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ABSTRACT

A multi-zone air-handling unit was popular several decades ago due to the convenience of small sized modular units, which were inexpensive to install and easily maintained in a mechanical room. The cost and convenience proved to be of little benefit as the units perform poorly from an energy usage perspective. A "three-deck" multi-zone unit is a hybrid of its kind, and it can be very efficient when controlled properly. In theory, there will not be simultaneous heating and cooling if its heating damper is controlled separately from the control of the cooling damper. When the zone load is neutral (not heating or cooling), for example, all the mixed air will be bypassed through the bypass deck. However, there are opportunities and challenges in this system.

This paper presents theoretical backgrounds of advantages and challenges in the system operation of the three-deck multi-zone unit and methods to optimize temperature and economizer control to improve energy efficiency. A case-study will be presented examining a medical facility's utilization of 35 three-deck multi-zone units serving most hospital areas, as well as illustrates a 10% savings in total gas and electric consumptions over the period of one year.

INTRODUCTION

A multi-zone air-handling unit was popular several decades ago due to the convenience of small size modular units. However, it performs poorly in its energy usage perspective. The lack of energy efficiency is a result of constant fan operation and simultaneous heating and cooling. The system generally serves three (3) to ten (10) zones from a centrally located air-handling unit. Since its number of serving areas is limited, the unit is typically too small for the zone and tends to operate with a constant speed fan. Each zone requires different heating or cooling load that is accommodated by mixing cold and warm air through a zone damper. [ASHRAE 2000] For each zone damper, the hot deck zone damper is interconnected with the cold deck zone damper in opposed direction. Therefore, simultaneous heating and cooling is not avoidable unless all the zones are exposed to full cooling or full heating. In fact, these units are currently installed as an option for small serving zones in many facilities, taking advantage of the centrally located system configuration leading to reduced installation costs and easy maintenance. As an alternative to the multi-zone air-handling unit, engineers may select a three-deck (3-deck) multi-zone unit or a Texas multi-zone unit depending on their budgets or preferences in order to improve energy efficiency, thus save energy costs.

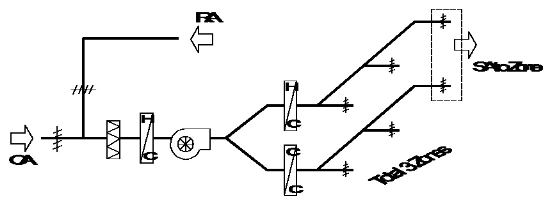


Figure 1. Schematic of a Typical Multi-Zone Unit

This paper discusses a case study of optimization of three-deck (3-deck) multi-zone units in a medical facility. The 3-deck multi-zone unit is a hybrid of its kind, and it can be very efficient when controlled properly. The system has three (3) distinct decks: a heating deck on the top, a cooling deck at the bottom, and a bypass deck in the middle. The heating deck heats up mixed air by a heating coil, and the cooling deck cools down the mixed air by a cooling coil. The bypass deck has no coils. In each zone, the hot deck damper is linked with a bypass damper in an opposed direction while the cold deck damper is linked with another bypass damper in an opposed direction. [ASHRAE 2000; McDowall 2007]

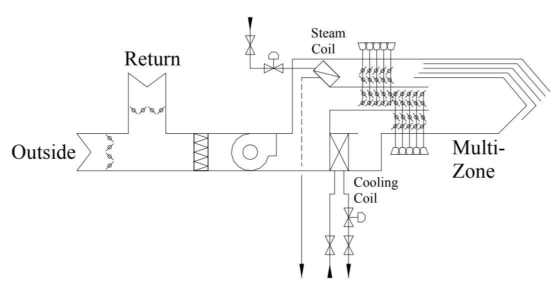


Figure 2. Schematic of a 3-deck Multi-Zone Unit

This paper presents theoretical backgrounds of advantages and challenges in the system operation of the 3-deck multi-zone unit and methods to optimize temperature and economizer control to improve energy efficiency. A case-study will be presented examining a medical facility's utilization of 35 three-deck, multi-zone units serving most hospital areas.

THEORETICAL BACKGROUND

The control strategies for conventional (2-deck) multi-zone units should not be applied to 3-deck multi-zone units because of three unique system characteristics. First in theory, no simultaneous heating and cooling occurs in the 3-deck multi-zone system because the heating damper is controlled separately from the control of the cooling damper. If the zone load is neutral (not heating or cooling), for example, all the mixed air will be bypassed through the bypass deck. [McDowall. 2007]

Second, the economizer may generate a higher cost penalty than the savings achieved from free cooling. The major advantages of the 3-deck multi-zone units over conventional 2-deck multi-zone units are negated when both the cold deck and bypass deck supply cold air during the period of economizer use. Then, the 3-deck units function like a conventional 2-deck multi-zone unit which works very similar to a dual-duct constant volume system. The heating coil is required to heat up the cold air to the hot deck. The economizer of a dual-duct system needs to be carefully selected. [Liu et. al. 1997; Joo 2004]

Finally unlike conventional 2-deck multi-zone units, 3-deck multi-zone units with different hot and cold deck temperature set points yield comparable heating and cooling consumptions, providing that in hot and humid season cold deck temperature is set constantly dehumidifying the air. The theory behind this statement is following.

In a 3-deck multi-zone unit, the heating and cooling are represented as Equations 1 and 2.

$$\dot{q}_h = \sum_{i=1}^{n} \dot{Q}_{h_i} \cdot \rho \cdot c_p (T_h - T_m) \qquad \text{Equation (1)}$$

$$\dot{q}_c = \sum_{i=1}^{n} \dot{Q}_{c_i} \cdot \rho \cdot c_p (h_m - h_c) \qquad \text{Equation (2)}$$

Where, $\dot{q}$ = heating or cooling

$\dot{Q}$ = airflow rate

ρ = air density

c_p = specific heat

T = temperature

Subscripts:

h, m, c and n = heating, mixed, cooling and number of boxes, respectively.

Equations 3 and 4 are examples comparing the heating consumptions for two different hot deck temperatures in a zone.

$$\dot{q}_{h_1} = \dot{Q}_{h_1} \cdot \rho \cdot c_p (T_{h_1} - T_m) \qquad \text{Equation (3)}$$

$$\dot{q}_{h_2} = \dot{Q}_{h_2} \cdot \rho \cdot c_p (T_{h_2} - T_m) \qquad \text{Equation (4)}$$

With conceivably constant volume, supply air temperature for the zone depends only on the zone load. A mass and heat balance theory can be applied as shown in Equations 5 and 6. The variations of air density and specific heat for different temperatures were negligible.

$$\dot{Q}_{h_1} + \dot{Q}_{m_1} = \dot{Q}_{h_2} + \dot{Q}_{m_2} \qquad \text{Equation (5)}$$

$$\dot{Q}_{h_1} \cdot T_{h_1} + \dot{Q}_{m_1} \cdot T_m = \dot{Q}_{h_2} \cdot T_{h_2} + \dot{Q}_{m_2} \cdot T_m \qquad \text{Eq. (6)}$$

When Equation 5 is inserted into Equation 6, the results of the heating consumptions for two different hot deck temperatures in Equations 3 and 4 are equal. Considering that the other zones work the same way, heating consumptions for the 3-deck unit with different hot deck temperatures do not differ. The same analogy can be applied to the cooling consumptions with two different cold-deck temperatures for dry-coil application. The cold deck temperature generally remains constant in wet-coil application for dehumidification.

CASE STUDY

Facility Information

The case study shows the implementation of Continuous Commissioning® (CC®)[1] in a full-service hospital with operating rooms (OR), emergency rooms (ER), a lab, a pharmacy, patient rooms, day clinics, and administrative offices. The facility began

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

operation in 1965, and has been renovated multiple times throughout the past 40+ years. The main hospital is a six story building and has a floor area of 439,834 square feet [40,862 square meter]. The facility operates 24 hours a day, 7 days a week, year round.

Steam and chilled water are provided by a utility plant located in a nearby separate building. The chiller plant has three chillers: a 500 ton chiller (Chiller 1), a 300 ton (Chiller 2) and a 750 ton (Chiller 3). The chilled water system has a primary-secondary loop. The central plant houses five boilers: two (2) large main boilers and three (3) small back-up boilers. The boilers produce 100 pound per square inch (PSIG) [689476 pascals (Pa)], high-pressure steam and reduces to 60 PSIG [413685 Pa] medium-pressure for sterilization and laundry, and to 15 PSIG [103421 Pa] low-pressure for heating.

There are total of 47 air-handling units serving the hospital. The majority of air-handlers are multi-zone (MZ) units: 35 three-deck multi-zone units and 2 old regular multi-zone units. Among the 35 three-deck units, 16 units were installed in 2004, and are serving exterior-zone patient rooms. The other system types include: (2) two dual-duct (DD) systems, (2) two single-duct constant volume (SDCV) systems serving the OR, as well as single-zone (SZ) systems and outside air make-up systems.

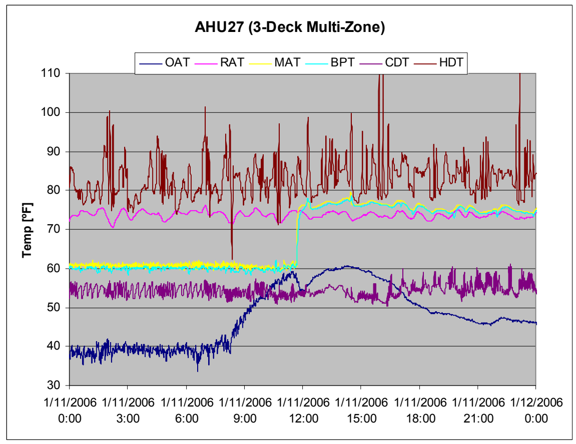
SAVINGS OPPORTUNITIES

Unoccupied Hours Set-back

The facility utilized an unoccupied-hour set-back strategy for 16 new three-deck multi-zone units serving patient rooms, but did not use it prior to the implementation of Continuous Commissioning® (CC®) due to technical difficulties in the control system. Therefore, a schedule was created to control the system based on building occupancy for the rest 19 units. The occupied period begins at 05:00, or 5 a.m., and ended at 17:00, or 5:00 p.m. During the unoccupied period, the cold and hot deck temperature set points are set back near room temperature, and the economizer was disabled to save heating energy. Detailed control sequences for temperature reset and economizer control are described in next sections.

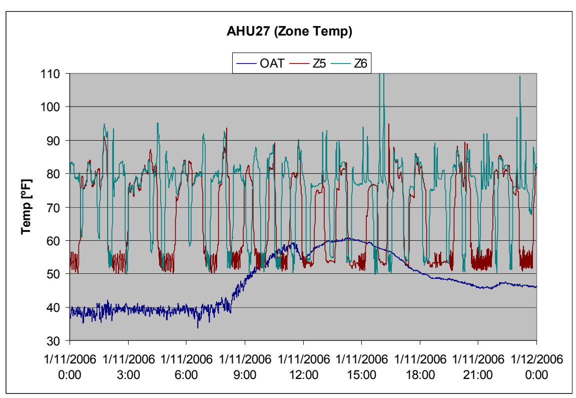
Hot-deck and Cold-deck Temperature Reset

Even if it was theoretically proven that different hot and cold deck temperature set points yield same heating and cooling consumptions, temperature reset had to be implemented in this facility because of the unstable hot deck temperature control. *Figure 3* shows a day of trended temperature data in a 3-deck multi-zone unit. The hot deck temperature was oscillating very rapidly ranging from 75 ºF [23.9 ºC] to over 100 ºF [37.8 ºC]. This occurs due to the use of steam coil as a heating source in the hot deck. The pressure inside steam coil could be below atmospheric pressure at the projected mixed and hot deck temperature. This unstable hot deck temperature yields instability in supply air temperature in some zones as shown in *Figure 4*. These zones responded the oscillation of hot deck temperature. Therefore, the temperature reset can help reduce the energy savings if the hot deck temperature is swinging at the lower range and vice versa for the cold deck.



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

Figure 3. A day of trended temperature data in a 3-deck multi-zone unit



OAT: outside air temperature
Z5 & Z6: zone supply air temperature

Figure 4. A day of trended zone supply air temperature data in a 3-deck multi-zone unit

In the previous control, the cold deck temperature set points were set at constant 55ºF [12.8 ºC] in most units. The hot deck temperature set points were set constant at 90ºF in most units, over 90ºF [32.2 ºC] in some units overridden by the facility engineer.

Basically, the cold deck temperature set point has been reset to maintain the hottest room at a maximum room temperature, and the hot deck temperature set point has been reset to maintain the coldest room at a minimum room temperature. This is called "warmest or coldest zone control." The warmest or coldest zone control monitors all or sampled zone temperatures served by individual AHU's, and adjusts the set point accordingly.

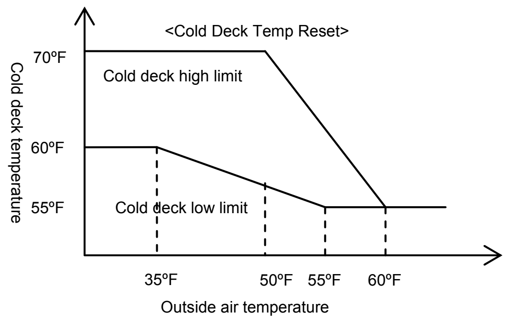


Figure 5. Cold deck temperature reset boundaries

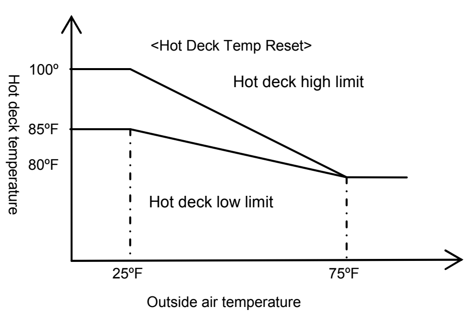


Figure 6. Hot deck temperature reset boundaries

The actual heating/cooling set points fall within the boundaries set by a low limit and the sum of the lower limit and reset band. These set points are adjusted up or down by the warmest/coldest zones mentioned in the previous paragraph. The cold deck low limit is reset between 55 ºF [12.8 ºC] and 60 ºF [15.6 ºC] based on outside air temperature as shown in *Figure 5* for occupied operation. The cold deck reset band is also reset between 0 ºF [0 ºC] and 10 ºF [5.6 ºC]. The hot deck low limit is reset between 80 ºF [26.7 ºC] and 85 ºF [29.4 ºC] based on outside air temperature for occupied operation as shown in *Figure 6*, and 70 ºF [21.1 ºC] and 75 ºF [23.9 ºC] for unoccupied operation. The hot deck reset band is set between 0 ºF [0 ºC] and 15 ºF [8.3 ºC].

Temperature Reset Set-Back during Unoccupied Hours

For unoccupied operation the cold deck is set at 72 ºF [22.2 ºC], and the cold deck reset band is set at 0. The hot deck temperature is set at 70 ºF [21.1 ºC] and 75 ºF [23.9 ºC]. Therefore, the units supply no heating or cooling, even though the fan is running, until the coldest zone reaches 55 ºF [12.8 ºC] or the hottest zone reaches 78 ºF [25.6 ºC] as set-back temperatures. If a zone reaches the set back temperature, the unit becomes an occupied mode.

Optimal (Smart) Economizer Scheduling

Most three-deck multi-zone units have set-back schedule from 17:00 to 05:00 following morning. When the units come back to the occupied mode at 05:00, most zones require heating during cold season. The heating penalty much exceeds free cooling with economizer operation during this warm-up period as described in an earlier section.

The economizer was turned off until more zones switched to cooling mode during the warm-up period. In the control systems, no airflow rates could be measured by any means. Therefore, the schedules for the warm-up period were set in different hours for different units and serving areas. This scheduling was critical and the major energy savings measure in this facility.

In order to avoid sudden opening of outside air dampers soon after the economizer is enabled, the optimal economizer scheduling added an artificial mixed air temperature set point during warm-up schedules to 78 ºF [25.6 ºC]. This occurs due to the control memorizing the previous mixed air temperature value (normally about 70 ºF [21.1 ºC]) compared to the mixed air temperature set point of 2 ºF [1.1 ºC] lower than cold deck temperature set point.

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 and gas consumptions by utility bills. Then, weather-dependent models are simulated by using EModel program. [Kissock. et. al. 1993]

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Whole Facility Energy Baseline

The baseline models of energy consumptions were derived from Year 2004 and 2005 monthly electricity and gas utility 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 CC® process according to the facility management.

Figures 7 and *8* illustrate the results of weather-dependent baseline models for electricity and gas usage, respectively. 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 or MCF [1 MCF = 28316 cubic meter]/day * number of day per month) to actual utility bills.

Three data points in *Figure 7* seem outliners, but they are valid. Unfortunately, the facility had bills which did not show exact reading dates. This may impact the accuracy of baseline calculations.

Whole Facility Energy Savings

The CC® activities began in March 2006, and the major CC® implementation started in June 2006 through January 2007. Therefore, the savings calculation starts from July 2006. Models normalized by outside air temperature, explained in an earlier section, derived the baseline data. The utility data were continuously collected until December 2007.

Figures 9 and *10* show monthly electricity and gas comparisons, respectively, between the baseline and actual consumption. The facility saved accumulative electric and gas consumption for the period of 18 months by 1,898,923 kWh and 6,899 MCF, respectively, which is about 9.1% of the baseline electricity and 10.0% of the baseline gas consumption.

ANALYSIS

The actual consumptions seem odd because the number of days per month in the utility bills was not shown, but the overall consumption remains comparable. Also note that the actual gas consumption from February 2007 to June 2007 increased drastically. It was discovered that the facility's staff had overridden the optimal economizer scheduling and used the economizer constantly in the winter. In August 2007, a workshop was conducted outlining the system characteristics and configurations to the facility's staff. Subsequently, the facility's staff had a better understanding of the system and resumed the optimal economizer sequence.

CONCLUSION

The 3-deck multi-zone unit has unique characteristics including fairly good energy efficiency and low capital costs when engineers design an air-handling system. Minor disadvantages include difficulty in humidity control in certain outdoor conditions and inefficient constant speed fan operation. If one is to install this unit in a facility, the accommodating equipments and control systems must be carefully considered in order to avoid additional obstacles and energy waste. With the major control schemes such as optimal temperature reset and economizer control in this study, the facility could save 9.1% of the baseline electricity and 10.0% of the baseline gas consumption.

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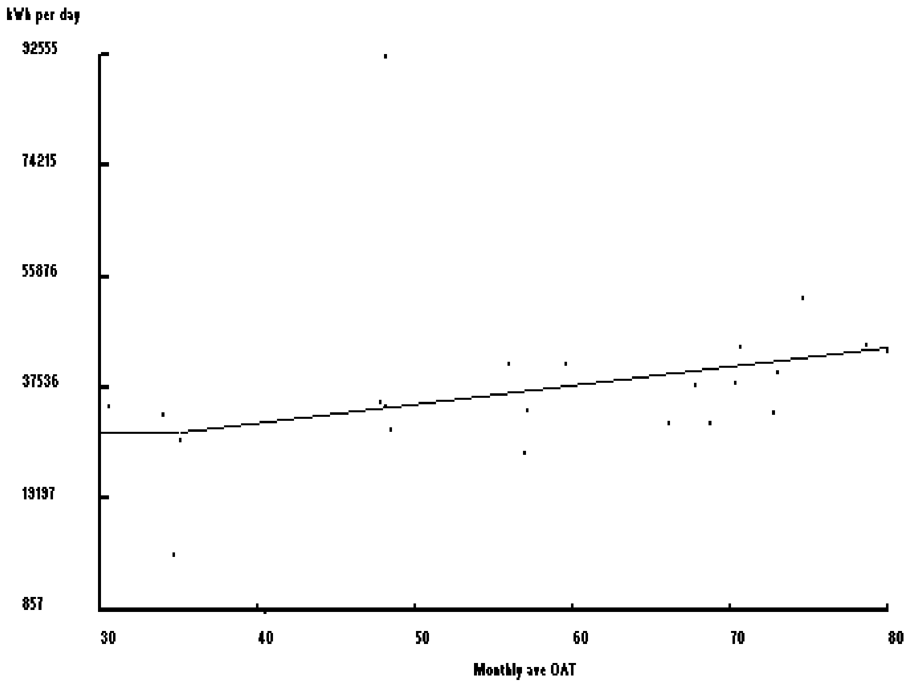

Figure 7. A baseline model for electricity usage as kWh/day vs. outside air temperature by using EModel

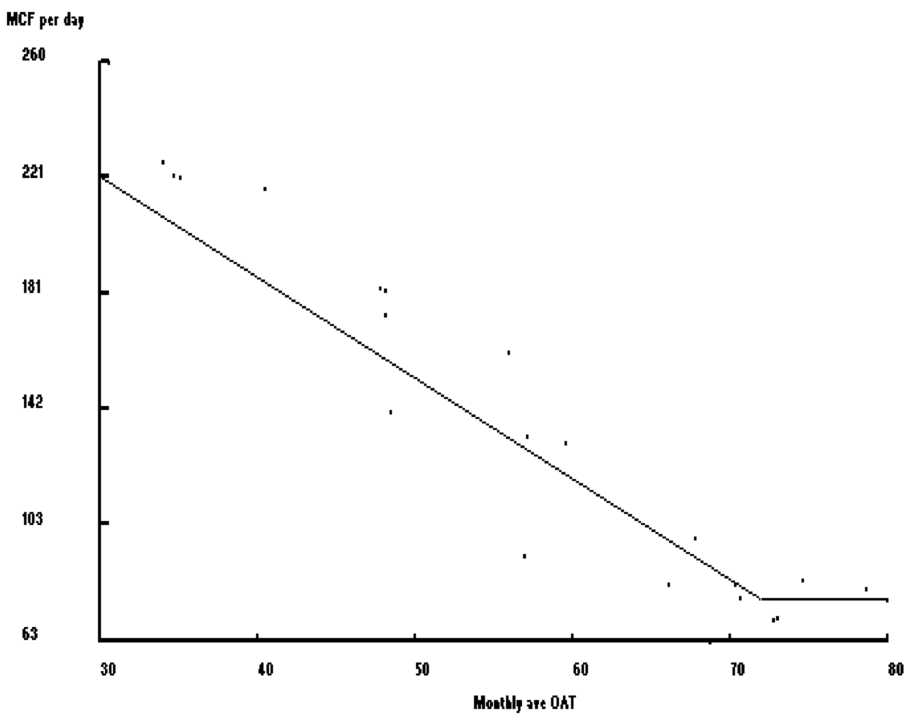

Figure 8: A baseline model for gas usage as MCF/day vs. outside air temperature by using EModel

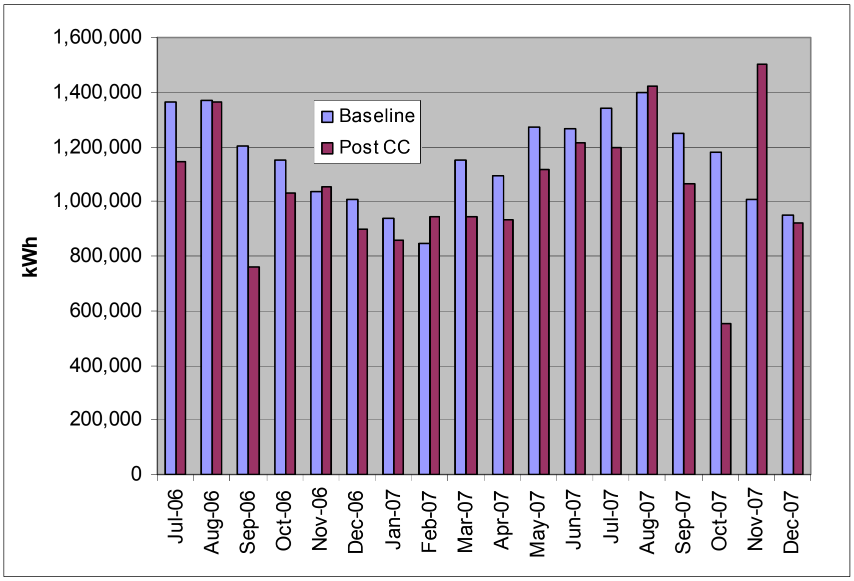

Figure 9. Monthly electricity consumptions comparison

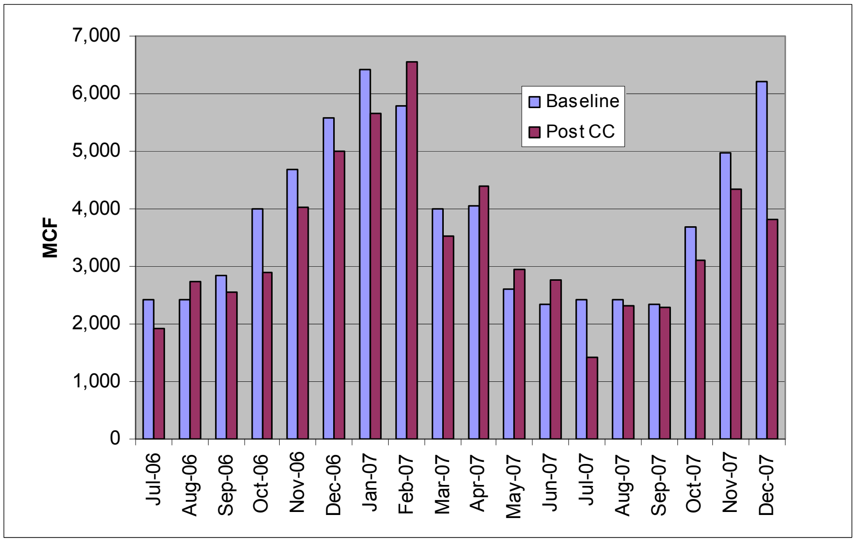

Figure 10. Monthly gas consumptions comparison