,106,085 kWh and 16,768 kLbs [7606 kg of steam], respectively, which is about 4.9% of the baseline electricity and 10% of the baseline steam consumption as shown in *Figures 8* and *9*.

ANALYSIS

There are two major factors having affected the results in heating consumptions which are not described in the paper. First, the maintenance personnel complained for slow response of heating up and overriding the hot deck temperature in December 2006. Most dual-duct systems were serving patient rooms, which generally require higher room temperatures in a short time upon patients' demand. With constant volume system configuration, this consumed too much steam for heating afterward. It was partially corrected during the next visit in April 2006, but the hot deck temperature reset had to be somewhat compromised for some units. Valve leakage was also noticed for hot water valves in four units due to high pressure across them, resulting in excessively high hot deck temperature. The other is that there can be a bias on baseline calculation in steam consumption because in year 2004 and 2005, the facility used both of the absorption chillers, while in 2006, they were used only for maintenance purposes. There are two reasons that prevent the use of absorbers: 1) reduced cooling capacity through Continuous Commissioning® and 2) upgrade of four centrifugal chillers from 605 tons to 635 tons.

CONCLUSION

With the advancement of control and network systems, the demand-based control is the most feasible way to optimize HVAC systems. This control scheme was applied to various HVAC systems in a 1.2 million square foot medical facility for the period of 15 months. The facility saved an accumulative electric and gas consumption of 4.9% of the baseline electricity and 10% of the baseline steam consumption, respectively.

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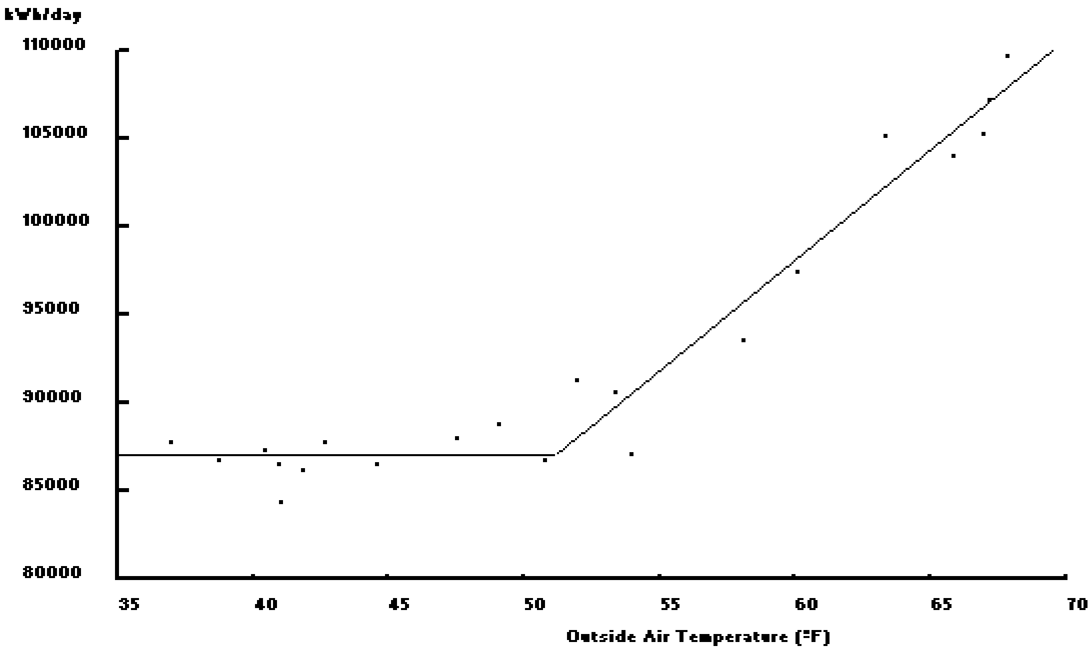

Figure 6. A baseline model for electricity usage (kWh/day vs. outside air temperature) by using EModel

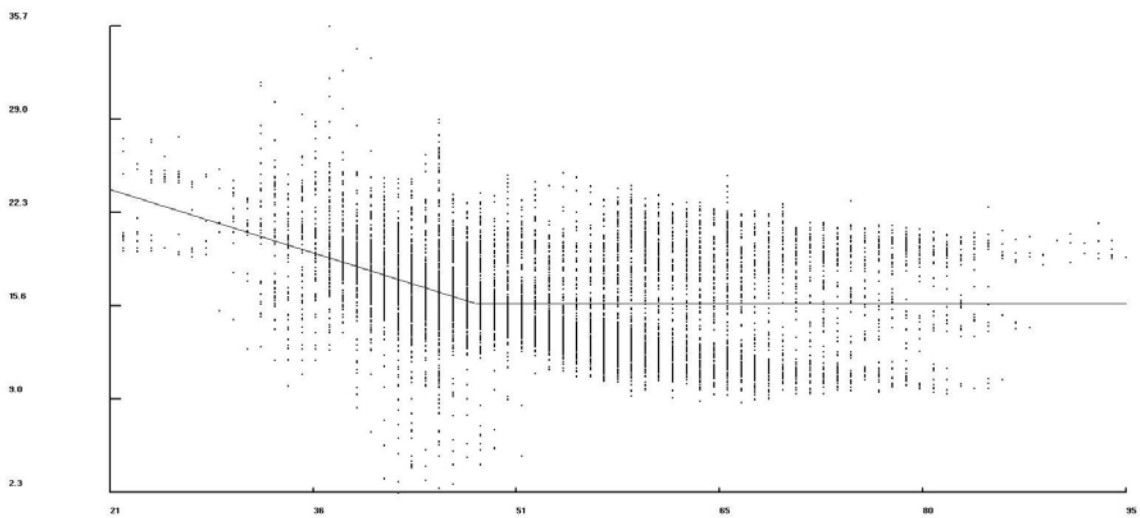

Figure 7. A baseline model for steam usage (kLbs/hour vs. outside air temperature) by using EModel

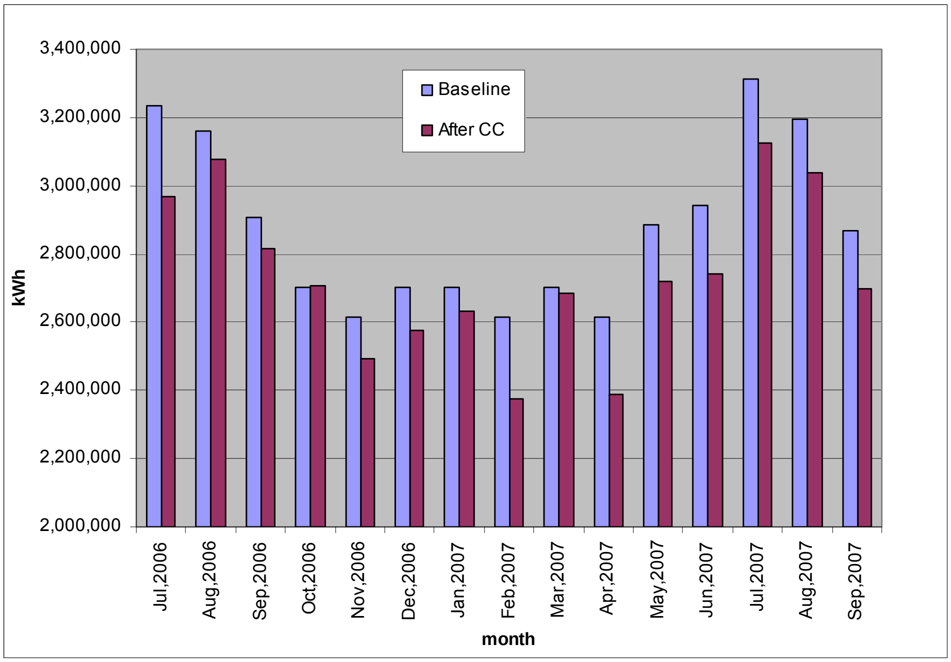

Figure 8. Monthly electricity consumptions comparison (baseline vs. actual consumption after CC®)

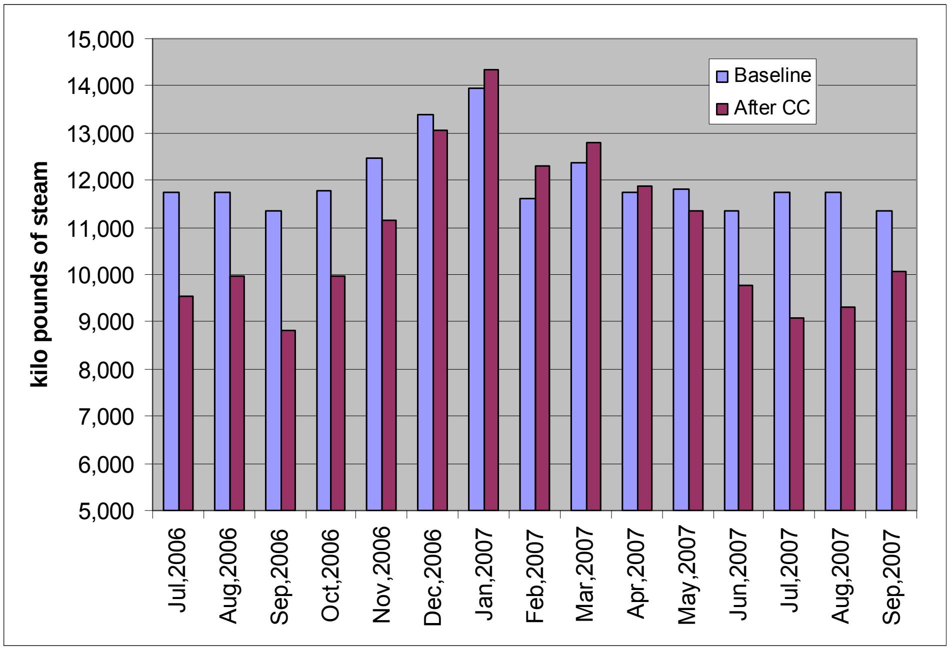

Figure 9. Monthly steam consumptions comparison (baseline vs. actual consumption after CC®)