Energy Efficiency and Solar PV for Klondike Gold Rush National Historical Park

January 2010

Prepared for:

University National Parks Energy Partnership Program
and
National Park Service

Prepared by:

Keith Boyd Fackler and Phil Malte
Energy and Environmental Combustion Laboratory
Department of Mechanical Engineering
University of Washington
Seattle, WA 98195-2600
Phone: 206-685-2171
E-mail: malte@u.washington.edu

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Introduction

The Klondike Gold Rush National Historical Park is a “bookend” park, with the southern Visitors’ Center in Seattle, Washington and the northern Visitors’ Center in Skagway, Alaska. The park provides the visitor with historical perspective and background on the Klondike Gold Rush of the late 1800s and early 1900s, which brought gold rush fever from Seattle to southern Alaska into the Yukon Territory of Canada. NPS web sites for the park are:
Seattle: http://www.nps.gov/klse/index.htm
Alaska: http://www.nps.gov/klgo/index.htm

This work treats the Seattle Center, which is housed in the Cadillac Hotel and is located in the historic Pioneer Square District of downtown Seattle. The Hotel is shown in the photograph (Photo 1) on the front cover of this report (taken early February, 2009). Photo 2 below is a photograph of the hotel ca. 1918, and Phoot 3 shows the damage following the Nisqually earthquake of February 28, 2001. The hotel was rebuilt and Klondike NHP and NPS moved in a few years ago. Although no longer a hotel, the hotel sign over the sidewalk has been retained for historical purposes. Presently, the hotel houses the Klondike Gold Rush Visitors’ Center on the first floor (ie, the street level) and in the basement, and NPS offices and library on floors two and three. In the photographs, the east side of the hotel is to the right, and south side of the hotel is to the left. There is an alley on the west side of the hotel, so only one side (the north side) is common with an adjacent building.

In this study, UNPEPP tasked the University of Washington with 1) examining energy usage by the Seattle Klondike National Historical Park (KHP) and 2) working with Klondike KHP to bring solar PV to the park. Energy usage at Klondike NHP and the Cadillac Hotel has been of concern, and thus, it was thought timely to examine this in detail, thereby understanding the building’s energy usage and allowing efficiency and conservation methods to develop from this understanding. Green generation of electricity by solar PV is of key interest to both the National Park Service and the City of Seattle. The partners for this the solar PV installation are Klondike NHP, Historic Seattle Inc, Seattle City Light, Bonneville Environmental Foundation, and the University of Washington. The major equipment for a 3.4 kw rated solar PV system, including PV panels and inverter, has been purchased and delivery taken, with installation planned during the first half of 2010.

In this final report of the University of Washington’s UNPEPP-funded work, 1) the energy usage study and findings are reported, and 2) the solar PV system designed and specified by the partnership is reported. Also, a summary of work specific to the last two months of the UNPEPP project is given first.

Photo 2. Cadillac Hotel ca. 1918

Photo 3. Cadillac Hotel February 28, 2001

Summary for November and December 2009

The UNPEPP contract to the UW ran from July 1 to December 31, 2009. Previously, summaries have been provided for July-August and September-October, 2009.

During the November-December period, the energy usage study for the Cadillac Hotel was completed by the UW. This included the analysis of information from several sources – the principal architect involved in the reconstruction of the hotel following the 2001 Nisqually earthquake, HVAC engineering involved in zone water loop heat pump systems as used in the Cadillac Hotel, maintenance and occupant information on the heat pump system, and electric and gas utility bills. The draft report on energy usage was written, the final version of which is included in this report as the "Energy Usage at Cadillac Hotel…." sections. The key findings are as follows:

- Energy usage appears to be high for a building of the size and function of the Cadillac Hotel.
- Two explanations of the apparent high energy usage stand out: 1) lack of insulation on the outer walls of the building, and 2) and a poorly maintained heat pump system.
- On the matter of lack-of-insulation, it was decided upon reconstruction of the Hotel following the 2001 Nisqually earthquake to withhold insulation from the brick walls so as to permit heat to readily flow from the interior of the building through the brick walls, thereby helping to maintain the brickwork dry during the cool rainy season. This was a controversial decision, not significantly supported by the architect.
- On the matter of heat pump system maintenance, we found several problems, including: 1) the preheater on one of the system's two air handlers has malfunctioned since the winter freeze of 2008-09, 2) the heat pump unit on the 1st floor (west end) that has not worked for some time and this heating function has been replaced with floor mounted electrical resistance heaters, 3) windows open on upper floor offices on winter days (40-50°F) presumably because of room overheating, and 4) electrical resistance heaters in upper floor offices presumable to provide auxiliary heat on cold days. Additionally, thermostats may be poorly placed respective to the heat pump units they control.
- Our estimate of the overall coefficient of performance (CoP) for the Cadillac Hotel heat pump system, based on the data available to us, is 0.6. Admittedly, there is significant uncertainty in our estimate. Nonetheless, it is well below the literature value of 5 for the CoP of a building heat pump system of this type and 2-3 for the CoP of residential air source heat pumps.

A decision on adding insulation to the building walls is not an easy one, though it should be carefully weighed. It will reduce energy consumption and add to comfort. A decision on the heat pump system is obvious. A first class contractor, skilled in the art of water loop zone heat pump systems in the Pacific Northwest (especially for small-medium sized buildings) should be retained and put to work repairing, tuning, and perhaps re-engineering the system.

The major equipment comprising the solar PV system was ordered on December 1, 2009 and received at Klondike NHP on December 22, 2009. This included 16 REC 210 watt rated solar PV panels and 1 SMA SB 3000US inverter rated at 3 kw AC output. Of this, the UW, through funds available via this contract, purchased 14 of the 16 REC solar PV panels at a total cost of $7500. The additional costs are being borne by Klondike NHP and Seattle City Light. These include the balance of the equipment and material costs, installation, and monitoring. Bonneville Environmental Foundation is coordinating the installation, which is planned to take place during the first half of 2010. The design and layout of the solar PV system is discussed in the last section of this report.

Energy Usage at Cadillac Hotel – Facility

Introduction

The purpose is to determine how energy is used at the Cadillac Hotel. Included in this analysis is a list of potential energy conservation projects, their approximate costs, and the estimated energy savings. This information can be used to support future performance contracting.

The recommendations within this report are based on a few years of submitted usage data including monthly natural gas and electrical consumption data and 15 minute electrical load data obtained for August and January of 2008. In addition to load data, the final recommendations are based on various site reviews, interviews with building inhabitants, and correspondence with the building architect. The energy savings and energy production figures are projected estimates based on conceptual project upgrades, information gathered at the site, and from the historical utility data provided. The actual savings may vary from these estimates for a variety of factors. The figures used for the cost of recommended upgrades are based on probable costs from 2009 RS Means Construction Data and are intended for feasibility purposes only. The recommended measures should proceed to detailed design and further reevaluation followed by competitive bidding.

The approach taken in this has included several comprehensive building walkthroughs and a review of the associated systems and equipment, including both process and building systems. The major areas covered in our audit include the building envelope, electrical systems, HVAC system, and operational and maintenance procedures. A major element of the audit also includes an initial interview and ongoing consultation with operational and maintenance personnel, as well as building occupants.

Structure

Shown in Figure 1 (same as Photo 1 on cover), the Cadillac Hotel is a three story brick building located at 319 Second Ave S in Seattle. The approximate floor area is 26,000 square feet. The exterior walls are constructed of brick with both wood and steel framing. The south, east and west walls are exposed, where as the north wall is shared with the adjacent building. The building was constructed in 1889, when fire damage was much more of a concern than earthquakes (MacIntosh, 2009). The walls were constructed of brick with wood and steel framing. The original interior walls were composed of lath and plaster. Some of the exterior and most of the interior walls were critically damaged by an earthquake that occurred on February 28th 2001. Damaged to the point of possible demolition, Historic Seattle Inc. stepped in and decided to restore and preserve the building as a historical structure.

Figure 1: The Cadillac Hotel from the southeast. (2009)

The specific rehabilitation to the building includes seismic reinforcement of the south, east, and west walls, and restoration of the exterior brick and wooden structure. The original single glazed windows were upgraded to double glazed with ¼'' gap on the second and third floors. The windows on the east wall and part of the south wall on the first floor were upgraded to double glazed with a ½'' gap. There were no upgrades made to the windows on western end of the south wall due to the desire to preserve ornamental glass that was part of the original construction shown in Figure 2.

Figure 2: Ornamental glass above single glazed windows on the 1st floor south facing wall.

As mentioned above, the interior walls were almost completely destroyed in the earthquake. Upon tearing out the lath and plaster, it was discovered that the existing insulation was quite damp and the brick had significant water damage.

Instead of replacing the broken interior with a fresh vapor barrier, new insulation, and modern sheet rock, it was decided to leave out the insulation. It was thought that the structural integrity of the brick was compromised and the building heat would help keep the brick dry (Phillips, 2009). This will be discussed more later, but it should be noted that this was a questionable decision as far as the energy efficiency of the building goes.

Despite the debatable decision to remove the wall insulation, the building received a brand new roof with new insulation there. Also, two large skylights were installed above the main lobby and library on the 3rd floor. These are double glazed with ½ inch gap.

<u>Heating and Cooling</u>

The heating and cooling of the building is accomplished by a multi-zone water-source heat pump system. In a multi-zone system, multiple heat pumps are placed in different zones of the building to provide individual space conditioning. These small heat pumps are connected to a common source, which is a pipe of water looped through the building. The closed water loop runs between each heat pump and a cooling tower and boiler on the roof to provide additional heat or cooling when necessary. The water in the loop can be thought as a heat source/sink for each of the heat pump units, and it can store the heat inside a building and meet the different requirements of cooling and heating of each heat pump unit (ie, zone) during a certain time of period. Typically, the heat from the inner zone (interior zone or core area) in the building can be transferred to its outer zone (exterior zone or perimeter area) by the circulation water (Lian, 2005). A schematic diagram of a generic water-source heat pump coupled to a boiler and cooling tower is shown in Figure 3.

Closed-loop water-source heat pumps are often used to condition large, multi-zone commercial buildings, such as schools and office buildings. These systems usually work quite well when a building needs to cool one section of the building and heat another. For example, during a bright sunny day, the rooms on the south side of a building may need to be cooled, while the rooms on the north side need to be heated. In an office building, the core of the building could be in cooling mode, while the perimeter is being heated (Fegan, 2003). These systems work most efficiently when some of the heat pumps are in cooling mode and others are in heating mode; thus heat is simply being transferred around the building and not created (boiler/cooling tower). Buildings in the far north and far south do not get as much benefit out of these systems, because there are large periods of time when all of the heat pumps are either in cooling or heating mode (*Heat Pump Mechanics*). Small and medium sized buildings might also not receive as much benefit.

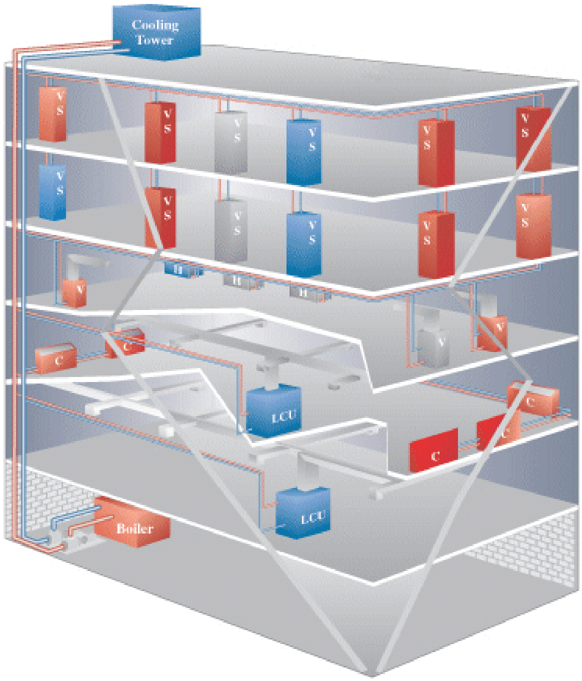

Figure 3: Schematic of a closed loop, water source heat pump with auxiliary boiler and cooling tower (*Heat Pump Mechanics*).

Unlike an air-source heat pump, which exchanges heat with the outdoor air, closed loop water-source heat pumps exchange heat with themselves, the boiler, and the cooling tower. When most of the heat pumps in a building are in heating mode, most of the heating comes from the boiler; thus, supplementary thermal energy must be supplied to the system in addition to the electrical energy that is powering the air handling units and heat pump compressors (Fegan, 2003).

The Cadillac Hotel has 5 heat pumps per floor (including the basement) and each heat pump has an associated thermostat. A couple of points will be illustrated by the following example. The mechanical drawing for the southwest corner of the 3rd floor is shown in Figure 4. Heat pumps 302, 303, and 305 are circled in red and their corresponding thermostats are circled in green. Heat pumps 302 and 305 condition the library which is one open room. Heat pump 303 conditions the office space to the south of the library.

The thermostats for heat pumps 302 and 305 are located only about twenty feet apart from each other. These two heat pumps are essentially conditioning the same space. If someone where to set these thermostats at different points, the two heat pumps would be competing against each other, which would definitely cause an increase in energy consumption. Another situation to note is the fact that the thermostat for heat pump 303 is not located in the same room as the heat pump itself. The thermostat is actually separated by two walls and a hallway. At the very least, the thermostat should be located in the same room of that which its corresponding heat pump is conditioning.

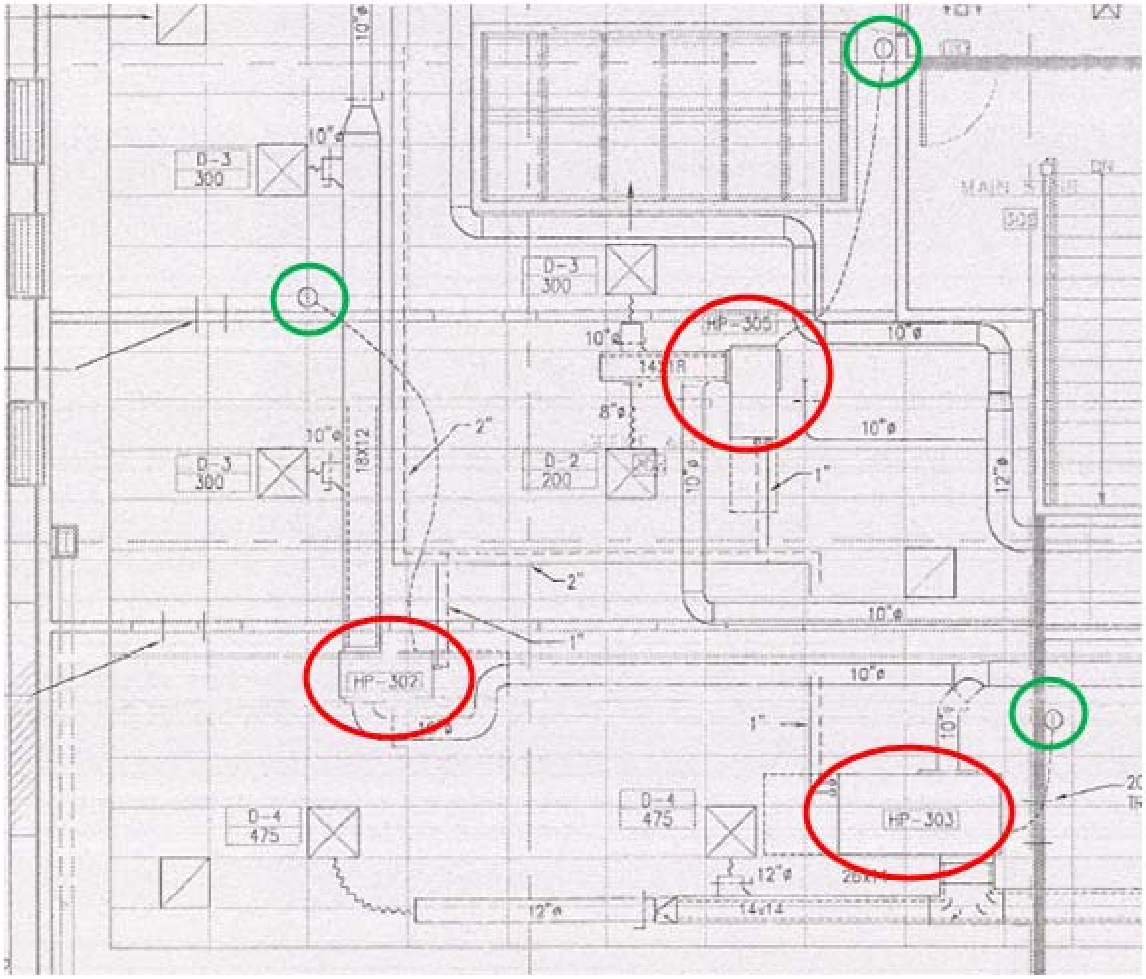

Figure 4: Mechanical drawings for the southwest corner of the 3rd floor.

The above discussion is just an example of some of the concerns with the existing HVAC system. There are similar examples of this throughout the building. The particular system is quite complicated, and it should be left to experienced professionals to make detailed changes.

The before mentioned cooling tower and boiler are both of high quality and efficiency. The evaporator is an Evapco ATW4830 high efficiency closed circuit cooling tower. This unit is an induced draft counter-flow cooler, as shown in Figure 5. Hot return water from the heat pump loop enters the cooling tower at near the top of the unit. Cool dry air enters through the base of the unit and moves up over the coils. At the same time, water is sprayed down over the coils opposite the flow of air. The evaporation of this water helps remove the heat from the closed circuit loop. Warm moist air then exits the top of the unit with the help of a fan, and liquid water that didn't evaporate along with some make up water is pumped up to the top of the cooler and back down over the coils (ATWB Closed Circuit Cooler).

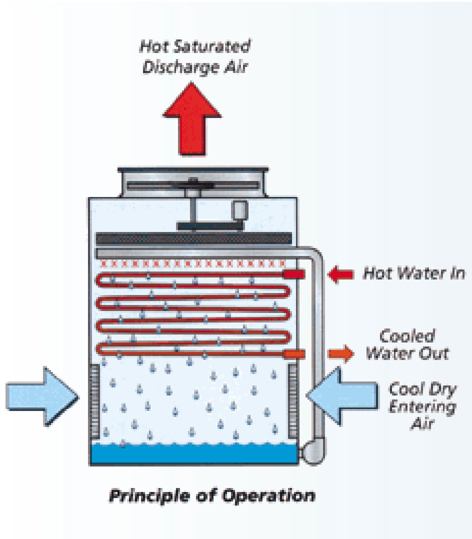

Figure 5: The operational basics behind the Evapco ATWB closed circuit cooling tower (ATWB Closed Circuit Cooler).

Because the system operates on the principle of evaporative cooling, there is very little electrical load. Really the only electrical consumption associated with this device is associated with the fan and the cooling water sump pump.

The boiler is a high efficiency natural gas fired boiler with a copper fin heat exchanger manufactured by the Lochnivar Corporation (model # CBN0645). This boiler meets the toughest air standard requirements reportedly producing less than 30 ppm NO_x (corrected to 3% O_2). Water is circulated at a high velocity through highly efficient finned copper tubes. The particular model on the Cadillac Hotel has a thermal efficiency of 81%; with a natural gas input of 645,000 Btu/hr, approximately 522,450 Btu/hr is transferred to the working fluid (Copper Fin Commercial Gas Boilers, 2007).

The concerns are how often the cooler and boiler are called on the maintain the water in the building water loop between the high and low set points

Lighting

The lighting on the second, third floor and part of the basement is primarily 28 and 32 watt T8 fluorescent lights with high efficiency ballasts. There are CFL's scattered throughout the building, and much of the building lighting is controlled by motion sensors. Additional lighting in the museum (located on the basement and first floors) includes 113 Phillips Mastercolor 30 L 70 watt metal halide lamps with reflector cans. For the most part, the building lighting system incorporates good energy efficient equipment. One possible improvement would be to replace the metal halide spot lights with something of higher efficiency. However, since these lights are located inside the museum part of the building, the quality and hue of the light may be more important than efficiency.

Other Equipment
The building supports a variety of light office equipment, including copy machines, servers, computers, and printers. The majority of this equipment is located on the second and third floors. Museum displays include several touch screen computers. It also supports an elevator.

Schedules
The Klondike NHP occupies the basement and 1st floor of the building. The park museum is open daily throughout the year from 9:00am to 5:00pm with the exception of Thanksgiving, Christmas, and New Year's Day. The National Park Service offices, located on the second and third floors of the building, are open from approximately 8:00 AM to 6:00 PM on Monday through Friday. Although the park offices are not open on the weekends, the park employees that work the museum have offices on the upper two floors. There is still an occupational electrical load on the upper two floors over the weekend. In addition, there is no reduced operational schedule in regard to space conditioning on the upper two floors over the weekend.

Energy Usage at Cadillac Hotel – Analysis

When performing an energy analysis, it is important to gather information on energy trends throughout the year. Information for both electrical and natural gas consumption for 2008 was obtained from Seattle City Light and Puget Sound Energy and is displayed below in Figure 6.

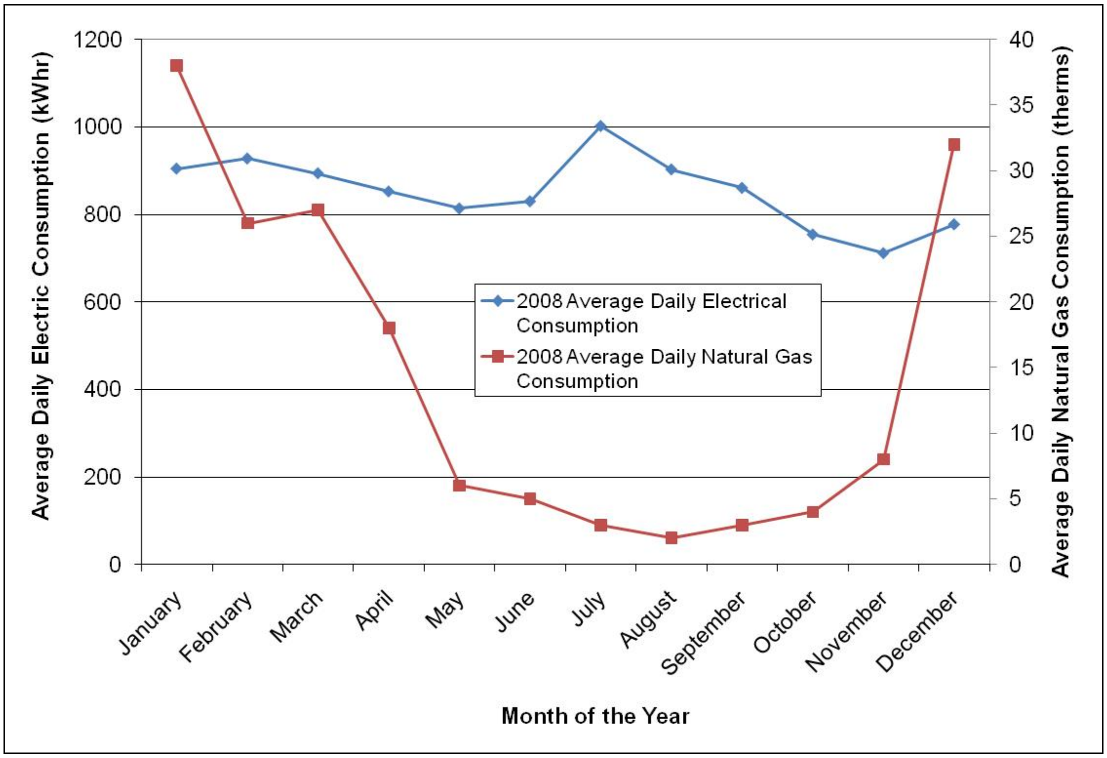

Figure 6: Actual Electrical and Natural Gas Consumption for 2008.

The trends shown above in Figure 6 are typical of a building in the Pacific Northwest. As expected, the natural gas usage is the greatest during the heating season. There are two peaks in the electrical consumption, one corresponding to an increased cooling load in the summer and the other corresponding to an increased heating load in the winter. Although there is no direct electrical heating (e.g. baseboard resistance heating), the increased electrical usage in the winter can be attributed to increased electrical consumption by the heat pump compressors.

One troubling point to take note of in regard to the natural gas consumption is the fact that there is still significant gas consumption during the summer months. The average daily consumption for August is approximately 2 therms and the consumption for July and September is approximately 3 therms. The equivalent energy consumption in kWh is between 58 and 88 kWh per day, which comprises about 10% of the daily building energy consumption. This would not be quite as disturbing if the hot water in the building was heated by a gas boiler; however, the sole hot water tank in the building is a 50 gallon electrically powered water heater. This relatively large amount of natural gas usage during the summer months is thought to be a direct result of the heat pump system. This issue should be further explored if the building is to become more efficient.

Another way to look at the data is to convert the natural gas energy usage into kWh as a common basis, as shown in Figure 7.

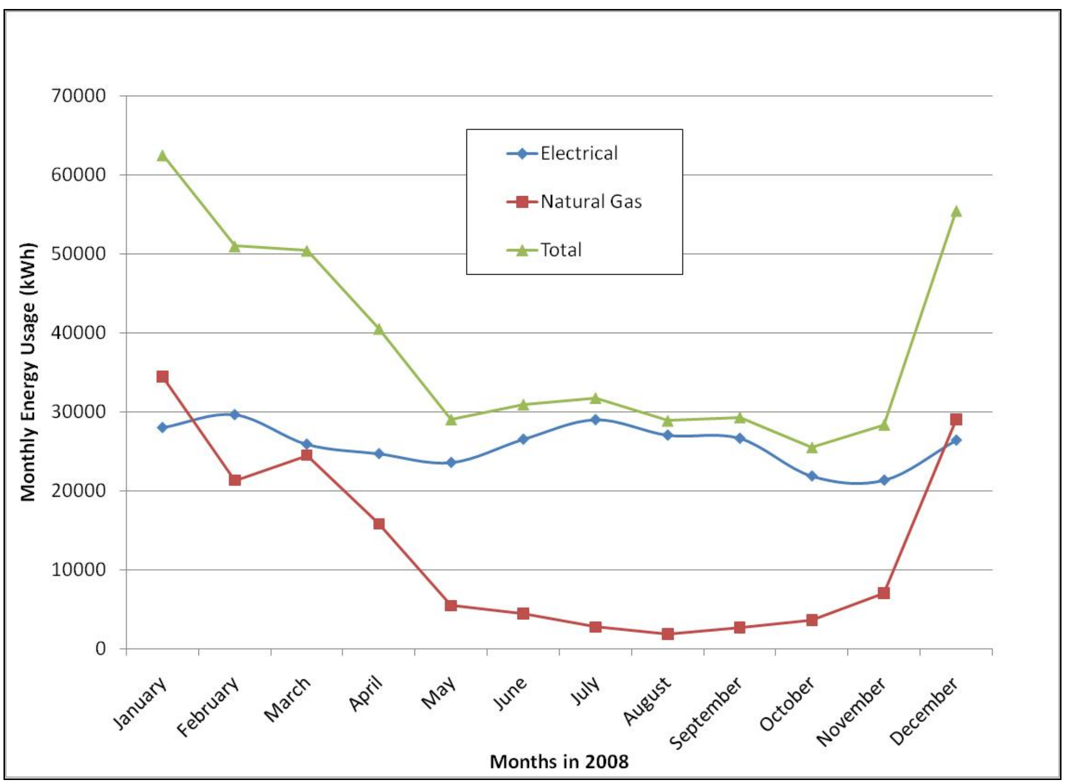

Figure 7: Monthly energy totals for both electricity and natural gas (expressed in kWh).

As shown above, energy use in the summer months is far less than the usage in the winter. The total monthly energy usage closely follows the electrical usage from May until October. As the temperature gets colder, the natural gas usage rivals the electrical usage, causing the totals to more than double in January and December.

During the winter months, the heat pumps will be operating in heating mode. In the Seattle area and much of the Pacific Northwest, the winter outdoor mean temperatures are much further away from the building set point than the mean outdoor temperatures in the summer. For example: winter: 68°F indoor temperature versus 40°F outdoor temperature, for a difference of 28°F; summer: 72°F indoor temperature and 77°F afternoon temperature, for a difference of 5°F. Heat pumps are far more efficient when they are operating within a small temperature differential. This picture holds well for an air source heat pump. In the present application, it is the temperature of the water in the building water loop relative to the indoor temperature that determines the efficiency of each individual heat pump. If the water is maintained between about 60-65°F (winter) and 80-85°F (summer afternoon), the individual heat pumps should be operating near best efficiency. However, a compromised building envelope and a poorly tuned and maintained heat pump system will be responsible for large energy consumption.

Determining the end use of utility expenditures can be a valuable tool in identifying where additional conservation efforts can be most effective. Figure 8 shows an estimate of the total utility cost allocated to the end use at the Cadillac Hotel. Utility costs are calculated based on \$0.059 per kWh for electricity and \$1.41 per therm for natural gas.

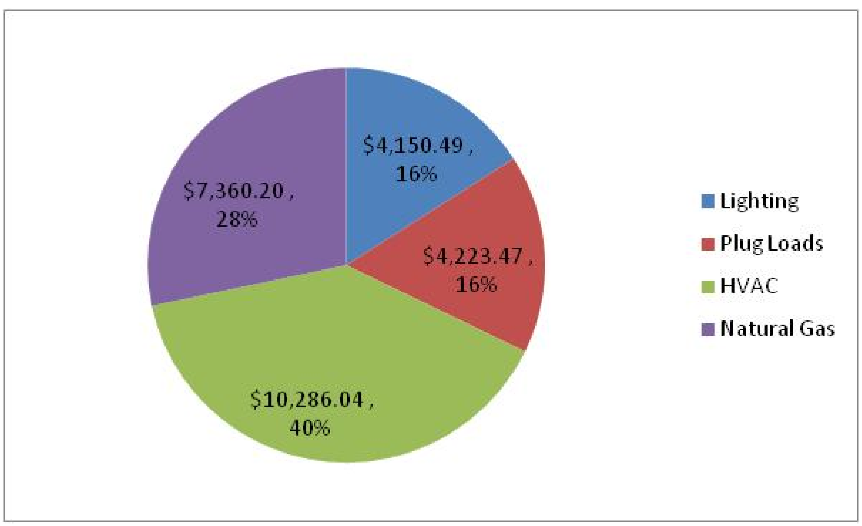

Figure 8: Yearly utility end use cost estimate.

As shown in Figure 8, more than 25% of the yearly energy cost is associated with natural gas usage, and almost 70% of the yearly energy cost is associated with building space conditioning in general.

The Cadillac Hotel already employs many high efficiency plug loads. There is currently a push to upgrade all of the personal computers to laptops (Wagner, 2009). However, since plug loads and lighting make up a relatively small portion of the yearly energy use, it appears it would be more cost effective to focus on reducing the cost associated with heating and cooling. This can be accomplished by both tuning the heat pump system to operate more efficiently as explained above and making thermal improvements to the building envelope.

In Figure 8 both the plug loads and the lighting yearly energy consumption have been estimated by counting the lights and each plug load. Since the lighting and plug loads are not operating at their full rated capacity 24 hours a day, some estimate must be made to the load schedule of each device. If the device is a copy machine, printer, microwave, or any other appliance that has a variable power draw throughout the day, then some estimate is made to the length of time that the device is operating in working mode, sleep, standby, etc. As an example, the methodology for a copy machine is explained below to clarify the process.

For example, there is a Canon Image Runner 5070 in the copy room on the third floor. When it is running at full capacity the power draw is 1354 watts. However, this copy machine does not run at full capacity throughout the entire day. Instead, the machine runs part time at full capacity, standby mode, suspend mode, and sometimes it is turned off. The environmental consulting firm Arthur D. Little Inc. has compiled estimates on copy machine power draw and number of hours per year that the copy machine is run in each particular mode based on the size of the copy machine, as shown in Table 1. The information displayed in Table 1 is based on a copy machine that has a copying capacity between 31 and 69 pages per minute. The Canon Image Runner 5070 has a copying capacity of 50 pages per minute (imageRUNNER 5070, 2009).

Table 1: Power draw and usage data for a generic copy machine with a copying capacity of 50 pages per minute (Roth, Goldstein, & Kleinman, 2002).

Copy Machine Mode	Active	Standby	Suspend	Off
Power Draw (watts)	1354	396	68	0.6
# of hours per year (hrs)	313	1408	1408	5631
Energy consumed per year (kWh)	423.8	557.6	95.7	3.4

The daily energy consumption for this copy machine is calculated as follows:

1. Add up the 4^{th} row to obtain energy per year:

   423.8 + 557.6 + 95.7 + 3.4 = 1080.5 kwh/yr

2. Divide this number by 365 to obtain energy consumption per day:

   [1080.5 kwh/yr] / [365 days/yr] = 2.96 kwh/day

The daily energy consumption of other like machines (computers, printers, fax machines, etc.) are calculated in a similar matter.

For components like lights that are either on or off, an estimate is made to the number of hours that the component is on. For example, the lights on the basement and first floors are estimated to be on 10 hours per day and seven days per week, which is one hour before the museum opens and one hour after the museum closes. The daily electrical consumption associated with lighting and plug loads is estimated to be 384 kWh/day.

It is obviously a difficult task to estimate non-HVAC daily energy consumption from plug loads and lighting. It is both time consuming and cost intensive to monitor each piece of equipment throughout the year. In order to triangulate on this estimate, the amount of yearly energy consumed by the HVAC system alone is estimated separately. Shown in Figure 9 is fifteen minute load data for the Cadillac Hotel from August 2^{nd} through August 5^{th} 2008 and from January 12^{th} through January 15^{th} 2008.

As expected, the electrical load oscillates throughout the day with peaks in the afternoon and troughs in the middle of the night for both the summer and winter months. The first 48 hours of each time stretch are Saturday and Sunday, while second 48 hours are Monday and Tuesday. As expected, the electrical consumption during the middle of the day is larger in the summer than in the winter. It is also expected that the electrical load will be greater during the weekdays than the weekend, since the park offices are closed on the weekends. The last important point to make is that the troughs for both months are almost identical. It is assumed that most of the lights are turned off at night and the plug loads are either turned off or are operating in “power save” mode. The building power draw at the trough can be associated with the HVAC system mainly. The average low power draw (associated with the daily trough) for the month of August is 19.5 kW and the average low power draw for the month of January is about 18 kW. The average low power draw for both the winter and the summer months is about 18.75 kW. Multiplying this number by 24 hours in a day and 365 days per year gives a yearly HVAC energy consumption of 164,250 kWh per year. Note that this energy consumption is representative of HVAC electrical alone -- that is, the air handling units and heat pump compressors.

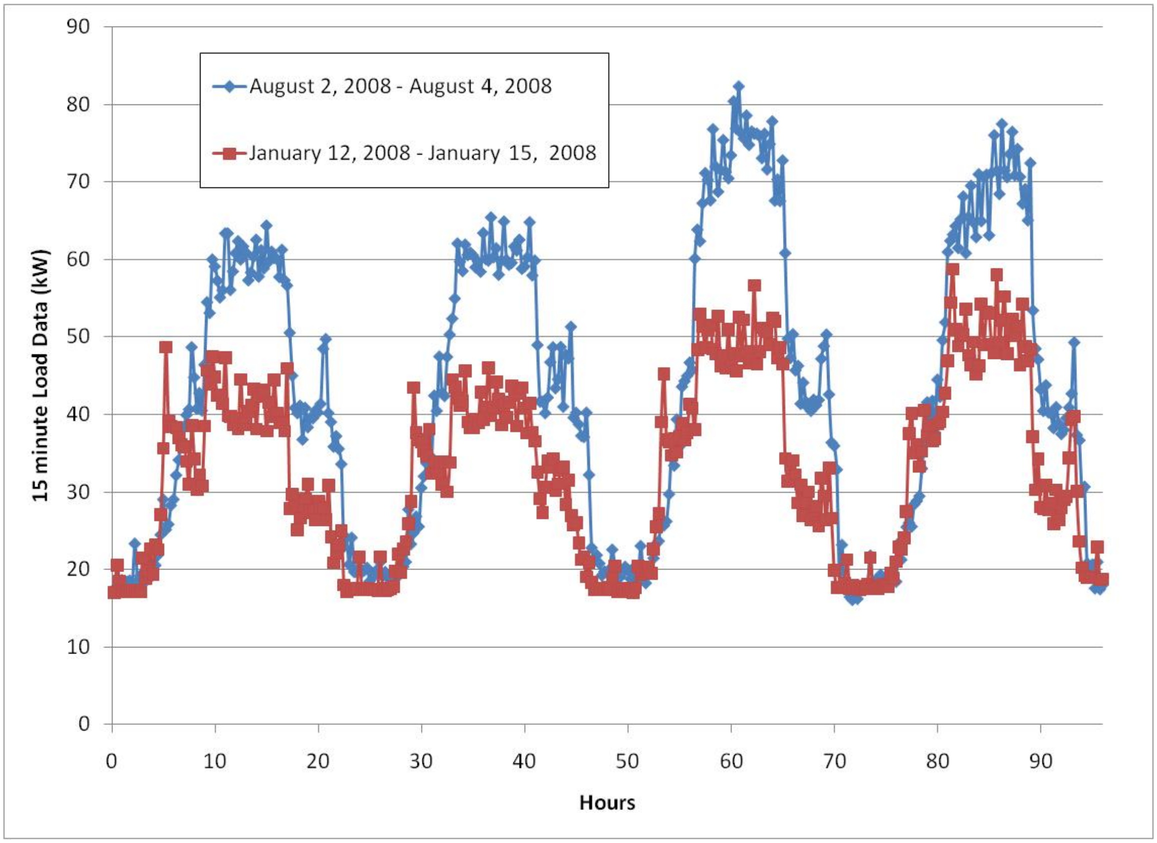

Figure 9: Fifteen minute load data between 8/2/2008 and 8/5/2008.

The measured yearly electrical consumption in 2008 was 311,000 kWh. The yearly energy consumption devoted to lighting and plug loads is then calculated as the difference of these two numbers, or 146,750 kWh per year. The yearly energy consumption for lighting and plug loads calculated by counting devices and estimating the amount of time each device is turned on as detailed above is found to be 139,978 kWh per year. The difference between these two estimates is only about 4.6%. Due to the dynamic nature of a building's electrical system and difficulty in calculating actual yearly consumption, it is thought that these estimates are sufficiently close to each other, and are thus validated.

The detailed breakdown of each light and plug load is displayed in Appendix A.

<u>Energy Usage at Cadillac Hotel – Building Load Coefficient</u>

The total heat transmission from the entire above ground portion of the building can be characterized by a building load coefficient, BLC. The BLC is an important parameter to classify when a building is being considered for an energy conservation project. The BLC describes how power consumption increases when the outdoor air temperature rises above or falls below a certain set point, and is expressed in units of power per degree of temperature (e.g. W/K, W/°F, etc.). Due to the uncertainty in measuring air infiltration loads in the summer, the BLC is often estimated using data from the heating season (Krarti, 2000).

The BLC can be directly calculated with detailed knowledge of the building envelope, or can be calculated indirectly with actual utility and weather data. In order to calculate the BLC directly, detailed architectural drawings are consulted. Once the major features of the building are identified, a thorough walkthrough is completed. Throughout the walkthrough the drawings are verified and notes are taken pertaining to the general health of the building components, inhabitant behavior, and general building comfort level. The BLC can be calculated indirectly by plotting actual utility energy use against outdoor air temperature. The slope of the regression line will be the BLC. Since there are many complex parameters that affect a building's performance, it is wise to calculate the BLC both ways as a sort of validity check.

BLC Calculated from Building Envelope

The building load coefficient can be calculated directly from Equation 1 (Krarti, 2000).

$$\mathbf{BLC = \sum A_i U_{Ti} + m_{inf} C_{pa}} \qquad \text{Equation 1}$$

Where:

- BLC is the building load coefficient: (W/K or W/°F)
- N_E is the number of different building elements i in the summation
- A_i is the area associated with each element: (m^2 or ft^2)
- U_{Ti} is the total overall heat transfer coefficient, or U-factor, associated with each element: (W/m^2-K or Btu/h-°F-ft^2)
- m_{inf} is the mass flow rate of infiltration air: (kg/s)
- C_{pa} is the specific heat of air, which is assumed to be constant at 1.005 kJ/kg-K.

Although the number of building elements and their associated areas can easily be counted and measured, the total heat transfer coefficient, U_T, and the mass flow rate of infiltration air, $\dot{m}_{inf}$, are more complicated. The basic concepts behind these two calculations are discussed below.

The envelope of the building includes everything that separates the building from the outside environment. The envelope includes windows, doors, walls, and the roof of the building. Improvements made to the building envelope can have a significant effect on its overall energy efficiency. This is especially the case with the Cadillac Hotel, because it is so dominated by heating and cooling loads. The envelope of a building loses and gains heat mainly by transfer through the envelope, solar and equipment gains, and air infiltration.

In above ground buildings, heat transfer is typically considered to be one-dimensional and is dominated by both conduction and convection. The convective losses are a result of the conditions of the air both inside and outside of the building and usually do not change very much throughout the year. For the purposes of this study, the convective conditions indoors are calculated assuming stagnant air at 65°F. The outdoor convective conditions are assumed to be associated with a 7.5 mph wind and 80°F for summer conditions and a 15 mph wind and 30°F for winter conditions, according to the ASHRAE Handbook (ASHRAE, 2005).

Although convection at the wall interface should not be neglected, the overall resistance to heat transfer from convection is generally quite small in comparison to the conductive resistance through the wall. The general equation for one dimensional heat transfer through a wall is given by Equation 2.

$$\mathbf{q = A(T_i - T_o)/R_T} \qquad \text{Equation 2}$$

where:

- q is the rate of heat transfer through the wall: (W or Btu/h)
- A is the area of the wall: (m^2 or ft^2)
- T_i is the indoor air temperature: (K or °F)
- T_o is the outdoor air temperature: (K or °F)
- R_T is the total resistance to heat transfer: (m^2-K/W or ft^2-°F-h/Btu)

It is the goal of any building designer to make R_T as large as possible; thus, reducing heat transfer from the inside out during the winter and from the outside in during the summer. The conductive resistance of a building element is defined as the thickness of the element divided by its thermal conductivity as shown in Equation 3, and the convective resistance is the reciprocal of the convective heat transfer coefficient as shown in Equation 4. The total resistance, R_T, for one dimensional heat transfer is sum of each of the resistances. For example, a building with convection on the inside and outside and a wall composed of one material will have a total resistance to heat transfer as shown in Equation 5.

$$\mathbf{R_{conduction} = L/k} \qquad \text{Equation 3}$$

$$\mathbf{R_{convection} = 1/h} \qquad \text{Equation 4}$$

$$\mathbf{R_T = R_{conduction} + R_{convection\ inside} + R_{convection\ outside}} \qquad \text{Equation 5}$$

where:

- L is the thickness of the material: (m or ft)
- k is the thermal conductance of the material: (W/m-K or Btu/h-ft-°F)
- h is the convective heat transfer coefficient: (W/m^2-K or Btu/h-ft^2-°F)

Often walls are composed of materials with greatly varying thermal conductivities. These walls must be treated as parallel thermal circuits. For example, consider a wall composed of cinderblocks on the exterior with wooden frames filled with insulation on the interior. All of the heat will pass through the exterior of the structure at the same rate since it is composed of a homogeneous material. This is not the case for the interior since the thermal resistances of the wooden studs will be far less than that of the insulation. The overall resistance is calculated by adding these resistances in parallel. This concept is illustrated in Table 2, where the overall heat transfer coefficient, U_T, is calculated assuming parallel heat transfer through both the studs and the space in between the studs. More information on parallel heat transfer in a composite wall and a deeper discussion of heat transfer in general is beyond the scope of this report. The interested reader should consult other sources (ASHRAE, 2005).

As before mentioned, the walls in the Cadillac Hotel are composed of 8 inches of red brick, a vapor barrier, wood studs, and gypsum sheet rock. As a rule of thumb for walls constructed with studs built 16 inches on center, the insulated cavity (or un-insulated cavity as is the case with the current construction) occupies approximately 75% of the transmission area; whereas, the studs, plates, sills and headers occupy the remaining area (ASHRAE, 2005). The thermal resistance for one wall in the Cadillac building is summarized as shown in Table 2. Note that the thermal resistance of the vapor barrier has been neglected since it is quite small. Also the studs, plates, sills, and headers are assumed to be constructed of solid wood and are grouped together.

Table 2: The thermal resistance breakdown for a generic wall in the Cadillac Hotel.

	Thermal Resistance (m^2/W-K)	
Building Element	Un-insulated Cavity	Studs, plates, sills, and headers
Outside Air with 15 mph wind	0.03	0.03
8" red brick	0.4064	0.4064
Un-insulated air cavity	0.28	0
stud	0	0.635
gypsum board	0.079	0.079
air	0.12	0.12
sum	0.9154	1.2704

The overall heat transfer coefficient, U_T, (or sometimes called the U-factor) is defined as the reciprocal of the total resistance for each path. Assuming area weighting factors as stated above, the total overall heat transfer coefficient for the above wall is given by Equation 6. The thermal resistances for each component have been estimated from the ASHRAE Handbook (ASHRAE, 2005).

$$\mathbf{U_T = 0.75U_1 + 0.25U_2 = 1.016\ w/m^2\text{-}K} \qquad \text{Equation 6}$$

The overall heat transfer coefficients for windows are calculated much in the same way as above. For double and triple glazed windows, the center of glass U-factor differs from the window's perimeter. There is an additional U factor associated with the framing style and material. The total U-factor for each style of window found in the Cadillac Hotel is shown below in Table 3. Note that this U-factor is calculated by including effects of the type of framing, edge effects, and whether or not the window is operable or fixed. The U-factors have been obtained from the ASHRAE Handbook (ASHRAE, 2005). As shown in Table 3, upgrading single glazed windows to double glazed nearly reduces the U-factor by half, which means that only half of the original heat lost through the single glazed will be lost through the double glazed windows. This could offer a significant energy improvement opportunity for a building with single glazed windows and a large window to wall area ratio.

Table 3: A summary of U factors for windows found in the Cadillac Hotel.

Type of Window	fixed or operable	air gap (in)	Type of Frame	U (W/m^2-K)
single glazed windows	fixed	NA	wood	5.55
double glazed windows	operable	0.25	wood	3.13
double glazed windows	operable	0.5	wood	2.87

The roofing contractor was not able to be contacted directly; however, it is assumed that the roof is built up with R-15 insulation. It is estimated that a built up roof with R-15 insulation has an overall heat transfer coefficient equal to 0.358 W/m^2-K (Vandalay, 2004).

Inherently, horizontal surfaces have larger losses than the same surface oriented vertically; thus, skylights have larger losses than vertical windows. The overall U-factor for double glazed skylights with a ½ inch air gap, and no thermal break is 4.59 W/m^2-K (ASHRAE, 2005).

Shown below in Table 4 is a breakdown of each of the U-factors associated with the components of the Cadillac Hotel. The U-factors are then multiplied by the area associated with each particular component and these values are summed to give a BLC based on heat transfer through the building envelope alone. Once some estimation of the mass flow rate of infiltration air is calculated, the BLC can be calculated from Equation 1.

Table 4: A breakdown of the U factors and surface areas used in calculating the BLC.

Part of Building Envelope	Component	Surface Area (m^2)	U (W/m^2-K)	U*A (W/K)
South Wall	wall	313.73	1.02	318.79
	single glazed windows	27.72	5.55	153.85
	double glazed windows (1/2 inch gap)	10.88	2.87	31.22
	double glazed windows (1/4 inch gap)	40.27	3.13	126.06
West Wall	wall	205.13	1.02	208.43
	single glazed windows	NA	NA	0.00
	double glazed windows (1/2 inch gap)	NA	NA	0.00
	double glazed windows (1/4 inch gap)	13.70	3.13	42.89
East Wall	wall	156.87	1.02	159.40
	single glazed windows	NA	NA	0.00
	double glazed windows (1/2 inch gap)	38.27	2.87	109.83
	double glazed windows (1/4 inch gap)	23.69	3.13	74.15
Roof	Built up roof	586.42	0.36	209.78
	Skylights	27.87	4.59	127.93
BLC_envelope (W/K)				1562.32

Air infiltration or exfiltration is when air flows in or out of the building through leaks in the envelope. This flow of air is rather unregulated and is independent of any mechanical means of moving air through a building. Because the infiltration air is unconditioned, significant building energy can be consumed trying to heat or cool this air to the building set point. According to some studies, air infiltration can account for as much as 40% of the total building heating load (Tuluca & Winter, 1997). In larger buildings, air infiltration is generally not much of an issue since the total building volume divided by the surface area is typically small and more robust mechanical systems generally keep the building pressure slightly larger than the outdoor pressure (Krarti, 2000). For residential and low-rise buildings, air infiltration can be significant, especially if there is structural damage to the building envelope, as is the case for the Cadillac Hotel (Krarti, 2000).

It is difficult to estimate the actual air infiltration rate of a building. There are two main techniques used to estimate air infiltration in a building: pressure difference techniques and tracer gas techniques. Pressure differences between the indoors and the outdoors of a building can be used to estimate volumetric air flow rate through the building envelope. This technique requires a significant amount of instrumentation. All outside doors and windows must be closed, and mechanical ventilation system must be disabled (Krarti, 2000). Since the Cadillac Hotel operates as a museum nearly all year long, this option does not seem feasible because it would disrupt daily operations.

Injecting and monitoring the concentration of a safe tracer gas can also be used to estimate the exchange rate of air through the building envelope. Similar to pressure techniques, tracer gas measurement requires a fair bit of time and sophisticated equipment; both of which are beyond the scope of this study. Instead, the air infiltration will be estimated from historical data.

Briggs et al. analyzed 20 different buildings around the country ranging in both height and year of construction (Emmerich, Persily, & VanBronkhorst, 1995). In order to obtain a good estimate for a 3 story building like the Cadillac Hotel, buildings with more than 4 stories and less than 2 stories have been omitted from the data, which is displayed below in Table 5. The data in Table 5 is obtained from buildings from many different climatic zones and years of construction. It is thought to be reasonable to average the air change rates shown in the last column as a crude estimate for the air change rate within the Cadillac Hotel. The air change rate is often expressed as the ACH, or air changes per hour. Note that the values in Table 5 were obtained with a disabled building ventilation system.

Table 5: Air Exchange rates for various buildings around the US.

Building	# of Floors	Year Built	Location	Air Change Rate w/ Fans off (h^{-1})
1	3	1920	Cleveland, OH	1
2	2	1970	Washington, DC	0.33
3	2	1969	Madison, WI	0.28
4	2	1953	Lake Charles, LA	0.42
5	4	1925	Des Moines, IA	0.65
6	2	1967	Las Vegas, NV	0.18
7	3	1967	Salt Lake City, UT	0.21
8	2	1986	Raleigh, NC	0.62
9	2	1986	Phoenix, AZ	0.58
10	2	1986	Pittsburgh, PA	0.53

The ACH can then be multiplied by the volume of the building and the density of air to obtain an estimate for the mass flow rate of infiltration air as shown in Equation 7 (Emmerich, Persily, & VanBronkhorst, 1995). The average ACH determined from Table 5 is 0.48 air changes per hour.

$$\mathbf{m_{inf} = V(ACH)\rho_{air}} \qquad \text{Equation 7}$$

where:

- V is the volume of the building: (m^3 or ft^3)
- ACH is the air change rate in changes per hour: (h^{-1})
- ρ_{air} is the density of air, which is assumed to be constant at STP.

The volume of the Cadillac Hotel is determined to be approximately 7265 m^3 and the density of air at standard conditions is 1.17 kg/m^3; thus, the mass flow rate of infiltration air is determined to be:

$$\mathbf{m_{air} = 7265m^3 \times 0.48\ changes/hr \times 1.17\ kg/m^3 \times 1\ hr/3600s = 1.136\ kg/s}$$

For a nominal specific heat of 1.005 kJ/kg-K, the second term in Equation 1 is determined to be:

$$\mathbf{m_{inf}C_{pa} = 1.136\ kg/s \times 1.005\ kJ/kg\text{-}K = 1142\ w/K}$$

This number can now be added to the BLC obtained from the above heat transfer analysis to give a BLC of 2704 W/K, or 1502 W/°F. This means that for every °F that the outdoor air temperature drops below the building set point temperature, 1502 additional watts must be supplied to the building to maintain this temperature.

BLC Calculated from Utility and Weather Data

In order to obtain the BLC from indirect estimation, historical utility data must be plotted against outdoor air temperature. The outdoor air temperature should be averaged over the same period at which the utility data is obtained, and all loads unassociated with the heating or cooling load should be subtracted. Obviously, the daily natural gas usage should be included in the heating load. Since the heating in the Cadillac building is accomplished via a multi-zone water-source heat pump system, the electrical usage associated with the HVAC system should also be included. As determined above, the daily electrical consumption from non-HVAC equipment is estimated to be 384 kWh per day. The average daily electrical consumption for each month in 2008 is displayed in Table 6. The daily electrical consumption associated with lighting and plug loads is then subtracted from this value, and the average daily energy consumption associated with natural gas is added to this column to give an average daily energy consumption associated with heating. The mean outdoor air temperature for each month in 2008 is shown in the final column. The last two columns of Table 6 are then plotted against each other shown in Figure 9. The slope of the regression line is then the corresponding BLC. This number can then be divided by 24 hours in a day to obtain a BLC in W/°F.

As shown in Figure 10, the BLC estimated from utility and weather data is determined to be 36.141 kWh/°F. Dividing this number by 24 hours/day and multiplying by 1000 W/kW gives a BLC of 1505.4 W/°F, which is nearly identical to the BLC obtained from direct calculation.

There are a couple of points to be made for the above analysis. First, the correlation shown in Figure 10 should be representative of more than one year of data. This was not feasible due to a difficulty in obtaining natural gas usage data for years other than 2008. In addition to this, an R^2 value of 0.668 is fairly low. More data and a higher R^2 value would give this analysis more statistical validity.

Table 6: Average Daily Energy Consumption and Outdoor Air Temperature for 2008.

Months of year	Average Daily Electrical Consumption (kWh)	Average Daily Electrical Consumption, non-HVAC loads subtracted (kWh)	Average Daily Natural Gas Consumption (therms)	Average Daily Natural Gas Consumption (kWh)	Average Daily Energy Consumption for heating (kWh)	Average Daily Outdoor Air Temperature (°F)
January	905	521	38	1103	1624	39
February	928	544	26	755	1299	44
March	894	510	27	784	1294	44
April	852	469	18	523	992	47
May	814	430	6	174	605	56
June	830	447	5	145	592	59
July	1001	618	3	87	705	65
August	903	519	2	58	577	66
September	861	477	3	87	564	61
October	754	371	4	116	487	52
November	712	329	8	232	561	49
December	778	394	32	929	1323	37

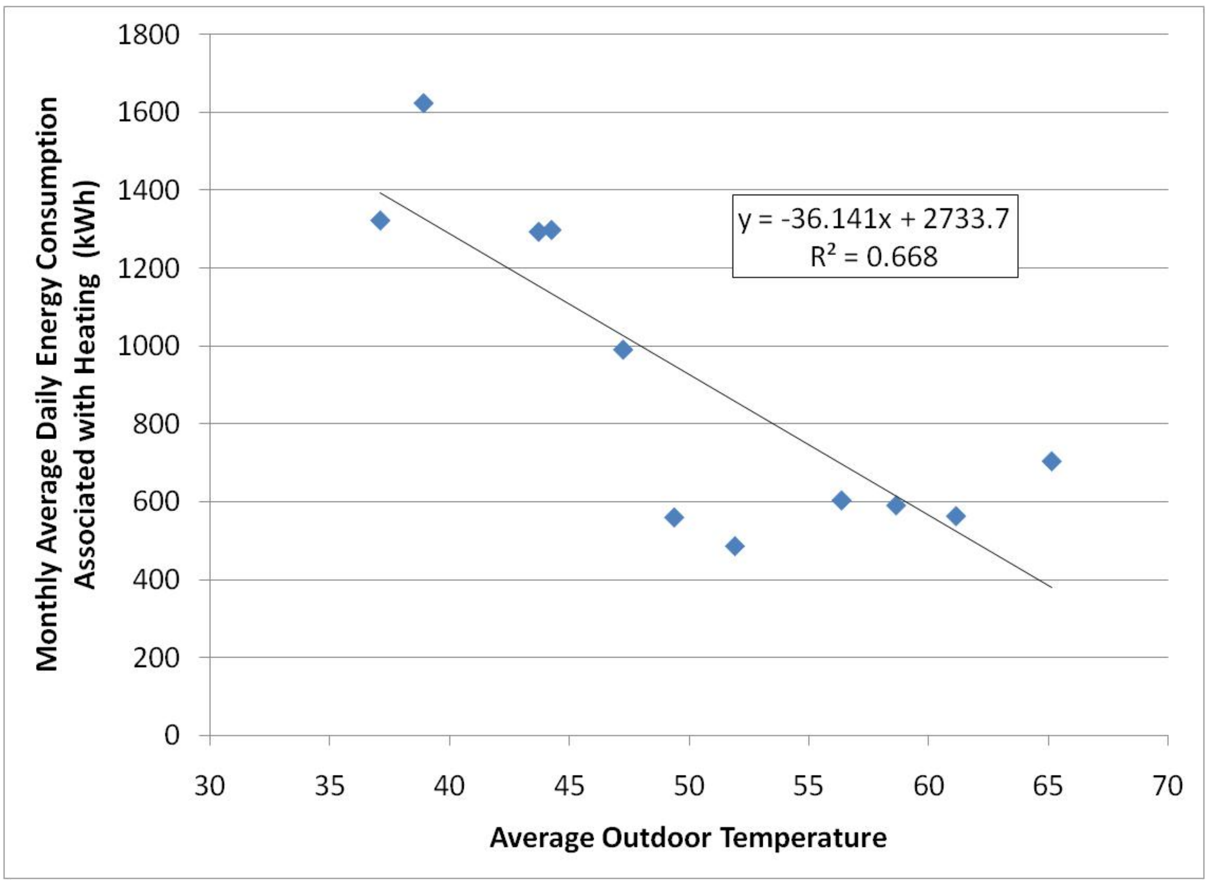

Figure 10: BLC based on Total Heating Energy Consumption

There is uncertainty in the above analysis. A more detailed and knowledgeable estimation of U-factors and air infiltration, coupled with a few more years of utility data would make the results more concrete. This being said, the important trends have been identified by different methods, the results are consistent, and it is fairly obvious that efficiency improvements should be made to the building envelope.

Energy Usage at Cadillac Hotel – Degree-Day Method and COPs

The Variable Base Degree-Day Method provides an estimation of the heating and cooling loads of a building based on the balance temperature, T_b, of the building. The balance temperature of the building is the temperature at which the building requires no additional heating or cooling input to maintain comfort. The balance temperature includes internal heat gains from equipment, infiltration, and the sun. Nominally, the balance temperature for most buildings in the United States is 65°F. The total heating load for a building over an entire heating season can be characterized by a parameter called heating degree-days, or DD_H, which is defined in Equation 8. In the same manner, cooling degree days, DD_C, is total cooling load for a building over an entire cooling season, as defined in Equation 9 (ASHRAE, 2005).

$$\mathbf{DD_H(T_b) = \sum(T_b - T_{o,i})} \qquad \text{Equation 8}$$

$$\mathbf{DD_C(T_b) = \sum(T_{o,i} - T_b)} \qquad \text{Equation 9}$$

where:

- DD_H is the heating degree-days, which is a function of balance temperature, T_b: (°F-days/year).
- DD_C is the cooling degree-days, which is a function of balance temperature, T_b: (°F-days/year).
- N_H is the number of heating days in the season I in the summation.
- T_b is the balance temperature of the building: (°F)
- $T_{o,i}$ is the mean outdoor air temperature for each day (°F).

An important point to make about Equations 8 and 9 is that negative values are not added to the summation. If the value is negative in Equation 8, then the building is in cooling mode and the load should not be associated with heating, and vice versa. Assuming a balance point of 65 °F and summing over the mean daily outdoor Seattle air temperatures for 2008, DD_H is determined to be 5062 °F-day/year and DD_C is 193 °F-days/year. Since the above calculation is only based on one year of data, it is decided to use an average value of for heating degree-days found in the literature, where DD_H = 4727 °F-days/year (Krarti, 2000). Unlike heating degree days, nominal values for DD_C are not reported in the literature. Due to building thermal hysteresis, mild weather, and large variability in air infiltration (e.g. open windows), the degree-days method is not quite as accurate in calculating loads associated with cooling (Krarti, 2000).

Instead of a nominal value for DD_C, the value from 2008 will be used in the forthcoming analysis. The yearly energy consumption for heating and cooling can then be estimated by Equations 10 and 11.

$$E_{heating} = BLC \times DD_H \times (24\ hr/day) \times (1\ kw/1000w)/\eta \qquad \text{Equation 10}$$

$$E_{cooling} = BLC \times DD_C \times (24\ hr/day) \times (1\ kw/1000w)/\eta \qquad \text{Equation 11}$$

where:

- $E_{heating}$ is the yearly energy consumption devoted to space heating (kWh)
- $E_{cooling}$ is the yearly energy consumption devoted to space cooling (kWh)
- η is the efficiency of the heating or cooling system
- DD_H, DD_C, and BLC are defined as above

The average building envelope retrofit generally doesn't alter the balance temperature significantly (Krarti, 2000). If there are no changes to the heating and cooling system (η remains constant) and the heating degree-days remain constant, the expected energy savings after a retrofit can be calculated by Equation 12 (Krarti, 2000).

$$\Delta E_{heating} = (BLC_E - BLC_R)(DD_H/\eta) \times (24\ hr/day) \times (1kw/1000w) \qquad \text{Equation 12}$$

where:

- BLC_E is the existing Building Load Coefficient: (W/°F).
- BLC_R is the Building Load Coefficient after the retrofit: (W/°F).
- $\Delta E_{heating}$ is the expected change in yearly energy consumption: (kWh)

In order to compare one building retrofit against another, the difference in the building load coefficient before and after the retrofit can be set equal to the difference in the U_T*A values before and after the retrofit. For example if just the windows in a building are to be replaced, $(BLC_E - BLC_R) = (U_T*A_{window,E} - U_T*A_{window,R})$ (Krarti, 2000).

In order to make any projections about energy savings, some estimate must be made for the efficiency, η, of the heating or cooling system. If the heating or cooling system in the Cadillac Hotel were more traditional, this would be an easy calculation. For example, if the building were heated by electric baseboard heaters, η could be assumed equal to about 1. If the heat pump system was a traditional air-source heat pump that exchanges heat with the outside air only, and electricity is the only input; then, nominal values for COP and EER could be chosen (for air based heat pumps COP ≈ 3 and EER ≈ 10) (Energy, 2007). The coefficient of performance, or COP, of a heat pump is equal to the amount of heat moved into the conditioned space in Btu/h divided by the amount of electricity supplied to the system in Btu/h. The energy efficiency ratio, or EER, of a heat pump is equal to the amount of heat moved into the conditioned space in Btu/h divided by the amount of electricity supplied to the system in watts. The DOE reports that a nominal water-source heat pump with a boiler and cooling tower as the heat source and sink, has a nominal COP of about 5 (Energy, 2007). A water-source heat pump with a COP equal to 5, is of course a heat pump system that is operating at its highest efficiency. A highly efficient water-source heat pump system does use the boiler and chiller on occasion. However, for the majority of the time it is moving heat around the building due to temperature gradients within the building. As mentioned throughout this report, there are many issues with the existing heat pump system. The most apparent issue is the amount of natural gas that it consumes both in the summer and especially in the winter.

It would be incorrect to use a COP equal to 5 without making some correction to the apparent inefficiencies. Instead, the COP of the heat pump system can be estimated by a more classical approach as shown in Equation 13.

$$\mathbf{COP = Desired\ Energy\ Output\ /\ Required\ Energy\ Input} \qquad \text{Equation 13}$$

If the building is assumed to be in thermal equilibrium with the outside environment, then the daily desired energy output of the system in the heating season will be equal to the building load coefficient multiplied by heating degree-days, or Desired Energy Output = $BLC*DD_H$. The required energy input is simply the daily amount of energy supplied to the heat pump system, which is the total daily electric consumption with the lighting and plug loads subtracted plus the total daily natural gas consumption. In order to account for the cooling loads in the summer, some estimate of cooling degree-days should be added to the heating degree days. The average monthly COP of the heat pump system throughout the year can then be calculated as shown in Equation 14.

$$\mathbf{COP = [BLC \times (DD_{H,avg} + DD_{C,avg})] / [E_{electrical\ for\ HVAC} + E_{nat\ gas}]} \qquad \text{Equation 14}$$

where:

- BLC is the existing building load coefficient: (W/°F).
- $DD_{H,avg}$ is the average daily heating degree-days for each month: (W/°F).
- $DD_{C,avg}$ is the average daily cooling degree-days for each month: (W/°F).

- $E_{electric\ HVAC}$ is the total daily electrical energy consumed averaged over the month minus the electrical for lighting and plug loads: (kWh).
- $E_{nat\ gas}$ is the total daily natural gas energy consumed averaged over the month: (kWh).

The results are shown below in Table 7. Note that the values in column 2 of Table 7 are the same values that were calculated in column 6 of Table 6. The average daily degree-days for heating and cooling are averaged over the entire month.

One interesting point to note is the fact that for much of the year in Seattle the cooling degree-days are equal to zero or are quite small. The BLC used in the above calculations is 1505 W/°F, calculated above. The COP is then calculated as column 7 divided by column 2. Columns 5, 6, and 7 have been converted to kWh by multiplying by 24 hours/day and dividing by 1000 W/kW.

Table 7: Estimation of COP from utility data and the calculated BLC.

Months in 2008	2008 Daily Required energy input for space conditioning (kWh)	DD_H	DD_C	$BLC*DD_H$	$BLC*DD_C$	$BLC*(DD_H+DD_C)$	COP
January	1624.37	26.06	0	939.80	0	939.80	0.58
February	1298.93	20.72	0	747.25	0	747.25	0.58
March	1294.26	21.26	0	766.50	0	766.50	0.59
April	991.56	17.73	0	639.41	0	639.41	0.64
May	604.51	9.06	0.45	326.84	16.28	343.12	0.57
June	591.69	7.53	1.20	271.63	43.27	314.90	0.53
July	704.99	1.35	1.52	48.85	54.67	103.52	0.15
August	577.25	1.71	2.84	61.65	102.35	164.00	0.28
September	564.26	4.10	0.27	147.83	9.62	157.45	0.28
October	487.13	13.06	0	471.06	0	471.06	0.97
November	560.79	15.60	0	562.49	0	562.49	1.00
December	1323.29	27.87	0	1004.94	0	1004.94	0.76

The monthly COP's calculated for the heat pump system are far less than 5. Moreover, the calculated COP's are less than unity and in the summer months these values are quite small. The low estimates of COP in the summer months are most likely due to the uncertainty associated with cooling degree-days as explained above. Since the COP values estimated for the months of July through September are associated with some uncertainty, they have been omitted, and the yearly average COP to be used in Equation 12 is calculated to be 0.69. Note that COP is a measure of efficiency and replaces η in Equation 12.

Energy Usage at Cadillac Hotel – Energy Savings

The ultimate goal of any building conservation measure is to provide a comfortable living environment for the building inhabitants, while reducing building operating costs; subsequently, reducing the environmental costs associated with energy usage. In order to determine the benefits of an energy conservation measure, the energy use savings must be estimated.

As illustrated above in this report, the non-HVAC electrical usage of the Cadillac Hotel makes up a relatively small contribution to the overall building energy consumption. In addition to this, most of the plug loads throughout the building have an energy star rating and much of the lighting in the building is highly efficient with light timers in most of the rooms. Although there are probably some minor improvements to make in this area, energy usage associated with heating and cooling far outweighs any savings that can be accomplished by making improvements to the lighting and plug loads. Although the HVAC system is not operating at optimal performance, it is a very complicated system and should be analyzed and tuned by professionals specializing in this area. This being said, the below economic analysis and recommendations are focused on improvements made to the existing building envelope.

Now that all of the variables have been estimated, the yearly energy savings from any one retrofit can be calculated. For example, consider replacing the single glazed windows on the 1st floor with double glazed windows with a ½ inch air gap. The surface area is the same and the U-factors for single and double glazed windows are 5.55 W/m^2-K and 2.87 W/m^2-K, respectively. The yearly energy savings can then be computed as shown below.

$$\Delta E_{heating} = [(BLC_E - BLC_R) \times DD_H / \eta] \times 24 \text{ hr/day} \times 1 \text{ kw/1000 w}$$

$$\Delta E_{heating} = [U \times (A_E - A_R) \times DD_H / COP] \times 24 \text{ hr/day} \times 1 \text{ kw/1000 w}$$

$$\Delta E_{heating} = [27.7 \text{ m}^2 \times (5.55 - 2.87) \text{ w/m}^2\text{-K} \times (1/0.69)] \times 4727 \text{ °F-days/yr} \\ \times 24 \text{ hr/day} \times 1 \text{ kw/1000 w} \times 1 \text{ K/1.8 °F}$$

$$= 6781 \text{ kwh/yr}$$

The cost of heat must be estimated in order to approximate the simple payback period for the retrofit. Similar to Figure 8, the yearly energy consumption for the HVAC system, lighting, plug loads, and natural gas are compared to each other on a kWh basis in Figure 11.

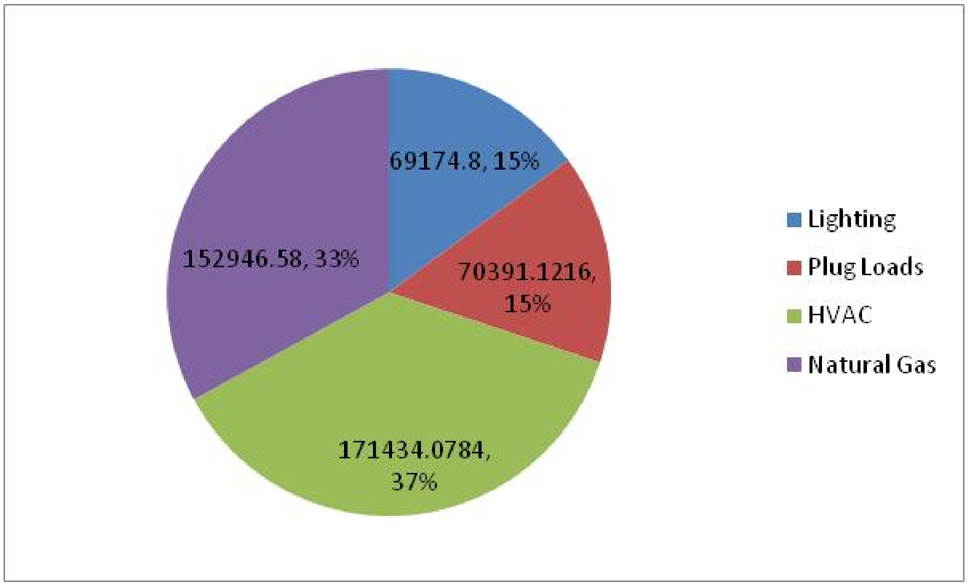

Figure 11: The yearly energy consumption by sector (kWh)

The total yearly consumption for space conditioning will be the HVAC consumption added to the natural gas consumption, or 324,381 kWh. As stated above, the assumed cost of natural gas is $1.41 per therm, or $0.048 per kWh equivalent, and the cost of electricity is $0.059 per kWh. The estimated cost for the heating mix will then be:

[152947 kwh x 0.048 $/kwh + 171434 x 0.059 $/kwh] / 324381 kwh = 0.054 $/kwh

A simple payback period can then be defined as shown in Equation 15.

Payback = [A x CPA] / [DE x COH] Equation 15

where:

- A is the surface area of the envelope improvement: (m^2).
- CPA is the cost per area; ($/$m^2$)
- ΔE is the energy savings per year: (kWh/year).
- COH is the cost of heat: ($/kWh).

The cost for installing double glazed picture windows is estimated to be approximately 305 $/$m^2$ (Means, 2009). The simple payback period for the window upgrade is then:

$$\textbf{Payback} = [27\ m^2 \times 305\ \$/m^2] / [6781\ kwh/yr \times 0.054\ \$/kwh] = 22.4\ yrs$$

This retrofit is obviously not worth it assuming the above assumptions are valid. The most obvious energy savings is thought to be associated with insulating the walls of the Cadillac Hotel. As stated above, the original walls have an estimated U-factor equal to 1.016 W/m^2-K. Installing R-21 blanket insulation to the walls will decrease the U-factor to 0.356 W/m^2-K. The total above ground wall area in the Cadillac Hotel is 675.7 m^2 (the window area has been subtracted). The cost of installing the insulation is estimated to be 9.09 $\$/m^2$ (Means, 2009). The cost associated of installing new sheet rock is estimated to be 15.82 $\$/m^2$ (Means, 2009). The cost associated with painting the new walls is estimated to be 16.15 $\$/m^2$ (Means, 2009). The total cost associated with installing new insulated walls is estimated to be 41.06 $\$/m^2$. The estimated annual energy savings is given as:

$$\Delta E_{heating} = (1/0.69) \times 675.7\ m^2 \times (1.016 - 0.356)\ w/m^2\text{-}K \times 4727\ °F\text{-}days/yr \times 24\ hr/day \times 1\ kw/1000\ w \times 1\ K/1.8\ °F$$

$$= 40736\ kwh/yr$$

The simple payback period is then estimated to be:

$$\textbf{Payback} = [675.7\ m^2 \times 41.06\ \$/m^2] / [40736\ kwh/yr \times 0.054\ \$/kwh] = 12.6\ yrs$$

The potential upgrades illustrated above, do not seem to be particularly cost effective based solely on thermal performance. Keep in mind that the cost associated with tearing out and installing insulation is dominated by the new drywall and painting costs. Also, it should be noted that an investment in insulation may be justified by an increase in thermal comfort within the working space.

A laundry list of potential envelope improvements could be made to the Cadillac Hotel. The above examples are estimates of potential energy savings –
the estimates can be used to guide a building manager to improve the building over its lifetime.

Energy Usage at Cadillac Hotel – Recommendations

The energy consumption of the Cadillac Hotel has been analyzed. The end use of energy has been examined, and specific areas of improvement have been discussed. The following recommendations are based on this study, historical utility trends, weather data, building envelope materials, and 2009 construction cost data.

HVAC System

The HVAC system is not operating at its optimal level. Without going onto too much detail about specifics the following recommendations are mostly based on personal observations and common sense.

1. The thermostats for multiple heat pumps conditioning the same room should be tied together at the very least. If possible, sections of the building should be tied together. For example, one could have a thermostat that controls the east and west side of the building, respectively. There is no need to have heat pumps that are competing against each other; thus wasting energy.

2. The thermostat for any one heat pump should not be located in a different room than the one that it is conditioning. This could lead to someone turning the temperature up or down in a room that they do not even inhabit. This has the potential to greatly increase energy use in a similar manner to the increase in energy associated with recommendation 1.

3. Historically, water-source closed loop heat pump systems are installed in large buildings located in very mild climates. First of all, the Cadillac Hotel can hardly be considered large. A large building generally has a core zone and a perimeter; whereas the layout of the Cadillac Hotel is such that most of it is actually a perimeter. Of course, there is still potential to move solar heat around; however, there is sufficient shade on the building (especially to the south and west) that there is only a small portion of the year when this feature can be utilized. Secondly, as shown in the above utility and weather data, there is a large portion of the year when the building is in heating mode only. A water-source heat pump system exchanges heat with the other heat pumps in the building and the boiler on the roof. If all of the heat pumps are in heating mode, the system is only exchanging heat with the boiler, thus reducing the system efficiency and greatly increasing overall energy consumption.

4. The heat pump system should be well maintained by the building operator. In this regard, we found several defects: 1) the preheater on air handler #1 has not worked since the hard freeze of winter 2008-09 and leads to degraded heating of the 3rd floor library area on cold days, 2) the heat pump unit for the west section of the 1st floor has not worked for some time and has been replaced by 110 volt floor resistance heaters, and 3) some upper floor windows are opened to provide cool air for offices on winter heating days.

Building Envelope

The thermal integrity of the Cadillac Hotel was significantly compromised when the insulation was removed from the walls during reconstruction following the

Nisqually earthquake. The total energy savings and overall thermal comfort of the building could be greatly improved.

For example, during several building walkthroughs, it was noticed that there were at least 10 electrical space heaters near the perimeter of the building on the 2^{nd} and 3^{rd} floors. A properly operating HVAC system coupled with an insulated building perimeter would make this auxiliary heat unnecessary.

Although the simple payback periods for the proposed retrofits in the Energy Savings section appear to be quite long, adding only R-21 insulation to the walls will reduce the total annual energy consumption of the Cadillac Hotel by about 40,000 kWh per year. The annual energy consumption including natural gas is about 470,000 kWh. Adding insulation would reduce the total building energy consumption by about 8.5%, which is not insignificant.

In addition to insulation, the above analysis indicates that over half of the building load coefficient is dominated by infiltration losses. Although the above analysis is based on simple assumptions, the order of magnitude is probably not far off. There are several low cost methods to help reduce energy consumption:

1. Caulking can be applied around plumbing and electrical penetrations as well as doors and windows.
2. In addition to caulking, weather stripping can also be applied to doors and windows to help seal them off from the outside environment.
3. If the sheet rock walls are torn out and insulation is installed, new high density vapor/air barriers can be applied to the inside of the brick; however this option is expensive (Krarti, 2000).

The above retrofits would most definitely improve the energy efficiency of the building. However, the original rationale for removing the building insulation was to effectively preserve the structural integrity of the outer brick by letting the building heat help dry it. If this is still of great concern, then no action should be taken. It is highly recommended that other methods to dry the brick are researched. Although it is a goal of the building owners to preserve the historical structure, it may be in the best interest of the inhabitants of the building that sections of the brick that have experienced significant water damage are replaced or weather sealed. Insulation could then be safely placed within the walls.

Selection and Specifics of the PV System for the Cadillac Hotel

Over the last half of year 2009, the partnership of Klondike NHP, Historic Seattle Inc., Seattle City Light, Bonneville Environmental Foundation, and the University of Washington designed and specified the solar PV system to be installed on the roof of the Cadillac Hotel. A grid-tied system rated at 3.36 kw was specified. The solar PV panels and inverter were purchased and received December 22, 2009. Table 8 lists the equipment and labor for installation of the system. The UW, using UNPEPP funding of $7500, has purchased 14 of the 16 REC solar PV panels for use by Klondike NHP. The balance of funding for the Table 8 costs is being provided by Klondike NHP and Seattle City Light, $7500 each. Additionally, Seattle City Light is providing monitoring of the system electrical performance (hardware and service) and covering Bonneville Environmental Foundation management fees.

Table 8. Solar PV system components and estimated costs

Component	Cost
16 REC 210AE solar PV panels	$8600
SMA Sunnyboy 3000us inverter	$1800
SMA RS-485 Comm Card	$100
Racking	$3500
Balance of system	$2000
Installation time	$3000
Structural engineering	$2500
Building permit	$1000
Total	$22500

Each of the REC panels is rated at 210 watts DC output (for AM1.5 sunlight of 1000 w/m^2, panel temperature of 25°C). The silicon is manufactured in Moses Lake, WA, while the panels are manufactured in Norway. The REC panels are rated for high static loading, about 3 times that of most other solar PV panels, in order to withstand wet, heavy snow which occurs in Seattle from time to time.

The inverted is rated at 3.0 kw AC output. Estimated annual electrical energy generation by the system for the site is 3560 kwh.

The data sheets for the REC panels and SMA inverter are attached as Appendix B.

Figures 12 and 13 (prepared by Bonneville Environmental Foundation) are drawings of the roof of the Cadillac Hotel with the 16 solar PV panels included. During most hours of daylight throughout the year, the panels will be fully illuminated. Only during periods of low sun angle will the panels be shaded by adjacent buildings.

Figure 14 (also by BEF) shows the electrical and monitoring hardware and pathway schematic.

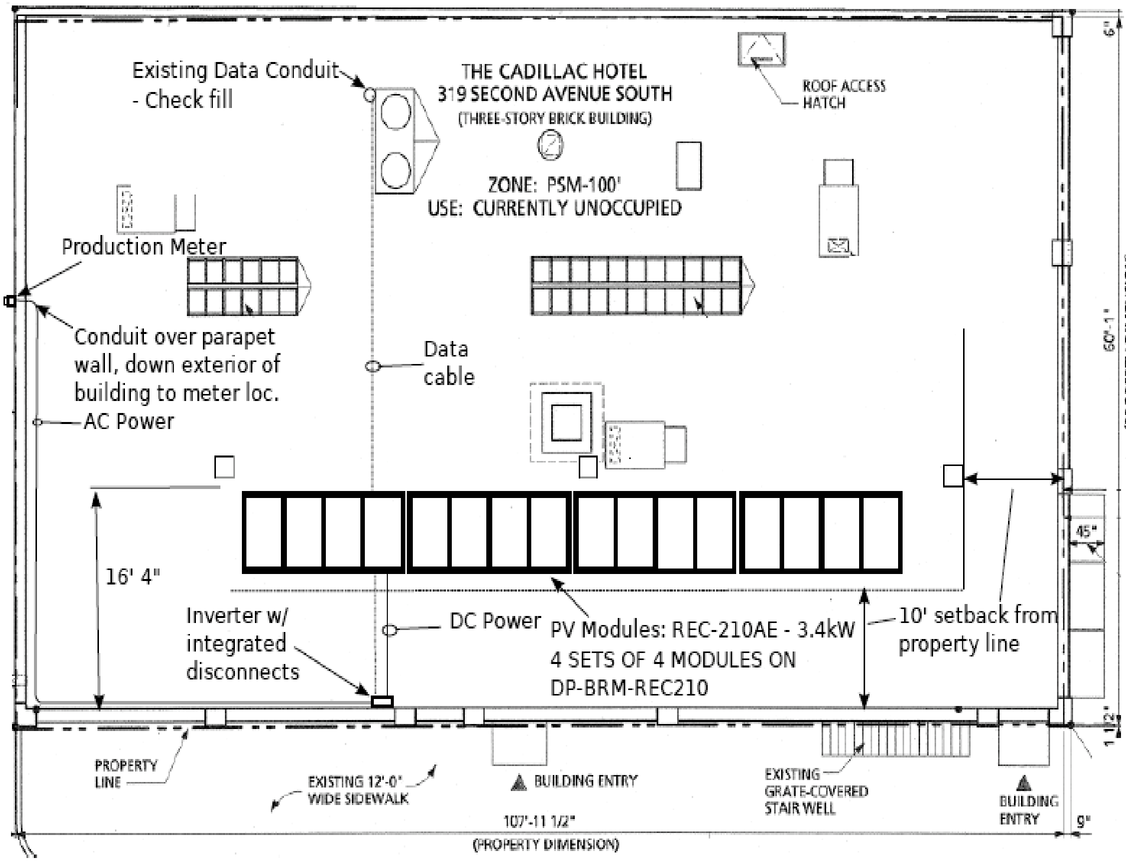

Figure 12. Layout of 16 solar PV panels on roof of Cadillac Hotel

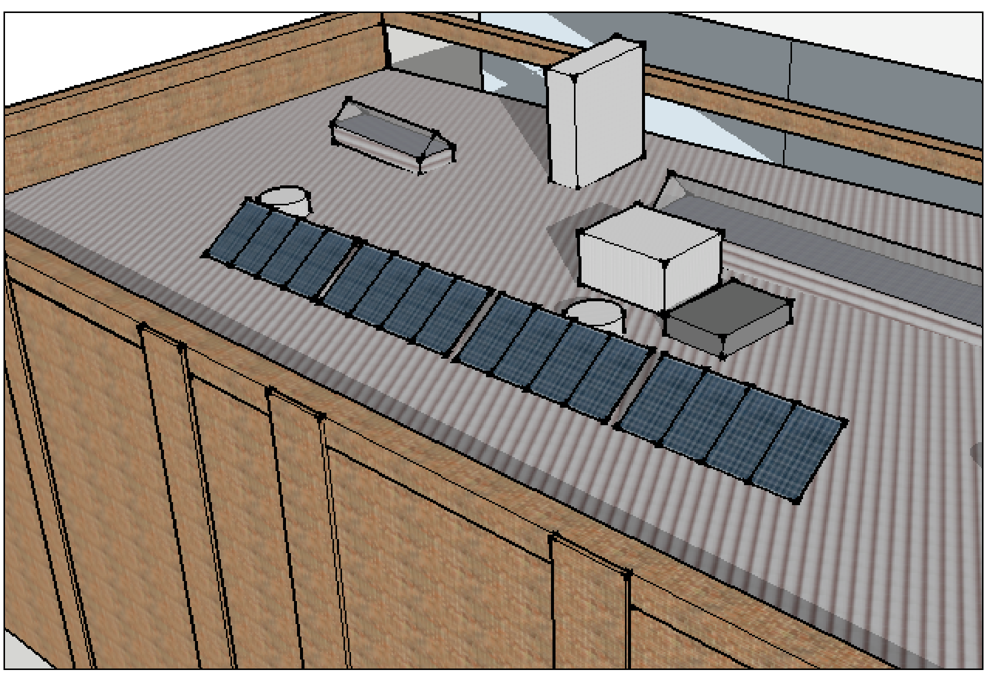

Figure 13. Drawing of 16 solar PV panels on roof of Cadillac Hotel.

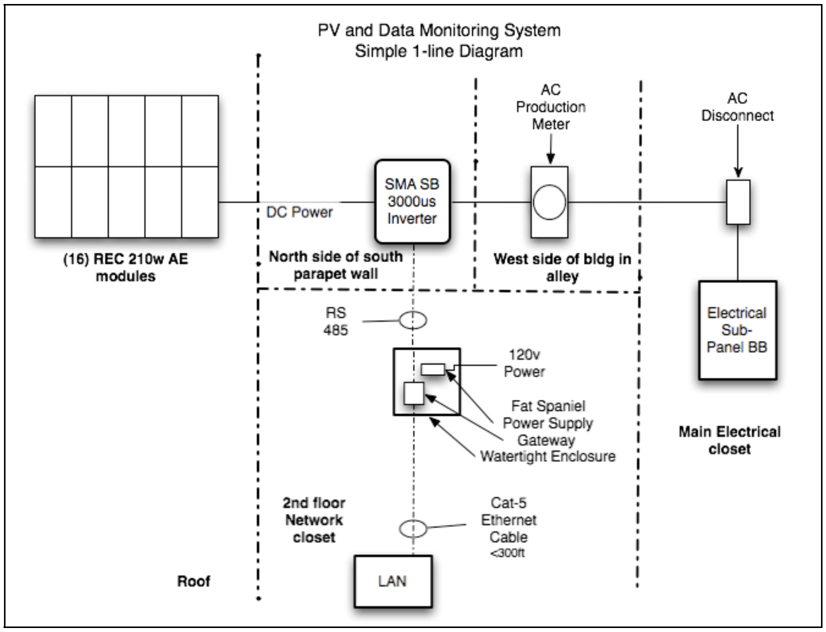

Figure 14. Solar PV electrical and monitoring pathways.

Installation of the solar PV system is planned for the first half of 2010. Bonneville Environmental Foundation will manage the installation for the partnership.

Acknowledgements

The University of Washington wishes to thank the following individuals and organizations, without whose support and assistance this project would not have been successful:

- University National Parks Energy Partnership Program, and Dr. Jamie Winebrake, Director, Rochester Institute of Technology.
- Mr. Sean O'Mara, Acting Superintendent, Klondike National Historical Park, Seattle, WA.
- Ms. Karen_Beppler-Dorn, Superintendent, Klondike National Historical Park, Seattle, WA.
- Mr. Rick Wagner, Chief, Land Resources Program Center, Pacific West Region, National Park Service, Seattle, WA.
- Mr. Steve Butterworth, Pacific West Region: Energy, Dams, Regular Cyclic, National Park Service, Seattle, WA.
- Mr. Kji Kelly, Director of Property and Asset Management, Historic Seattle Inc., Seattle, WA.
- Mr. Randy Batchelor, Project Manager, Bonneville Environmental Foundation, Portland, OR.
- Mr. Jack Brautigam, Solar Generation Program Manager, Seattle City Light, Seattle, WA.

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Appendix A: Lighting and Plug Loads

Table 8: Basement Electrical Loads

Load Description	Model	quantity	Power (W)	Total Power Consumed (W)	Hours of operation per day	Daily Energy Consumption (kWh)	Yearly Energy Consumption (kWh)
spot light	Phillips Master Color 30 L	53	70	3710	10	37.1	13541.5
2' by 4' surface mounted compact fluorescent lighting fixtures each with three 28 watt fluorescent tubes in museum	28 watt high efficiency	10	84	840	10	8.4	3066
Touch screen kiosk	iGO Panel PC	5	80	400	24	9.6	3504
50'' Flat screen	don't know	1	100	100	10	1	365
Compact Fluorescent lighting fixtures each with two 28 watt fluorescent tubes in museum	Sylvania	25	56	1400	1	1.4	511
totals						57.5	20987.5

Table 9: First Floor Electrical Loads

Load Description	Model	quantity	Power (W)	Total Power Consumed (W)	Hours of operation per day	Daily Energy Consumption (kWh)	Yearly Energy Consumption (kWh)
spot light	Phillips Master Color 30 L	60	70	4200	10	42	15330
hanging lights	28 watt high efficiency	6	28	168	10	1.68	613.2
Touch screen kiosk	iGO Panel PC	4	80	320	24	7.68	2803.2
hallway lights	28 watt high efficiency	12	28	336	10	3.36	1226.4
Auditorium Lights	42 watt high efficiency	10	28	280	2	0.56	204.4
Gift shop lighting	60 W Incandescent Lights	4	60	240	8	1.92	700.8
Spot Lights	28 watt high efficiency	5	28	140	8	1.12	408.8
totals						58.32	21286.8

Table 10: Second Floor Electrical Loads

Load Description	Model	quantity	Power (W)	Total Power Consumed (W)	Hours of operation per day	Daily Energy Consumption (kWh)	Yearly Energy Consumption (kWh)
2' by 4' surface mounted compact fluorescent lighting fixtures each with three 28 watt fluorescent tubes in museum	Sylvania	70	84	5880	9	52.92	19315.8
Copy Machine	RICOH Aticio 1022 active	1.00	1044.00	1044.00	0.62	0.65	236.99
	RICOH Aticio 1022 standby	1.00	179.00	179.00	3.95	0.71	258.12
	RICOH Aticio 1022 suspend	1.00	42.00	42.00	4.00	0.17	61.32
	RICOH Aticio 1022 off	1.00	0.50	0.50	15.43	0.01	2.82
Fax Machine	Canon CFX-L4500 IF in use	1.00	38.00	38.00	0.50	0.02	6.94
	Canon CFX-L4500 IF standby	1.00	4.80	4.80	23.50	0.11	41.17
PC's	dell optiplex 360	6.00	255.00	1530.00	9.00	13.77	5026.05
monitors	dell	9.00	29.00	261.00	9.00	2.35	857.39
printer	HP Laser Jet 5000 N printing	4.00	365.00	1460.00	0.50	0.73	266.45
printer	HP Laser Jet 5000 N idle	4.00	40.00	160.00	8.50	1.36	496.40
laptops	dell	2.00	28.00	56.00	9.00	0.50	183.96
Air conditioner	Emerson	1.00	500.00	500.00	24.00	12.00	4380.00
server	Poweredge 2800	1.00	930.00	930.00	24.00	22.32	8146.80
servers	Poweredge 2600	2.00	930.00	1860.00	24.00	44.64	16293.60
PC's	dell optiplex 360	1.00	255.00	255.00	9.00	2.30	837.68
monitors	dell	2.00	29.00	58.00	9.00	0.52	190.53
laptops	dell	1.00	28.00	28.00	9.00	0.25	91.98
PC's	dell optiplex 360	7.00	255.00	1785.00	9.00	16.07	5863.73
monitors	dell	9.00	29.00	261.00	9.00	2.35	857.39
printer	HP Laser Jet 5000 N printing	1.00	365.00	365.00	0.50	0.18	66.61
printer	HP Laser Jet 5000 N idle	1.00	40.00	40.00	8.50	0.34	124.10
laptops	dell	1.00	28.00	28.00	9.00	0.25	91.98
monitors	dell	1.00	29.00	29.00	9.00	0.26	95.27
PC	dell optiplex 360	1.00	255.00	255.00	9.00	2.30	837.68
monitors	dell	1.00	29.00	29.00	9.00	0.26	95.27
laptops	dell	1.00	28.00	28.00	9.00	0.25	91.98
monitors	dell	1.00	29.00	29.00	9.00	0.26	95.27
totals						177.84	64913.23

Table 11: Third Floor Electrical Loads

Load Description	Model	quantity	Power (W)	Total Power Consumed (W)	Hours of operation per day	Daily Energy Consumption (kWh)	Yearly Energy Consumption (kWh)
2' by 4' surface mounted compact fluorescent lighting fixtures each with three 28 watt fluorescent tubes in museum	Sylvania	55	84	4620	9	41.58	15176.70
laptops	dell	2	28	56	9	0.50	183.96
PC's	dell Optiplex 360	3	255	765	9	6.89	2513.03
monitors	dell	5	29	145	9	1.31	476.33
printer	HP Laser Jet 5000 N printing	1	365	365	0.5	0.18	66.61
printer	HP Laser Jet 5000 N idle	1	40	40	8.5	0.34	124.10
laptops	dell	2	28	56	9	0.50	183.96
PC's	dell Optiplex 360	2	255	510	9	4.59	1675.35
monitors	dell	4	29	116	9	1.04	381.06
printer	HP Laser Jet 5000 N printing	1	365	365	0.5	0.18	66.61
printer	HP Laser Jet 5000 N idle	1	40	40	8.5	0.34	124.10
Copy Machine	Canon Image Runner 5070 active	1	1354	1354	0.85753425	1.16	423.80
	Canon Image Runner 5071 standby	1	396	396	3.85753425	1.53	557.57
	Canon Image Runner 5072 suspend	1	68	68	3.85753425	0.26	95.74
	Canon Image Runner 5073 off	1	0.6	0.6	15.4273973	0.01	3.38
Fax Machine	Canon CFX-L4500 IF in use	1	38	38	0.5	0.02	6.94
	Canon CFX-L4500 IF standby	1	4.8	4.8	23.5	0.11	41.17
Copy Machine	Lexmark C920	1	400	400	0.85753425	0.34	125.20
	Lexmark C920	1	13	13	23.1424658	0.30	109.81
Microwave	Emerson	1	1350	1350	0.5	0.68	246.38
Refrigerator	whirlpool	1	57	57	24	1.37	499.32
PC's	dell Optiplex 360	1	255	255	8	2.04	744.60

monitors	dell	1	29	29	8	0.23	84.68
printer	HP Laser Jet 5000 N printing	1	365	365	8	2.92	1065.80
printer	HP Laser Jet 5000 N idle	1	40	40	8	0.32	116.80
PC's	dell Optiplex 360	1	255	255	8	2.04	744.60
monitors	dell	1	29	29	8	0.23	84.68
printer	HP Laser Jet 5000 N printing	1	365	365	8	2.92	1065.80
printer	HP Laser Jet 5000 N idle	1	40	40	8	0.32	116.80
PC's	dell Optiplex 360	1	255	255	8	2.04	744.60
monitors	dell	1	29	29	8	0.23	84.68
printer	HP Laser Jet 5000 N printing	1	365	365	8	2.92	1065.80
printer	HP Laser Jet 5000 N idle	1	40	40	8	0.32	116.80
PC's	dell optiplex 360	3	255	765	8	6.12	2233.80
monitors	dell	3	29	87	8	0.70	254.04
printer	HP Laser Jet 5000 N printing	1	365	365	8	2.92	1065.80
printer	HP Laser Jet 5000 N idle	1	40	40	8	0.32	116.80
Totals						89.83	32787.19

Appendix B: Data Sheets for REC Solar PV Panels and SMA Inverter

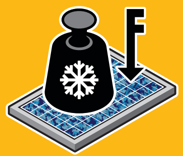
REC

HIGH PERFORMANCE SOLAR MODULES

REC AE-US

REC AE-Series modules are the perfect choice for building solar systems that combine long lasting product quality with reliable power output. REC combines high quality design and manufacturing standards to produce high-performance solar modules with uncompromising quality.

US - MADE SILICON

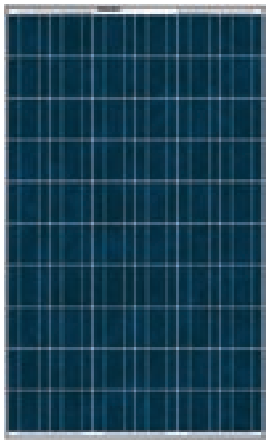


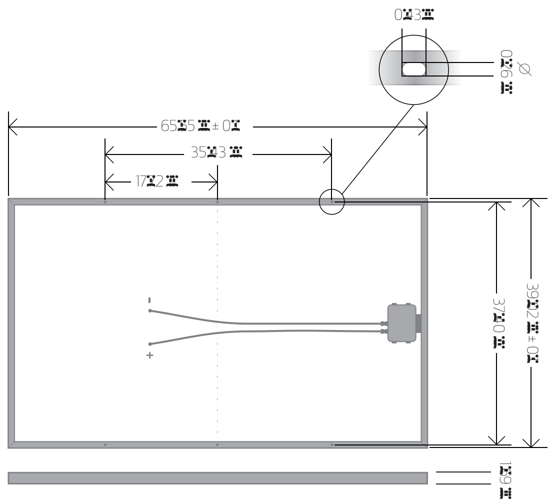
ROBUST AND DURABLE DESIGN

ENERGY PAYBACK TIME OF ONE YEAR

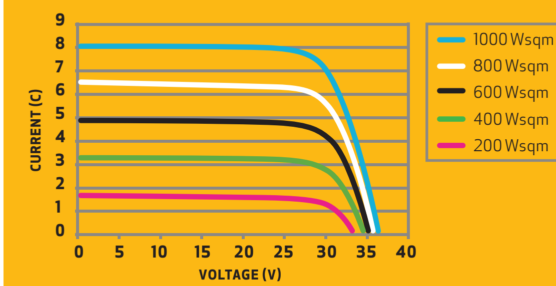
OPTIMIZED ALSO FOR CLOUDY WEATHER

REC AE-US





IV CHARACTERISTICS 215W MODULE



13.9 EFFICIENCY

63 MONTHS WORKMANSHIP WARRANTY

25 YEAR POWER OUTPUT WARRANTY

ELECTRICAL DATA @ STC	REC205 AE-US	REC210 AE-US	REC215 AE-US	REC220 AE-US	REC225 AE-US	REC230 AE-US
Peak Power Watts - P_{MAX} (Wp)	205	210	215	220	225	230
Power Output Tolerance - P_{MAX} (%)	±3	±3	±3	±3	±3	±3
Maximum Power Voltage - V_{MPP} (V)	27.2	27.6	28.0	28.4	28.8	29.1
Maximum Power Current - I_{MPP} (A)	7.6	7.6	7.7	7.8	7.8	7.9
Open Circuit Voltage - V_{OC} (V)	36.0	36.1	36.3	36.4	36.6	36.8
Short Circuit Current - I_{SC} (A)	8.3	8.3	8.3	8.4	8.4	8.4
Module Efficiency (%)	12.4	12.7	13.0	13.3	13.6	13.9

Values at Standard Test Conditions STC (Air Mass AM 1.5, Irradiance 1000 W/m², Cell temperature 25 °C)

GENERAL DATA

Cell Type	156 x 156 mm multi-crystalline cells 60 cells in series
Glass	High-transparency solar glass with antireflection surface treatment by Sunarc Technology
Back sheet	PTE foil
Frame	Anodized aluminium
Connectors	2 x 61 in (1.55m) solar cables with MC4 connectors

TEMPERATURE RATINGS (230W RATED MODULE)

Nominal Operating Cell Temperature (NOCT)	48.6°C (±2°C)
Temperature Coefficient of P_{MPP}	-0.497%/°C
Temperature Coefficient of V_{OC}	- 0.370%/°C
Temperature Coefficient of I_{SC}	0.109%/°C

WARRANTY

10 years limited warranty of 90% power output
25 years limited warranty of 80% power output
63 months limited product warranty

CERTIFICATION

C US

Certification / Standards
Certified according to UL 1703 - 3rd edition

CEC/CSI listed

MECHANICAL DATA

Dimensions	65.55" x 39.02" x 1.69"
Area	17.76 ft²
Weight	48.50 lbs

MAXIMUM RATINGS

Operational Temperature	-40 ... +176°F [-40 ... +80°C]
Maximum System Voltage	600V
Maximum Load	112 lbs/ft² (5400 Pa)
Wind Speed	122 mph (safety factor 3, 197 km/h)
Max Series Fuse Rating	15A
Max Reverse Current	15A

Note! Specifications subject to change without notice.

REC is the leading vertically integrated player in the solar energy industry. REC Silicon and REC Wafer are among the world's largest producers of polysilicon and wafers for solar applications. REC Solar is a rapidly growing manufacturer of solar cells and modules. In 2008, REC Group had revenues of NOK 8 191 million and an operating profit of NOK 2 529 million. About 2 400 employees work in REC's worldwide organization.

For further information, contact your local distributor or visit our web site: **www.recgroup.com**

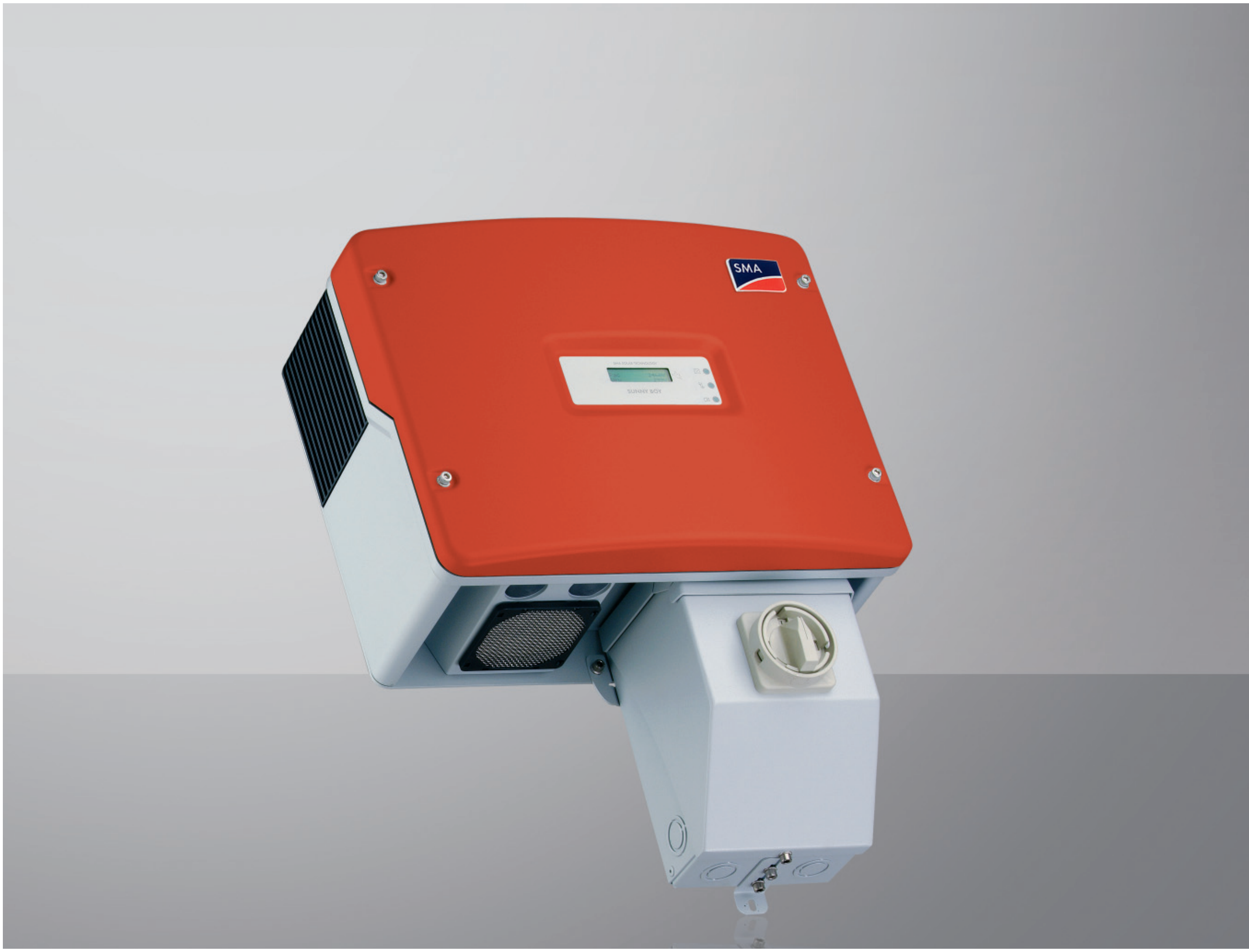
REC

REC Solar AS
Kjørboveien 29
1329 Sandvika
Norway

www.recgroup.com/usa

July 2009, version B

SUNNY BOY 3000US / 4000US



- Highest CEC efficiency in its class
- Integrated load-break rated lockable DC disconnect switch
- Integrated fused series string combiner

- Sealed electronics enclosure & Opticool™
- Comprehensive SMA communications and data collection options

- Ideal for residential or light commercial applications
- Rugged cast aluminum outdoor rated enclosure
- UL 1741/IEEE-1547 compliant
- 10 year standard warranty

SUNNY BOY 3000US / 4000US

The best in their class

Our US series inverters utilize our proven technology and are designed specifically to meet IEEE-1547 requirements. Compact design makes them ideal for residential use and the integrated lockable DC disconnect makes installation more cost effective. They are field-configurable for positive ground systems making them more versatile than ever. Increased efficiency means better performance and shorter payback periods. Throughout the world, Sunny Boy is the benchmark for PV inverter performance and reliability.

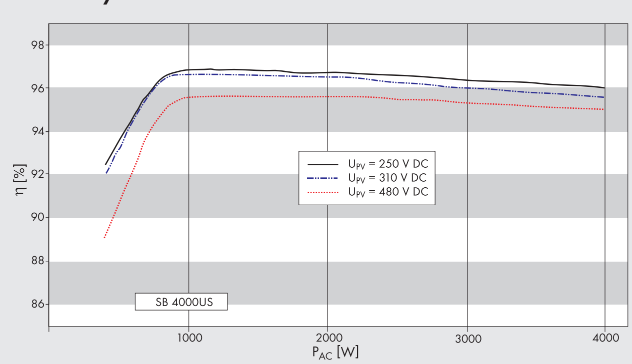
Technical Data

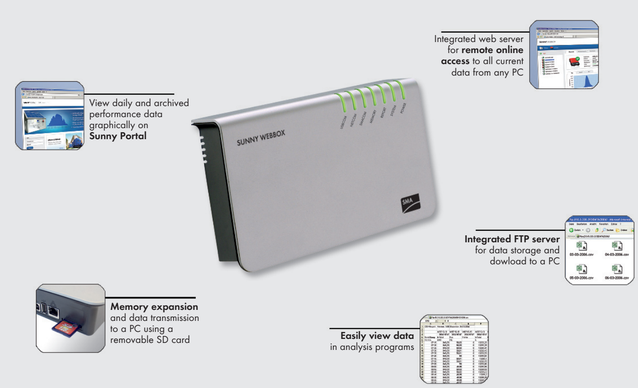
	SB 3000US	SB 4000US
Recommended Maximum PV Power (Module STC)	3750 W	4375 W @ 208 V / 5000 W @ 240 V
DC Maximum Voltage	500 V	600 V
Peak Power Tracking Voltage	175 – 400 V @ 208 V 200 – 400 V @ 240 V	220 – 480 V @ 208 V 250 – 480 V @ 240 V
DC Maximum Input Current	17 A	18 A*
DC Voltage Ripple	< 5%	< 5%
Number of Fused String Inputs	2 (inverter), 4 x 20 A (DC disconnect)	2 (inverter), 4 x 20 A (DC disconnect)
PV Start Voltage	228 V	285 V
AC Nominal Power	3000 W	3500 W @ 208 V / 4000 W @ 240 V
AC Maximum Output Power	3000 W	3500 W @ 208 V / 4000 W @ 240 V
AC Maximum Output Current	15 A @ 208 V, 13 A @ 240 V	17 A
AC Nominal Voltage Range	183 – 229 V @ 208 V 211 – 264 V @ 240 V	183 – 229 V @ 208 V 211 – 264 V @ 240 V
AC Frequency: nominal / range	60 Hz / 59.3 – 60.5 Hz	60 Hz / 59.3 – 60.5 Hz
Power Factor (Nominal)	0.99	0.99
Peak Inverter Efficiency	96.5%	96.8%
CEC Weighted Efficiency	95.0% @ 208 V 95.5% @ 240 V	95.5% @ 208 V 96.0% @ 240 V
Dimensions: W x H x D in inches	17.8 x 13.8 x 9.3	17.8 x 13.8 x 9.3
Weight / Shipping Weight	84 lbs / 97 lbs	84 lbs / 97 lbs
Ambient Temperature Range	−13 to 113 °F	−13 to 113 °F
Power Consumption: standby / nighttime	< 7 W / 0.1 W	< 7 W / 0.1 W
Topology	Low frequency transformer, true sinewave	Low frequency transformer, true sinewave
Cooling Concept	OptiCool™, forced active cooling	OptiCool™, forced active cooling
Mounting Location: indoor / outdoor (NEMA 3R)	●/●	●/●
LCD Display	●	●
Communication: RS485 / wireless	❍/❍	❍/❍
Warranty: 10-year	●	●
Compliance: IEEE-929, IEEE-1547, UL 1741, UL 1998, FCC Part 15 A & B	●	●
Specifications for nominal conditions	● Included ❍ Optional	

*Also available in 16 A current limited configuration for use in 100 A service applications.

NOTE: US inverters ship with gray lids.

Efficiency Curves





Tel. +1 916 625 0870
Toll Free +1 888 4 SMA USA
www.SMA-America.com

SMA America, LLC