Energy Performance and Economic Evaluations of the Geothermal Heat Pump System used in the KnowledgeWorks I and II Buildings, Blacksburg, Virginia

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(ABSTRACT)

Heating, Ventilating and Air Conditioning Systems (HVAC) are not only one of the most energy consuming components in buildings but also contribute to green house gas emissions. As a result often environmental design strategies are focused on the performance of these systems. New HVAC technologies such as Geothermal Heat Pump systems have relatively high performance efficiencies when compared to typical systems and therefore could be part of whole-building performance design strategies.

In collaboration with the Virginia Tech Corporate Research Center, Inc., this research studies the energy consumption and cost benefits of the Geothermal Heat Pump System that has been integrated and operated in the KnowledgeWorks I and II buildings located on the Virginia Tech campus.

The purpose of this thesis is to understand the energy and cost benefits of the Geothermal Heat Pumps System when compared to the conventional package variable air volume (VAV) with hot water coil heating and air-source heat pump systems using computer simulation and statistical models. The quantitative methods of building energy performance and life-cycle cost analyses are applied to evaluate the results of simulation models, the in-situ monitoring data, and the associated documents. This understanding can be expanded to the higher level of architectural systems integration.

Acknowledgements

I always remember my first class at Virginia Tech. Instead of sitting in a dark lecture room, Dr. Jones took us outside for a walk under the shade of trees. It was a memorable moment for experiencing thermal comfort and human dwelling with surrounding nature. Dr. Jones is a teacher who would like his students to be a learner not a grade chaser. Followed his philosophy, this thesis and my study are all about learning experience that I enjoy very much. Thank you, Dr. Jim Jones, for your inspired teaching both in profession and moral principles. Thank you for your time, patience, generous, encouragement, and supports. I would not be at this point without the opportunities you have given.

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This thesis is dedicated to my parents, my sisters, and my wife. Mom and dad, your example has taught me to be an empty glass and not to give up. I am proud of being your son and love you very much. Earn and Ing, thanks to keep in touch while I am away from home. Mam, thanks for your brave resigning from your successful career to become a domestic engineer and stay with me in the U.S. You are my inspiration and I am privileged to be your husband and best friend.

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Chapter 1: Introduction

Today in the United States and in most industrialized countries, there is growing concern for fossil fuel depletion due to inefficient energy consumption and an increasing body of evidence linking these issues to global climate change. The growth of population, the economy, and consequent energy use are typically dependent on nonrenewable fossil energy resources. The use of nonrenewable resources such as oil, coal, and natural gas is limited and contributes significantly to greenhouse gas (GHG) emissions and pollution. Since the building sector annually consumes about 48% of energy in the U.S. and is responsible for 46% of U.S. carbon emissions (Mazria 2003), improving the efficiency of building energy consumption has significant impact potential. Consistent with LEED (Leadership in Energy and Environmental Design), the Energy and Atmosphere (EA) credit 1, Optimize Energy Performance, is the single largest opportunity to obtain points in the LEED rating system. Therefore, energy conservation and efficiency and the use of renewable energy are among the most important concerns in today's building industry.

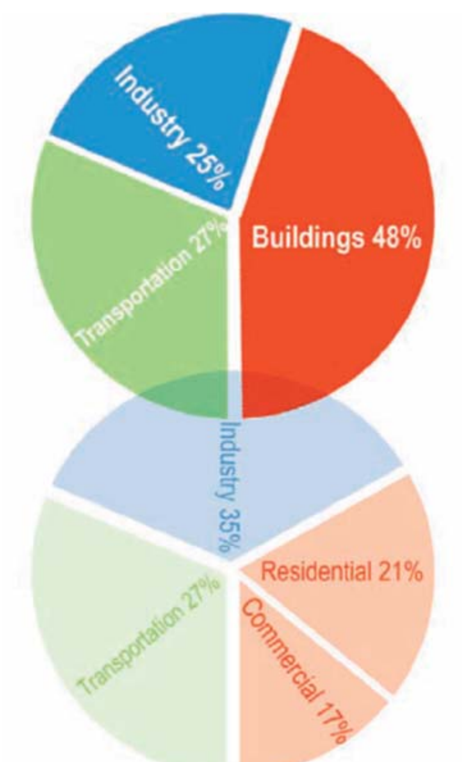


Figure 1.1 - Combining the annual energy required to operate residential, commercial, and industrial buildings along with the embodied energy of industrial, commercial, and industrial buildings along with the embodied energy industry-produced building materials like carpet, title, glass, and concrete exposes buildings as the largest energy consuming and greenhouse gas emitting sector from (AIA 2006).

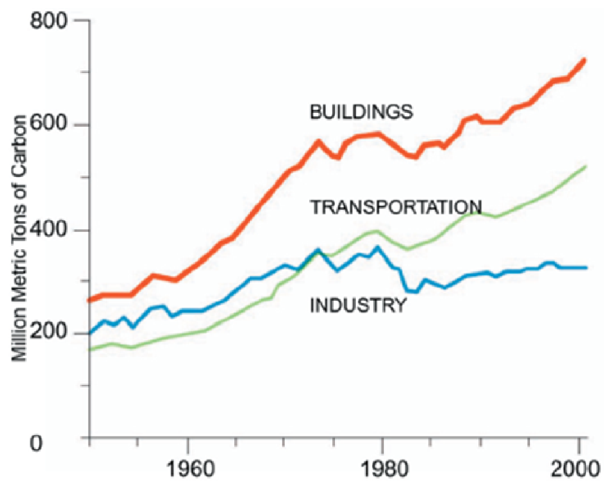


Figure 1.2 - U.S. CO2 emissions by sector from (AIA 2006).

In the U.S., over 100 years of total energy use, buildings typically consume 5-10% for materials and construction related activities and the remaining 90-95% for operating energy (Randolph and Masters 2007). Energy conservation, improved efficiency, and the use of renewable energy sources can result in a huge energy reduction in buildings. Operating energy includes but is not limited to energy used for electrical lighting, electrical appliances, and heating, ventilating and air-conditioning (HVAC) systems. Among all systems, HVAC systems consume approximately 43% of total operating energy in residential buildings and about 33% of operating energy in commercial buildings (DOE 2007). HVAC systems are the main source of GHG emissions in buildings. As a result, the reduction of energy consumption and GHG emissions in HVAC systems can significantly improve the environment.

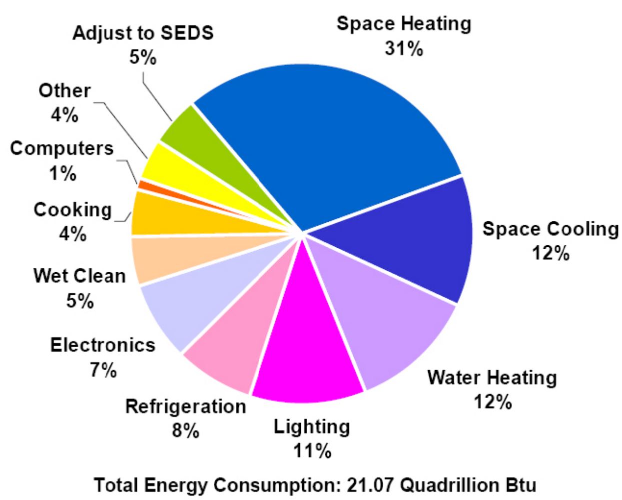


Figure 1.3 - U.S. residential buildings primary energy end-use splits, 2005 from (DOE 2007)

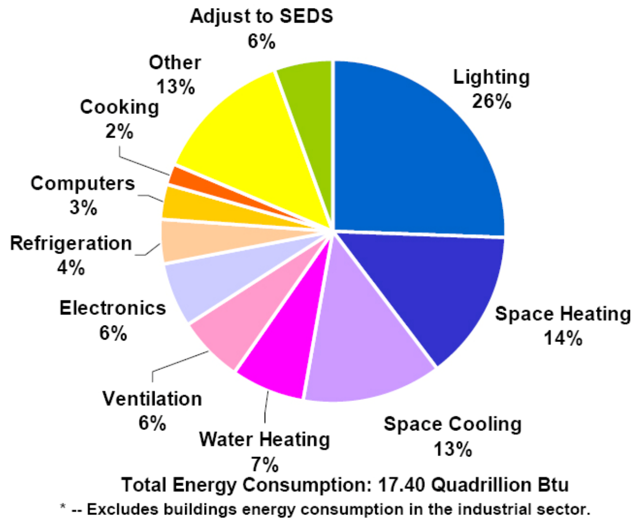


Figure 1.4 - U.S. commercial buildings primary energy end-use splits, 2005 from (DOE 2007)

With today's technology, the most commonly used HVAC systems are forced-air central heating, hydronic, compressive air conditioners, evaporative and absorption cooling, heat pump, and geothermal heat pump systems. Systems may use combustion based energy sources such as fossil fuel, oil, coal, and natural gas or non-combustion based energy resources such as electricity to provide heating or cooling in the conditioned space. Although electricity is a non-combustion energy source, the primary source of electricity could be combustion based. From an energy and environmental consciousness perspective, the geothermal heat pump system is considered one of the most energy efficient and low GHG emitting systems available. It not only consumes less electricity but also has no direct GHG emitting parts.

Heat pumps use electricity to transfer heat from one place to another to provide heating or cooling. The basic components of heat pumps include a compressor, condenser, expansion valve, and evaporator. Heat pumps can be operated in two cycles: cooling and heating. In the cooling cycle, the refrigerant enters the compressor as a low pressure, low temperature saturated vapor and is compressed to the condenser pressure. It leaves the compressor as a high temperature, high pressure, and superheated vapor and cools down and condenses as it flows through the coils of the condenser by releasing heat to the surrounding medium. Then, it enters an expansion valve or capillary tube where its pressure and temperature decrease drastically due to the throttling effect. The low pressure, low temperature, low quality vapor refrigerant then enters the evaporator, where it evaporates by absorbing heat from the conditioned space. The cycle is completed as the refrigerant leaves the evaporator and reenters the compressor. In the heating cycle, the refrigerant is processed in the reverse order.

Combining heat pumps with the geoexchange process, also called geothermal or ground-source heat pump (GSHP), takes advantage of more stable earth temperatures as the heat source or heat sink. The GSHP cooling cycle captures heat from inside the conditioned space, compresses, and exchanges heat with the earth. In the GSHP heating cycle, heat is captured from the earth through underground loops, compressed, and then released to the conditioned area, as shown in Figure 1.5 and Figure 1.6. According to the U.S. Environmental Protection Agency (EPA), GSHP can reduce energy consumption and corresponding emissions up to 44% when compared to air-source heat pumps, and up to 72% when compared to electric resistant heating with standard air-conditioning equipment (DOE 2005). A GSHP can improve the indoor-air-quality and humidity control in the conditioned area as well. Furthermore, 70% of energy used in GSHP is renewable energy from the ground (GHPC 2003).

Geothermal heat pumps exchange heat with the earth through underground loops. Underground loops include horizontal, vertical, and pond loops. The loop installation results in the GSHP having the highest initial cost when compared to air-source heat pumps and other competitive systems. However, underground loops are reported to last 25-50 years, while air to air heat pumps usually only last about 20 years. In addition to the system's extended life, GSHP has only a few moving parts all of which are installed inside the building thus reducing maintenance costs. GSHP equipment is durable, highly reliable, and requires less building space than other HVAC systems. GSHP also consumes less energy than most other HVAC systems which helps reduce the energy cost over its lifetime. It is believed that lower energy and maintenance costs of the GSHP system make it potentially cost-competitive with other HVAC systems, which is a question for this thesis.

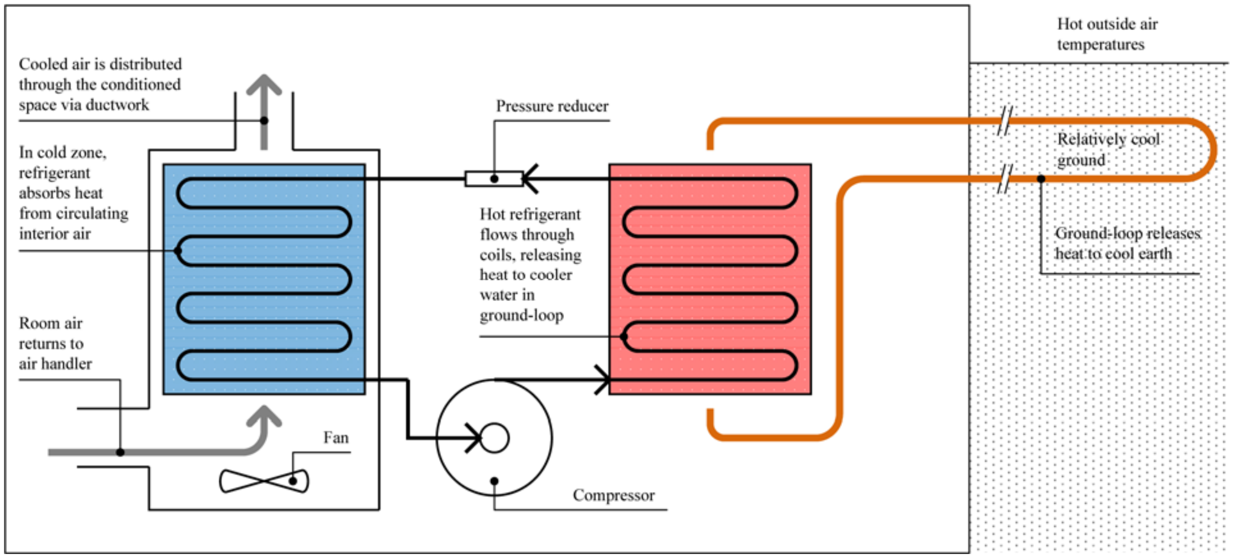


Figure 1.5 - The cooling cycle of geothermal heat pumps system.

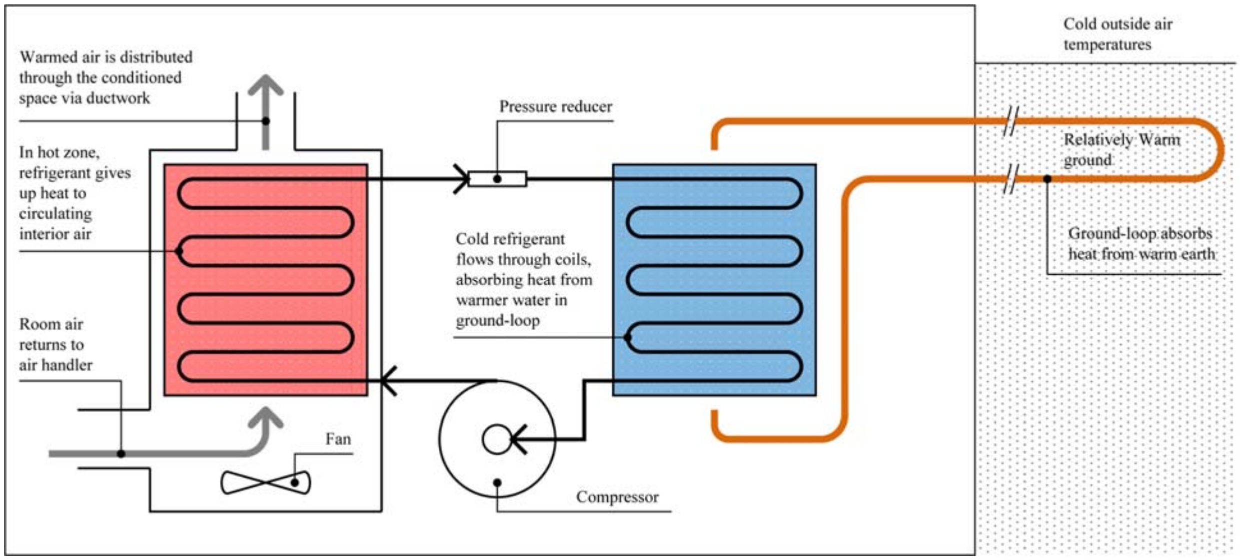


Figure 1.6 - The heating cycle of geothermal heat pumps system.

In summary, the geothermal heat pump system is considered one of the most energy efficient and environmentally friendly systems among today's HVAC technologies. However, the initial cost of the system is high when compared to most other competitive HVAC systems. Also, the cost effectiveness of geothermal heat pumps could be diverse in different building contexts such as efficiency of the GSHP system, electricity rates, and maintenance and repair costs. Those factors could affect the economic benefits of GSHP systems and make them too expensive for investment. Therefore, the performance and life-cycle cost of the geothermal heat pump systems should be evaluated.

1.1 Background

The Virginia Tech Corporate Research Center (VTCRC) is an outstanding university-based research facility in the U.S. Mid-Atlantic region. The campus has about 19 completed buildings and experiences continual growth. In these structures, there are two connected buildings, KnowledgeWorks I and II which were constructed in 2005 and 2006, which share a geothermal heat pump system. The system employs both ground and surface water for heat sources and sinks. After two years of full operation, the building operator has been interested in evaluating the energy and economic performances of geothermal heat pumps system. With the collaboration of Dr. Paul Fleming, Director of VTCRC Facilities and Data Services, and Dr. James R. Jones, Associate Professor of the Department of Architecture, College of Architecture and Urban Study (CAUS), Virginia Tech, the geothermal heat pump system energy and economic performances evaluation was brought to this master's thesis.

Figure 1.7 - VT KnowledgeWorks I and II (photograph by Kongkun Charoenvisal, 2007).

Figure 1.8 - Water pond, a part of the geothermal heat pump system (photograph by Kongkun Charoenvisal, 2007).

1.2 Objectives

The intent of this quantitative research was to understand the energy consumption and cost benefits of the geothermal heat pump system used in the KnowledgeWorks I and II buildings. The research uses hourly whole building energy consumption simulation and actual building data to develop a baseline model of the KnowledgeWorks I and II buildings. The baseline model was then used for comparative evaluations of energy consumptions for alternative HVAC systems. In cooperation with the design engineers of these buildings, the costs of HVAC system installations were estimated. The simulated energy consumption and estimated system costs were input to life-cycle costs analyses.

1.3 Assumptions

The building energy simulation software, eQUEST, has internal algorithms for the ground-source heat pump (GSHP) system, and is validated and acceptable for simulating the GSHP system used in buildings.

Conventional HVAC systems are limited to the package VAV with hot water coil heating, and conventional air-source heat pump systems.

1.4 Hypotheses

Hypothesis 1: Energy Performance Evaluation

Hypothesis 1.1: The annual energy consumption (AE) in the case study buildings will be lower for the geothermal heat pump system when compared to conventional HVAC systems. The annual energy saving (AES), Equation 1.1, is used to explain the method of comparison.

$$AES_{GSHP}\,(kWh/Year) = AE_{ALT}\,(kWh/Year) - AE_{GSHP}\,(kWh/Year) > 0 \quad (1.1)$$

Where:

AES_{GSHP} = Annual energy saving from using the GSHP system

AE_{GSHP} = Annual energy consumption in buildings with the GSHP system

AE_{ALT} = Annual energy consumption in buildings with the alternative systems

Hypothesis 2: Economic Evaluations

Hypothesis 2.1: The life-cycle costs of the ground-source heat pump system used in the case study buildings will be lower than the life-cycle costs of the conventional HVAC system scenarios. Equation 1.2 demonstrates the life-cycle costs comparison.

$$LCC_{ALT}\,(\$) - LCC_{GSHP}\,(\$) > 0 \quad (1.2)$$

Where:

LCC_{GSHP} = Life-cycle costs of the GSHP system in present value $

LCC_{ALT} = Life-cycle costs of the alternative systems in present value $

In addition to the life-cycle costs comparisons, the supplementary measures of economic evaluation are used to provide a better understanding of economic benefits of the GSHP system. The supplementary measures used in this research are the saving-to-investment ratio (SIR) and discounted payback (DPB).

Hypothesis 2.2: The Saving-to-Investment Ratio (SIR) is a ratio of operational savings to difference in capital investment costs. The SIR of the GSHP system will be higher than one when compared to conventional systems. Equation 1.3 shows the hypothesis equation.

$$SIR_{GSHP:ALT} > 1 \quad (1.3)$$

Where:

$SIR_{GSHP:ALT}$ = SIR of the GSHP system when comparing to the alternative systems

Hypothesis 2.3: The Discounted Payback (DPB) is the number of times required for the cumulative savings from the GSHP system to recover its initial investment costs and other accrued costs, taking into account the time value of money. The DPB of the GSHP system when comparing to the alternative systems should be lower than the analysis period. Equation 1.4 provides the method of comparison.

$$Analysis\ years\ -\ DPB_{GSHP:ALT}\ (years)\ >\ 0 \tag{1.4}$$

Where:

$DPB_{GSHP::ALT}$ = DPB of the GSHP system when comparing to the alternative systems

1.5 Research Design Approach Summary

In summary, Chapter 1: Introduction has introduced the overview of this research, its objectives, and its hypotheses. The next chapter, Chapter 2: Literature Review provides the background necessary to understand the contents of this research. Chapter 3: Research Methodology describes the data gathering, analysis, and evaluation methods. Then, Chapter 4: Data Analysis and Results, applies the methods from Chapter 3 to acquire the results. Finally, Chapter 5: Conclusions and Summary, discusses the outcome of this research and introduces the application of this research in real world situations.

Chapter 2: Literature Review

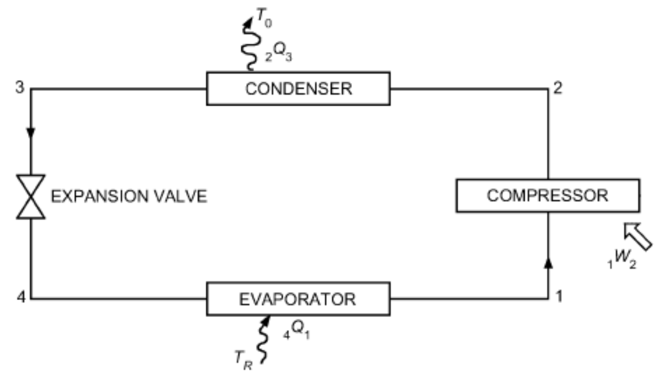
2.1 Principles and Applications of Geothermal Heat Pumps

From the second law of thermodynamics, heat is transferred in the direction of decreasing temperature from high temperature mediums to low temperature mediums. It is impossible to reverse this phenomenon without introducing mechanical devices such as air conditioners and heat pumps. Air conditioners and heat pumps share the same characteristic which is the ability to transfer heat from a low temperature medium to a high temperature medium. Air conditioners and heat pumps serve different objectives. An air conditioner extracts heat from conditioned space and releases it to the outside to cool the space while a basic heat pump extracts heat from outside the conditioned space and releases it to the inside to provide heating. Using today's technology, several heat pumps combine both cooling and heating capacities to serve as air conditioner and heat source for conditioned spaces.

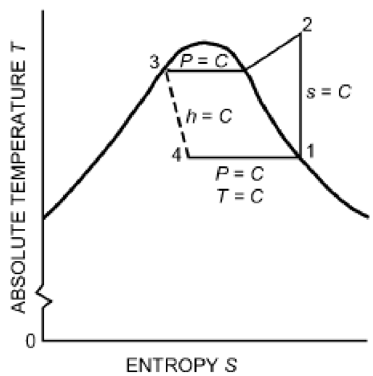
There are three different types of heat pumps defined by their heat sources or sinks, and distribution fluids including air-air, water-air, and water-water heat pumps. The geothermal or ground-source heat pump (GSHP) systems are heat pump systems that use water as their heat source or sink. The source water is used to exchange heat with the Earth, or water sources such as ponds, lakes, or well water. The source water may be mixed with an anti-freeze, and can be circulated directly to heat pumps or indirectly through an intermediate fluid in a closed loop. Inside the conditioned space, heat can be distributed by air or water depending on the distribution systems. GSHPs are also referred to as ground-coupled heat pumps, water-source heat pumps, or a Geoexchange system.

To best understand the thermodynamic cycles of heat pumps, the ideal vapor-compression refrigeration cycle is normally used to explain the heat transfer processes of air conditioners as well as heat pumps. The cycle consists of the circulating refrigerant and four components including the compressor, condenser, expansion device, and evaporator as shown in Figure 2.1(a). The cycle considers heat transfer in the condenser and evaporator without pressure losses. The components are connected by piping that has neither pressure loss nor heat transfer with the surroundings. The cycle has four processes: isentropic compression in the compressor $(1-2)$, constant-pressure heat rejection in the condenser $(2-3)$, throttling in the expansion device

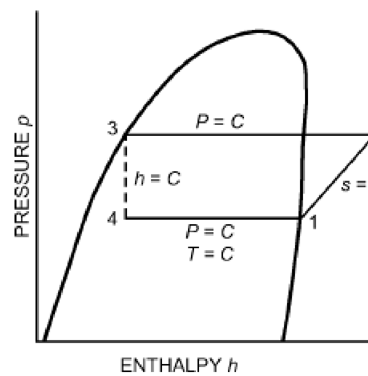
$(3-4)$, and constant-pressure heat absorption in the evaporator $(4-1)$. The changes in thermodynamic properties of each process occurring in the cycle are commonly illustrated in the absolute temperature to entropy $(T-S)$, and the pressure to enthalpy $(p-h)$ diagrams, Figure 2.1(b) and Figure 2.1(c).



(a) An ideal vapor-compression refrigeration cycle components diagram



(b) Absolute temperature to entropy $(T-S)$ diagram



(c) Pressure to enthalpy $(p-h)$ diagram

Figure 2.1 - Theoretical single-stage vapor compression refrigeration cycle from (ASHRAE 2005)

In the first process of an ideal vapor-compression refrigeration cycle, the refrigerant leaves the evaporator and enters the compressor at $state$ 1 as a low pressure, low temperature saturated vapor. The reversible and adiabatic (isentropic) compression process in the compressor $(1-2)$

raises the refrigerant vapor pressure by applying the mechanical work to the compressor. The process also increases the refrigerant temperature and enthalpy. As a result, the refrigerant vapor becomes the superheat vapor at $state$ 2 with a temperature above the surrounding medium. From the first law of thermodynamics, the law of the conservation of energy, the energy balance equation can be derived as in Equation 2.1, where $\dot{W}$ is the work applied to the compressor, h is an enthalpy at each state of the refrigerant, and $\dot{m}$ is the mass flow rate of the refrigerant. In this equation, the negative quantity expresses that the work is done on the refrigerant by the compressor.

$$ {}_1\dot{W}_2 = -(h_2 - h_1)\dot{m} \tag{2.1} $$

The second process occurring in the cycle is the constant-pressure heat rejection process in the condenser $(2 - 3)$. The refrigerant leaves the compressor and enters the condenser as a high temperature, high pressure superheated vapor. It is first desuperheated and then condensed at a constant pressure by releasing heat to the surroundings. Although, the refrigerant pressure remains constants, the enthalpy is lowered by decreasing the refrigerant's internal energy. For this process, the energy balance equation can be derived as in Equation 2.2, where an additional term $\dot{Q}$ is the heat flow rate occurring in the condenser.

$$ {}_2\dot{Q}_3 = -(h_2 - h_3)\dot{m} \tag{2.2} $$

In the third process, the refrigerant leaves the condenser and enters the expansion device as a high pressure, medium temperature saturated liquid. The throttling process in the expansion device $(3 - 4)$ reduces the refrigerant pressure by expanding the refrigerant liquid irreversibly and adiabatically (constant enthalpy) which makes its temperature drop below the conditioned space. Since this device is fundamentally a flow restrictor that prevents both work and any significant amount of heat to transfer, there is only a slight change in entropies from $state$ 3 to $state$ 4 which is assumed to be zero as derived in Equation 2.3.

$$ h_3 = h_4 \tag{2.3} $$

The forth process is the constant-pressure heat absorption process in the evaporator $(4 - 1)$. In this process, the refrigerant leaves the expansion valve and enters the evaporator as a low pressure, low temperature low quality vapor. The refrigerant vapor is evaporated reversibly and adiabatically at constant pressure to the saturated state at $state$ 1 by absorbing heat from the refrigerated space. Then the refrigerant leaves the evaporator as a low pressure, low temperature saturated vapor and reenters the compressor, completing the cycle. The energy balance equation can be derived as in Equation 2.4.

$$_4\dot{Q}_1 = (h_1 - h_4)\dot{m} \tag{2.4}$$

The heat transfer processes in the heating cycle of heat pumps is simply the reverse cycle of the ideal vapor-compression refrigeration cycle described above. In the heating cycle, heat pumps draw heat from the low temperature surroundings using the evaporator, and discharge heat to the high temperature conditioned space using the condenser. This reverse cycle can operate without rearranging the refrigeration cycle components. There are two basic methods used to reverse heat pump cycles between cooling and heating. In air-source heat pumps, the air dampers are used to control the outside air to flow through the evaporator in winter months, and allow the outside air to pass the condenser in summer months. Another method is used in the water-source heat pumps. The refrigerant fluid circulation can be reversed by using the reversing valve to control the fluid to either flow to the evaporator or the condenser. This arrangement also requires a suction-line accumulator to protect the compressor from refrigerant floodback during the heating-cooling changeover cycles. As a result, in the same mechanical system, heat pumps can provide both heating and cooling capacities for the conditioned space.

Since heat pumps are mechanical systems that require work input to produce the cooling or heating output, it is important to understand the efficiency of energy used to operate the systems. The energy efficiency performances of heat pumps can be determined by the coefficient of performance (COP), the energy efficiency ratio (EER), and the seasonal energy efficiency ratio (SEER).

The coefficient of performance (COP) is the ratio of the rate of heat removal from or delivered to the conditioned space to the rate of energy input, in consistent units, for a complete operating refrigerating or heat pump plant, or some specific portion of that plant under designed operating conditions. According to Equations 2.1-2.4, the COPs of refrigeration and heat pump cycles can be derived as expressed in Equation 2.5 and 2.6, where COP_R is the COP of the refrigeration cycle, COP_{HP} is the COP of the heat pump cycle, $\dot{Q}_L$ is the rate of heat removed from the conditioned space (Q_L) at temperature T_L, $\dot{Q}_H$ is the rate of heat delivered to the conditioned space (Q_H) at temperature T_H, and $\dot{W}_{net,in}$ is the rate of energy input.

$$COP_R = \frac{\text{Rate of heat removal}}{\text{Rate of energy input}} = \frac{\dot{Q}_L}{\dot{W}_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1} \tag{2.5}$$

$$COP_{HP} = \frac{\text{Rate of heat delivered}}{\text{Rate of energy input}} = \frac{\dot{Q}_H}{\dot{W}_{net,in}} = \frac{h_2 - h_3}{h_2 - h_1}$$
$$= \frac{\dot{Q}_L}{\dot{W}_{net,in}} + 1 = COP_R + 1 \tag{2.6}$$

From equations 2.5 and 2.6, COP_R is a positive quantity when the refrigeration cycle is in operation, thus the minimum quantity of COP_{HP} is, in most cases, greater than 1 as also graphically illustrated in Figure 2.2. The COP_{HP} of heat pumps ensure that, in a worst case scenario, the energy efficiency performance of heat pumps is still better than the conventional electric resistance heaters which normally produce the same amount of heating energy as the amount of work input. The COPs are used in systems comparison, where the higher COPs are the more energy efficient of a system.

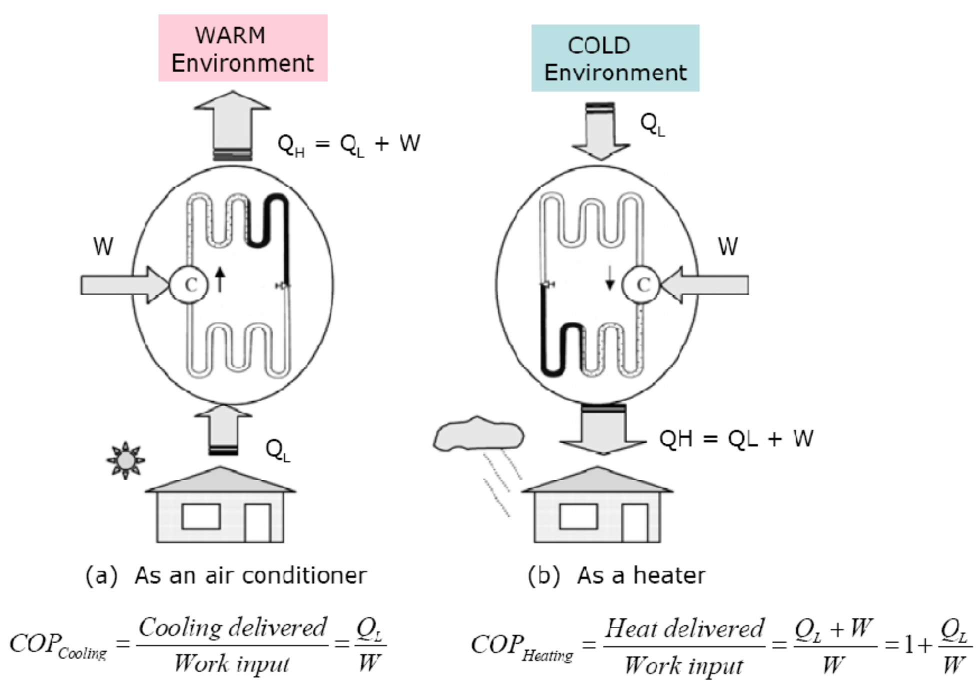


Figure 2.2 - The $COP_{Cooling}$ and $COP_{Heating}$ of an air-to-air heat pump from (Masters 2007).

In the United States, the performance of air conditioners is usually expressed in terms of the energy efficiency ratio (EER). The EER is the ratio of net cooling capacity in Btu/h to the total rate of electric input in Watts (W) under designed operation conditions, Equation 2.7. The greater the EER, the less electricity is consumed in an air conditioner for the same cooling output. Given that 1W = 3.412 Btu/h, the EER can be derived in terms of the EER relationship as shown in Equation 2.8. The EER can be applied to determine the energy efficiency of heat pumps by replacing the net cooling capacity and electric power consumed by air conditioners with the net heating capacity and electric power input of heat pumps.

$$EER = \frac{\text{Net cooling capacity } (Btu/h)}{\text{Total rate of electric input in } (W)} \tag{2.7}$$

$$EER = \frac{\dot{Q}_L(Btu/h)}{\dot{W}_{net,in}(Btu/h)} \times \frac{3.142\ Btu/h}{1\,W} = 3.412COP_R \tag{2.8}$$

In addition to the EER, the seasonal energy efficiency ratio (SEER) is used to determine the energy efficiency of air conditioners over a period of time. The SEER is a ratio of the total cooling output of an air conditioner during its normal annual usage period for cooling, in Btu, to the total electric energy input, in Watt-hour (Wh), during the same cooling period, Equation 2.9. The greater SEER indicates less electricity consumption during the considered season. Similar to the EER, the SEER can be applied to determine the seasonal energy efficiency of heat pumps during the heating season which is known as the heating season performance factor (HSPF), Equation 2.10 (Masters 2007).

$$SEER = \frac{\text{Total seasonal cooling output } (Btu)}{\text{Total electric energy input of the cooling season} (Wh)} \tag{2.9}$$

$$HSPF = \frac{\text{Total seasonal heating output } (Btu)}{\text{Total electric energy input of the heatling season} (Wh)} \tag{2.10}$$

However, in reality, the actual performances of refrigeration and heat pump cycles are different from the ideal cycles. This is because the efficiency is lowered due to fluid friction, which is the cause of pressure drops in the systems, and heat transfer from or to the surroundings via the delivery systems and system components

One common problem caused for air-air heat pumps is frost built up on the evaporator coil when the surrounding temperature is close to or below the freezing-point temperature. In freezing conditions, an air-air heat pump requires a defrost cycle, which simply shuts down the evaporator fan and turns the heat pump back to the refrigeration cycle. When the heat pump is in the refrigeration cycle, the evaporator coil becomes the condenser coil, where a high temperature, high pressure superheated vapor from the compressor can melt the undesirable frost. During the defrost cycle, auxiliary heating systems are necessary to maintain the space heating setpoint. Auxiliary heaters can be electric resistance, or fuel combustion type. These additional heating systems reduce the energy efficiency, economic, and environmental benefits of air-air heat pumps, and make heat pumps less useful when compared with certain other heating systems in cold weather regions.

Geothermal heat pumps typically do not experience frost built up and do not require the use of auxiliary heating systems due to the more stable temperature of the ground or water sources. The ground and water sources have large thermal energy storage capacities or thermal mass which can be defined as thermal energy reservoirs. Referring to the second law of thermodynamics, a reservoir is a hypothetical body with a relatively large thermal energy capacity, a product of mass and specific heat, which can absorb or release finite amounts of heat (Çengel and Boles 2008).

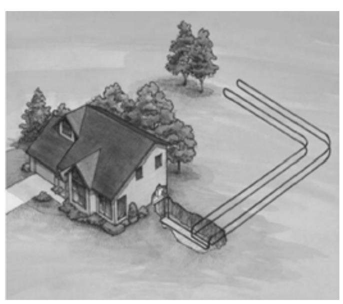
In the United States, below the surface the ground temperature is relatively constant. It ranges from 40 ˚F to 70 ˚F depended on the geographical location and soil properties. The heat transfer process of the soil is transient and can be determined by the soil thermal diffusivity. Thermal diffusivity is the ratio of thermal conductivity to the product of density and specific heat. Among all soil contents, moisture is the most influential factor when determining the thermal conductivity.

For relatively large water sources, the temperature may range from 40 ˚F in northern to 70 ˚F in southern regions of the United States. The water sources used in GSHP systems include, but are not limit to, well, surface, and waste water. The heat transfer process is not only influenced by the heat capacity of the water but also the depth of the body of water. The source water must be deep enough to prevent the temperature equalization across the surface and the bottom of the water. For example, in residential applications, the water body should be larger than 1 acre surface and 10 feet depth or more.

According to the Geothermal Heat Pump Consortium (1997), ground sources can absorb about half of the clean and renewable energy from the sun. In heating operation of GSHP systems, this amount of absorbed solar energy can supply at least three units per one unit of electric energy consumed by the system. In cooling operation, the heat from the systems can be released to the surrounding ground sources that equilibrate with the atmosphere. Therefore, GSHP systems are not only one of the possible solutions for eliminating frost build-up and the use of auxiliary heating systems, but are also one of the most energy efficient and environmental friendly systems available.

In practice, the ground-source heat pump is a common technical term used for the geothermal heat pump. The ground-source heat pump (GSHP) systems may include the ground-coupled heat pump (GCHP), groundwater heat pump (GWHP), and surface water heat pump (SWHP) systems (ASHRAE 2003). These systems share similarities in applications and installation methods, but are different in the use of thermal energy reservoirs including ground, groundwater, and surface water sources. Related to the systems used in the case study buildings; KnowledgeWorks I and II, this research has focused only on the GCHP and SWHP.


(a)


(b)


(c)


(d)

Figure 2.3 - Pictures (a) and (b) are the GCHPs, picture c is the GWHP, and picture d is the SWHP from (WaterFurnace 2006).

Ground-coupled heat pump (GCHP) systems use the ground as the thermal energy reservoir. A GCHP system is composed of a reversible compression cycle connected with a closed ground heat exchanger buried underground. The systems include the direct-expansion (DX) and the ground-coupled GCHP systems. The DX GCHP is the refrigerant-air/water heat pump that uses an underground copper piping network, in which the refrigerant is circulated, as the buried heat exchanger unit. This system is rarely used because it has high maintenance costs due to leak repair and requires large quantities of the refrigerant.

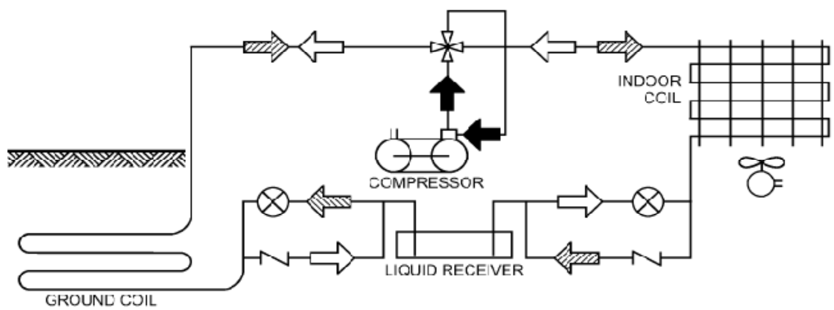


Figure 2.4 - The diagram of the direct-expansion (DX) GCHP from (ASHRAE 2004).

Instead of coupling the refrigerant with underground soil, the GCHP is typically a water-air/water heat pump that circulates water or antifreeze-water solution through the closed thermoplastic piping loops buried underground. This system is also known as closed-loop GSHP system. The thermoplastic pipe used in GCHP applications is usually made from the high-density polyethylene (HDPE), which can be installed in vertical or horizontal configurations depended on available resources. In the heat exchanger, the solutions can directly exchange heat with a reversible compression cycle, or indirectly through the transitional refrigerant usually found in a liquid-refrigerant heat exchanger. In comparison, the GCHP system is more wildly used than the DX GCHP because it requires less maintenance cost, and less quantities of the refrigerant.

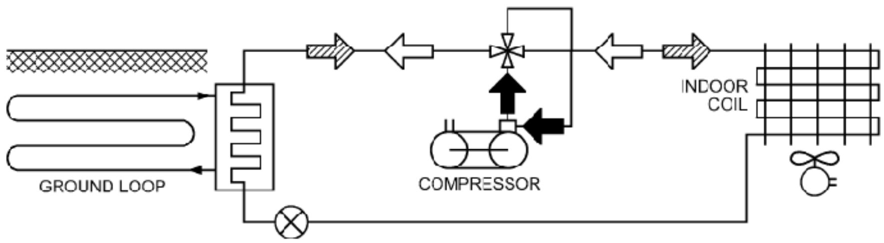


Figure 2.5 - The diagram of the ground-couple heat pump from (ASHRAE 2004).

The vertical-loop GCHPs are usually composed of two HDPE tubes with tube diameters ranging from 0.75 to 1.5 inches. The tubes are installed in a vertical borehole filled with a solid medium such as bentonite or sand mixtures to prevent tubes from damaging and the lost of heat transfer capacities due to changes in surrounding soil conditions. The bottom-end of the tubes is heat fused or clamped by a stainless steel clamp to a close return U-bend fitting. The boreholes can be configured either in series as a single row, or in the parallel as a grid pattern. Bore depths can range from 60 to 600 feet depended on the designed heating or cooling capacity, and soil contents of the site. A recommended spacing between the bores is at least 20 feet in distance for the grid pattern and can be less for a single row. The vertical-loop GCHP systems can yield the most efficient GCHP system. They also require less ground area and the least amount of pipe and pumping energy. However, the equipment needed to drill the borehole is very expensive, and the contractors, who can perform such installation, are very limited.

In the horizontal-loop GCHPs, the HDPE tubes are laid horizontally 3-6 feet below the ground surface, where the soil temperature is not significantly influenced by the ambient air temperature. Frequently, the horizontal loop GCHPs is used in three different settings including the single pipe, multiple pipe, and spiral pipe installations. The single pipe system employs a single HDPE pipe line installed 4 feet below the surface. This system occupies the largest area for the installation, which could be a limiting factor in many building construction sites. The system occupied area can be reduced for the multiple and spiral pipe installations. The multiple pipe system consists of two, four, or six small-diameter HDPE tubes buried in a trench or multiple trenches. A similar approach to the multiple pipe system, the spiral pipe system applies the spiral-bended HDPE tube laid in an underground trench or trenches. Both systems can reduce about half or more than half of the system occupied area, but they still require the same length of pipe used in the single pipe configuration based on required heating or cooling capacity. The horizontal-loop GCHP systems are less expensive than the vertical-loop systems. On the downside, these systems require more installation space and have lower efficiency than vertical-loop systems.

Surface water source heat pump (SWHP) systems include the water-to-water, or water-air heat pumps that use a body of water such as a pond or lake as the thermal energy reservoir. The antifreeze and water solution can be circulated directly through the heat pumps, or indirectly through intermediate fluids. The system can be found in two different configurations: open-loop and close-loop. The open-loop system applies the same principles used in the cooling tower excluding the needs for fan operating energy and frequent maintenance. This system is more applicable in warm climate zones rather than cold. This is because the system may not be able to serve as a heat source when the water temperature is less than 45 ˚F. Therefore, the open-loop SWHP system gains less interest than other systems.

For the close-loop SWHPs, the systems apply the same principles as used in GCHPs, but with HDPE pipe with ultraviolet (UV) protection. They may be structured in a variety of configurations, and are sunken to the bottom of the water body. The water depth is the critical factor of the system performance. The depth should be deeper than 10 feet, and the HDPE should be installed at least 1 foot above the bottom. This system is relatively low cost when compared to the GCHPs. The system requires low pumping-energy, infrequent maintenance, and has low operating costs. Nevertheless, underwater coil erosion can occur in public lakes. Also, the variations of the surface temperature can cause unwanted variations of the system efficiency and capacity, but still less than for air-air heat pumps.

Although GSHPs consume less energy and produce less environmental pollution than other alternatives, the growth of their applications has been relatively slow since they were first introduced to the market. The primary barrier to growth has been the very high initial cost of the system. Only in the past few decades has the GSHP system become more attractive because of the rapid ascent of global energy cost and awareness of the environmental crises. The crises have caused increasing energy prices and growing concerns for the environmental impacts from human activities.

There are a number of investigations related to GSHPs that have been conducted in the past thirty years. The studies showed the possibility that the initial costs of the GSHPs can be offset by the low operating costs. Also, the initial costs can be compensated by low maintenance cost because these systems typically have only a small number of moving parts and the heat pump units are usually installed inside the building and most GSHP system components are buried underground or sunken underwater which protect them from environmental damage. In addition, the GSHPs do not directly use combustion fuels, and require no or little refrigerant fluids. These features can minimize the environmental impacts.

One study by Stien, et al. (2006) focused on the installed system at the Wildlife Center of Virginia at Waynesboro. This research was conducted in the 1990’s, for the 5,700 square feet Wildlife teaching – research hospital. The facility is served by four GSHPs connected to 11,350 feet of underground spiral pipe, also called slinky, buried in about 2,500 feet of trench. The energy simulation studies show that the annual energy consumption of the GSHPs can save up to 47% of energy consumption compared to the air-air heat pumps and electrical water heaters.

Another case study by Stien, et al. (2006) was the GSHPs used in Daniel Boone High School near Johnson City, Tennessee. The study focused on the advantages of GSHP Systems in a retrofit project. The 160,000 square feet school replaced its two-pipe chilled water system and electric resistance heaters with a vertical-loop GCHP. The project won a 1998 ASHRAE Technology Award.

More recently research was conducted by the U.S. Department of Energy (DOE) under the Federal Energy Management Program (FEMP). The research by FEMP (2001) reinforced previous work on the energy performance and cost benefits of GSHP systems. This research was based on a baseline computer model of a typical two-story office building located in Washington, DC. The prototype has the total floor area of 25,000 square feet with the occupant density of one person per 200 square feet. The prototype building gross wall area has 40% window coverage, and all of the designs and specifications meet the requirements of ASHRAE standard 90.1-99.

The baseline model of the prototype building was developed for the DOE2 building energy simulation software package. DOE2 was developed by James J. Hirsch & Associates (JJH) in collaboration with Lawrence Berkeley National Laboratory (LBNL) supported by the U.S. Department of Energy. DOE2 is considered one of the most validated programs to perform building energy performance simulations both in federal and industrial building projects. The model is used to estimate the quantities of energy consumed by four different HVAC systems including air-source heat pump, gas furnace with air-source air conditioner, recommended efficiency level GSHP, and best available GSHP applied to the prototype building. Then the results from DOE2 are used to evaluate the cost benefits of the three alternative systems when compared to the air-source heat pump base case by using the building life cycle cost (BLCC) software, developed by FEMP. The simulation studies show that for this building the recommended and the best available GCHPs will be cost-effective when their prices are less than 17,000 and 23,000 dollars above the cost of air-source heat pump base case.

The FEMP conducted another study based on a real world retrofit project at Fort Polk, Louisiana. In this study, 4,000 units of the military family housing which occupied 5.6 million square feet, had used various types of HVAC systems including air-air heat pumps, and gas furnaces combined with air conditioners that provided 6,600 tons of total cooling capacity. The existing HVAC systems were replaced by the GSHPs integrated with the hot gas desuperheaters to provide domestic hot water. Other energy efficiency improvements also added to each unit such as compact fluorescent lamps (CFLs), low-flow shower heads, and attic insulation. All together, the approximation of total retrofit cost was about 19 million dollars.

The results of the study show that the retrofit can save up to 25.6 million kWh of electric energy or about 33% of the present annual electric energy consumption. For the peak demand, the retrofit can reduce more than 6.5 MW, which is about 43% of the previous peak demand. The retrofit also eliminated the average 260,000 therms of natural gas per year. Moreover, it was estimated that the Army could save about 77% of the money used to invest on the previous HVAC system’s maintenance. As a result, this retrofit project could save approximately 3 million dollars per year with a simple cost recovery of only a few years.

In summary, the thermodynamic principles and the system engineering of geothermal heat pumps can save operating energy by moving heat rather than creating it by burning fossil fuels.

The use of alternative natural thermal energy reservoirs such as the ground and water sources helps improve the system's energy efficiency, and reduce the use of CFC content refrigerants. These are reasons that GSHPs are becoming a popular HVAC technology. In the past, GSHPs seem to have a lack of interest because of the high initial cost and limited knowledge by designers and installers. Under the recent global energy and environmental situations, GSHPs have rapidly gained more interest than before. Several studies claimed that previous barriers can be offset by the GSHPs high energy efficiency features and low additional costs. Since the system performance often relies on natural resources available in different locations, it is important to understand the strengths and weaknesses of each GSHP alternative.

2.2 Life-cycle Cost Analysis and the Economics of Geothermal Heat Pumps

Geothermal heat pumps are like other energy conserving technologies such as high efficiency windows, photovoltaics (PV), and wind turbines in that they usually have high initial costs, and such costs are claimed to be offset by the energy savings. There are several economic measures that are used to evaluate the feasibilities of investments on building projects such as return on investment (ROI), and payback period. These measures are capable of, and still appropriate for, comparing investment alternatives, but unable to compare the cost benefits of energy efficiency investments. This is because these measures omit either the future costs appearing after the first investment or the value of money over a project life-time.

Investments in energy efficiency require economic evaluation procedures that take into account future costs including, but not limited to, operating costs, maintenance costs, and repair costs (OM&R). They also require an evaluation method that considers "time value of money" such as inflation and opportunity cost. These parameters are important to determine how the lower future costs can compensate for the higher initial costs of the alternatives. As a result, there are a number of organizations such as the DOE's Federal Energy Management Program (FEMP), the American Society for Testing and Materials (ASTM), and the National Institute of Standards and Technology (NIST) that have developed standards for the economic evaluation for energy efficiency investments. Life-cycle cost analysis (LCCA) is one of the most recognized procedures available during the past few decades. In addition to LCCA, commonly used economic measures have been developed to include the future costs and time value of money. According to Fuller (2007), the supplementary economic measures include net savings (NS, or net benefits), saving to investment ratio (SIR, or saving benefits to cost ratio), adjusted internal rate of return (IRR), simple payback (SPB), and discounted payback (DPB).

Economic evaluation procedures can be used for three purposes (S. Fuller 2007). First is for determining the cost effectiveness of alternatives where the procedures are the LCCA and NS. Second is to rank the alternatives in which the methods are the SIR and AIRR. The third purpose is for screening the alternatives where the procedure can be either SPB or DPB. Related to the intents of this research, these economic evaluation methods were used to assess the cost benefits of the GSHPs system when compared to a conventional VAV HVAC system. The economic evaluations used in this research follow the FEMP/NIST procedures since they are validated and acceptable for both government and industrial projects. This research uses the LCCA method to determine the cost effectiveness of alternatives because it captures and includes both first and future costs over time, while the NS captures all costs during the project lifetime but focuses on the differences between operational savings and capital investment costs between a base case and a single alternative. For alternative ranking, the SIR is used since it allows more flexibility in alternative comparison between the savings and investments when compared to the AIRR which uses discount rate as the comparison basis. Furthermore, the DPB is used to screen the alternatives because it considers the time value of money which is omitted in the SPB approach.

Life-cycle cost analysis (LCCA) is an economic evaluation method that has been developed to perform economic assessments for energy efficiency investments. The LCCA considers the cost-effectiveness between alternatives by comparing their life-cycle costs, where the lowest life-cycle cost determines the highest cost-effectiveness. Life-cycle cost (LCC) is the overall life-time costs of an energy efficient alternative consisting of two cost categories: investment related costs and operating costs. The costs that related to investments include initial investment costs (i.e., land acquisition, and construction or installation costs), capital replacement costs, and residual values. The operating costs include energy costs, water costs, and the operation, maintenance, and repair costs (OM&R). In addition to investment related and operating costs, there are other costs that are external to the LCCA but can be added, if they have significant effects on the evaluations. The other costs include the finance charges (i.e., loan interest payments, and contract costs such as for the energy saving performance contract (ESPC), or utility energy services contract (UESC)), and the non-monetary benefits or costs. It is also important that the costs considered as future costs are discounted to their present values (PV). The summary of the LCC can be demonstrated in Equation 2.11.

$$
\begin{aligned}
\text{Life-cycle costs } (LCC) \;=\;& \text{Initial investment costs}^{a} \text{ (I)} \\
&+ \text{Present value (PV) capital replacement costs (Repl)} \\
&- \text{PV residual values (Res)} \\
&+ \text{PV of energy costs (E)} \\
&+ \text{PV of water costs (W)} \\
&+ \text{PV OM\&R} \\
&+ \text{PV of other costs (O)}
\end{aligned} \tag{2.11}
$$

a. The initial investment costs can be discounted to the present values, if the costs are not incurred at the base date.

From Equation 2.11, the present values are the costs of the LCC parameters at the time when an evaluation is made, discounted from the time they will be expensed in the future. In FEMP/NIST method, the present values are categorized into two categories of costs: goods or services, and energy. The method considers that the goods or services costs have the same inflation rate which is different from that of the energy costs. The first step in estimating the PVs of goods or services is to identify their future costs. The future cost (F_t) of a particular item can be identified by the calculation method in Equation 2.12, where P_0 is the present value of goods or services, i is the assumed rate of general inflation, and t is the assumed future year in the calculation. It is important to note that the FEMP/NIST method limits the maximum study period to 25 years.

$$F_t = P_0 \times (1 + i)^t \tag{2.12}$$

After the first cost was estimated, the next step is to determine the discount rate used in the PV calculations. The discount rates can be defined as the nominal discount rate and real discount rate. The nominal discount rate is the minimum rate of return on an alternative investment for borrowed capital such as the loan rate (Addison 1999). The nominal discount rate is considered sensible for an investment if it is greater than the inflation rate. In the FEMP/NIST method, the discount rate is required to be real rather than nominal, where the inflation is permitted to factor out. As a result, the real discount rate (d) for an assumed nominal discount rate (D) and inflation rate (i) can be calculated by the procedure in Equation 2.13.

$$d = \frac{1 - D}{1 - i} - 1 \tag{2.13}$$

From Equation 2.12 and 2.13, the present value (PV) equation of goods or services for a future cost (F_t) and real discount rate (d) during the study period (t) can be derived as expressed in Equation 2.14, where the real discount rate used in the calculation is annually revised by the FEMP.

$$PV = F_t \times \frac{1}{(1 + d)^t} \tag{2.14}$$

For the present value of energy costs, the FEMP/NIST LCCA method uses the annual energy price index which is revised yearly and available through the U.S. Department of Energy. This index is calculated from a specific discount rate for energy costs recognized as the energy escalation rate. The present value of energy costs is simply the product of the energy price at the base date and the energy price index. Since the energy escalation rate is real, the present value of energy costs is constant and can be used in the LCCA and the supplemental economic measures that require the energy costs to be discounted.

In addition to the LCCA, the saving to investment ratio and the discounted payback are the supplemental economic measures used in this research. The saving to investment ratio (SIR) is the ratio of the operational savings to the difference of capital investment costs. The SIR is used for evaluating an individual alternative against a base case project solution, where an SIR greater than one is considered economically acceptable. For multiple alternatives, each single alternative is evaluated against the base case to obtain its particular SIR value. The SIRs can be used for ranking the project alternatives to determine which has the highest saving potential for a limited investment fund. It is important to recognize that the highest SIR alternative is not necessarily the most cost effective one, since it might not be an alternative that has the lowest LCC. The SIR equation is illustrated in Equation 2.15, where all related costs used in the equation are discounted to their present values.

$$SIR_{A:BC} = \frac{Operational\ savings}{Captital\ investment\ costs} = \frac{\Delta E + \Delta W + \Delta OM\&R}{\Delta I_0 + \Delta Repl - \Delta Res} \tag{2.15}$$

Where

$SIR_{A:BC}$ = Saving to investment ratio, computed for the alternative relative to the base case

ΔE = Saving in energy costs attributable to the alternative $(E_{BC} - E_A)$

ΔW = Saving in water costs attributable to the alternative $(W_{BC} - W_A)$

$\Delta OM\&R$ = Difference in $OM\&R$ costs $(OM\&R_{BC} - OM\&R_A)$

ΔI_0 = Additional initial investment costs required for the alternative relative to the base case $(I_A - I_{BC})$

$\Delta Repl$ = Difference in replacement costs $(Repl_A - Repl_{BC})$

ΔRes = Difference in residual value $(Res_A - Res_{BC})$

The discounted payback (DPB) is a supplementary economic measure used for determining the time required to recover the initial investment costs. This payback method is relevant to use in energy efficient investment projects because it allows the annual cash flows to be discounted to present value before collecting them as savings and costs. The method considers that a project is cost effective when its DPB is shorter than the length of the service period used in an analysis. In practice, the DPB is usually compared with the chosen time period which may be shorter than the expected service period. Hence, it is possible that the future costs such as OM&R occurring after the considered payback period cannot compensate with the investment related cost such as initial investment costs. This may cause reduction in the project cost effectiveness. The DPB can be used for screening the project alternatives rather than for determining the cost effectiveness and rank of the alternatives. The DPB calculation method is summarized in Equation (2.16)

$$\sum_{t=1}^{y} \frac{[\Delta E_t + \Delta W_t + \Delta OM\&R_t - \Delta Repl_t - \Delta Res_t]}{(1+d)^t} \geq \Delta I_0 \tag{2.16}$$

Where

ΔE_t = Saving in energy costs per year t $(E_{BC} - E_A)_t$

ΔW_t = Saving in water costs per year t $(W_{BC} - W_A)_t$

$\Delta OM\&R_t$ = Difference in $OM\&R$ costs per year t $(OM\&R_{BC} - OM\&R_A)_t$

$\Delta Repl_t$ = Difference in replacement costs per year t $(Repl_A - Repl_{BC})_t$

ΔRes_t = Difference in residual value per year t $(Repl_A - Repl_{BC})_t$; this value is usually equal to zero in all years excluding in the last year

d = Discount rate

ΔI_0 = Additional initial investment costs required for the alternative relative to the base case $(I_A - I_{BC})$

Since economic evaluations of energy efficiency investment projects have to deal with future costs, it is possible that some of the estimated costs can be mistaken from the actual costs at the time of occurrences. Therefore, the evaluation methods, usually the LCCA, require procedures that are capable of treating this uncertain. The procedures can be either deterministic or probabilistic. The deterministic approaches focus on how the changes of a key-value or a combination of key-values affect the evaluation outcomes. Conversely, the probabilistic approaches try to determine all possible alternative outcomes that can occur in a risk investment by accumulating a large number of outcomes to a complex probability study. In comparison, no one is better than another because they are suitable for the different types of projects. However, the deterministic approaches are preferred in the FEMP/NIST LCCA methods because they use the same information as used in the LCCA, and do not require sophisticated tools to perform the evaluations.

The deterministic approaches used in the FEMP/NIST LCCA are the sensitivity and breakeven analyses. The sensitivity analysis studies the effects of a single-value input or multiple-value inputs on the economic evaluation outcomes. The analysis can be performed by identifying critical inputs, estimating a range of outcomes, and testing possible alternative scenarios. The first process is used for identifying which inputs are crucial to the outcomes. This process can be done changing each input by a certain percentage then investigating the inputs that make the critical changes on the outcomes. After the critical parameters are known, the next step takes them to estimate the lower and upper bounds of an individual parameter. In this process a key-value such as energy costs is adjusted to several levels of decreasing and increasing percentages. Then the adjusted values are evaluated with their effects on the relative outcomes. This process helps determine a range that the estimated costs are still economically acceptable. The third process is used for extending the sensitivity analysis to test various possible alternative scenarios. This process uses the same analysis methods as in processes one and two, and tests one combination of key-values at a time. It is important that some combinations might not occur in real world situations, which can mislead the economic decision makings. A limitation of the sensitivity analysis is that it does not indicate information about the distinctions between the different alternatives.

The breakeven analysis is used for determining the minimum and maximum values necessary to make the project breakeven. This analysis can be performed by constructing an equation that the operational costs and investment related costs are equal for a given alternative, as shown in Equation 2.16, where all costs are in present value terms. Then the next step is to specify the values of all inputs excluding the breakeven variable. Afterward, the equation is solved algebraically to obtain the breakeven cost of the element of interest. This method can be used for benchmarking and comparing against the estimated performance of uncertainty variables. The analysis is also used to determine the bounds of the nonmonetary benefits that are usually external to the LCCA. The breakeven analysis has the same limitation as sensitivity analysis that it cannot provide information about the different possible alternatives.

$$S = \Delta C$$

$$[\Delta E + \Delta OM\&R + \Delta W] = [\Delta I_0 + \Delta Repl - \Delta Res] \tag{2.17}$$

Where

S	= Operational savings for the alternative relative to the base case
ΔC	= Investment-related additional costs for the alternative relative to the base case
ΔE	= Saving in energy costs in year t $(E_{BC} - E)_t$
$\Delta OM\&R$	= Difference in $OM\&R$ costs $(OM\&R_{BC} - OM\&R_A)$
ΔW	= Saving in water costs in year t $(W - W_A)_t$
ΔI_0	= Additional initial investment costs required for the alternative relative to the base case $(I_A - I_{BC})$
$\Delta Repl$	= Difference in replacement costs $(Repl_A - Repl_{BC})$
ΔRes	= Difference in residual values $(Repl - Repl)_t$

Economic evaluation methods for energy efficiency projects such as LCCA and other supplementary economic measures require a significant amount of cost-related information. The DOE's FEMP has useful hardcopy worksheets and a computational software package, also known as FEMP building life-cycle cost analysis (BLCC) that can help perform the economic evaluations. For more detailed information concerning economic assessments, the NIST handbook 135 is recommended as a useful resource (Fuller and Petersen 1995).

2.3 Related Studies on Energy Performance and Economic Evaluations of Geothermal Heat Pumps

Using Existing Standards to Compare Energy Consumption of Ground Source Heat Pump with Conventional Equipment

In his paper titled *Using Existing Standards to Compare Energy Consumption of Ground Source Heat Pump with Conventional Equipment*, Kavanaugh (1992) investigates the energy use and demand characteristics of ground-source heat pumps (GSHPs). The GSHPs are compared to the conventional equipment based on the existing standards used to rate unitary heating and cooling equipment. The conventional equipment includes single speed heat pumps (SSHPs), furnace with DX air cooling (F/AC), and variable-speed heat pumps (VSHPs). There are five standards used to rate the conventional equipment and GSHPs. The *ASHRAE 103-1988, Methods of Testing for Heating Seasonal Efficiency of Central Furnaces and Boiler* is used to rate the F/AC. The Air conditioning and Refrigeration Institute (ARI) standard 210/240-89 is used to rate the SSHPs and VSHPs. The ARI 320-86, ARI 325-85, and ARI 330-90 is used to rate the GSHPs.

The paper points out that using only the single-measure estimates for the energy efficiency rating expressed in the standards such as the annual fuel utilization efficiency (AFUE), the seasonal performance factor (SEER), and the heating seasonal performance factor (HSPF) are inaccurate for equipment comparison. These measures fail to incorporate weather-driven energy calculations which can account for specific system operational characteristics. Therefore, the author suggests that the bin calculation method is a better procedure for calculating seasonal energy and demand requirements of heating and cooling equipment than the single-measure estimate.

The bin analysis procedure summarizes the total number of hours that fall in a bin of 5°F outside air temperature (OAT) range. For example, in observing a given year, a bin which OAT ranges from 73°F to 77°F (5°F difference) may have a total of 839 hours falling in the bin. The bin hours are used to determine the total annual heating and cooling loads and capacities. Then, the total heating and cooling capacities are used for estimating the annual energy use and demand. This bin calculation method can apply to the equipment that use outside air as a heat source or heat sink. For GSHPs, the author uses a computer program to estimate the annual energy use and demand of the system. The program has been developed to simulate the ground coil as a function of the ground coil design, thermal properties of the soil, and building load pattern.

In this study, the author used a bin analysis to compare the energy consumption of four types of heating and cooling equipment in two different climates: Atlanta and Chicago. The weather data are used to compare F/AC, SSHP, VSHP, and GSHP equipment. The Atlanta home heating and cooling loads are 36,000 Btu/h (10.6 kW), and the Chicago home's load is 48,000 Btu/h (14.1 kW) for heating and 30,000 Btu/h (8.8 kW) for cooling. Table 2.1 summarizes the rating, energy use, and demand of the conventional equipment and GSHPs. Table 2.2 shows the cost comparison of conventional equipment with ground-source heat pump system.

Table 2.1 - Rating, energy use, and demand of conventional equipment and ground-source heat pumps from Table 4, page 604 (Kavanaugh 1992).

	Atlanta				Chicago			
	SSHP	F/AC	VSHP	GSHP	SSHP	F/AC	VSHP	GSHP
Rating								
SEER	9.60	9.60	15.2		9.70	9.30	15.1	
HSPF	7.45		8.80		7.25		9.4	
AFUE		81%				81%		
EER-330				15.2				14.9
COP-330				3.40				3.40
Cooling Energy and Demand								
kWh	3740	3740	2627	2556	1902	1773	914	1053
kW-peak	4.05	4.05	4.03	2.77	5.12	3.23	4.12	2.92
Heating Energy and Demand								
kWh-hp	6201		4437	3498	12891		8815	7718
kWh-aux	536		472	13	3707		3759	352
kWh-tot	6737	529	4909	3511	16598	942	12574	8070
CCF-gas		616				1306		
kW-peak	13.5	0.59	13.0	6.0	18.9	0.60	18.4	8.5
Annual Energy Requirement								
kWh	10477	4269	7536	6067	18500	2715	13488	9123
CCF-gas		616				1306		

Table 2.2 - Cost comparison of conventional equipment with ground-source heat pump system from Table 5, page 604 (Kavanaugh 1992).

	Atlanta		Chicago		
	Cost/kWh		Cost/kWh		
Unit Type	$0.06	$0.08	$0.06	$0.08	Nat. Gas Cost
SSHP	$629	$838	$1110	$1480	
F/AC	$564*	$650*	$816*	$870*	$0.50
VSHP	$452	$603	$809	$1079	CCF
GSHP	$364	$485	$547	$729	
SSHP	$629	$838	$1110	$1480	
F/AC	$626*	$711*	$947*	$1001*	$0.60
VSHP	$452	$603	$809	$1079	CCF
GSHP	$364	$485	$547	$729	

According to this author, the mid-efficiency SSHP uses the most electric energy in all situations. The VSHP uses 27% less energy than the SSHP while the GSHP uses 42% and 51% less energy than the SSHP. The VSHP costs less to operate than the F/AC, except in Chicago, where electricity costs are high and gas costs are low. The GSHP is the least expensive to operate in all cases. The VSHP shows no demand improvements over the SSHP while the GSHP reduces peak demand by 32% in cooling and 55% in heating. Furthermore, a high-efficiency vertical GSHP rated under ARI 300 consumes 19.5% less (Atlanta) and 32.4% less (Chicago) energy than a VSHP with equal ARI 210/240 ratings. The GSHP is much more effective in reducing both winter and summer demand when compared to the SSHP and VSHP.

Geothermal Central System

In addition to the previous studies, this research intends to study the energy performance and economy of the continual generation of a geothermal heat pump system. In their paper, *Geothermal Central System*, Durkin and Cencil (2007) investigate the potential of using the geothermal central system in schools. The paper provides the background knowledge of the geothermal heat pump central system and the case study of Lexington Elementary School.

The geothermal central system has a single heat recovery chiller/heater or Geo-H/C unit located in a central mechanical room. The Geo-H/C can be connected to a two- or a four-pipe building system which are both common systems used in school buildings. For the air-side equipment, the Geo-H/C can be used with the standard air handlers and unit ventilators or fan coils. The outstanding advantage of this system configuration is an opportunity to operate two types of air-side economizers: dry bulb and enthalpy.

In this paper, the Geo-H/C is one of HVAC system alternatives proposed for the Lexington Elementary School's HVAC system retrofit project. Lexington Elementary School is located in Scott County, Indiana. The school has 300 students and occupies 45,000 square feet (4181 square meters) of floor area. The school was built in 1925 and renovated in 1985 with air conditioning. The system selected for the 1985 school renovation was the self-contained air-source heat pump that was installed in each class room rather than the conventional furnace system. The air-source heat pump system was chosen because the natural gas used in the furnace system is not available in that part of the state at that time. By 2005, the air-source heat pumps were failing, O&M costs were increasing, and poor indoor air-quality was a concern; hence, the system upgrade was required, and the options for the change are listed as follows:

1. Direct replacement of the air-to-air heat pumps with the addition of a decoupled makeup air system to address the IAQ concerns.

2. Geothermal heat pumps with a decoupled makeup air system.

3. A modern two-pipe unit ventilator system designed per the E-Source pamphlet, with an air-cooled chiller and the following boiler option: (a) propane and (b) fuel oil.

4. An air-cooled chiller system with electric resistance heat.

5. A geothermal system with a central heater/chiller (Geo-H/C) and two-pipe unit ventilators serving the classrooms.

From five options, the school owner and design team agree with options 3a, 4, and 5. In the paper, the incremental procedure is used to estimate the overall efficiency of the Geo-H/C system (COP=3.09), and the payback method is used for the economic evaluation. The results of the study show that when the air-cooled chiller system with electric resistance heat (option 4) is considered as the baseline, the propane (option 3a) will pay back in 10.5 years, while the Geo-H/C will pay back in 13.2 years from their cost saving. And, if the natural gas is available for this site, the payback period of the system using natural gas will be 1 year.

In addition to the Lexington Elementary School case, the authors also provide interesting information about well field thermal imbalance of the conventional geothermal heat pump system. The information illustrates that the ground-coupled heat pumps may experience rising ground temperature which could be 1°F to 1.5°F (0.6°C to 0.8°C) per year. In theory, the Geo-H/C requires less bore-field than ground-coupled heat pumps which helps reduce the amount of heat rejected to the earth. Therefore, the authors believe that the Geo-H/C concept produces less long-term thermal imbalance than a conventional decentralized heat pump system.

Furthermore, the authors conclude that the geothermal heat pump system is more efficient than other conventional systems. However, investment costs of this system are high, and the paybacks can be lengthy depending on the baseline system which the geothermal heat pump is compared to. The authors suggest that the partial or hybrid geothermal heat pump system can reduce the high investment costs of the conventional GSHP system with minimal loss of operation efficiency. The authors also conclude that the Geo-H/C system can be cost effective as a result of the high efficiency of the system based on the principle that the system can operate with two types of cooling economizers.

Chapter 3: Methodology

3.1 Methodology Overview

This research aims to study the energy performance and economy of the geothermal heat pump system used in the KnowledgeWorks I and II buildings in comparison with two alternative HVAC systems. The research is based on a quantitative paradigm using a simulation and modeling research methodology.

In this research, a building energy performance simulation software package is used as a tactical tool to perform the simulation of a building baseline model that accurately replicates the thermodynamics and energy performance of actual buildings. The baseline model is used for simulating the buildings operated with alternative HVAC systems to determine the energy performance of the buildings with alternative systems. The information gathered from the baseline and alternative model simulations are used as inputs for the economic evaluations including the LCCA, SIR, and DPB. Subsequently, the outcomes of the building energy performance simulations and economic evaluations are applied to the hypotheses testing.

3.2 Simulation Research and Building Energy Performance

Simulation and modeling is a research method that has been used in architecture related studies that are concerned with the replication of real-world realities. This research method aims to replicate a real-world context, and study the dynamic interactions between a real-world context and manipulated factors, where the results can be collected as data and applied to real-world applications (Wang 2002). Simulation is also used as the primary tactic in several experimental research approaches since they share the characteristic that they isolate the context and study how the manipulated variables affect performance outcome. Sometimes the simulation can be used as a part of a qualitative research approach such as in the interpretive-historical research, where the past events or conditions are reconstructed.

Before computers became the leading aids in simulation and modeling research, the simulation models were defined in four different types including iconic, analog, operational, and mathematical models (Clipson 1993). The iconic models refer to models that are typically used for testing materials or products. Analog models deal with the dynamic simulation of an actual or proposed physical system. Operational models are used for replicating the human interaction within physical contexts. Mathematical models are systems of numerical coding that capture real world relationships in quantifiable abstract values. Then with the capabilities of computers there is a tendency to merge the boundary of these models and transform them to the operational level (Wang 2002). For pure mathematical models such as those used in building energy simulation programs, the mathematical-spatial models such as the geographic information systems (GIS), and the virtual reality environments such as the computer assisted virtual environments (CAVE) are examples of such computer capabilities.

In order to perform simulation and modeling research, it is important to understand important tactical concerns including the accuracy of replication, completeness of input data, and cost and workability (Wang 2002). When applying simulation research, the accuracy of replication to the real-world context is the most imperative concern, thus a simulation must closely replicate the real-world context. The accuracy of the simulation is influenced by the completeness of the input data, the second concern. It is important that the data are well collected and logically translated into meaningful input data. All the previous concerns have raised the last tactical concern of the simulation research which is concerned with cost and workability. The cost and workability can be important issues for the physical and computer simulation models, and in the data collection process. For example, the data gathering process may require an expensive instrument for collecting data that could be used to calibrate the model; this cost could be difficult to recover. Therefore, it is important to apply simulation research when it is affordable and workable for meeting the research objectives.

Since building energy conservation and efficiency are important concerns, there are opportunities to apply simulation and modeling methods as a tactical tool for estimating the consumption of energy in buildings. Building energy performance simulation is one of the applications of the simulation and modeling research approach. Based on mathematical models, building energy simulation programs are capable of replicating the thermodynamics of buildings. Such programs allow users to explore the affects of manipulated factors and generate results which can be applied to a real-world context.

Building energy performance simulation also has the tactical concerns recommended by the Energy Design Resources (EDR) (2000). In the design process, the simulation should be incorporated in the early stages, and subsequently refined as the design develops. The simulation models should be as simple as possible while containing enough key-information required for accurate representation of the building. For example, for some programs, the results of simulating a floor by floor full model of a ten-story building may not be significantly different from a simulation of the same building while modeled only the first floor, top floor, and multiplication of second through ninth floors. The simulation requires the completeness of inputs, which is always the major concern when applying simulation research. Furthermore, the building energy simulation may be inappropriate when the design is in the late stages where changes may be expensive, and when the costs of performing the simulation are not justified, when the important questions cannot be answered by the simulation, and when detailed design information is required that goes beyond questions of energy consumption.

There is a number of building energy simulation software packages available today with more than 300 listed in the DOE website. In 2005, there was a study conducted by a group of researchers for the DOE: Contrasting the Capabilities of Building Energy Performance Simulation Programs (Crawley, et al. 2005). The study was a comparison on the capabilities of twenty leading building energy simulation programs. In the study, there are about half of the programs that are capable of simulating the HVAC systems including GSHPs to a high level. The programs include DeST, EnergyPlus, eQUEST, ESP-r, HAP, IDA ICE, IES <VE>, Trace, and TRNSYS. From the cost and workability concern, there are only two programs out of these nine that are appealing for this research: eQUEST and EnergyPlus, because they have no cost for a license. The summary table of HVAC system simulation capability is provided in Appendix A, Table a.1

eQUEST and EnergyPlus are derived from the DOE2 software program, which is considered one of the most validated energy modeling programs. DOE2 is commonly used to perform building energy performance simulations both in federal and industrial building projects. DOE2 was developed by James J. Hirsch & Associates (JJH) in collaboration with Lawrence Berkeley National Laboratory (LBNL) supported by the U.S. Department of Energy. DOE2 version E has been used as the modeling engine in other programs such as EnergyPlus, Building Design Advisor (BDA) and SPARK. In addition to DOE2.1E, version DOE2.2 is used as the computation engine in building energy simulation programs such as eQUEST and PowerDOE.

Unfortunately, there was a conflict of interest between the developers; as a result, the latest official version of DOE is still the DOE2.1E.

eQUEST was developed by JJH with funding support from California Utilities. The program is a combination of the DOE2.2 simulation engine, easy to use wizards, and integrated user friendly LCCA program. The DOE2.2 engine performs an hourly simulation of the proposed building for a one year period. DOE2.2 calculates heating and cooling dynamically for 8,760 hours based on factors including, but not limited to, building shell, occupancy, lighting, equipment, and air ventilation. DOE2.2 also calculates the energy consumption of mechanical and energy-consuming systems in the building. The wizards consist of the schematic design, design development (DD), and energy efficiency measure (EEM) wizards which help the users perform the building energy simulations from the beginning of the design phase to the detailed analysis of state-of-the art building technologies. eQUEST can generate a set of professional-level results in graphical and text report formats with reasonable user effort. The program also has the built-in graphic user interface (GUI) that can be used for describing building parameters and viewing the mechanical plant diagrams and proposed building in 2D and 3D views. The integrated LCCA program used in eQUEST is the FEMP's BLCC spreadsheets which are validated to perform LCCA. The down side of eQUEST is its relative inflexibility of data input at detailed levels. The detailed parameters can only be edited, where the generated outcomes can be saved, but the detailed inputs cannot be saved.

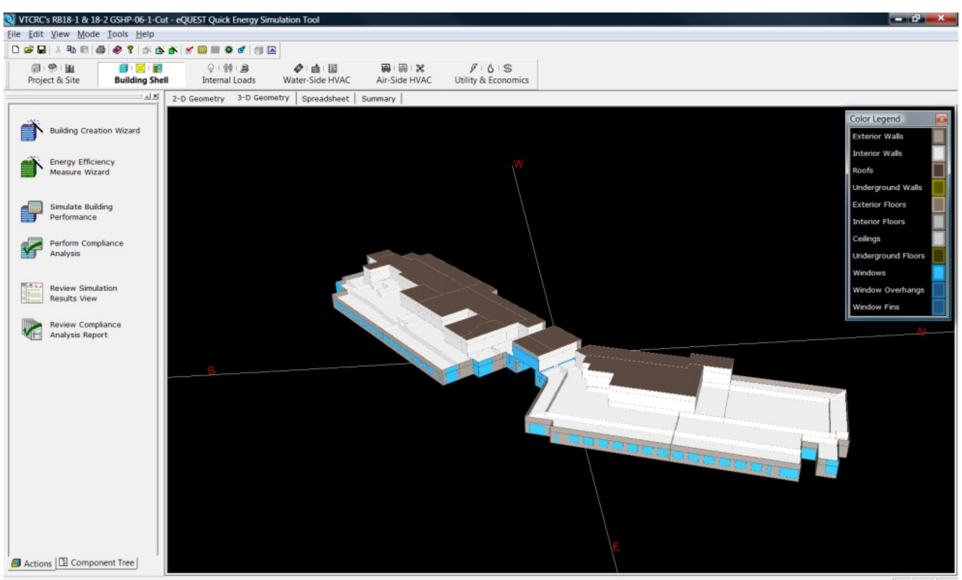


Figure 3.1 - eQUEST screenshot.

EnergyPlus was developed with funding supported from LBNL, University of Illinois, and U.S. Army Construction Engineering Research Laboratory. The program is based on the well-known features of BLAST and DOE2.1E. EnergyPlus consists of two basic components: a heat balance simulation module and a building system simulation module. The heat balance module performs the loads calculation and the building systems module calculates heating and cooling systems, and plant and electrical system response. These two modules are controlled by the simulation manager which is able to control communication between the two modules. The heat balance module is an important feature of this program. The heat balance engine is based on BLAST which couples loads and systems simulation. Another strong feature is that EnergyPlus has a detailed daylighting simulation system, DElight, which is based on the radiosity interreflection method that is capable of analyzing complex fenestration systems characterized by bi-directional transmittance data. A weakness of EnergyPlus is that it requires advance user knowledge and experience with the BLAST and DOE2 programs to be able to perform the simulations effectively. The program does not provide user-friendly graphic or program interfaces. Although there are several GUI programs for EnergyPlus, they are still underdeveloped and have not been validated.

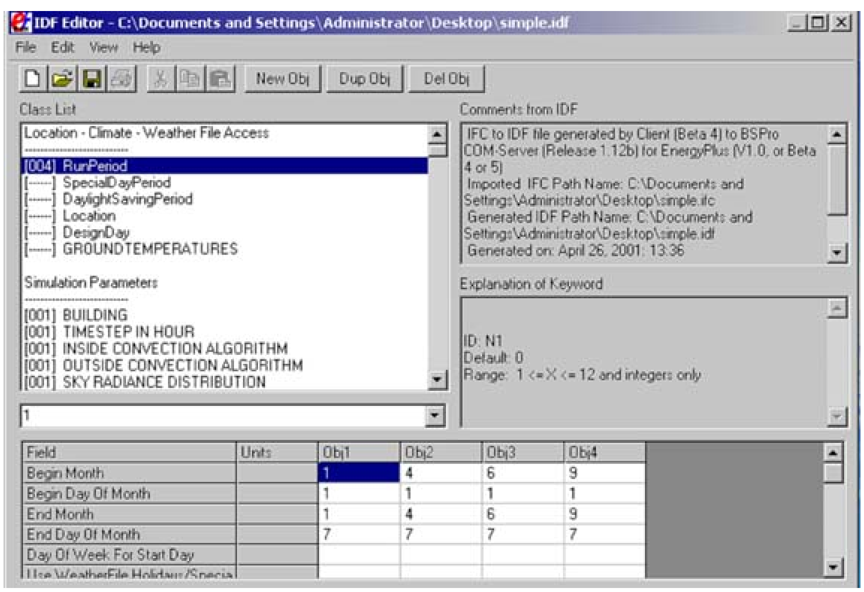


Figure 3.2 - EnergyPlus screenshot from (DOE 1996-2007).

In comparison, eQUEST and EnergyPlus are both partially based on the simulation engine and calculation algorithms of DOE2. On the other hand, they use different loads calculation methods, the eQUEST hour-by-hour method with de-coupled loads and systems and the EnergyPlus building heat balance method. Both methods are validated and are satisfactory for this research. When comparing program capabilities, eQUEST has five fewer HVAC subcategories than EnergyPlus. The HVAC subcategories include discrete HVAC components, idealized HVAC systems, automatic sizing; air loop flow, outside air, and zone airflow, zonal air distribution unit; VAV reheat/variable speed fan (UFAD), and zone forced air unit; energy recovery ventilator (stand alone). However, the specifications of these HVAC subcategories do not affect the simulations conducted in this research with regard to the accuracy of replication and completeness of input data. For cost and workability concerns, both programs are freeware. The strength of eQUEST is that it has the internal GUI and user-friendly program interface, which are still under development for EnergyPlus. As a result, eQUEST was selected as the appropriate tool to be used for this research. Moreover, the program has the integrated FEMP's User-friendly Life-Cycle Costs Analysis spreadsheet for life-cycle costs analysis which can be used to perform the economic evaluations in this research.

3.3 eQUEST Modeling Approach

eQUEST performs the building energy performance simulation utilizing the DOE-2.2 engine which is capable of performing an hourly building energy simulation for 8,760 hours per year. Inside eQUEST, DOE-2.2 calculates the hour-by-hour building loads as well as simulates the performance of various building system components and energy-consuming devices. eQUEST not only utilizes the full capabilities of the DOE-2.2 engine, but also extends and expands the capabilities of DOE-2.2 to include the interactive operation and dynamic/intelligent defaults. Seeing eQUEST as a tactical tool for this research, it is necessary to understand some important aspects of the program which include the computation procedures, calculation algorithms for the ground-source heat pumps, economic evaluation procedures and modeling methods used.

3.3.1 eQUEST Computation Procedures

eQUEST computation procedures are based on the four primary computation programs from DOE-2: LOADS, SYSTEMS, PLANT, and ECONOMICS as shown in Figure 3.3. In LOADS, DOE-2 calculates the hourly space heating and cooling loads, also called space load, from building external gains/losses through building envelopes and internal gains from people, lighting, and equipment which are named as the instantaneous gain. Proceeding to the

SYSTEMS program, the space loads pass from the LOADS program with a list of user-defined system characteristics such as air flow rates and set point temperatures of each HVAC zone. The system loads are then passed to the PLANT program which performs the computation of the performance of the primary energy conversion equipment, where the operation of each plant component such as boiler, compression chiller, and cooling tower is simulated on the basis of operating conditions and part-load performance characteristics. Afterward, the ECONOMICS program is used for estimating the utility costs from the simulated building energy consumption and the user-defined utility rates. The steps are shown in Figure 3.3.

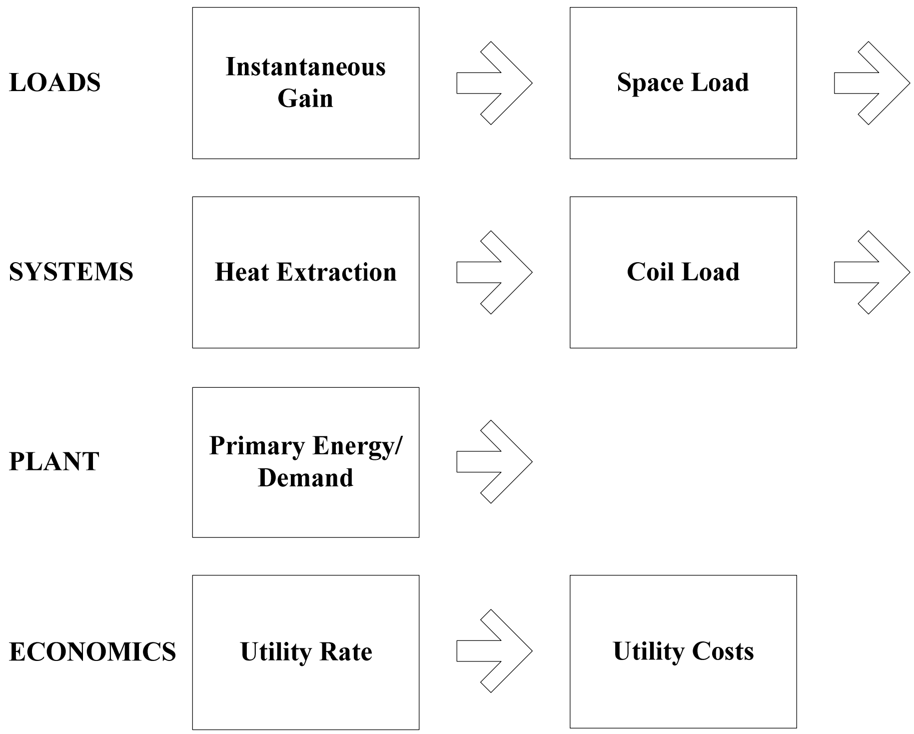


Figure 3.3 - Computational steps in eQUEST/DOE2

In addition to the four DOE-2 computation subprograms, it is also important to understand the methods that DOE-2 uses to determine heat transfer through building surfaces and space internal loads used in the computation subprograms as they influence the eQUEST energy modeling. DOE-2 allows four types of heat transfer surfaces: light-transmitting, exterior, interior, and underground. The light transmitting surfaces such as glass doors and windows, glass block walls, and skylights are considered to be the same type of heat transfer surface, a WINDOW. Exterior surfaces such as opaque exterior floors, walls, and roofs are all treated as an EXTERIOR WALL.

The interior surfaces such as opaque interior walls, floors, walls, and ceilings are considered as an INTERIOR WALL. Underground surfaces such as basement floors and walls, and slab-on-grade are treated as an UNDERGROUND WALL.

For internal loads, DOE-2 has three different categories defined by how the loads are seen through the calculations. The first type of internal loads corresponds to those loads that can be seen by a thermostat and directly impact the utility meter. These include receptacle/plug loads, and lighting. The second type of internal load can only be seen by a thermostat without impacting the utility meter. These include occupants and process loads. The third internal load type is only seen by the utility meter and not by a thermostat. This includes outdoor parking or sign light. The eQUEST wizards may only support these typical load sources.

Furthermore, in order to efficiently perform the building energy simulation using eQUEST, it is important to understand the limitations of the program especially as they relate to the HVAC systems. Currently, DOE-2 is not capable of performing the simulation of solar thermal systems, vertical self-contained fan-powered VAV systems, steam loop systems, plate-and-frame economizers, two or more HVAC systems serving the same zone, comfort-controlled radiant cooling or heating systems, and water-use and emissions. Also, eQUEST has been developed to be user-friendly and easy to use which can be limiting as in some cases it is difficult for users to input the specific data related to the simulation objectives.

3.3.2 eQUEST Ground-Source Heat Pumps Simulation

eQUEST is one of the few building energy simulation programs that is capable of modeling and simulating ground-source heat pumps. In DOE-2.2 within eQUEST, ground-source heat pump systems are considered as a special form of water-loop heat pump systems, where the calculation procedures are based on the water-to-air California heat pump (subroutine HTPUMP) algorithms which were used in DOE-2.1E and validated against field measurements. This subroutine simulates the heat and moisture exchange of a water-to-air (California) heat pump system utilizing subroutine TEMDEV calculations. The subroutine TEMDEV is the interface between LOADS and SYSTEMS programs that calculates the room air temperature and the heat addition or extraction rate by applying the room air temperature weighting factors and constant temperature loads passed from the LOADS program. The basic components of the water-to-air heat pump are composed of a pipe loop with a circulating fluid that is utilized as a heat source and heat sink and individual water-to-air heat pump units in each zone connected to the system.

The subroutine HTPUMP has two calculation steps, where the first step is for each zone connected to the system and the second is for simulating the central fluid loop. For each zone connected to the system, the subroutine HTPUMP calculates the maximum cooling and heating rates and the hourly room air temperature together with the hourly heat addition/extraction rate. Subroutine TEMDEV then simulates the zone heat pump performance in both cooling and heating modes. For simulating the central fluid loop, the subroutine HTPUMP calculates the temperature of the fluid in the loop and any heating or cooling of the loop fluid, if either is required. The subroutine HTPUMP requires more than forty calculation algorithms which are explained in detail in the *DOE-2 Engineer Manual Version 2.1A* written by the Energy and Environmental Division, Building Energy Simulation Group, Lawrence Berkeley Laboratory and Group Q-11, Solar Energy Group, Energy Division, Los Alamos National Laboratory, 1982 (page IV.143-IV.153).

Currently, eQUEST version 3.6, 2007, with DOE-2.2 version 44 has two new significant features: water-loop and ground-source enhancements, and wizard HVAC selection of GSHPs. The enhancements increase the program capability in modeling the existing water-source heat pump and vertical well ground source heat exchangers systems in certain ways. DOE-2.2 version 44 can model and simulate the water-source heat pumps with the package variable volume and temperature (PVVT), package variable air volume (PVAV), and package single-zone air conditioner (PSZ) systems in water-loop heat pump configurations. Also, the new version can model and simulate the vertical well ground source heat exchangers more accurately, in which users are able to specify grout properties and fluid type with the additional library of 42 well configurations. For the HVAC selection wizard of GSHPs, users can select and define the details of a ground loop heat pump circulation loop and ground heat exchanger system. The new features have made eQUEST more flexible for modeling and simulating ground-source heat pumps than most other simulation tools.

3.3.3 eQUEST Economic Evaluation Procedures

eQUEST can be used to evaluate the economy of energy efficiency investments by passing energy costs calculated by DOE2's ECONOMICS program to an integrated User-friendly Life-cycle Costing spreadsheet to perform the life-cycle costs and supplementary measures of economic evaluation analyses. The spreadsheet (developed by M.S. Addison and Associates, Tempe, AZ) calculation procedures are derived from the widely used life-cycle economics methodology recommended by the DOE's Federal Energy Management Program (FEMP) and the National Institute of Standards and Technology (NIST) discussed in the previous chapter. Users can directly input the economic parameters related to the LCCA into the spreadsheet, where the results of evaluation are presented in graphical and text formats together with other simulation results.

The recent version of the spreadsheet within eQUEST allows users to select the secondary fuel type which can be any non-electric fuel. This version also has the capability to calculate and report the results of saving-to-investment ratio (SIR), adjusted internal rate of return (AIRR), and discounted payback (DPB) onto the results summary sheet in addition to the life-cycle costs (LCC), net saving (NS), and simple payback (SPB) provided in previous versions. This version also adds the net saving graph that illustrates the cumulative net saving of all project alternatives over their maximum life-time of 25 years.

The User-friendly Life-cycle Costing spreadsheet has several limitations for the LCC calculation factors including energy type, cash flow convention, and time steps. The spreadsheet accepts only two energy source types: electricity and non-electricity such as natural gas, oil, and coal. The spreadsheet allows only end-of-year cash flow. For time steps, the spreadsheet assumes only whole year time steps.

3.3.4 eQUEST Modeling Methods

eQUEST simulates building energy performance based on a virtual model that replicates the important thermodynamics of the proposed building. The model should be the simplest that captures the critical details related to the simulation objectives. In eQUEST, there are two basic modeling procedures that help users develop a workable model, the building blocks of simulation and HVAC zoning procedures. The building blocks of simulation procedure helps users address the necessary information required in the simulation model, and the HVAC zoning procedure helps users define the HVAC zones of the modeled building.

Building Blocks of Simulation

The building blocks of simulation helps address the following information.

Analysis objectives:

The clear understanding of simulation inquiries can help identify the critical aspects required in the simulation model. Thus, the simulation objectives must be clearly addressed before developing the simulation model.

Building site information and weather data:

eQUEST requires site information consisting of the site characteristics (i.e., latitude, longitude, and elevation) and adjacent structures or landscaping that cast shadows on the proposed building. The program also requires the weather data of the building site in TMY2 file format which can be downloaded from the program support website.

Building Shell, structure materials, and shades:

eQUEST calculates heating and cooling loads hourly from the heat transferring through the building envelope. Therefore, the program requires the description of materials and configurations of heat transfer surfaces such as walls, openings, overhangs, and fins. The various settings of heat transfer surface components can be selected from the building component libraries provided with the program.

Building operation and scheduling:

The building operation and scheduling information includes, but is not limited to, building occupancy, occupied indoor thermostat setpoints, and HVAC and internal equipment operations schedules. Several useful building operations and scheduling defaults have been provided in eQUEST based on the selection of building type.

Internal loads:

People, lights, and equipment can significantly contribute to the energy requirements in large buildings. These influences may be direct from their power requirements or indirect from their effects on the heating and cooling requirements. For the information related to internal loads, eQUEST provides reasonable default values based on the selected building type. Furthermore, the industrial standard data can be found in the *ASHRAE Handbook of Fundamentals* (published every four years). Also, resent research related to this topic is available from Lawrence Berkeley National Laboratory (LBNL).

<u>HVAC equipment and performance:</u>

Heating, Ventilating, and Air Conditioning (HVAC) equipment are among the most energy-intensive systems in buildings. The information related to HVAC equipment efficiencies are very important for determining the accuracy of the simulation model. The default values for HVAC equipment performance in eQUEST are derived from the *California Title 24* energy standard. The program also allows for manual inputs.

<u>Utility rates:</u>

eQUEST is capable of estimating hourly electrical demand profiles which can be coupled with full details of the applicable utility rates (tariffs). In the program, the default values of utility rates are derived from the California utilities, where users outside California can create the utility rate descriptions using eQUEST's DOE-2-derived Building Description Language (BDL) that can be saved as text files for use in eQUEST. The program developers have provided a template in a file named "BDL Utility Rate Documentation.pdf" that explains the syntax and structure of BDL utility rate files and the instructions overview in a file named "Readme.txt". Both files are located in the "C:\Program Files\eQUEST\Rates" folder.

<u>Economic parameters:</u>

eQUEST considers the economic parameters on a life-cycle costs basis. The life-cycle costs and the supplementary economic measures including NS, SIR, AIRR, SPB, and DPB are calculated in an integrated User-friendly Life-cycle Costing spreadsheet. The spreadsheet default calculation factors such as fuel escalation region, analysis sector, and secondary fuel type are based on the information described in the eQUEST input wizards. The spreadsheet uses the

FEMP discount rates and DOE energy price escalation rates, and allows users to customize the rates manually. The types of costs that can be described in the spread sheet are non-recurring costs including investment-related costs and operation-related costs in the year of occurrence, energy costs in constant dollar, and annual recurring costs in constant dollars.

In order to gather the useful data required in the building blocks of simulation, the program developers have provided worksheets that help users identify the type of data that should be assembled before developing the simulation model, or confirmed in the course of modeling. The worksheets also help users organize and manage the data to other project members. The worksheets are shown in Appendix A.

HVAC Zoning

In addition to the building blocks of simulation method, HVAC zoning influences the accuracy of the simulation model. HVAC zone refers to a group of conditioned spaces controlled by a single thermostat, where such spaces share similar load and usage characteristics. At present, the program has two zoning schemes including one-zone-per-floor and simple core-vs.-perimeter zoning that help automatically zone the simulation model based on the user selection. Also, the eQUEST developers have provided some basic criteria for HVAC zoning in the *Energy Simulation Training for Design & Construction Professionals* (2004), page 10 of 142, retrieved as follows:

• When modeling existing buildings, refer to the actual zoning indicated by the HVAC plans, if available

For new buildings and when simplifying the zoning of an existing building considers:

• Magnitude and schedule of internal loads

• Magnitude and schedule of solar gains

• Schedule of fan system operations

• Outside air requirements

• Intended efficiency measures (ECM's)

• Location of thermostats called out on the HVAC plans

In general, the users should provide the basic zones listed as follows:

• One exterior zone per major orientation (12 to 18 feet deep)

• One internal zone per use schedule

• One plenum zone (if plenum returns) for each air handler to be modeled separately

• One zone each for special uses (e.g., conference rooms, cafeterias, etc.)

• Separate ground and top floor zones

Furthermore, the program will accept detailed criteria for HVAC zoning that can help the simulation model satisfy both model simplicity and key information requirements. Additional criteria are retrieved from the *Energy Simulation Training for Design & Construction Professionals* (2004), page 13 of 142, listed as follows:

• Tenant and leasing flexibility may dictate that the building be divided in a manner that facilitates flexible leasing of space assignment requirements.

• Ceiling space limitations or manufacture's terminal equipment size limitations may cause a larger number of smaller units to be specified than strictly required by the rules on the previous page.

• Acoustical privacy requirements may separate supply to adjacent areas.

• Code requirements may separate supply to adjacent areas (e.g., separate return for smoking areas).

Common ways that modelers simplify the zoning and size of their models include the following:

• In multiple floor high rise-type buildings, intermediate "typical" floors are modeled as only one floor in the simulation model and a floor multiplier is applied in the model to permit the modeled typical floor to represent the true, larger, number of floors.

• All actual perimeter zones along similar orientations are combined into one zone with the same common orientation. This assumes that all of the perimeter zones so combined behave in a very similar manner.

• Separate core zones are usually combined, again, on the assumption that the separate core zones actually behave in an indistinguishable manner.

It is important to note that the zoning simplification may cause a reduction of the number of modeled HVAC air-handler systems from the number of HVAC systems in the actual building. Therefore, two or more HVAC systems may be combined in the model, where the performance characteristics of the systems are the average of the combined systems.

3.4 Baseline Model

General Information

KnowledgeWorks I and II are two-story office and research laboratory buildings located in the Virginia Tech Corporate Research Center featuring space for pre-launch operations, semi-custom office suites, laboratory space, and shared conference rooms of various sizes. KnowledgeWorks I (RB XVIII), completed in 2005, has a gross floor area of 45,439 square-feet consisting of office, laboratory, and shared conference room space. KnowledgeWorks II (RB XIX) is the expansion construction of the first building which has 40,245 square-feet gross floor area supporting large office space. The two buildings are connected via the shared conference room space on the second floor of the existing KnowledgeWorks I building. Both buildings use the GSHP system as their major HVAC system covering about 94% of the building conditioned space. GSHPs are supported by the combined water loop with antifreeze solution installed in the pond, also used for collecting rain water, and installed underground in the vertical-loop configuration.

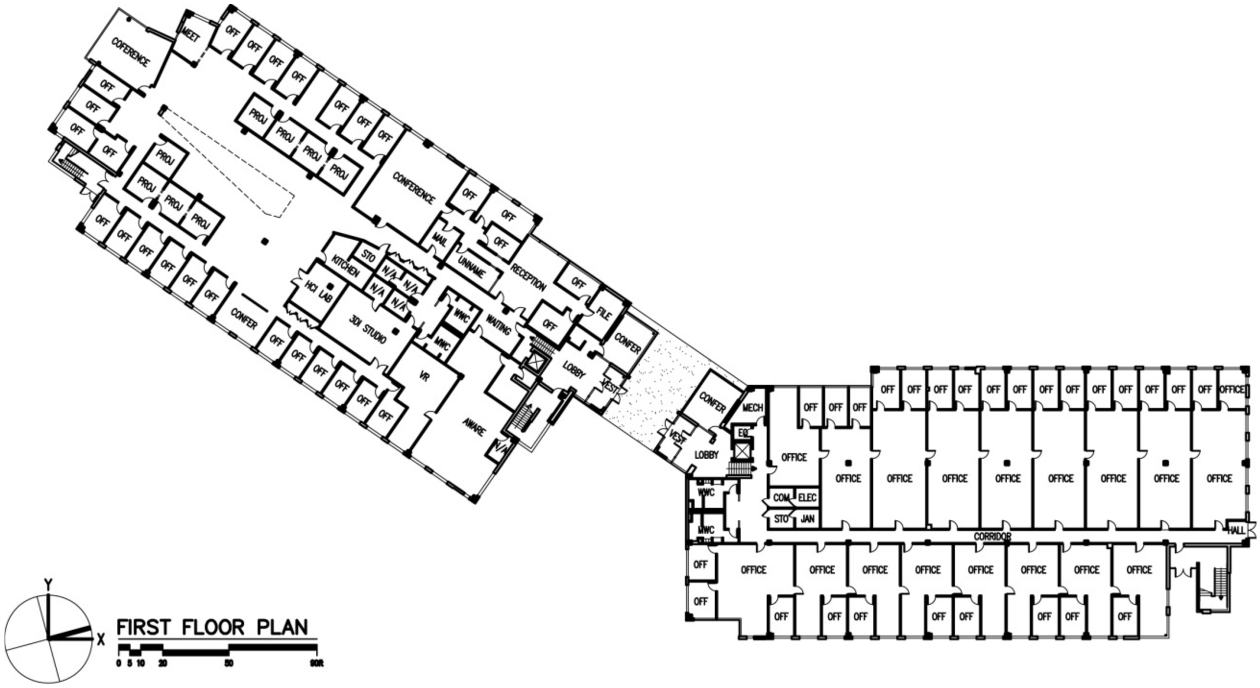


Figure 3.4 - First floor plan of the KnowledgeWorks I & II buildings redrawn from the original drawing of SMBW Architects, P.C.

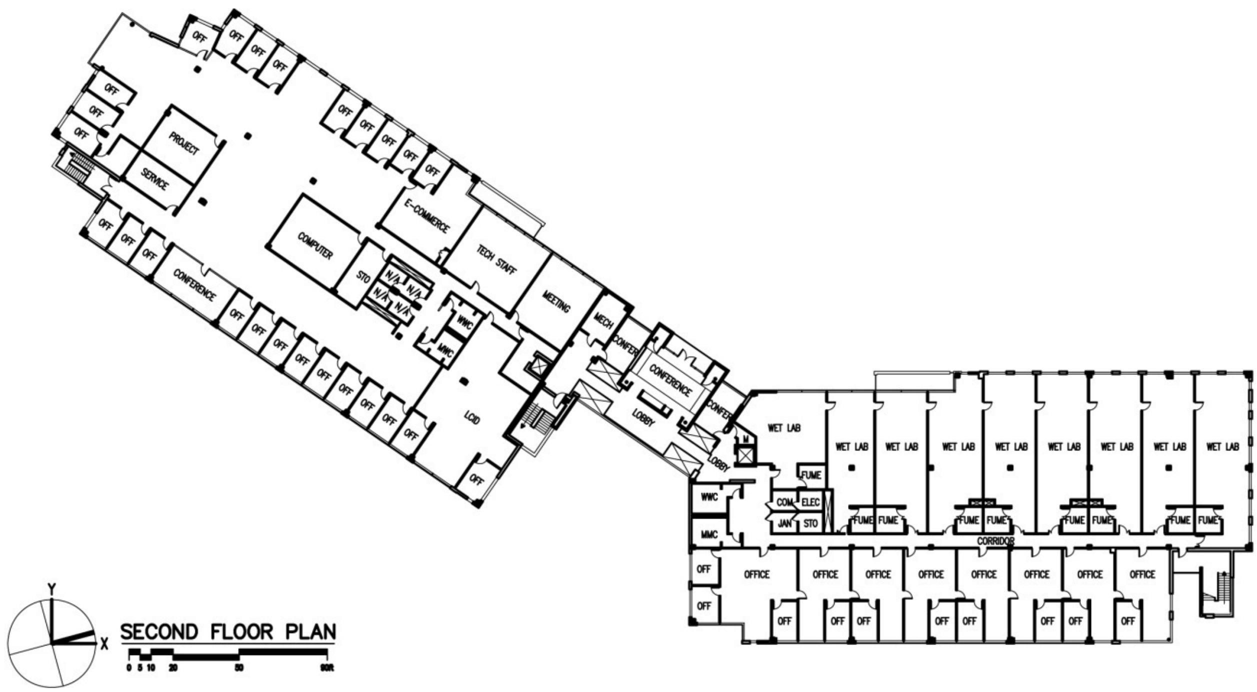


Figure 3.5 - Second floor plan of the KnowledgeWorks I & II buildings redrawn from the original drawing of SMBW Architects, P.C.

Figure 3.6 - KnowledgeWorks I & II buildings (Photograph by Kongkun Charoenvisal, 2007)

Building Blocks of Simulation

Analysis objectives:

In this research, the baseline model is developed for replicating the thermodynamics and energy performance of the VT CRC's KnowledgeWorks I & II buildings. The main focus of utilizing this baseline model is to study the energy performance and life-cycle costs benefits of the GSHP system that has been integrated and operated in the buildings in comparison with the alternative models consisting of air-source heat pump and DX cooling with hot water heating systems.

Building site information and weather data:

KnowledgeWorks I and II buildings are located on 2200 Kraft Drive, Blacksburg, VA 24060, at the 37˚12'01.44" North latitude and 80˚24'29.76" West longitude, and at 2,140 feet above sea level. Based on on-site observation, the building site has no adjacent structures or landscaping that cast shadows on the buildings. For this location, eQUEST automatically obtains the Roanoke TMY2 weather file which is based on data from the Roanoke VA Regional Airport,weather station. The detailed building site information and weather data are summarized in Appendix B, Table B.1.

Building Shell, structure materials, and shades:

A description of materials and configurations of heat transfer surfaces used in the baseline model were based on the original architectural and structural drawings provided by the CRC facility manager, and program for an office building. The detailed assumptions for all building components are summarized in Appendix B, Table B.2.

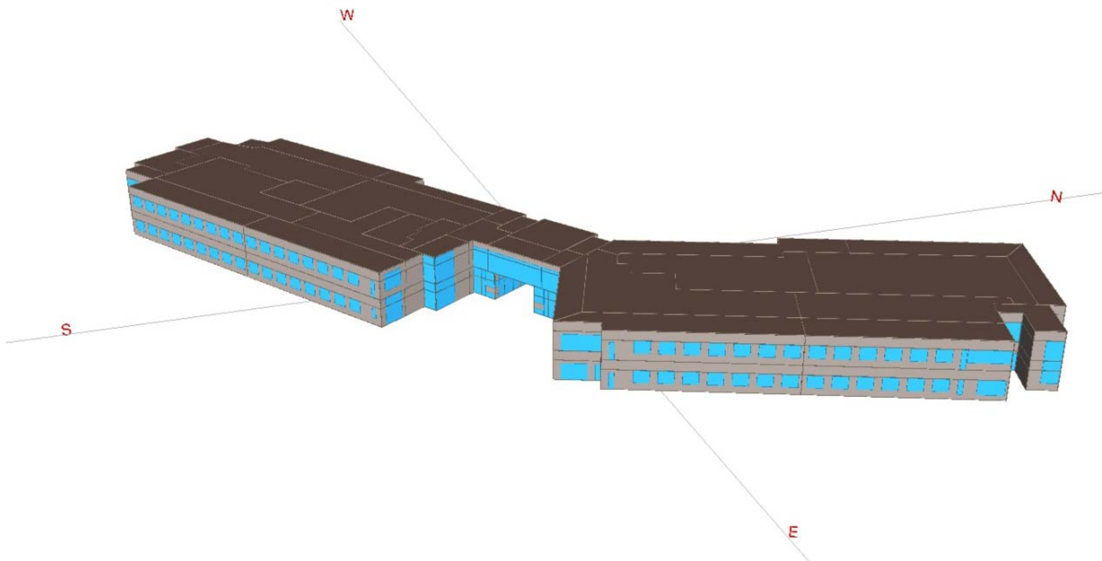


Figure 3.7 - eQUEST model representing building geometry

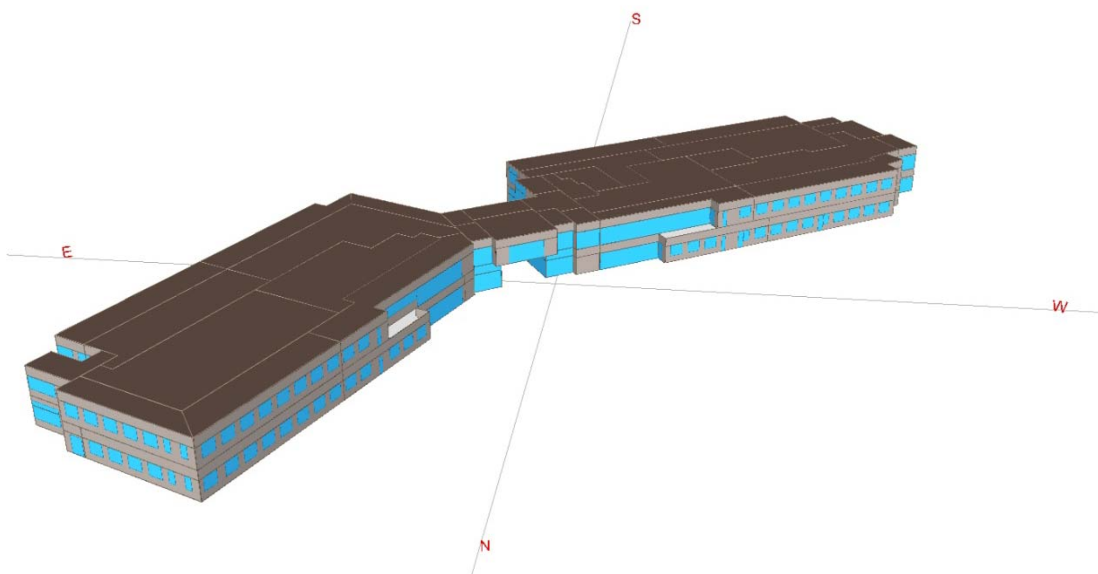


Figure 3.8 - eQUEST model representing building geometry

<u>Building operation and scheduling:</u>

Building operation and scheduling information were provided by the CRC mechanical systems maintenance staff based on the Trane Tracer Summit program that has been used for monitoring and controlling the CRC HVAC systems. The regular operating hours of the KnowledgeWorks I building are Monday through Friday, from 6:00 a.m. to 6:00 p.m. excluding four laboratory units on the second floor that are operated twenty-four hours per day, seven days per week. The regular operating hours of KnowledgeWorks II building are Monday through Sunday, from 6:00 a.m. to 10:00 p.m. excluding some office space and computer rooms on the second floor that are operated twenty-four hours per day, seven days per week. The buildings are closed on national holidays except for the 24/7 spaces. The detailed information of building operation and scheduling are summarized in Appendix B, Table B.3.

<u>Internal loads:</u>

Input data for internal loads (e.g. people, lights, and equipment) are based on the given mechanical, electrical and plumbing (M/E/P) drawings, program defaults for building type, and related recommendations from the ASHRAE *Handbook of Fundamentals* (published every four years). The detailed information concerning internal loads can be found in Appendix B, Table B.4.

<u>HVAC equipment and performance</u>:

Data inputs for HVAC equipment and performance are based on mechanical drawings, the mechanical equipment operation and maintenance information handbooks provided by mechanical and electrical contractors, G.J. Hopkins, Inc., information given by CRC mechanical systems maintenance staff based on Trane Tracer Summit program, program default values for the applicable HVAC system types, and the manufacturer website.

The major HVAC system of the buildings is the GSHP system consisting of nine package rooftop water-air heat pumps and eight indoor water-air heat pumps sharing a single water loop that exchanges heat with a rainwater collection pond and underground heat exchangers. In this system, water with antifreeze solution enters and leaves the buildings through a single SDR 17 HDPE main pipe forced by three variable speed drive (VSD) pumps that run in standby mode. Heat pumps extract heat from and discharge heat to the water loop through the intermediate R-22 refrigerant circuit. From Appendix B, Table B.5, the average design cooling and heating efficiencies of heat pump units are around 10.70 EER for cooling and 3.78 COP for heating. The system provides cooling and heating capacities for more than 76,000 square feet covering about 94% of the building conditioned space.

In addition to the GSHPs, some spaces including five laboratory units on the second floor of KnowledgeWorks I and one of the computer rooms on the second floor of KnowledgeWorks II are supported by DX cooling with electric resistant heater package rooftop units which are independent from the GSHP system. The detailed input data for HVAC equipment and performance are summarized in Appendix B, Table B.5.

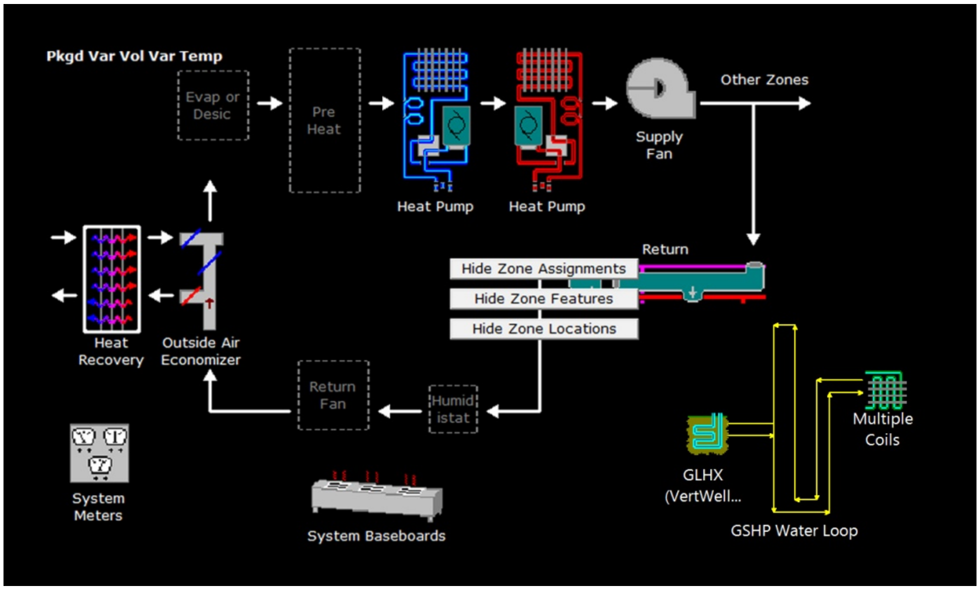


Figure 3.9 - eQUEST model representing air-side and water-side of the GSHP system

<u>Utility rates:</u>

Utility rate inputs are based on utility bill information provided by the facility manager and the Virginia Tech Electric's tariff structure for Medium General Service consumers. The tariff structure also requires additional information related to local tax and state tax rates for electricity consumption which are provided by the Town of Blacksburg and State of Virginia. Table 3.1 show the 2006 Virginia Tech Electric's Tariff Structure. The detailed Utilities rate assumptions are explained in Appendix B, Table B.6.

Table 3.1 - 2006 Virginia Tech Electric's Tariff Structure[a] for Medium General Service consumers

Type of Service	Minimum Monthly Service Charge	First 900 kWh Energy Charge per kWh	Over 900 kWh Energy Charge per kWh	Demand Charge per kW	Reactive Demand Charge per kVAR
Medium Gen	$18.00	$0.042690	$0.042690	$3.42	-

a. The tariff structure above has been changed to the new rate since July 1[st], 2007.

Economic parameters:

Data inputs for economic evaluation of the GSHP system are based on the costs provided by project mechanical and electrical contractors; G.J. Hopkins, Inc., and information from R.S. Means Mechanical Cost Data books. G.J. Hopkins has given the total initial investment costs and cost per square-foot of the GSHP system constructed between 2003 and 2006 which were $2,192,983 and $28.51/square-foot respectfully. In the baseline model, these costs are assumed to be the costs in analysis year 2006 which was the year that the buildings construction was completed, and both buildings were occupied. Other costs of the system such as OM&R costs that have not been provided by G.J. Hopkins are estimated from the breakeven analysis following the FEMP's procedures. In the baseline model, these costs are assumed to be zero. For energy costs, eQUEST generates energy costs from energy consumed by the buildings and input utilities rates. The economic evaluation is assumed to start in year 2006 and have a 25 year analysis life.

HVAC Zoning

HVAC zoning in the baseline model was developed to follow the eQUEST HVAC zoning procedure based on mechanical drawings, information from Trane Tracer Summit program, and detailed information provided by CRC mechanical systems maintenance staff. The HVAC zones described in eQUEST are illustrated in the following Figure 3.7-3.8.

Baseline Model Calibration

Since this research was focused on existing buildings that have been operated and occupied, there is an opportunity to calibrate the results of the baseline model simulation including monthly energy consumption, energy peak demand, and energy cost with the utility bills. The utility bills given by the facility manager were based on the Virginia Tech Electric's tariff structure for Medium General Service consumers consisting of two billing accounts. The first billing account started in November 2004, when KnowledgeWorks I was completely constructed and the second billing account started in March 2006, when KnowledgeWorks II was completed. This calibration uses the 2007 utility bills, representing the pattern of energy used in the buildings after being fully occupied.

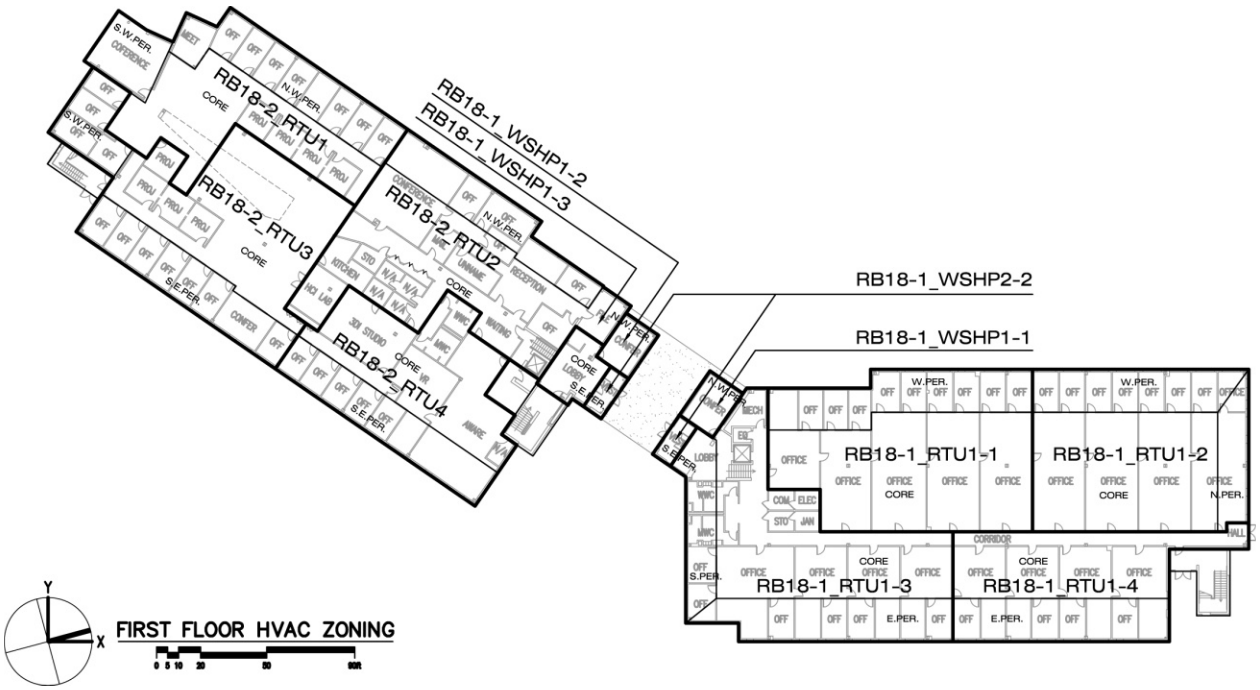


Figure 3.10 - First floor HVAC zoning of the KnowledgeWorks I & II buildings (drawing by Kongkun Charoenvisal, 2008)

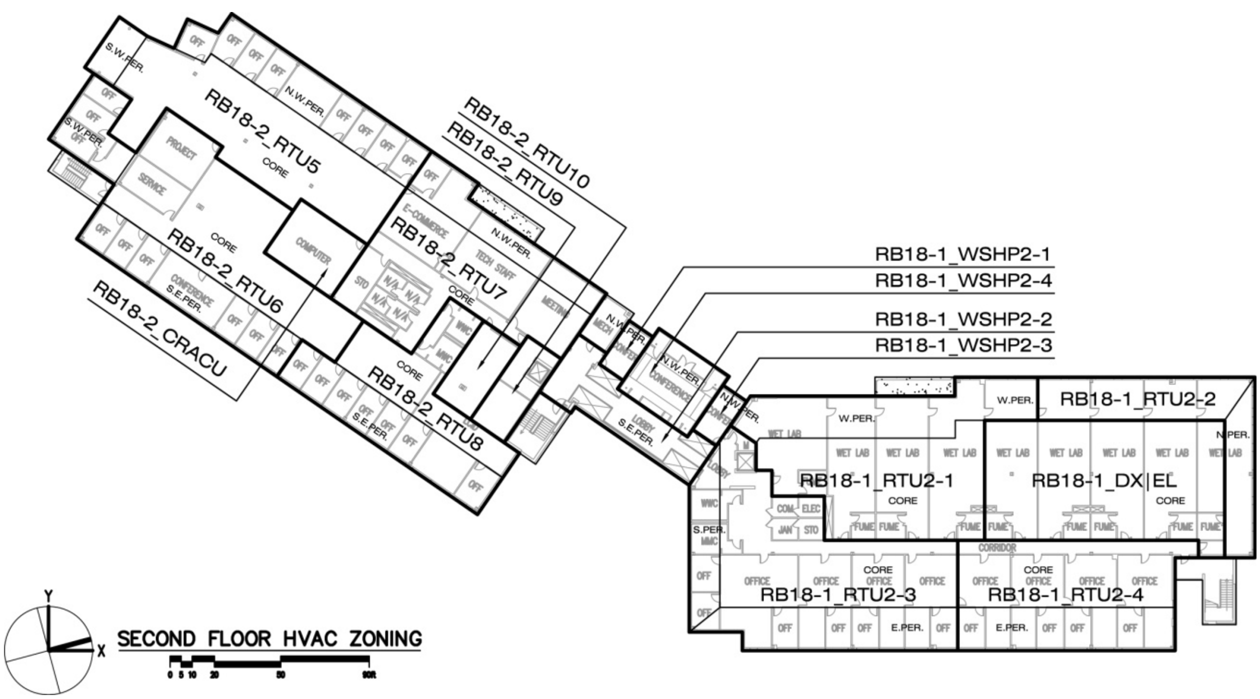


Figure 3.11 - Second floor HVAC Zoning of the KnowledgeWorks I & II buildings (drawing by Kongkun Charoenvisal, 2008)

3.5 Alternative Models

Alternative model I

Alternative model I is developed to simulate the KnowledgeWorks I and II buildings with a conventional VAV HVAC system. The typical HVAC system used in the most recently constructed buildings in CRC before changing to GSHPs in KnowledgeWorks I and II buildings is DX coil cooling source with gas furnace heating. In this study, for comparison the existing system is replaced by a packaged VAV with a hot water reheat. It was assumed that packaged VAV with hot water reheat is representative of a typical HVAC system in non-residential buildings.

Two HVAC systems were compared; the first was a ground-source heat pump system with DX coils as the source for heating and cooling. The second was a packaged VAV with hot water reheat. The VAV system uses DX coils as the cooling source and hot water coils for heating. The hot water is supplied by boilers that use natural gas. The custom control settings used in the baseline model (i.e., thermostat setpoints) were not changed for the VAV simulation. Default values were used for HVAC equipment performance inputs as shown in Table 3.2. The CRC purchases natural gas from ATMOS Energy Corp. The natural gas rate inputs are based on the ATMOS Energy's Virginia Rates tariff structure for Small Commercial and Industrial Gas Service consumers as shown in Table 3.3. The tariff structure also requires additional information related to local and state tax rates for natural gas consumption which are provided by the Town of Blacksburg and State of Virginia. The detailed Utilities rate assumptions are explained in Appendix B, Table B.7. The electric rate was unchanged for both HVAC simulations.

Table 3.2 - HVAC equipment performance parameters of the packaged VAV with a hot water reheat system

Unit Size	water-air heat pump		package VAV w/HW reheat	
<65 kBtuh	**EER**	11.0528	**EER**	8.4200
or 5.4 tons	**COP**	3.7394	**Boiler Efficiency**	80%
90-135 kBtuh	**EER**	12.5703	**EER**	8.9000
or 7.5-11.25 tons	**COP**	3.8397	**Boiler Efficiency**	80%
135-240 kBtuh	**EER**	9.9198	**EER**	8.5000
or 11.25-20 tons	**COP**	3.7885	**Boiler Efficiency**	80%

Table 3.3 - 2005 ATMOS's Gas Tariff Structure[b] for Small Commercial and Industrial Gas Service

Charge	Detail
Customer Charge:	A monthly customer charge of $14.50 is payable regardless of the usage of gas.
Monthly Rate:	All Consumption, Per Ccf $.1121
Activation Charge:	When a customer applies to initiate service, a charge of $40.00 will be assessed to cover the cost of activating service.

b. The tariff structure above has been changed to the new rate since August 1st, 2008.

To determine the appropriate economic parameters, G.J. Hopkins provided estimates for the total initial investment costs and cost per square-foot for the DX coils cooling with gas furnace heating system installed between 2001 and 2002 in building RB XVI which were $718,000 and $13.94/square-foot respectfully. The KnowledgeWorks I and II buildings have the total floor area of 84,963 square feet. Therefore, this system cost approximately $1,184,384 in year 2002. This amount is assumed to be in the same range as the packaged VAV with hot water reheat system used in this comparison. From R.S. Mean Mechanical Cost Data 2006, the 2002 system cost can be adjusted to the 2006 cost using the historical cost indexes method which results in a cost of $1,433,879 in year 2006 (see Equation 3.1). Since G.J. Hopkins only provided the initial investment costs, other LCC parameters of the system are estimated from the breakeven analysis following the FEMP procedures.

$$Cost\ in\ Year\ A = \frac{Index\ for\ Year\ A}{Index\ for\ Year\ B} \times Cost\ in\ Year\ B \tag{3.1}$$

$$Cost\ in\ Year\ 2006 = \frac{Index\ for\ Year\ 2006}{Index\ for\ Year\ 2002} \times Cost\ in\ Year\ 2002$$

$$= \frac{100}{82.6} \times \$1{,}184{,}384 = \$1{,}433{,}879$$

Alternative Model II

Alternative model II was developed to simulate KnowledgeWorks I and II buildings with an air-source heat pump system rather than a ground source system. For this alternative the baseline HVAC system, the ground-source heat pump with DX coils was changed to a packaged single zone air-source heat pump with DX coils for cooling and heating. The custom control settings used in baseline model were not altered, while default values were used for other HVAC equipment and performance inputs as shown in Table 3.4. Since air-source heat pumps use electricity as the energy source, the baseline utility rates were used.

Table 3.4 - HVAC equipment performance parameters of the ASHP system

Unit Size	water-air heat pump		air-air heat pump	
<65 kBtuh	**EER**	11.0528	**EER**	8.4200
or 5.4 tons	**COP**	3.7394	**COP**	2.8100
90-135 kBtuh	**EER**	12.5703	**EER**	8.9000
or 7.5-11.25 tons	**COP**	3.8397	**COP**	3.0000
135-240 kBtuh	**EER**	9.9198	**EER**	8.5000
or 11.25-20 tons	**COP**	3.7885	**COP**	2.9000

For the economic parameters for this alternative, an assumption was made that air-source heat pumps do not have the underground exchanger. From R.S. Mean Mechanical Cost Data 2006, the estimated cost for 4-6" diameter well borings were estimated to be about $36.24 per linear foot for Roanoke, VA. The vertical-loop heat exchanger used to supply the as-built GSHP system has 64 wells each 500 feet deep and 6" diameter. The wells cost $1,159,984 which is about 52.88% of the system costs. This amount is in the same range with the costs recommended by the California Energy Commission (2006). The California Energy Commission estimates the costs of an air-source system around 50% lower than a GSHP system. Therefore, the initial investment costs of the air-source heat pump system used in this simulation model was assumed to be about 50% of $2,192,983 or $1,096,492. Other costs of the system except energy cost are estimated from the breakeven analysis following the FEMP procedures.

Chapter 4: Data Analysis and Results

4.1 Baseline Model

Baseline Model Calibration

The baseline model was simulated and calibrating with 2007 utility bills. The given eleven months of utility bills are based on the Virginia Tech Electric's tariff structure for a medium general service consumer. The information retrieved from utility bills include the twelve calendar months of electricity consumption, ten calendar months of peak electricity demand, and ten calendar months of electricity cost. From the utility bills, the KnowledgeWorks I and II buildings consumed 3,225,002 kWh of total annual electric energy, of which the buildings' total peak electricity demand was 5,603 kW, and the total electricity cost from the available ten months information was $137,333.80

The results of the simulation show that the annual electricity consumption for the baseline model was 3,215,500 kWh, which represents a -0.29% difference from the 2007 utility bills. The simulated peak electricity demand was 5,850 kW, which represents a +4.41% difference from the utility bills. The electricity cost is $136,334.00, which is within -0.73% of the utility bills. Using the available data, the simulation baseline seems to be a reasonably accurate approximation of the as-built condition. The results of this comparison are summarized in Table 4.1.

Table 4.1 - 2007 utility bills and baseline model annual electricity consumption, peak electricity demand, and electricity cost summary

Information	2007 Utility Bills	Baseline Model	% Difference	Note
Electricity consumption	3,225,002	3,215,500	-0.29%	Annual
Peak electricity demand	5,603	5,850	+4.41%	10 Months
Electricity cost	$137,333.80	$136,334.00	-0.73%	10 Months

Using monthly data, the baseline model simulation also closely predicts electricity consumption, peak electricity demand, and electricity cost; the results are presented in table 4.2. The accuracy of the model in predicting past electricity consumption varies significantly from month to month. However, the model predictions are most accurate for the most recent 5 months falling within 10% of the 2007 billed electricity use, while the predictions for April, October, and December fall within 5% of the billed usage.

Table 4.2 - 2007 utility bills and baseline model monthly electricity consumption, peak electricity demand, and electricity cost summary

Month	2007 Utility Bills				Baseline Model				Differences		
	Energy		Demand	Price	Energy		Demand	Price	Energy	Demand	Price
	Electric (kWh)	Intensity (Btu/ft^2)	Electric (kW)	Electricity ($)	Electric (kWh)	Intensity (Btu/ft^2)	Electric (kW)	Electricity ($)	%	%	%
Jan-07	277,485	11,050	753	14,655.23	235,300	9,370	521	12,032.00	-15.20	-30.78	-17.90
Feb-07	305,732	12,174	652	15,538.06	214,200	8,530	530	11,148.00	-29.94	18.75	-28.25
Mar-07	322,760	12,853	625	16,185.14	258,400	10,290	580	13,236.00	-19.94	7.27	-18.22
Apr-07	277,045	11,032	474	13,684.01	263,500	10,493	581	13,463.00	-4.89	-22.51	-1.62
May-07	259,299	10,325	432	12,771.01	290,900	11,584	601	14,721.00	12.19	-38.97	15.27
Jun-07	240,471	9,576	443	11,988.35	296,000	11,787	623	15,014.00	23.09	-40.77	25.24
Jul-07	252,233	10,044	-	-	305,300	12,157	621	15,412.00	21.04	-	-
Aug-07	259,334	10,327	483	12,946.54	320,200	12,751	614	16,039.00	23.47	-27.03	23.89
Sep-07	253,943	10,112	463	12,643.62	279,000	11,110	624	14,286.00	9.87	-34.72	12.99
Oct-07	272,849	10,865	555	13,779.33	276,200	10,998	618	14,140.00	1.23	-11.28	2.62
Nov-07	258,763	10,304	-	-	239,000	9,517	574	12,374.00	-7.64	-	-
Dec-07	245,088	9,760	721	13,142.50	237,500	9,457	558	12,255.00	-3.10	22.65	-6.75
Total	**3,225,002**	**128,422**	**5,603**	**137,333.80**	**3,215,500**	**128,044**	**7,045**	**164,120.00**	-	-	-
Average	**268,750**	**10,702**	**560**	**13,733.38**	**267,958**	**10,670**	**587**	**13,676.67**	-	-	-

The following figure compares monthly electric consumption from the baseline model to actual consumption from electricity billing records for 2007.

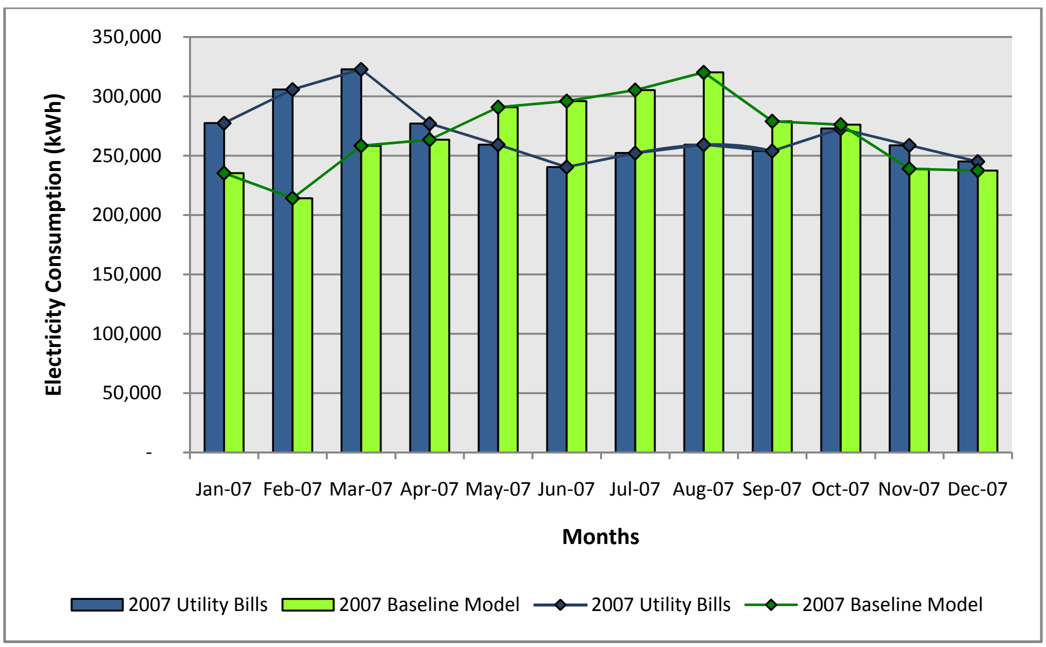


Figure 4.1 - Comparison of baseline model electricity consumption to 2007 billing records

To establish confidence concerning the accuracy of the baseline model, the simulated monthly electric consumption and billed data were evaluated using a statistical approach to compare two sample means. The pooled t-test was used to test the equality of the billing records mean (μ_1) and the baseline model mean (μ_2). The test hypothesis was $H_0: \mu_1 = \mu_2$ versus the null hypothesis, $H_a: \mu_1 \neq \mu_2$, where H_0 suggests that the means of the two samples are equal. This statistical procedure is based on the assumption that the samples are independent random samples drawn from a normal population, in which standard deviations, σ_1 and σ_2, are unknown but are approximately equal. From this assumption, the samples are assumed independent and fit in a normal distribution curve, where the equality of standard deviations is required.

The statistical procedure used for testing standard deviations is the F-test for two sample variances based on independent random samples drawn from a normal population. The test hypothesis is $H_0: \sigma_1 = \sigma_2$ versus $H_a: \sigma_1 \neq \sigma_2$. H_0 suggests that the standard deviations of two samples are equal, σ_1 and σ_2 are the standard deviations of the billing records and the simulated electric consumption.

Figure 4.2 shows the results of the two sample F-test for variances for the hypothesis $H_0: \sigma_1 = \sigma_2$ versus $H_a: \sigma_1 \neq \sigma_2$, where the H_0 rejection region for the 95% level of test ($\alpha = 0.05$) can be either $F < \mathrm{F}_{11,11,0.975} = 0.29$, or $F > \mathrm{F}_{11,11,0.025} = 3.48$. From the test results, the null hypothesis (H_0) is accepted since the test statistic $F = 0.57$ is neither less than 0.29, nor greater than 3.48. In addition, Figure 4.3 shows the plot of the F distribution curve which indicates that the test statistic $F = 0.57$ does not fall in the rejection regions colored in blue. Therefore, the electricity billing records and the simulated baseline model electricity consumption standard deviations are approximately equal, which supports the pooled t-test procedure.

Figure 4.4 illustrates the pooled t-test for the hypothesis $H_0: \mu_1 = \mu_2$ versus $H_a: \mu_1 \neq \mu_2$, in which the H_0 rejection region for the 95% level of test ($\alpha = 0.05$)) is $|t| > t_{22,0.025} = 2.074$. The results in Figure 4.4 show that the null hypothesis is accepted because the test statistic $|t| = 0.068$ is less than 2.074. Also, Figure 4.5 shows that the test statistic $|t| = 0.068$ does not fall into the rejection regions colored in blue. As a result, the means of the electricity utility bills and the simulated baseline model are assumed equal which supports the accuracy of the baseline model in predicting monthly electricity consumption of the KnowledgeWorks I and II buildings. It is important to note that the procedure should be repeated when more utility bills are available.

```
                 KnowledgeWorks I&II
             Hypothesis Test on Variances
            Utility Bills VS Baseline Model
  Two Sample Test for Variances of A_kWh and B_kWh

                  Sample Statistics

 Group          N      Mean     Std. Dev.   Variance
 ---------------------------------------------------
 A_kWh         12   268750.2       24450    5.9782E8
 B_kWh         12   267958.3       32277    1.0418E9

                   Hypothesis Test

  Null hypothesis:    Variance 1 / Variance 2 =  1
  Alternative:        Variance 1 / Variance 2 ^= 1

               - Degrees of Freedom -
     F          Numer.     Denom.         Pr > F
   ----------------------------------------------
    0.57          11          11          0.3709

95% Confidence Interval of the Ratio of Two Variances

            Lower Limit     Upper Limit
            -----------     -----------
               0.1652          1.9933
```

Figure 4.2 - Two sample F-test for variances of billing records and baseline model electricity consumption results from SAS9.1 program.

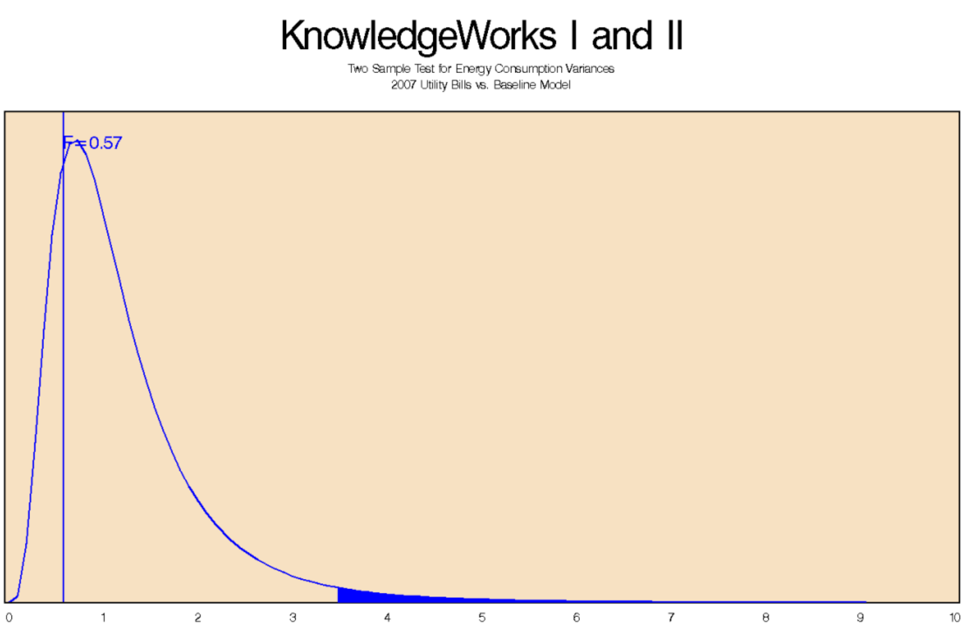


Figure 4.3 - F-distribution plot of billing records and baseline model electricity consumption results from SAS9.1 program.

```
                      KnowledgeWorks I&II
              Hypothesis Test on Means (Pooled t-Test)
                  Utility Bills VS Baseline Model
          Two Sample t-test for the Means of A_kWh and B_kWh

                          Sample Statistics

      Group           N       Mean     Std. Dev.   Std. Error
      ----------------------------------------------------
       A_kWh         12   268750.2       24450       7058.2
       B_kWh         12   267958.3       32277       9317.7

                           Hypothesis Test

            Null hypothesis:    Mean 1 - Mean 2 =  0
            Alternative:        Mean 1 - Mean 2 ^= 0

      If Variances Are    t statistic      Df       Pr > t
      ----------------------------------------------------
      Equal                     0.068         22       0.9466
      Not Equal                 0.068      20.50       0.9466

  95% Confidence Interval for the Difference between Two Means

                   Lower Limit     Upper Limit
                   -----------     -----------
                    -23450.1        25033.76
```

Figure 4.4 - Two sample t-test for the means of billing records and baseline model electric consumption results from SAS9.1 program.

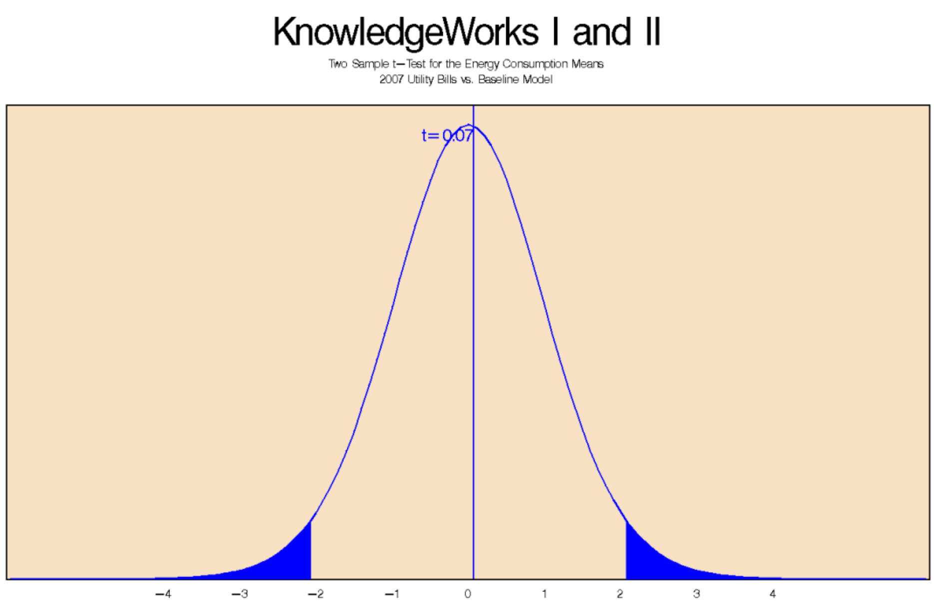


Figure 4.5 - t-distribution plot of billing records and baseline model electricity consumption results from SAS9.1 program.

Unlike the prediction of annual electricity consumption, the simulation results for peak energy demand in Table 4.2 show that only March's energy demand falls within 10% of the measured values. In addition, there are 4 months February, April, October, and December that fall within the range of 15-25% of the measured values.

The determination of the peak energy demand for both the actual utility bills and the simulated values are based on a similar structure which is the highest value from a 15-minute demand meter in kilowatts. The comparison results in Figure 4.6 illustrates that the baseline model simulation may be over estimating summer building uses. It is possible that the assumed internal load schedules (occupancy, lighting, and equipment schedules) were different from the real situation. It is important to note that the buildings have been occupied by many clients. The onsite detailed observation for internal load schedules were based on observation and this may account for the overestimation.

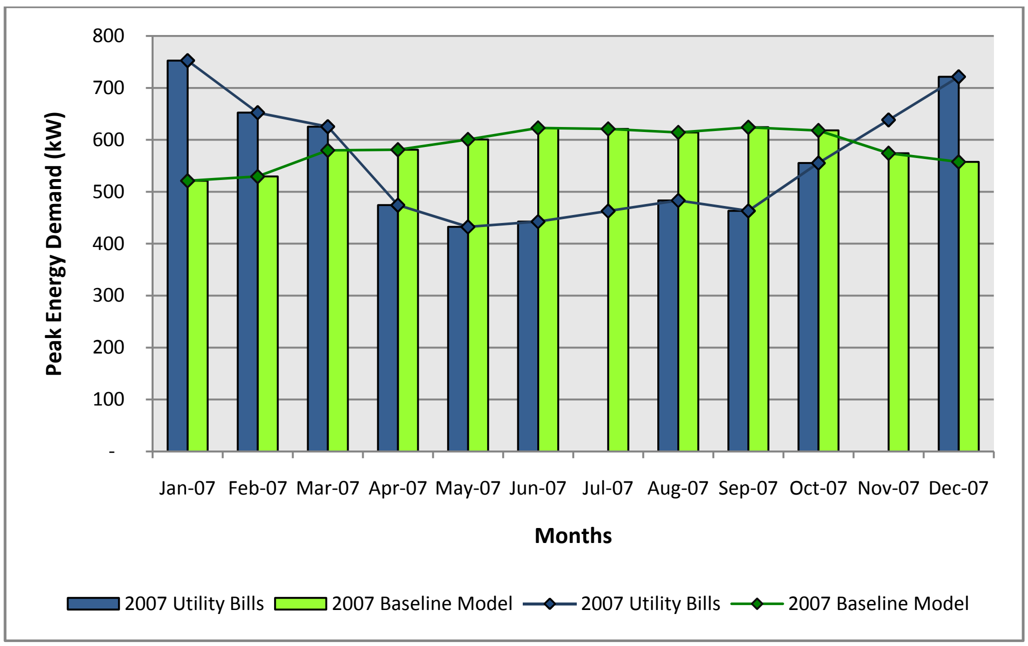


Figure 4.6 - Comparison of baseline model peak electric demand to 2007 billing records

Figure 4.7 shows the results of the two sample test for variances, where the hypothesis was $H_0: \sigma_1 = \sigma_2$ versus the alternative hypothesis $H_a: \sigma_1 \neq \sigma_2$. In this case, the H_0 rejection region for the 95% level of test ($\alpha = 0.05$) can be either $F < \mathrm{F}_{9,11,0.975} = 0.27$, or $F > \mathrm{F}_{9,11,0.025} = 3.59$. From the test results, the test variable falls within the rejection region ($F = 10.85$ is greater than 3.59). The F distribution plot, show in Figure 4.8, also indicates that the test statistic

$F = 10.85$ falls within the rejection region. Therefore, the electricity records and the simulation baseline model peak electric demand standard deviations are unequal and therefore the pooled t-test procedure is inappropriate.

Nevertheless, there is another statistic procedure that can be applied to this case. The two sample Welch t-test for two means, also called the Satterthwaite t-test in the SAS9.1 program, is a procedure used for testing the equality between the billing records mean (μ_1) and the simulation mean (μ_2) based on the assumption that the samples are independent and randomly drawn from a normal population, in which case standard deviations, σ_1 and σ_2, are completely unknown.

```
                 KnowledgeWorks I&II
              Hypothesis test on Variances
             Utility Bills VS Baseline Model
      Two Sample Test for Variances of A_kW and B_kW

                    Sample Statistics

  Group          N      Mean    Std. Dev.   Variance
  --------------------------------------------------
  A_kW          10     560.3      119.54    14288.88
  B_kW          12  587.1167      36.291    1317.047

                     Hypothesis Test

   Null hypothesis:    Variance 1 / Variance 2 =  1
   Alternative:        Variance 1 / Variance 2 ^= 1

                 - Degrees of Freedom -
       F         Numer.    Denom.        Pr > F
    ----------------------------------------------
     10.85           9        11         0.0005

95% Confidence Interval of the Ratio of Two Variances

                Lower Limit    Upper Limit
                -----------    -----------
                   3.0238        42.443
```

Figure 4.7 - Two sample F-test for variances of billing records and baseline model peak electric demand results from SAS9.1 program.

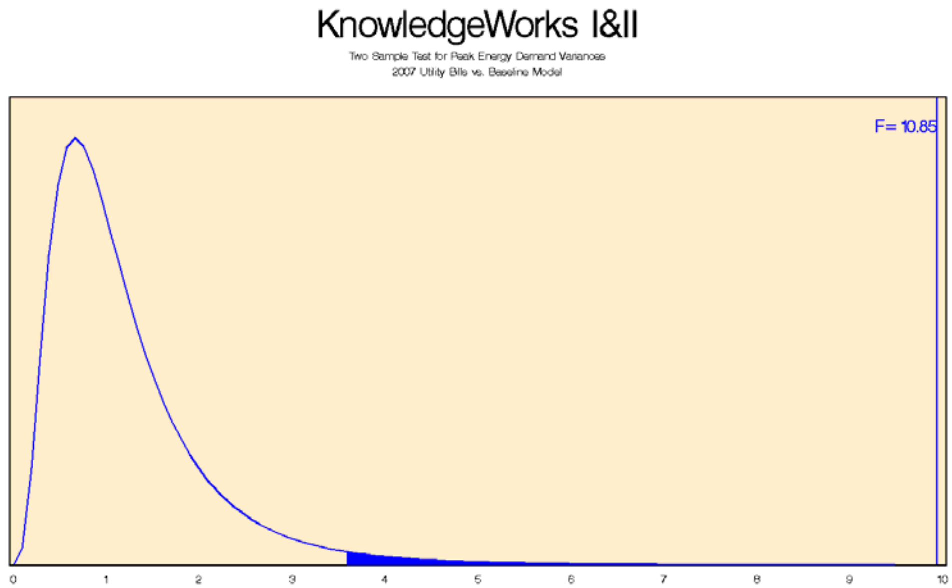


Figure 4.8 - F distribution plot of billing records and baseline model peak electricity demand results from SAS9.1 program.

Figure 4.9 shows the two sample Welch t-test for the hypothesis H_0: $\mu_1 = \mu_2$ versus H_a: $\mu_1 \neq \mu_2$, where the H_0 rejection region for the 95% level of test ($\alpha = 0.05$) is $|t| > t_{10.4,0.025} \approx 2.21$. The results in Figure 4.9 indicate that the null hypothesis is accepted because the test statistic $|t| = 0.68$ is less than 2.16. In addition, the Welch distribution plot in Figure 4.10 indicates that the test statistic $|t| = 0.68$ does not fall within the rejection regions colored in blue. Therefore, the means of peak electric demand from the utility bills and the simulation baseline model assumed to be equal. The outcome supports the accuracy of the baseline model in predicting the monthly peak electric demand of the KnowledgeWorks I and II buildings.

For electricity cost, the results in Table 4.2 demonstrate that there are two of ten months of available information falling within 5% of the actual billed electricity cost. The results show that the simulation model is accurate enough for predicting the electricity cost. Figure 4.11 compares the monthly electricity cost from the baseline model to the actual cost from electricity records for 2007.

```
                          KnowledgeWorks I&II
                 Hypothesis Test on Means (Welch t-Test)
                     Utility Bills VS Baseline Model
        Two Sample Welch T-test for the Means of A_kW and B_kW

                           The TEST Procedure

                               Statistics

                            Lower CL            Upper CL  Lower CL            Upper CL
Variable  HS             N      Mean     Mean       Mean  Std Dev   Std Dev   Std Dev   Std Err

score     D             12    564.06   587.12     610.17    25.708    36.291    61.618    10.476
score     S             10    474.79    560.3     645.81    82.221    119.54    218.23    37.801
score     Diff (1-2)          -48.73   26.817     102.36    64.711    84.584    122.14    36.216

                                T-Tests

            Variable     Method           Variances      DF     t Value     Pr > |t|

            score        Pooled           Equal          20        0.74       0.4676
            score        Satterthwaite    Unequal      10.4        0.68       0.5092
```

Figure 4.9 - Two sample Welch t-test for means of billing records and baseline model peak electric demand results from SAS9.1 program.

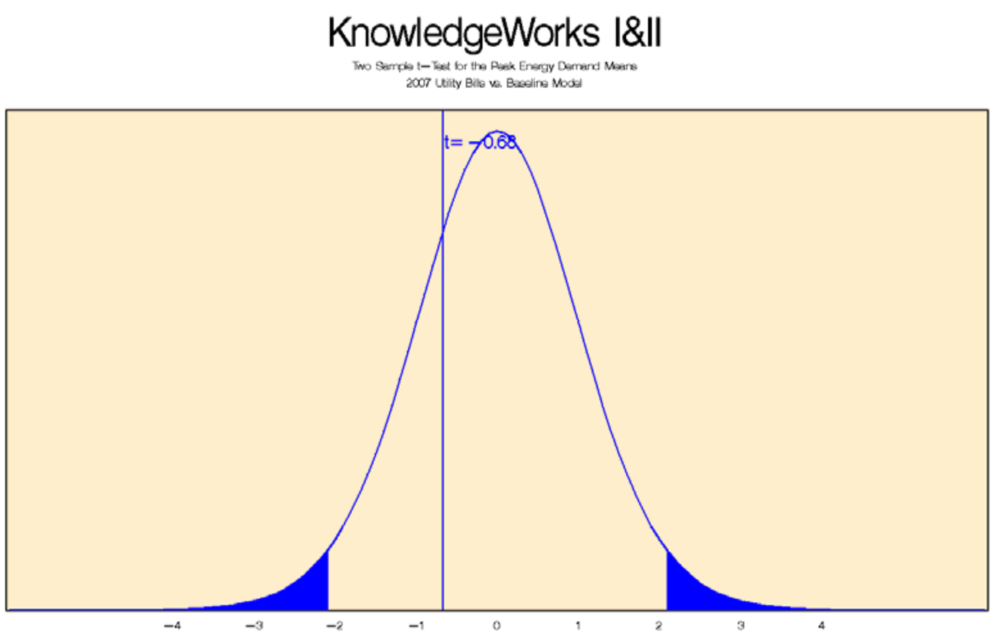


Figure 4.10 - Welch-distribution plot of billing records and baseline model peak electricity demand results from SAS9.1 program.

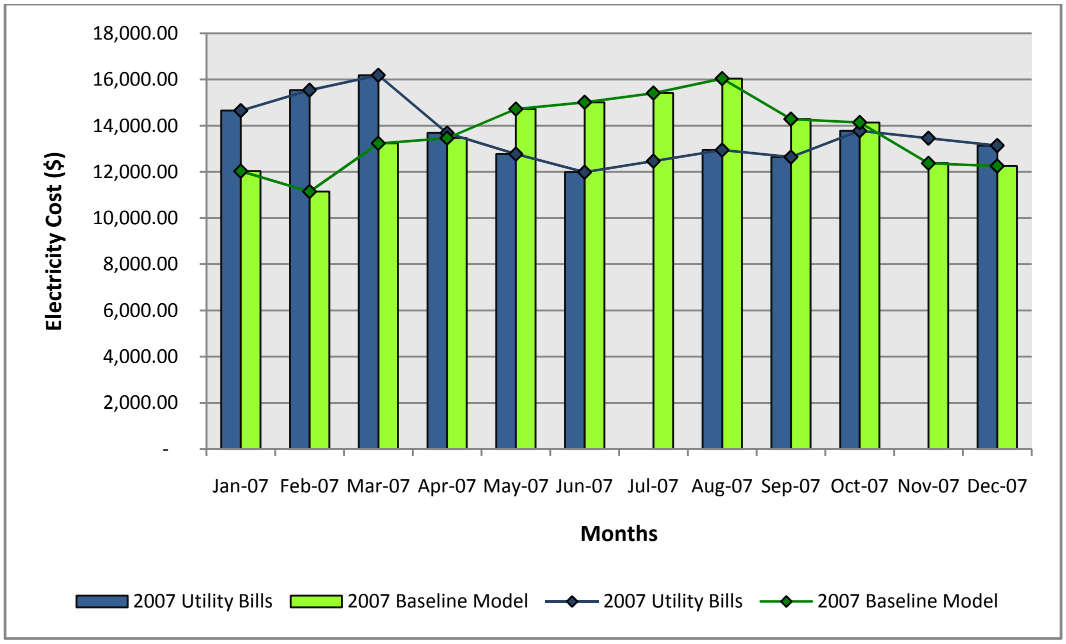


Figure 4.11 - Comparison of baseline model electricity cost to 2007 billing records

In Figure 4.12, the H_0 rejection region of the hypothesis $H_0: \sigma_1 = \sigma_2$ versus $H_a: \sigma_1 \neq \sigma_2$ for the 95% level of test $(\alpha = 0.05)$ can be either $F < \mathrm{F}_{9,11,0.975} = 0.27$, or $F > F_{11,11,0.025} = 3.59$. From the test results in Figure 4.8, the null hypothesis (H_0) is accepted because the test statistic $F = 0.79$ is neither less than 0.27, nor greater than 3.59. In addition, Figure 4.13 shows that the test statistic $F = 0.79$ does not fall within the rejection regions shown in the blue areas. Thus, electric billing records and the simulated baseline model electricity cost standard deviations are approximately equal and subsequently the pooled t-test procedure is appropriate for performing two sample test for the means.

Figure 4.14 presents the results of the pooled t-test for the hypothesis $H_0: \mu_1 = \mu_2$ versus $H_a: \mu_1 \neq \mu_2$. The H_0 rejection region for the 95% level of test $(\alpha = 0.05)$ is $|t| > t_{20,0.025} = 2.086$. The test results show that the null hypothesis is accepted because the test statistic $|t| = 0.093$ is less than 2.086. Figure 4.15 shows that the test statistic $|t| = 0.093$ does not fall within the rejection regions shown in the blue areas of the t distribution curve. Therefore, the means of electricity cost from the utility bills and the simulated baseline model values are assumed equal which supports the accuracy of the baseline model in predicting the monthly electricity cost of the KnowledgeWorks I and II buildings.

```
                  KnowledgeWorks I&II
              Hypothesis Test on Variances
              Utility Bills VS Baseline Model
     Two Sample Test for Variances of A_$ and B_$

                   Sample Statistics

 Group           N       Mean     Std. Dev.   Variance
 --------------------------------------------------
 A_$            10   13733.38      1343.7     1805496
 B_$            12   13676.67        1511     2283096

                    Hypothesis Test

  Null hypothesis:    Variance 1 / Variance 2 =  1
  Alternative:        Variance 1 / Variance 2 ^= 1

                - Degrees of Freedom -
     F          Numer.     Denom.         Pr > F
   ----------------------------------------------
    0.79             9         11           0.7362

95% Confidence Interval of the Ratio of Two Variances

               Lower Limit    Upper Limit
               -----------    -----------
                 0.2204         3.0937
```

Figure 4.12 - Two sample F-test for variances of billing records and baseline model electricity cost results from SAS9.1 program.

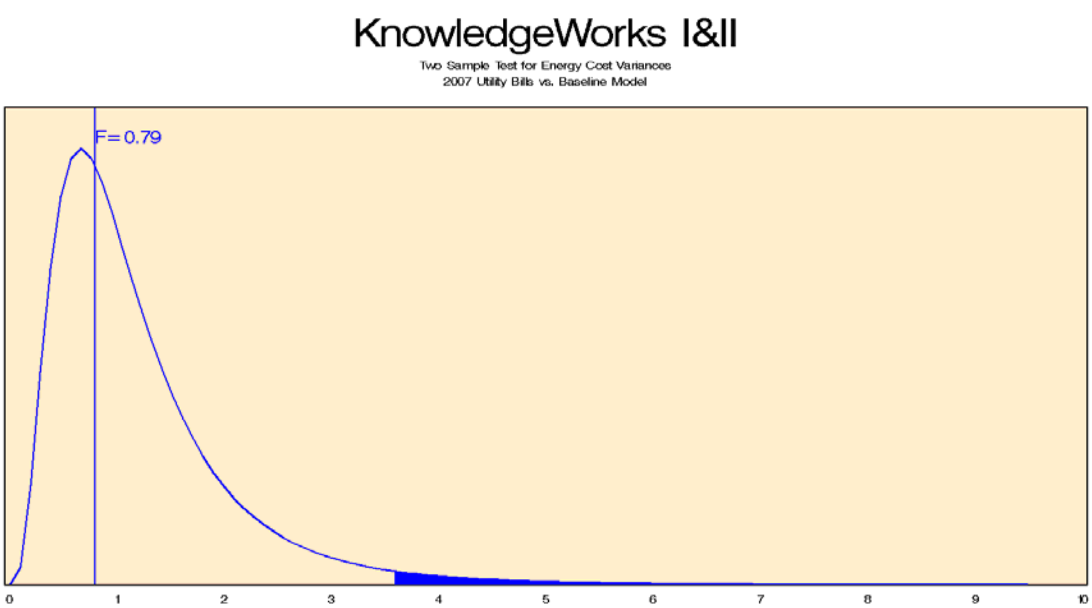


Figure 4.13 - F-distribution plot of billing records and baseline model electricity cost results from SAS9.1 program.

KnowledgeWorks I&II
Hypothesis Test on Means (Pooled t-Test)
Utility Bills VS Baseline Model
Two Sample t-test for the Means of A_$ and B_$

Sample Statistics

Group	N	Mean	Std. Dev.	Std. Error
A_$	10	13733.38	1343.7	424.91
B_$	12	13676.67	1511	436.19

Hypothesis Test

Null hypothesis: Mean 1 - Mean 2 = 0
Alternative: Mean 1 - Mean 2 ^= 0

If Variances Are	t statistic	Df	Pr > t
Equal	0.092	20	0.9275
Not Equal	0.093	19.89	0.9267

95% Confidence Interval for the Difference between Two Means

Lower Limit	Upper Limit
-1227.75	1341.17

Figure 4.14 - Two sample t-test for the means of billing records and baseline model electric consumption results from SAS9.1 program.

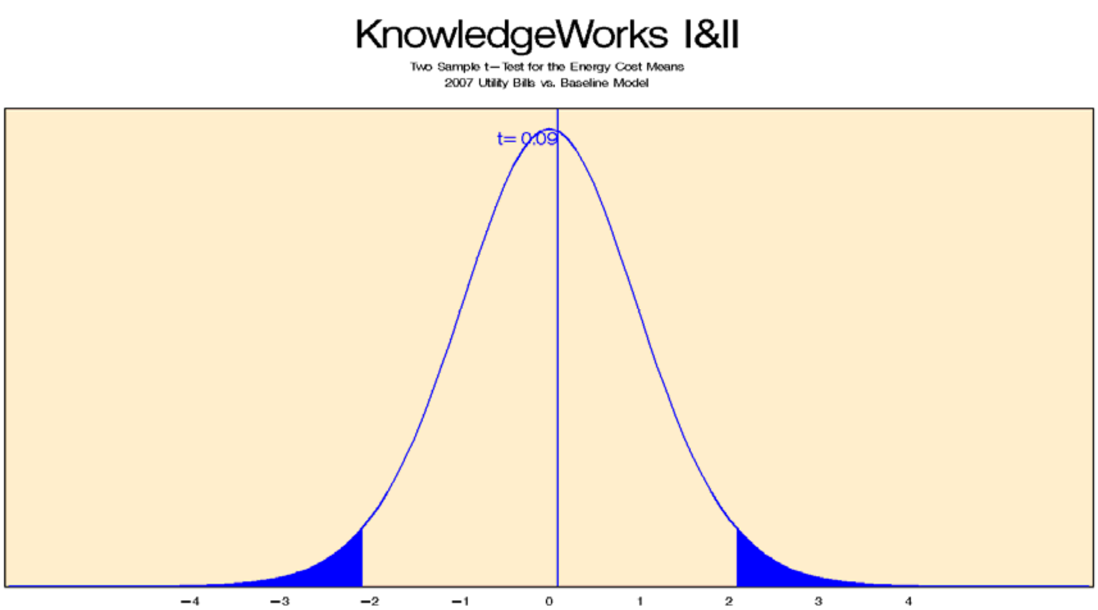


Figure 4.15: t-distribution plot of billing records and baseline model electricity cost results from SAS9.1 program.

In summary, both the comparison of direct results and statistical tests on the means of electric consumption, peak electric demand, and electricity cost, between the actual utility records and the simulated baseline model predictions indicate that the simulated base model is acceptable for simulating the KnowledgeWorks I and II buildings with the alternative HVAC system scenarios. It is important to note that the variations from simulated and actual values may be due to using TMY2 weather files and not actual data for the utility record period. Also, the variations may be due to using the TMY2 file for Roanoke location rather than Blacksburg.

Baseline Model Results

The 2006 baseline model indicates that the KnowledgeWorks I and II buildings annually consumes 3,214,400 kWh. The total energy consumption can be broken down into end-use categories including, but not limited to, miscellaneous equipment, area lighting, and space heating and cooling. Figure 4.16 shows that, on average, equipment use accounted for 53% of the total electricity consumption in 2006. After equipment, the greatest proportion of the buildings' energy consumption is attributed to area lighting (27%), space cooling (14%), and ventilation fans (4%). Space heating accounts for only 1% of the electricity consumption and all other uses combine to less than 3%.

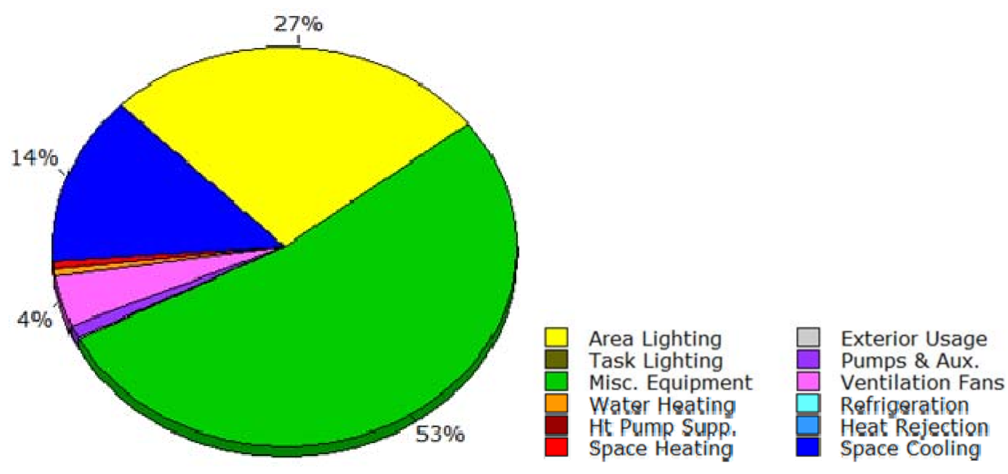


Figure 4.16 - Predicted annual energy consumption by end-use.

Table 4.3 shows the predicted monthly electricity consumption by end-use categories (Equipment, Lighting, HVAC, and Other). The relative proportion of electricity consumption by end-use differ little over the course of the year, with the obvious exceptions of space heating and space cooling, which are influenced by the seasonal weather conditions.

Figure 4.17 indicates that the HVAC system consumes less energy in the heating months than in the cooling months, in which the highest energy consumption occurs in August. In contrast, the total electricity consumption of other end-use categories varies relatively little over the course of the year. The totals for equipment, lighting, and others are relatively constant, with some small variability based on the hours of occupancy per month.

Table 4.3 - 2006 baseline model monthly energy consumption by end-use categories (kWh × 1,000)

Category	End-use	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Equipment	Misc. Equip.	138.3	129.1	150.1	139.8	145.5	144.5	140.7	150.1	137.8	142.8	135.5	140.6	1,694.9
Equipment	Sub-total	138.3	129.1	150.1	139.8	145.5	144.5	140.7	150.1	137.8	142.8	135.5	140.6	1,694.9
	% of total	59.8%	60.5%	57.2%	53.8%	50.1%	48.0%	46.7%	46.8%	48.4%	52.7%	56.5%	58.8%	52.7%
Lighting	Ext. Usage	0.5	0.4	0.5	0.4	0.3	0.3	0.3	0.5	0.5	0.5	0.5	0.5	5.4
Lighting	Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Lighting	Area Lights	70.4	66.1	77.5	71.9	74.7	74.6	72.0	77.5	70.6	73.2	69.1	71.9	869.3
Lighting	Sub-total	70.9	66.5	78.0	72.3	75.0	74.9	72.3	78.0	71.1	73.7	69.6	72.4	874.7
	% of total	30.7%	31.2%	29.7%	27.8%	25.8%	24.9%	24.0%	24.3%	24.9%	27.2%	29.0%	30.3%	27.2%
HVAC	Space Cool	0.8	0.7	17.7	32.5	53.9	64.8	71.5	74.9	60.2	38.9	19.1	6.8	441.7
HVAC	Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
HVAC	Space Heat	7.6	4.5	1.4	0.2	0.1	-	-	-	-	0.2	1.5	5.3	20.8
HVAC	HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
HVAC	Vent. Fans	10.9	10.0	11.5	10.9	11.3	11.1	11.0	11.5	10.7	11.1	10.6	11.0	131.7
HVAC	Pumps & Aux.	1.2	1.1	1.9	2.5	3.4	3.9	4.2	4.6	4.0	3.1	2.1	1.5	33.3
HVAC	Sub-total	20.5	16.3	32.5	46.1	68.7	79.8	86.7	91.0	74.9	53.3	33.3	24.6	627.5
	% of total	8.9%	7.6%	12.4%	17.7%	23.6%	26.5%	28.8%	28.4%	26.3%	19.7%	13.9%	10.3%	19.5%
Other	Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Other	Hot Water	1.5	1.5	1.8	1.6	1.6	1.5	1.3	1.4	1.2	1.3	1.3	1.5	17.4
Other	Sub-total	1.5	1.5	1.8	1.6	1.6	1.5	1.3	1.4	1.2	1.3	1.3	1.5	17.4
	% of total	0.6%	0.7%	0.7%	0.6%	0.6%	0.5%	0.4%	0.4%	0.4%	0.5%	0.5%	0.6%	0.5%
Total		231.1	213.4	262.2	259.8	290.7	300.8	301.0	320.4	285.0	271.1	239.7	239.1	3,214.4

The energy consumption for the HVAC system was 627,500 kWh in 2006. The vast majority of the HVAC electricity consumption is space cooling which accounts for 70% of the total HVAC consumption. The second most influential energy consuming part of the system is the ventilation fans which account for 21% of the consumption. Other components include pumps and auxiliary equipment, and space heating that consume 5% and 3% respectively. This pattern of energy consumption reflects that the KnowledgeWorks I and II buildings are dominated by internal loads, which is common for this building type. It is interesting to note that the buildings have laboratory space that require a high rate of air change which could be the cause of high energy consumption associated with fans operation.

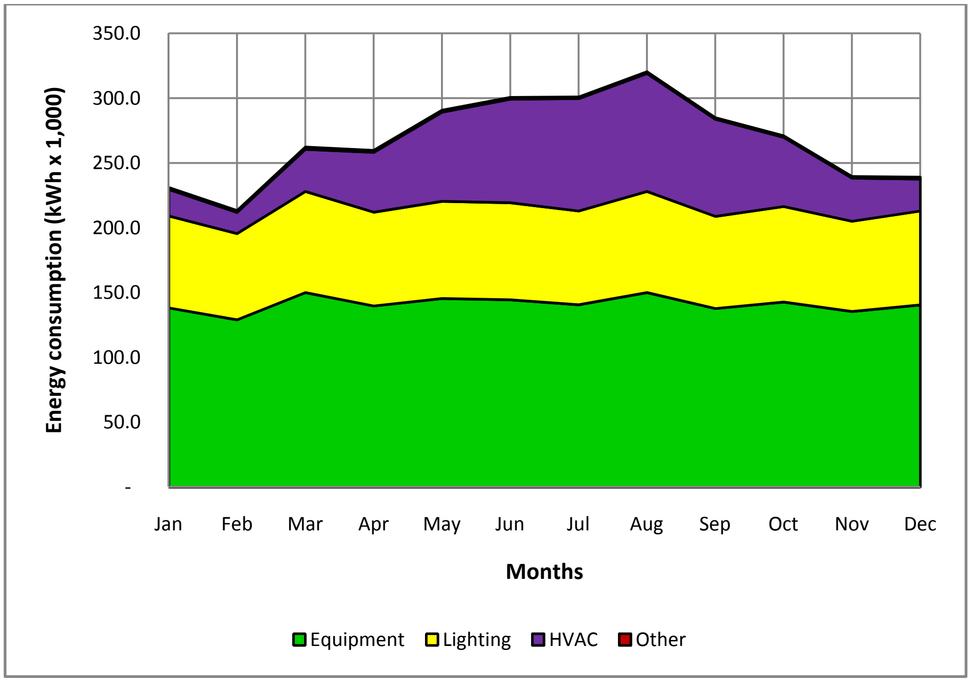


Figure 4.17 - Monthly energy consumption by end-use category

In addition to the total electricity consumption results, Figure 4.18 shows the predicted annual peak energy demand by end-use occurs in September. Figure 4.18 illustrates that equipment energy use is the greatest determinant of the demand (39%) followed by space cooling (34%) and area lighting (20%), with ventilation fans making up the remainder.

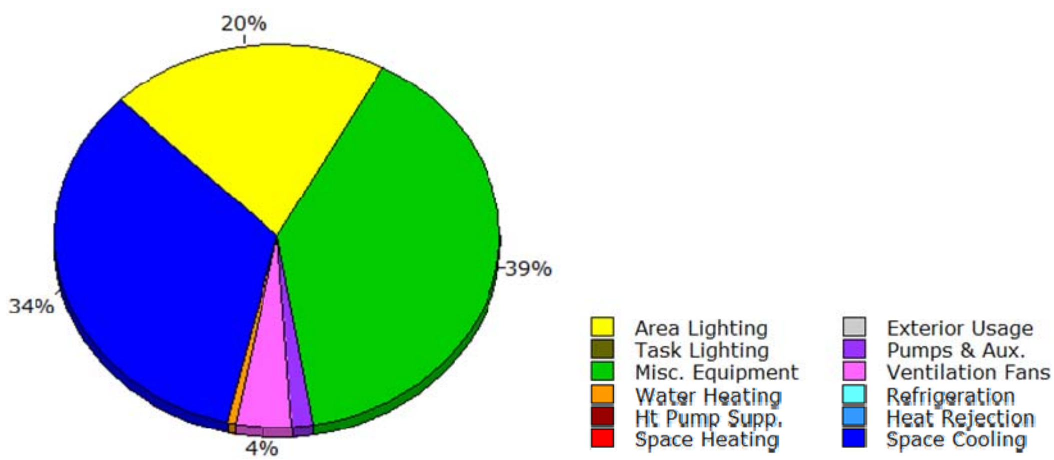


Figure 4.18 - Predicted annual peak energy demand by end-use.

The following figure illustrates the peak day load profiles for winter (January) and summer (September) loads representing the lowest and the highest peak energy demand months in a one year period. The peak day load profiles for all months are shown in Appendix C. Peak day load profiles indicate that the majority of load types are relatively constant for all months, while space cooling and heating change dynamically corresponding to the seasonal weather changes.

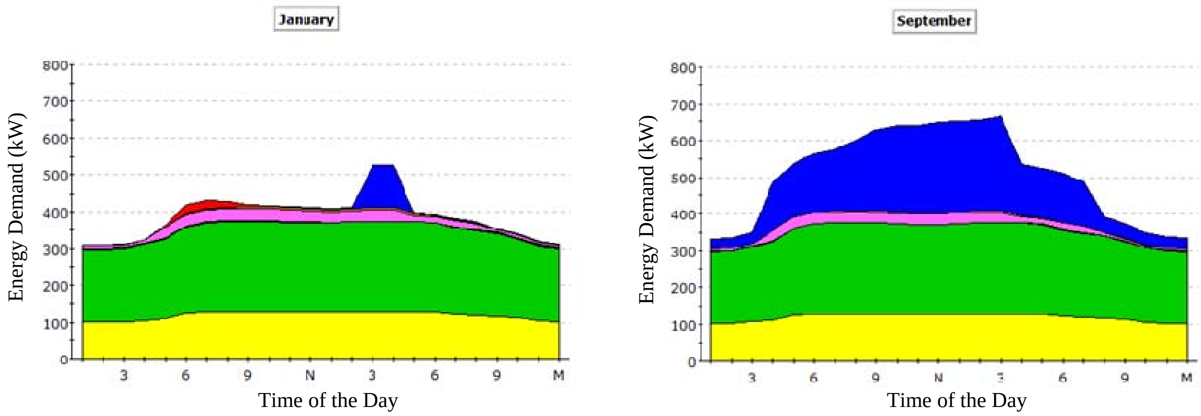


Figure 4.19: Monthly peak day load profiles

Table 4.4 shows the monthly peak energy demand by end-use categories (Equipment, HVAC, Lighting, and Other). It is interesting to note that equipment energy use is the most dominant peak demand factor for the total peak demand, whereas the HVAC system energy use dominates the peak energy demand in September.

Table 4.4 - 2006 baseline model monthly peak energy demand by end-use categories (kW)

Category	End-use	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Equipment	Misc. Equip.	246.1	246.1	246.1	246.1	243.7	246.1	246.1	246.1	246.1	246.1	243.7	245.6	2,947.9
Equipment	Sub-total	246.1	246.1	246.1	246.1	243.7	246.1	246.1	246.1	246.1	246.1	243.7	245.6	2,947.9
	% of total	47%	46.5%	42.2%	42.2%	40.2%	39.5%	39.8%	40.0%	39.4%	39.6%	42.4%	43.7%	41.7%
HVAC	Space Cool	110.8	119.1	169.8	169.4	194.5	210.0	205.3	203.3	211.9	209.0	164.1	148.2	2,115.4
HVAC	Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
HVAC	Space Heat	0.2	0.2	-	-	-	-	-	-	-	-	-	-	0.5
HVAC	HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
HVAC	Vent. Fans	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	307.4
HVAC	Pumps & Aux.	5.4	5.5	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	103.6
HVAC	Sub-total	142.0	150.4	204.7	204.3	229.4	244.9	240.2	238.2	246.8	243.9	199.0	183.1	2,526.9
	% of total	27.3%	28.4%	35.1%	35.0%	37.9%	39.3%	38.8%	38.7%	39.5%	39.2%	34.6%	32.6%	35.8%
Lighting	Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Lighting	Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Lighting	Area Lights	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	1,535.7
Lighting	Sub-total	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	1,535.7
	% of total	24.6%	24.2%	21.9%	21.9%	21.1%	20.5%	20.7%	20.8%	20.5%	20.6%	22.3%	22.8%	21.7%
Other	Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Other	Hot Water	4.8	5.0	5.0	4.9	4.8	4.2	4.0	3.8	3.8	3.9	4.4	4.7	53.2
Other	Sub-total	4.8	5.0	5.0	4.9	4.8	4.2	4.0	3.8	3.8	3.9	4.4	4.7	53.2
	% of total	0.9%	0.9%	0.9%	0.8%	0.8%	0.7%	0.6%	0.6%	0.6%	0.6%	0.8%	0.8%	0.8%
Total		521.0	529.4	583.8	583.2	605.9	623.2	618.3	616.0	624.6	621.9	575.0	561.4	7,063.7

For energy costs, the simulation results summarized in Table 4.5 show that the lowest value occurs in February ($11,113) and the highest occurs in August ($16,054), with an annual total of $164,137. The energy cost is higher during the cooling months when compared to the heating months, corresponding to energy required for space cooling. For example, the energy costs in August increase from February by about 44%, when space cooling energy consumption increases from 0.7 MWh to 74.9 MWh and peak energy demand increases from 150.4 kW to 238.2 kW.

Table 4.5 - 2006 baseline model monthly energy cost ($)

Energy Type	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Electricity	11,852	11,113	13,417	13,307	14,729	15,223	15,218	16,054	14,544	13,934	12,409	12,337	164,137

4.2 Alternative Model I

Alternative model I simulates the KnowledgeWorks I and II buildings with a package VAV cooling and hot water coil heating system. Table 4.6 presents the simulation results for monthly energy consumption for the baseline model and alternative model I. In table 4.6, natural gas consumed by the VAV system for hot water heating is converted to an equivalent electricity unit to make comparisons more direct. From the table, the GSHP scheme can reduce the overall building energy consumption by 123,700 kWh (3.7%) annually when compared to the VAV scheme. This electric energy savings corresponds to a reduction of 69.3 metric tons of carbon dioxide emission each year (for an emission rate of 1,232 lb per MWh EPA's eGRID database for Virginia's power generation). This energy savings comes only from the HVAC end-use category.

Table 4.6 - Monthly energy consumption in kWh ×1,000 from baseline model and alternative model I

End-use	Case	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Equipment	Baseline	138.3	129.1	150.1	139.8	145.5	144.5	140.7	150.1	137.8	142.8	135.5	140.6	1694.9
	Alternative I	138.3	129.1	150.1	139.8	145.5	144.5	140.7	150.1	137.8	142.8	135.5	140.6	1694.9
	Saving	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%
Lighting	Baseline	70.9	66.5	78.0	72.3	75.0	74.9	72.3	78.0	71.1	73.7	69.6	72.4	874.7
	Alternative I	70.9	66.5	78.0	72.3	75.0	74.9	72.3	78.0	71.1	73.7	69.6	72.4	874.7
	Saving	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		0.0%	0.0%	0.0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
HVAC	Baseline	20.5	16.3	32.5	46.1	68.7	79.8	86.7	91.0	74.9	53.3	33.3	24.6	627.5
	Alternative I	46.6	37.4	41.6	49.5	73.5	86.3	96.5	97.8	79.6	57.7	41.0	44.0	751.2
	Saving	26.1	21.1	9.1	3.4	4.8	6.5	9.8	6.8	4.7	4.4	7.7	19.4	123.7
		56%	56%	22%	7%	6%	8%	10%	7%	6%	8%	19%	44%	16%
Other	Baseline	1.5	1.5	1.8	1.6	1.6	1.5	1.3	1.4	1.2	1.3	1.3	1.5	17.4
	Alternative I	1.5	1.5	1.8	1.6	1.6	1.5	1.3	1.4	1.2	1.3	1.3	1.5	17.4
	Saving	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%
Total	Baseline	231.2	213.4	262.4	259.8	290.8	300.7	301.0	320.5	285.0	271.1	239.7	239.1	3214.5
	Alternative I	257.3	234.5	271.5	263.2	295.6	307.2	310.8	327.3	289.7	275.5	247.4	258.5	3338.2
	Saving	26.1	21.1	9.1	3.4	4.8	6.5	9.8	6.8	4.7	4.4	7.7	19.4	123.7
		10.1%	9.0%	3.3%	1.3%	1.6%	2.1%	3.1%	2.1%	1.6%	1.6%	3.1%	7.5%	3.7%

Table 4.7 shows that the GSHP system consumes 123,700 kWh or 16.5% less energy than the VAV system. In addition, the monthly results in Table 4.7 illustrate that the GSHP system consumes less energy than the VAV system for space heating, ventilation fans, and space cooling. It is interesting to note that the GSHP system consumes more energy than the VAV sytem for pumps and auxilary end-use, especially during the cooling months.

Table 4.7 - Monthly energy consumption (HVAC end-use) in kWh ×1,000 from baseline model and alternative model I

HVAC End-use	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Baseline Model (Ground Source Heat Pump)													
Space Cool	0.8	0.7	17.7	32.5	53.9	64.8	71.5	74.9	60.2	38.9	19.1	6.8	441.7
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	7.6	4.5	1.4	0.2	0.1	-	-	-	-	0.2	1.5	5.3	20.8
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	10.9	10.0	11.5	10.9	11.3	11.1	11.0	11.5	10.7	11.1	10.6	11.0	131.7
Pumps & Aux.	1.2	1.1	1.9	2.5	3.4	3.9	4.2	4.6	4.0	3.1	2.1	1.5	33.3
Total	20.5	16.3	32.5	46.1	68.7	79.8	86.7	91.0	74.9	53.3	33.3	24.6	627.5
Alternative model I (Package VAV Cooling with Hot Water Coil Heating)													
Space Cool	0.8	0.7	18.2	33.1	56.8	70.6	80.3	81.5	63.6	39.8	19.1	6.8	471.3
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	30.9	22.7	7.0	0.9	0.9	0.1	0.9	-	0.9	2.1	7.1	21.9	95.3
Electricity	11.7	7.7	2.7	0.6	0.2	0.1	0.1	-	0.2	0.7	3.0	8.4	35.3
Natural Gas	19.2	15.0	4.3	0.3	0.7	-	0.8	-	0.7	1.4	4.1	13.5	60.0
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	12.8	12.1	15.0	14.8	15.5	15.4	15.1	16.1	14.9	15.2	13.6	13.4	173.8
Pumps & Aux.	2.1	1.9	1.4	0.7	0.3	0.2	0.2	0.2	0.2	0.6	1.2	1.9	10.8
Total	46.6	37.4	41.6	49.5	73.5	86.3	96.5	97.8	79.6	57.7	41.0	44.0	751.2
Saving	26.1	21.1	9.1	3.4	4.8	6.5	9.8	6.8	4.7	4.4	7.7	19.4	123.7
	56.0%	56.4%	21.8%	6.9%	6.5%	7.5%	10.1%	7.0%	5.9%	7.6%	18.9%	44.1%	16.5%

Figure 4.20 shows that the GSHP system consumes about half of the energy when compared to the VAV system in December, January, and February, when space heating is required. The GSHP saves 22,200 kWh (52.2%), on average, during these three months. For the rest of the year, the GSHP system can save 6,400 kWh (10.3%) when compared to the VAV system. The results show that the GSHP system can save more energy on space heating than space cooling.

In addition to the energy consumption, Table 4.8 illustrates the simulation results for the baseline model peak energy demand versus alternative model I. The results indicate that the GSHP scheme can save 1,876.7 kW of the overall peak energy demand annually, which is about 21% reduction. It is interesting to note that for the equipment end-use category, the GSHP case consumes more energy than the VAV case during the peak period.

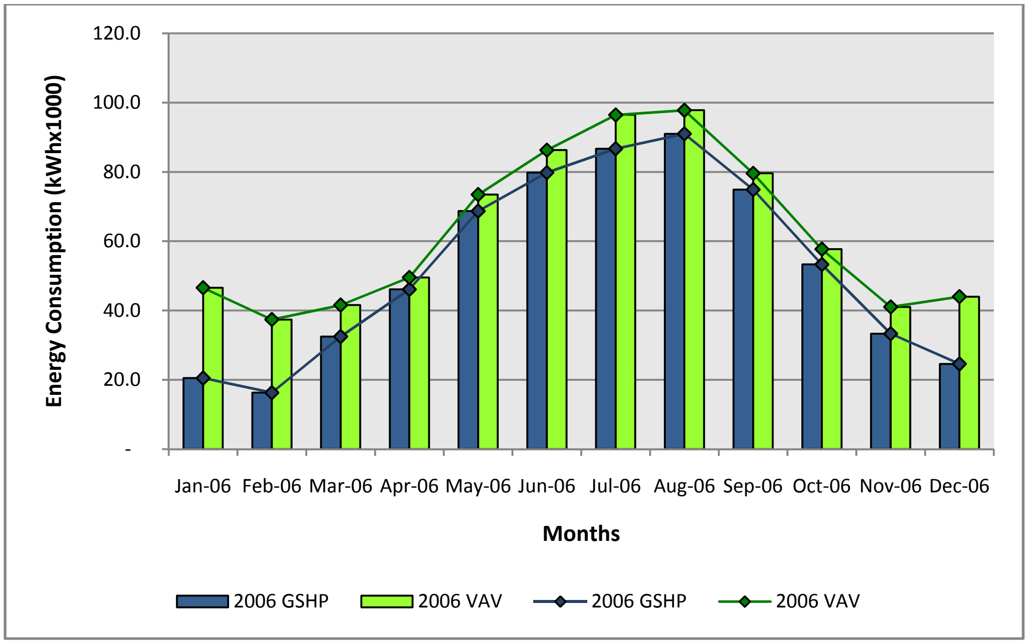


Figure 4.20 - Comparison of baseline model energy consumption (HVAC end-use) to alternative model I

Table 4.8 - Monthly peak energy demand in kW from baseline model and alternative model I

End-use	Case	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Equipment	Baseline	246.1	246.1	246.1	246.1	243.7	246.1	246.1	246.1	246.1	246.1	243.7	245.6	2,947.9
	Alternative I	245.6	246.1	243.7	246.1	243.7	246.1	243.7	246.1	246.1	245.6	246.1	245.6	2,944.4
	Saving	-0.5	0.0	-2.4	0.0	0.0	0.0	-2.4	0.0	0.0	-0.5	2.4	0.0	-3.5
		-0.2%	0.0%	-1.0%	0.0%	0.0%	0.0%	-1.0%	0.0%	0.0%	-0.2%	1.0%	0.0%	-0.1%
HVAC	Baseline	142.0	150.4	204.7	204.3	229.4	244.9	240.2	238.2	246.8	243.9	199.0	183.1	2,526.9
	Alternative I	306.8	312.5	390.0	382.7	402.2	296.7	438.9	287.9	441.2	432.9	360.7	353.4	4,406.4
	Saving	164.8	162.1	185.3	178.4	172.8	51.8	198.7	49.7	194.4	189.0	161.7	170.3	1,879.5
		53.7%	51.9%	47.5%	46.6%	43.0%	17.5%	45.3%	17.3%	44.1%	43.7%	44.8%	48.2%	42.7%
Lighting	Baseline	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	1,535.7
	Alternative I	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	1,535.9
	Saving	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.2
		0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%
Other	Baseline	4.8	5.0	5.0	4.9	4.8	4.2	4.0	3.8	3.8	3.9	4.4	4.7	53.2
	Alternative I	5.0	5.0	5.2	4.9	4.8	4.2	4.1	3.8	3.8	4.1	4.2	4.7	53.7
	Saving	0.2	0.0	0.2	0.0	0.0	0.0	0.1	0.0	0.0	0.2	-0.2	0.0	0.5
		4.0%	0.0%	3.8%	0.0%	0.0%	0.0%	2.4%	0.0%	0.0%	4.9%	-4.8%	0.0%	0.9%
Total	Baseline	520.9	529.5	583.8	583.3	605.9	623.2	618.3	616.1	624.7	621.9	575.1	561.4	7,063.7
	Alternative I	685.4	691.6	766.9	761.7	778.7	675.0	814.7	665.8	819.1	810.6	739.0	731.7	8,940.4
	Saving	164.5	162.1	183.1	178.4	172.8	51.8	196.4	49.7	194.4	188.7	163.9	170.3	1,876.7
		24.0%	23.4%	23.9%	23.4%	22.2%	7.7%	24.1%	7.5%	23.7%	23.3%	22.2%	23.3%	21.0%

Table 4.9 shows that the GSHP system can save 1,879.5 kW (12.6%) of peak energy demand for the HVAC end-use category annually when compared to the VAV system. While the peak energy demands of other GSHP end-use components are less than the VAV system, it is interesting to note that the GSHP system requires energy for operating pumps and auxiliary equipment during the peak period which is not required with the VAV system operation.

Table 4.9 - Monthly peak energy demand (HVAC end-use) in kW from baseline model and alternative model I

HVAC End-use	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Baseline Model (Ground Source Heat Pump)													
Space Cool	110.8	119.1	169.8	169.4	194.5	210.0	205.3	203.3	211.9	209.0	164.1	148.2	2,115.4
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	0.2	0.2	-	-	-	-	-	-	-	-	-	-	0.5
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	307.4
Pumps & Aux.	5.4	5.5	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	103.6
Total	142.0	150.4	204.7	204.3	229.4	244.9	240.2	238.2	246.8	243.9	199.0	183.1	2,526.9
Alternative Model I (Package VAV with Electric Reheat)													
Space Cool	114.2	122.9	197.2	194.8	229.1	259.2	262.4	251.2	265.5	242.6	168.3	157.9	2465.2
Heat Reject.	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Space Heat	157.4	155.0	159.7	154.2	138.5	0.0	140.2	0.0	138.1	154.3	159.5	161.8	1518.8
Electricity	1.1	1.3	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.2	0.1	2.8
Natural Gas	156.3	153.7	159.7	154.2	138.5	0.0	140.2	0.0	138.1	154.3	159.3	161.7	1516.0
HP Supp.	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Vent. Fans	35.0	34.4	32.9	33.5	34.4	37.3	36.1	36.5	37.4	35.8	32.7	33.5	419.5
Pumps & Aux.	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	2.9
Total	306.8	312.5	390.0	382.7	402.2	296.7	438.9	287.9	441.2	432.9	360.7	353.4	4406.4
Saving	164.8	162.1	185.3	178.4	172.8	51.8	198.7	49.7	194.4	189.0	161.7	170.3	1,879.5
	53.7%	51.9%	47.5%	46.6%	43.0%	17.5%	45.3%	17.3%	44.1%	43.7%	44.8%	48.2%	42.7%

Figure 4.21 shows that peak energy demands of the GSHP system are less than the VAV system during the cooling months. For most of the year, the GSHP system can save on average, 177.8 kW or 46.5% of the VAV system peak energy demand. Only in June and August, the GSHP system saves on average only 50.75 kW (17.4%) of the peak energy demand. This suggests that the benefit of the GSHP system in reducing peak energy demand would be highest when utility demand charges are high.

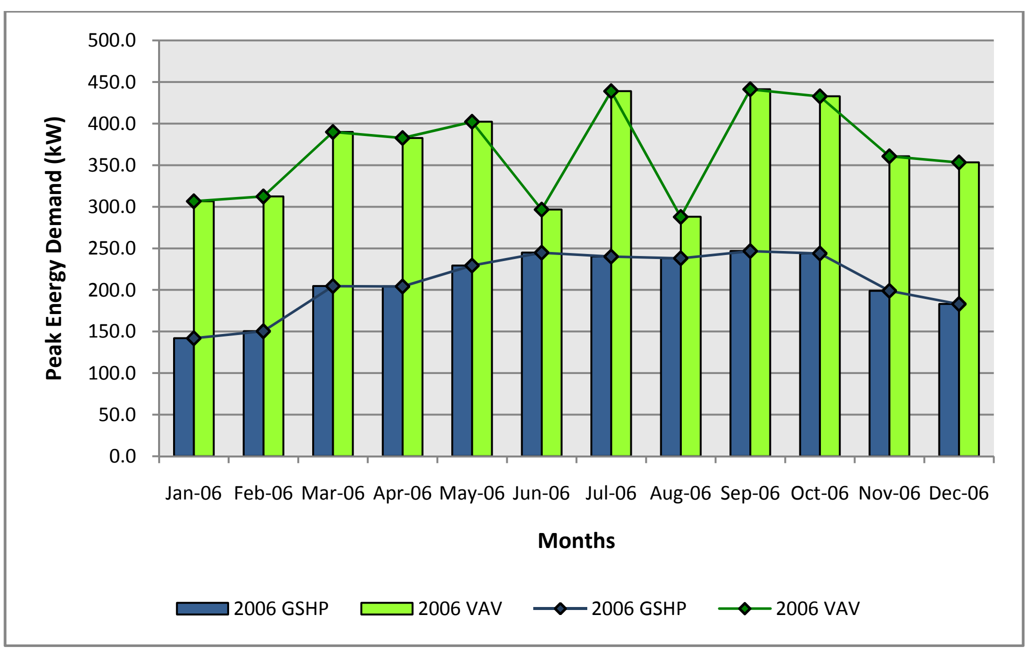


Figure 4.21 - Comparison of baseline model peak energy demand (HVAC end-use) to alternative model I

For energy costs, Table 4.10 shows the monthly energy cost from the baseline model and alternative model I. The simulation results indicate that the GSHP system can save $4,517 a year or about 2.7% of the annual energy costs when compared to the VAV system.

Table 4.10 - Monthly energy cost ($) from baseline model and alternative model I

Energy Type	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Baseline Model (Ground Source Heat Pump)													
Electricity	11,852	11,113	13,417	13,307	14,729	15,223	15,218	16,054	14,544	13,934	12,409	12,337	164,137
Alternative Model I (Package VAV with HW Coil Heating)													
Electricity	12,180	11,406	13,718	13,531	15,033	15,678	15,800	16,521	14,910	14,181	12,586	12,628	168,172
Natural Gas	103	88	41	17	19	15	20	15	19	24	40	81	482
Total	12,283	11,494	13,759	13,548	15,052	15,693	15,820	16,536	14,929	14,205	12,626	12,709	168,654
Saving	431	381	342	241	323	470	602	482	385	271	217	372	4,517
	3.5%	3.3%	2.5%	1.8%	2.1%	3.0%	3.8%	2.9%	2.6%	1.9%	1.7%	2.9%	2.7%

The following figure (Figure 4.22) compares the baseline model energy cost for all end-use categories to alternative model I. In each month, the energy cost for the GSHP is lower than the VAV system.

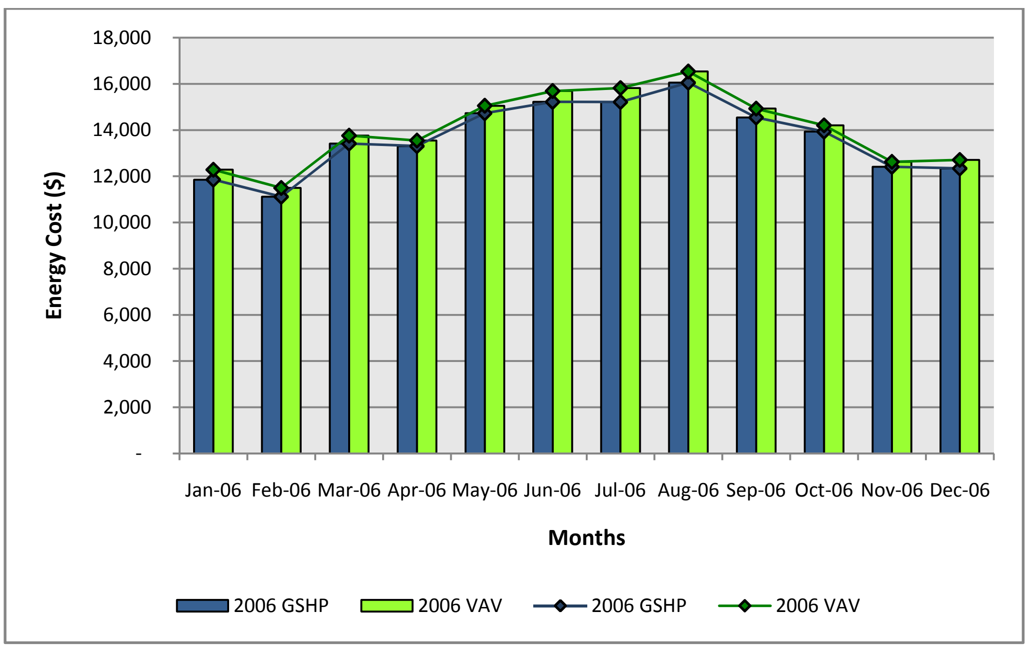


Figure 4.22 - Comparison of baseline model energy cost to alternative model I

In addition to energy consumption, peak energy demand, and energy cost comparisons, the life-cycle costs (LCC) and supplementary economic analyses are used for evaluating the economy of GSHP system when compared to the VAV system. Since only first cost and energy costs of the GSHP and VAV systems are available, the breakeven analysis procedure is used to determine at what level of other possible costs (OM&R, water, residual, and replacement costs) would result in an economically viable choice for the GSHP. Figure 4.23 shows that the present value of the difference (VAV-GSHP) in OM&R, water, and residual costs would need to be at least $732,842 for an investment in the GSHP system to break even. This $732,842 was input to the FEMP LCCA spreadsheet integrated within eQUEST to predict the life-cycle cost benefits of the GSHP system.

KnowledgeWorks I&II
Breakeven Analysis
Baseline Model VS Alternative Model I

$$
\begin{aligned}
S &= \Delta C \\
[\Delta E + \Delta OM\&R + \Delta W] &= [\Delta I_0 + \Delta Repl - \Delta Res] \\
[\Delta OM\&R + \Delta W - \Delta Repl + \Delta Res] &= [\Delta I_0 - \Delta E] \\
&= [759{,}104 - 26{,}262] \\
&= 732{,}842
\end{aligned}
$$

Where

S = Operational savings for the alternative relative to the base case

ΔC = Investment-related additional costs for the alternative relative to the base case

ΔE = Saving in energy costs in year t $(E_{BC} - E)_t$

$\Delta OM\&R$ = Difference in $OM\&R$ costs $(OM\&R_{BC} - OM\&R_A)$

ΔW = Saving in water costs in year t $(W - W_A)_t$

ΔI_0 = Additional initial investment costs required for the alternative relative to the base case $(I_A - I_{BC})$

$\Delta Repl$ = Difference in replacement costs $(Repl_A - Repl_{BC})$

ΔRes = Difference in residual values $(Repl - Repl)_t$

Figure 4.23 - Breakeven analysis between baseline model and alternative model I

Table 4.11 summarizes the results from the FEMP's LCCA spreadsheet. The cost difference of $732,842 was input as the maintenance present value of the VAV system as presented in Table 4.9. The results indicate that the LCC of the GSHP system will be less than for the VAV system, when the present value of maintenance and operating costs of the GSHP system is less than the VAV system by at least $732,842 over the 25 year life of the system. From the table, when the condition is met, the discount payback (DPB) of the GSHP investment will be less than 25 years, and the saving to investment ratio (SIR) will be greater than 1.

Table 4.11 - Life-cycle Costs analysis summary of baseline model vs. alternative model I (breakeven)

Life-Cycle Costs								
Case	**Description**	**One-Time Costs**		**Total Utility**		**Maintenance**		**Total**
		1st year $	**LCC PV$**	**1st year $**	**LCC PV$**	**1st year $**	**LCC PV$**	**LCC PV$**
Alternative I	VAV w/HW	$1,433,879	$1,433,879	$168,656	$986,026	$114,400	$732,846	$2,913,300
Baseline	GSHP	$2,192,983	$2,192,983	$164,137	$959,764	$0	$0	$2,896,398

Cumulative Life-Cycle SAVINGS (negative entries indicate increased costs)												
Case	**Description**	**One-Time Costs**		**Total Utility**		**Maintenance**		**NS $**	**SPB yrs**	**DPB yrs**	**SIR**	**AIRR**
		1st year $	**LCC PV$**	**1st year $**	**LCC PV$**	**1st year $**	**LCC PV$**					
Baseline	GSHP	($-759104)	($-759104)	$4,519	$26,262	$114,400	$732,846	$16,902	168	23.7	1	10.00%

To establish confidence concerning the economy of the GSHP, the replacement and maintenance costs were estimated and input to the LCCA spreadsheet. For the replacement costs it was assumed that all unitary HVAC equipment excluding the underground heat exchanger would need to be replaced after 15 years of operation according to their median service lives (Rosenquist, et al. 2004). The maintenance cost estimates were based on the procedure provided in the *2003 ASHRAE Application Handbook* (ASHRAE 2003). The detailed information of replacement and maintenance cost estimates are presented in Appendix D.

Table 4.12 summarized the replacement and maintenance costs of the GSHP and package VAV systems as estimated using the ASHRAE procedure. The results show that the GSHP system can save $400,880 or 27.96% on the replacement costs, and $29,874 or 62.48% on the maintenance costs. The replacement costs presented in Table 4.10 were input to the LCCA spreadsheet as the non-annual replacement costs and the maintenance costs were input as the annual recurring costs in constant dollars.

Table 4.12 - Replacement and maintenance costs estimation for baseline and alternative model I models

Case	**Description**	**Replacement (2006 $)**		**Maintenance (2006$/Year)**
Alternative I	VAV w/HW	System replacement (year 15th)	1,433,879.00	47,814.63
Baseline	GSHP	System replacement excluding underground loop (year 15th)	1,032,999.00	17,939.77
		Saving	400,880.00	29,874.86
			27.96%	62.48%

The LCCA results in Table 4.13 show that the LCC of the GSHP system is greater than the VAV system, the DPB is longer than 25 years, and the SIR is less than 1. As a result, the investment on the GSHP system in the KnowledgeWorks I and II buildings may not be feasible when compared to the VAV system.

Table 4.13 - Life-cycle costs analysis summary of baseline model vs. alternative model I

Life-Cycle Costs								
Case	**Description**	**One-Time Costs**		**Total Utility**		**Maintenance**		**Total**
		1st year $	**LCC PV$**	**1st year $**	**LCC PV$**	**1st year $**	**LCC PV$**	**LCC PV$**
Alternative I	VAV w/HW	$1,433,879	$1,606,649	$168,656	$986,026	$47,814	$306,296	$2,857,652
Baseline	GSHP	$2,192,983	$2,317,450	$164,137	$959,764	$17,939	$114,917	$3,251,160

Cumulative Life-Cycle SAVINGS (negative entries indicate increased costs)												
Case	**Description**	**One-Time Costs**		**Total Utility**		**Maintenance**		**NS $**	**SPB yrs**	**DPB yrs**	**SIR**	**AIRR**
		1st year $	**LCC PV$**	**1st year $**	**LCC PV$**	**1st year $**	**LCC PV$**					
Baseline	GSHP	($-759104)	($-710802)	$4,519	$26,262	$29,875	$191,379	($-393507)	168	0	0.3	4.90%

4.2 Alternative Model II

Alternative model II was developed for simulating the KnowledgeWorks I and II buildings with an air-source (rather than ground-source) heat pump (ASHP) system. Similar to the GSHP system, the ASHP system uses electricity as the system energy source. Table 4.14 shows that the GSHP case can save $23,700 kWh or 0.7% of the energy consumed by the ASHP case which corresponds to approximately 13.3 metric tons of carbon dioxide emission per year. The simulation results in Table 4.8 indicate that all energy savings comes from the HVAC end-use category.

Table 4.14 - Monthly energy consumption in kWh ×1,000 from baseline model and alternative model II

End-use	Case	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Equipment	Baseline	138.3	129.1	150.1	139.8	145.5	144.5	140.7	150.1	137.8	142.8	135.5	140.6	1,694.9
	Alternative II	138.3	129.1	150.1	139.8	145.5	144.5	140.7	150.1	137.8	142.8	135.5	140.6	1,694.9
	Saving	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%
Lighting	Baseline	70.9	66.5	78.0	72.3	75.0	74.9	72.3	78.0	71.1	73.7	69.6	72.4	874.7
	Alternative II	70.9	66.5	78.0	72.3	75.0	74.9	72.3	78.0	71.1	73.7	69.6	72.4	874.7
	Saving	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%
HVAC	Baseline	20.5	16.3	32.5	46.1	68.7	79.8	86.7	91.0	74.9	53.3	33.3	24.6	627.5
	Alternative II	24.1	19.5	34.5	46.9	69.4	82.0	90.8	92.8	75.1	53.5	34.5	27.8	651.4
	Saving	3.6	3.2	2.0	0.8	0.7	2.2	4.1	1.8	0.2	0.2	1.2	3.2	23.7
		14.9%	16.4%	5.8%	1.7%	1.0%	2.7%	4.5%	1.9%	0.3%	0.4%	3.5%	11.5%	3.6%
Other	Baseline	1.5	1.5	1.8	1.6	1.6	1.5	1.3	1.4	1.2	1.3	1.3	1.5	17.4
	Alternative II	1.5	1.5	1.8	1.6	1.6	1.5	1.3	1.4	1.2	1.3	1.3	1.5	17.4
	Saving	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%
Total	Baseline	231.2	213.4	262.4	259.8	290.8	300.7	301.0	320.5	285.0	271.1	239.7	239.1	3,214.5
	Alternative II	234.8	216.6	264.4	260.6	291.5	302.9	305.1	322.3	285.2	271.3	240.9	242.3	3,238.4
	Saving	3.6	3.2	2.0	0.8	0.7	2.2	4.1	1.8	0.2	0.2	1.2	3.2	23.7
		1.5%	1.5%	0.8%	0.3%	0.2%	0.7%	1.3%	0.6%	0.1%	0.1%	0.5%	1.3%	0.7%

Table 4.15 shows that the GSHP can save $23,700 kWh or 3.6% of the energy consumed by the ASHP system. The GSHP system consumes less energy than the ASHP system for ventilation

fans, space heating, and space cooling respectively. It is interesting to note that the GSHP system consumes about 4 times the energy for pumps and auxiliary operations when compared to the ASHP. However, the GSHP system does not require energy for heat pump supplementary heating which is needed by the ASHP system.

Table 4.15 - Monthly energy consumption (HVAC end-use) in kWh ×1,000 from baseline model and alternative model II

HVAC End-use	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Baseline Model (Ground Source Heat Pump)													
Space Cool	0.8	0.7	17.7	32.5	53.9	64.8	71.5	74.9	60.2	38.9	19.1	6.8	441.7
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	7.6	4.5	1.4	0.2	0.1	-	-	-	-	0.2	1.5	5.3	20.8
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	10.9	10.0	11.5	10.9	11.3	11.1	11.0	11.5	10.7	11.1	10.6	11.0	131.7
Pumps & Aux.	1.2	1.1	1.9	2.5	3.4	3.9	4.2	4.6	4.0	3.1	2.1	1.5	33.3
Total	20.5	16.3	32.5	46.1	68.7	79.8	86.7	91.0	74.9	53.3	33.3	24.6	627.5
Alternative Model II (Air Source Heat Pump)													
Space Cool	0.8	0.7	18.4	33.5	56.1	69	78	79.4	62.5	39.9	19.4	6.9	464.7
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	8.7	5.3	1.6	0.3	0.1	0	0	0	0	0.2	1.8	6.1	24.1
HP Supp.	0.2	0.2	0	0	0	0	0	0	0	0	0	0.4	0.8
Vent. Fans	12.6	11.7	13.4	12.6	13.1	13	12.8	13.4	12.5	13	12.3	12.8	153.3
Pumps & Aux.	1.8	1.6	1.1	0.5	0.1	0	0	0	0.1	0.4	1	1.6	8.3
Total	24.1	19.5	34.5	46.9	69.4	82.0	90.8	92.8	75.1	53.5	34.5	27.8	651.2
Saving	3.6	3.2	2.0	0.8	0.7	2.2	4.1	1.8	0.2	0.2	1.2	3.2	23.7
	14.9%	16.4%	5.8%	1.7%	1.0%	2.7%	4.5%	1.9%	0.3%	0.4%	3.5%	11.5%	3.6%

Figure 4.24 compares the GSHP system energy consumption to that of the ASHP system. On average, the GSHP system consumes 3,300 kWh (14.3%) less electricity than the ASHP in December, January, and February. The GSHP system energy consumption for the GSHP system was slightly less than that for the ASHP for the remaining months. On average, the GSHP system can save 1,500 kWh (24%) during the remaining nine month period.

The simulation results shown in Table 4.16 indicate that the GSHP case can reduce the overall peak energy demand by 203.3 kW (2.8%) when compared to the ASHP case. The GSHP case consumes less energy than the ASHP during the peak period for the HVAC (206 kW) and other (0.4 kW) operations. In contrast, the GSHP case uses more energy on the peak period than the ASHP for operating equipment (3 kW).

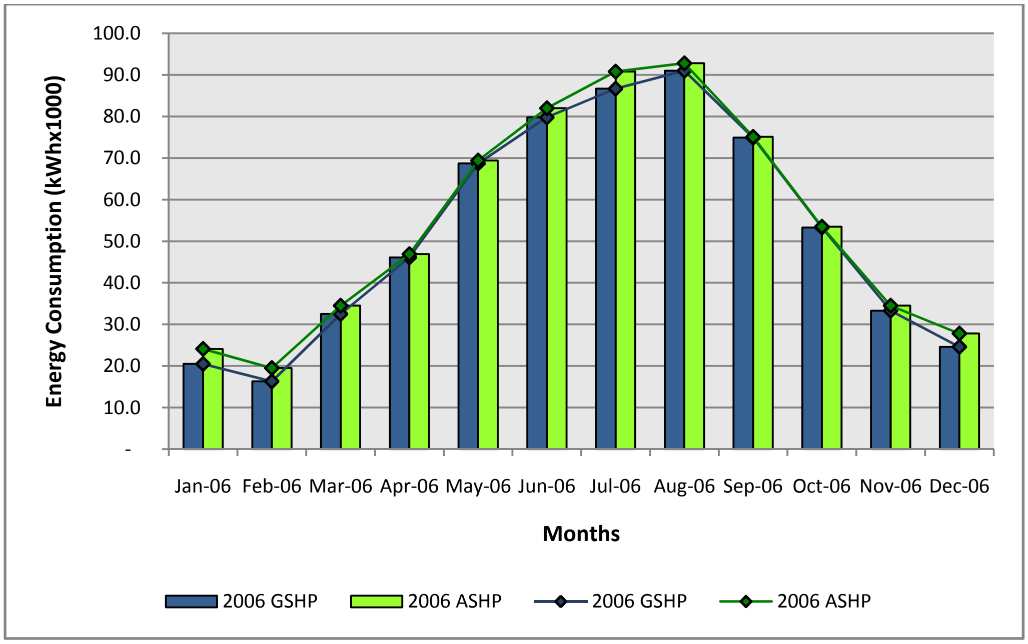


Figure 4.24 - Comparison of baseline model energy consumption (HVAC end-use) to alternative model II

Table 4.16 - Monthly peak energy demand in kW from baseline model and alternative model II

End-use	Case	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Equipment	Baseline	246.1	246.1	246.1	246.1	243.7	246.1	246.1	246.1	246.1	246.1	243.7	245.6	2,947.9
	Alternative II	246.1	246.1	243.7	246.1	243.7	246.1	243.7	246.1	246.1	245.6	246.1	245.6	2,944.9
	Saving	0.0	0.0	-2.4	0.0	0.0	0.0	-2.4	0.0	0.0	-0.5	2.4	0.0	-3.0
		0.0%	0.0%	-1.0%	0.0%	0.0%	0.0%	-1.0%	0.0%	0.0%	-0.2%	1.0%	0.0%	-0.1%
HVAC	Baseline	142.0	150.4	204.7	204.3	229.4	244.9	240.2	238.2	246.8	243.9	199.0	183.1	2526.9
	Alternative II	145.8	153	218.4	216.4	249	277.6	281.9	270.3	284.7	262.6	191.1	182	2732.9
	Saving	3.8	2.6	13.7	12.1	19.6	32.7	41.7	32.1	37.9	18.7	-7.9	-1.1	206.0
		2.6%	1.7%	6.3%	5.6%	7.9%	11.8%	14.8%	11.9%	13.3%	7.1%	-4.1%	-0.6%	7.5%
Lighting	Baseline	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	1,535.7
	Alternative II	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	128.0	1,535.8
	Saving	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	-0.1
		0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%
Other	Baseline	4.8	5.0	5.0	4.9	4.8	4.2	4.0	3.8	3.8	3.9	4.4	4.7	53.2
	Alternative II	4.8	5.0	5.2	4.9	4.8	4.2	4.1	3.8	3.8	4.1	4.2	4.7	53.6
	Saving	0.0	0.0	0.2	0.0	0.0	0.0	0.1	0.0	0.0	0.2	-0.2	0.0	0.4
		0.0%	0.0%	3.8%	0.0%	0.0%	0.0%	2.4%	0.0%	0.0%	4.9%	-4.8%	0.0%	0.7%
Total	Baseline	520.9	529.5	583.8	583.3	605.9	623.2	618.3	616.1	624.7	621.9	575.1	561.4	7,063.9
	Alternative II	524.7	532.1	595.3	595.4	625.5	655.9	657.7	648.2	662.6	640.3	569.4	560.3	7,267.2
	Saving	3.8	2.6	11.5	12.1	19.6	32.7	39.4	32.1	37.9	18.4	-5.7	-1.1	203.3
		0.7%	0.5%	1.9%	2.0%	3.1%	5.0%	6.0%	5.0%	5.7%	2.9%	-1.0%	-0.2%	2.8%

In Table 4.17, the simulation results show that the GSHP system will reduce the annual peak energy demand by 206 kW (7.5%). For most end-use HVAC components (space cooling, heat rejection, space heating, heat pump supplementary equipment, and ventilation fans), the GSHP consumes less energy than the ASHP system during the peak period.

Table 4.17 - Monthly peak energy demand (HVAC end-use) in kW from baseline model and alternative model II

HVAC End-use	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Baseline Model (Ground Source Heat Pump)													
Space Cool	110.8	119.1	169.8	169.4	194.5	210.0	205.3	203.3	211.9	209.0	164.1	148.2	2,115.4
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	0.2	0.2	-	-	-	-	-	-	-	-	-	-	0.5
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	307.4
Pumps & Aux.	5.4	5.5	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	103.6
Total	142.0	150.4	204.7	204.3	229.4	244.9	240.2	238.2	246.8	243.9	199.0	183.1	2,526.9
Alternative Model II (Air Source Heat Pump)													
Space Cool	115.6	122.9	188.6	186.6	219.2	247.8	252.1	240.5	254.9	232.8	161.3	152.2	2,374.5
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	0.4	0.3	-	-	-	-	-	-	-	-	-	-	0.7
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	29.8	29.8	29.8	29.8	29.8	29.8	29.8	29.8	29.8	29.8	29.8	29.8	357.7
Pumps & Aux.	-	-	-	-	-	-	-	-	-	-	-	-	-
Total	145.8	153.0	218.4	216.4	249.0	277.6	281.9	270.3	284.7	262.6	191.1	182.0	2,732.9
Difference	3.8	2.6	13.7	12.1	19.6	32.7	41.7	32.1	37.9	18.7	(7.9)	(1.1)	206.0
	2.6%	1.7%	6.3%	5.6%	7.9%	11.8%	14.8%	11.9%	13.3%	7.1%	-4.1%	-0.6%	7.5%

Figure 4.25 shows that the GSHP system consumes less energy than the ASHP system during the peak period from March to September. The GSHP system can save 36.1 kW (12.9%) in the four month period from June to September. It was somewhat surprising that the GSHP system consumes more energy than the ASHP in November and December. This is due to the additional pump and auxiliary equipment operation. This suggests that the GSHP may be capable of reducing the peak energy demand, and utility costs may be lower in the regions where demand charges are high.

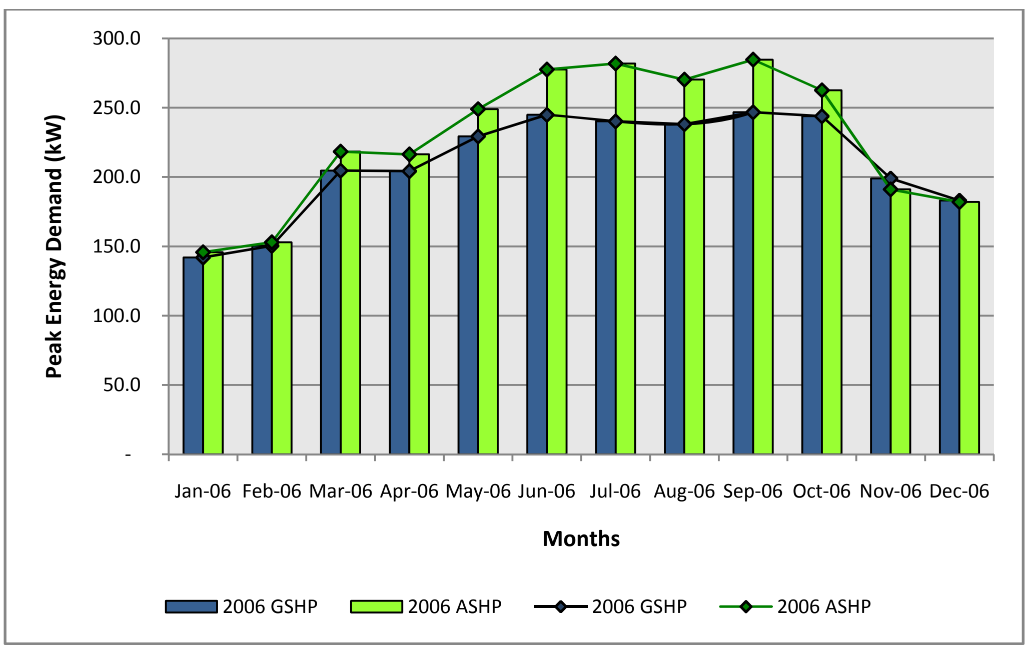


Figure 4.25: Comparison of baseline model peak energy demand (HVAC end-use) to alternative model II

The simulation results shown in Table 4.18 indicate that the monthly energy cost from the GSHP system can save $1,757 or about 1.1% of the annual cost when compared to the ASHP system.

Table 4.18 - Monthly energy cost ($) from baseline model and alternative model II

Energy Type	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Baseline Model (Ground Source Heat Pump)													
Electricity	11,852	11,113	13,417	13,307	14,729	15,223	15,218	16,054	14,544	13,934	12,409	12,337	164,137
Alternative Model II (Air Source Heat Pump)													
Electricity	12,031	11,266	13,554	13,395	14,834	15,431	15,532	16,245	14,680	14,005	12,445	12,476	165,894
Difference	179	153	137	88	105	208	314	191	136	71	36	139	1,757
	1.5%	1.4%	1.0%	0.7%	0.7%	1.3%	2.0%	1.2%	0.9%	0.5%	0.3%	1.1%	1.1%

Figure 4.26 illustrates the comparison of the baseline and alternative model II for energy cost for all end-use categories. As shown, the energy cost of the GSHP system is less than the ASHP system for each month.

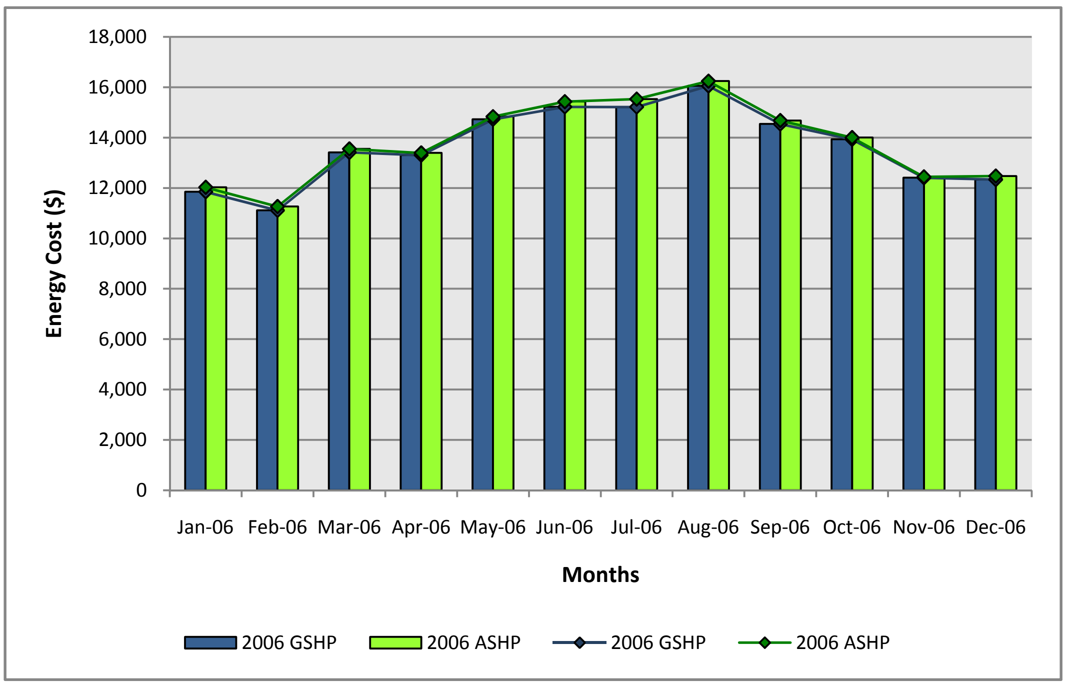


Figure 4.26: Comparison of baseline model energy cost to alternative model II

Figure 4.27 shows that the present value of the difference (ASHP-GSHP) in OM&R, water, and residual costs, subtracting the difference of replacement costs ($\Delta OM\&R + \Delta W + \Delta Res - \Delta Repl$) would need to be at least $1,085,691 for an investment in the GSHP system to break even. This $1,085,691 was input to the FEMP LCCA spreadsheet integrated within eQUEST to predict the life-cycle cost benefits of the GSHP system.

KnowledgeWorks I&II

Breakeven Analysis

Baseline Model VS Alternative Model II

$$
\begin{aligned}
S &= \Delta C \\
[\Delta E + \Delta OM\&R + \Delta W] &= [\Delta I_0 + \Delta Repl - \Delta Res] \\
[\Delta OM\&R + \Delta W - \Delta Repl + \Delta Res] &= [\Delta I_0 - \Delta E] \\
&= [1{,}096{,}491 - 10{,}800] \\
&= 1{,}085{,}691
\end{aligned}
$$

Where

S = Operational savings for the alternative relative to the base case

ΔC = Investment-related additional costs for the alternative relative to the base case

ΔE = Saving in energy costs in year t $(E_{BC} - E)_t$

$\Delta OM\&R$ = Difference in $OM\&R$ costs $(OM\&R_{BC} - OM\&R_A)$

ΔW = Saving in water costs in year t $(W - W_A)_t$

ΔI_0 = Additional initial investment costs required for the alternative relative to the base case $(I_A - I_{BC})$

$\Delta Repl$ = Difference in replacement costs $(Repl_A - Repl_{BC})$

ΔRes = Difference in residual values $(Repl - Repl)_t$

Figure 4.27 - Breakeven analysis between baseline model and alternative model II

Table 4.19 shows the summary of the LCCA results, in which the cost difference of $1,085,691 is used as the LCC present value of maintenance costs. The results indicate that the LCC of the GSHP system will be less than that of the ASHP system, when the present value of the cost difference between the ASHP-GSHP is greater than $1,085,691. Also, the discounted payback will be less than 25 years, and the savings to investment ratio will be greater than 1 for this scenario for cost difference.

Table 4.19 - Life-cycle costs analysis summary of baseline model vs. alternative model II (breakeven)

Life-Cycle COSTs

Case	Description	One-Time Costs		Total Utility		Maintenance		Total
		1st year $	LCC PV$	1st year $	LCC PV$	1st year $	LCC PV$	LCC PV$
Alternative II	ASHP	$1,096,492	$1,096,492	$165,984	$970,564	$169,481	$1,085,695	$2,920,896
Baseline	GSHP	$2,192,983	$2,192,983	$164,137	$959,764	$0	$0	$2,896,398

Cumulative Life-Cycle SAVINGS (negative entries indicate increased costs)

Case	Description	One-Time Costs		Total Utility		Maintenance		NS $	SPB yrs	DPB yrs	SIR	AIRR
		1st year $	LCC PV$	1st year $	LCC PV$	1st year $	LCC PV$					
Baseline	GSHP	($-1096491)	($-1096491)	$1,847	$10,800	$169,481	$1,085,695	$24,498	593.7	23.7	1	10.00%

Table 4.20 illustrates the replacement and maintenance cost estimates for the ASHP and GSHP system. The GSHP can save $63,493 (5.79%) on replacement costs and $891 (4.73%) on maintenance costs when compared to the ASHP.

Table 4.20 - Replacement and maintenance costs estimation for baseline and alternative II models

Case	Description	Replacement (2006 $)		Maintenance (2006 $/Year)
Alternative II	ASHP	System replacement (year 15th)	1,096,492.00	18,831.19
Baseline	GSHP	System replacement excluding underground loop (year 15th)	1,032,999.00	17,939.77
		Saving	63,493.00	891.42
			5.79%	4.73%

When the replacement and maintenance cost values were input to the LCCA spreadsheet, the results in Table 4.17 show that the LCC of the GSHP system is greater than for the ASHP. Table 4.17 indicates that the DPB is over 25 years, and the SIR is close to zero. The LCCA results lead to the conclusion that using the GSHP system is not economically feasible when compared to the ASHP.

Table 4.21 - Life-cycle costs analysis summary of baseline model vs. alternative model II

Life-Cycle Costs

Case	Description	One-Time Costs 1st year $	One-Time Costs LCC PV$	Total Utility 1st year $	Total Utility LCC PV$	Maintenance 1st year $	Maintenance LCC PV$	Total LCC PV$
Alternative II	ASHP	$1,096,492	$1,228,610	$165,984	$970,564	$18,831	$120,631	$2,280,867
Baseline	GSHP	$2,192,983	$2,317,450	$164,137	$959,764	$17,939	$114,917	$3,251,160

Cumulative Life-Cycle SAVINGS (negative entries indicate increased costs)

Case	Description	One-Time Costs 1st year $	One-Time Costs LCC PV$	Total Utility 1st year $	Total Utility LCC PV$	Maintenance 1st year $	Maintenance LCC PV$	NS $	SPB yrs	DPB yrs	SIR	AIRR
Baseline	GSHP	($-1096491)	($-1088841)	$1,847	$10,800	$892	$5,714	($-970293)	593.7	0	0	n/a

In this chapter, the baseline model simulation results for energy consumption, peak energy demand, and energy cost were summarized and calibrated with the 2007 utility records. The calibration results indicate that the simulated baseline model is accurate for simulating the KnowledgeWorks I and II buildings with alternative HVAC systems including 1) a package VAV with hot water heating system and 2) an air-source heat pump system. The results of the GSHP, VAV, and ASHP system simulations presented in this chapter will be used for hypotheses testing in the next chapter, Chapter 5: Conclusions and summary.

Chapter 5: Conclusions and Summary

5.1 Hypotheses Testing

Hypothesis 1: Energy performance evaluation

In hypothesis 1.1 it was proposed that the annual energy consumption (AE) of the geothermal heat pump system used in the KnowledgeWorks I and II building would be less than the AE for the alternative HVAC systems including 1) a package VAV with hot water heating and 2) an air-source heat pump system. From Chapter 1, Equation 1.1, the annual energy saving (AES) was used for evaluating the energy performance of the geothermal heat pump system when compared to the alternative systems.

The results presented in Chapter 4, Table 4.6. and 4.10 show that the annual energy saving from the GSHP system is greater than zero for either the package VAV with hot water coil heating system or the air-source heat pump system. Therefore, the claim that the geothermal heat pump system consumes less energy than the VAV and ASHP systems is accepted.

$$AES_{GSHP}\ (kWh/year) = AE_{ALT}\ (kWh/year) - AE_{GSHP}\ (kWh/year) > 0 \tag{5.1}$$

$$\begin{aligned} AES_{GSHP}\ (kWh/year) &= AE_{VAV}\ (kWh/year) - AE_{GSHP}\ (kWh/year) \\ &= 751{,}200\ (kWh/year) - 627{,}500 (kWh/year) \\ AES_{GSHP}\ (kWh/year) &= 123{,}700\ (kWh/year) > 0 \end{aligned}$$

$$\begin{aligned} AES_{GSHP}\ (kWh/year) &= AE_{ASHP}\ (kWh/year) - AE_{GSHP}\ (kWh/year) \\ &= 651{,}200\ (kWh/year) - 627{,}500\ (kWh/year) \\ AES_{GSHP}\ (kWh/year) &= 23{,}700\ (kWh/year) > 0 \end{aligned}$$

Where:

AES_{GSHP} = Annual energy saving from the geothermal heat pump system in kWh

AE_{GSHP} = Annual energy consumption of the geothermal heat pump system in kWh

AE_{ALT} = Annual energy consumption of the alternative system in kWh

AE_{VAV} = Annual energy consumption of the package VAV system in kWh

AE_{ASHP} = Annual energy consumption of the air-source heat pump system in kWh

Hypothesis 2: Economic evaluation

In addition to the energy performance evaluation, economic evaluation procedures were used to determine the economy of the geothermal heat pump system used in the KnowledgeWorks I and II buildings compared to the alternative systems. The life-cycle costs and supplementary economic measures analyses were used to test the following hypotheses.

Hypothesis 2.1 states that the life-cycle costs (LCC) of the geothermal heat pump system will be less than those for the package VAV with hot water coil and air-source heat pump systems. Equation 1.2 presented in Chapter 1 is represented below and used for evaluating the life-cycle cost savings from the implementation of the geothermal heat pump system.

The results of the LCC comparisons shown in Chapter 4, Table 4.11 and 4.19 indicate that the LCC of the GSHP system will be greater than the LCCs of the VAV and ASHP systems when the present value of OM&R, water, and residual cost differences minus the difference of replacement costs ($\Delta OM\&R + \Delta W + \Delta Res - \Delta Repl$) is greater than $732,842 for the package VAV case. This value would need to be at least $1,085,691 for the ASHP case.

$$LCC_{ALT}\ (\$) - LCC_{GSHP}(\$) > 0 \tag{5.2}$$

$$LCC_{VAV}\ (\$) - LCC_{GSHP}(\$) > 0$$

$$\text{when the PV of } [\Delta OM\&R + \Delta W + \Delta Res - \Delta Repl] > \$732{,}842$$

$$LCC_{ASHP}\ (\$) - LCC_{GSHP}(\$) > 0$$

$$\text{when the PV of } [\Delta OM\&R + \Delta W + \Delta Res - \Delta Repl] > \$1{,}085{,}691$$

Where:

LCC_{GSHP} = Life-cycle costs of the geothermal heat pump system in present value $

LCC_{ALT} = Life-cycle costs of the alternative system in present value $

LCC_{VAV} = Life-cycle costs of the package VAV system in present value $

LCC_{ASHP} = Life-cycle costs of the air-source heat pump system in present value $

However, the replacement and maintenance cost estimates presented in Chapter 4, Table 4.12 and 4.20 show that the present values of $732,842 and $1,085,691 are unlikely to be met. The LCCA results shown in the previous chapter, Table 4.13 and 4.21, indicate that the GSHP system would not be effective in terms of LCC when compared to the package VAV and ASHP systems. As a result, Hypothesis 2.1 is rejected.

Hypothesis 2.3 proposes that the SIR of the GSHP system should be greater than one when compared with the alternative systems. From the previous chapter, the SIR for the GSHP system investment will be greater than one when the present value of OM&R, water, and residual cost differences minus the difference of replacement costs $(\Delta OM\&R + \Delta W + \Delta Res - \Delta Repl)$ is greater than $727,640 for the package VAV case. For the ASHP case, the SIR of the GSHP system will be acceptable only when the present value of cost differences between the GSHP and the ASHP systems is greater than $1,085,691. The results of the hypothesis testing on the SIRs are shown in the following summary based on Equation 1.3 presented in Chapter 1.

$$SIR_{GSHP:ALT} > 1 \tag{5.3}$$

$$SIR_{GSHP:VAV} > 1,$$
when the PV of $[\Delta OM\&R + \Delta W + \Delta Res - \Delta Repl] > \$732{,}842$

$$SIR_{GSHP:ASHP} > 1,$$
when the PV of $[\Delta OM\&R + \Delta W + \Delta Res - \Delta Repl] > \$1{,}085{,}691$

Where:

$SIR_{GSHP:ALT}$	= SIR of the GSHP system when comparing the alternative systems
$SIR_{GSHP:VAV}$	= SIR of the GSHP system when comparing the package VAV system
$SIR_{GSHP:ASHP}$	= SIR of the GSHP system when comparing the package VAV system
$\Delta OM\&R$	= Difference in $OM\&R$ costs $(OM\&R_{BC} - OM\&R_{A})$
ΔW	= Saving in water costs in year t $(W - W_A)_t$
ΔRes	= Difference in residual values $(Repl - Repl)_t$
$\Delta Repl$	= Difference in residual values $(Repl - Repl)_t$

From Chapter 4, Table 4.12 and 4.20, the present values of $732,842 and $1,085,691 are unlikely to occur. The LCCA results presented in Chapter 4, Table 4.13 and 4.20, show that the SIR of the GSHP system when compared to the package VAV and ASHP are less than 1. This can be concluded that Hypothesis 2.3 fails to be accepted.

Hypothesis 2.3 proposes the discount payback (DPB) of the GSHP system investment should be less than the analysis period of 25 years. Similar to the hypothesis testing on the SIRs, the DPBs of the GSHP system investments for the package VAV and ASHP system cases are based on the assumption that the present values of OM&R, water, and residual cost differences, subtracting the difference of replacement costs ($\Delta OM\&R + \Delta W + \Delta Res - \Delta Repl$), are greater than \$732,842 and \$1,08,591. The following results based on Equation 1.4 presented in Chapter 1 shows that, only when these assumptions are met, the DPBs of the GSHP system will be less than 25 years for both package VAV and ASHP system cases.

$$25 - DPB_{GSHP:ALT} > 1 \tag{5.4}$$

$$25 - DPB_{\ GSHP:VAV} > 1,$$
when the PV of $[\Delta OM\&R + \Delta W + \Delta Res - \Delta Repl] > \$732{,}842$

$$25 - DPB_{GSHP:ASHP} > 1,$$
when the PV of $[\Delta OM\&R + \Delta W + \Delta Res - \Delta Repl] > \$1{,}085{,}691$

Where:

$DPB_{GSHP:ALT}$	= DPB of the GSHP system when comparing the alternative systems
$DPB_{GSHP:VAV}$	= DPB of the GSHP system when comparing the package VAV system
$DPB_{GSHP:ASHP}$	= DPB of the GSHP system when comparing the package VAV system
$\Delta OM\&R$	= Difference in $OM\&R$ costs ($OM\&R_{BC} - OM\&R_A$)
ΔW	= Saving in water costs in year t $(W - W_A)_t$
ΔRes	= Difference in residual values $(Repl - Repl)_t$
$\Delta Repl$	= Difference in residual values $(Repl - Repl)_t$

Referring to Chapter 4, Table 4.12 and 4.20, the replacement and maintenance cost estimates of the GSHP when compared to the VAV and ASHP systems are improbable to meet the values of \$732,842 and 1,025,691. The LCCA results (as shown in Table 4.13 and 4.20) illustrate that the DPB of the GSHP when compared to the VAV and ASHP systems take longer than 25 years for recovering the initial investment costs. Therefore, Hypothesis 2.3 is rejected.

5.2 Conclusions and Summary

This study aims to understand the energy performance and economy of the geothermal heat pump system used in the KnowledgeWorks I and II buildings located in the Virginia Tech Corporate Research Center in Blacksburg, Virginia. The study is based on the quantitative paradigm, using simulation and modeling as a methodology. In this research, the eQUEST energy simulation software package is used for performing energy performance simulations and life-cycle costs analyses of the buildings with the actual GSHP system and two alternative HVAC systems: the package VAV with hot water heating and air-source heat pump.

When compare to the package VAV with hot water coil heating alternative, the GSHP system can save 123,700 kWh (16.5%) annually. This savings would results in a reduction of 69.3 metric tons of carbon dioxide emissions each year. The GSHP system can reduce the peak energy demand per annum by 365 kW or about 12.6%. Furthermore, the GSHP system can save on average $4,517 or 2.7% of the annual energy bills.

The investment in the GSHP system when compared to the package VAV with hot water coil heating system will be recovered when the present value of OM&R, water, and residual cost differences minus the difference of replacement costs ($\Delta OM\&R + \Delta W + \Delta Res - \Delta Repl$) are greater than $732,842. When this condition is met, the life-cycle costs of the GSHP system will be less than that of the VAV system. Also, the saving to investment ratio (SIR) for the GSHP investment will be greater than 1, and the discount payback (DPB) of the GSHP will be less than the analysis period of 25 years.

In order to determine the life-cycle costs effectiveness of the GSHP system, the replacement and maintenance costs were estimated and input to the LCCA spreadsheet. The results of the LCCA indicate that the cost estimates can meet only $479,066 in their present value. The results lead to the conclusion that the LCC of the GSHP system would be greater than the VAV system. Also, the SIR for the GSHP investment would be less than 1, and the DPB would take longer than 25 years for the investment to be recovered.

For the air-source heat pump alternative, the GSHP would save, on average, 23,700 kWh or 3.6% of the energy consumed by the ASHP system, which equals to 13.3 metric tons of carbon dioxide emissions. The GSHP system can reduce the peak energy demand by 206 kW or 7.5%. Moreover, the GSHP system can save $1,757 or 1.1% of the ASHP system annual energy cost.

The investment in the GSHP system when compared to the air-source heat pump system will be recovered when the present value of OM&R, water, and residual cost differences, subtracting the difference of replacement costs ($\Delta OM\&R + \Delta W + \Delta Res - \Delta Repl$), are greater than $1,085,691. When this condition is met, the life-cycle costs of the GSHP system will be less than that of the ASHP system. Also, the SIR of the GSHP investment will be greater than 1, and the DPB of the GSHP will be less than the analysis period of 25 years.

When the replacement and maintenance costs were estimated and input to the LCCA spreadsheet, the results of the LCCA indicate that the cost estimates can meet only $252,749 in their present value. The results lead to the conclusion that the LCC of the GSHP system would be greater than the ASHP system. Also, the SIR for the GSHP investment would be less than 1 and the DPB would take longer than 25 years for the investment to be recovered when compared to the ASHP system.

Referring to the energy performance evaluation results, it is interesting to note that the energy consumption and peak energy demand of the building HVAC end-use are dominated by the space cooling subcategory. This characteristic may lessen a benefit of GSHP system which has higher energy efficiency on space heating than space cooling ($COP_{HP} = COP_{R} + 1$). Also, the cooling efficiency of conventional GSHPs may continue to decrease over their service life-time due to increasing soil temperature as stated by Durkin and Cencil (2007).

For economic evaluations, the results show that the cost savings from using the GSHP system are unlikely to overcome the initial investment cost differences between the GSHP system and the alternative systems. In some cases, the high drilling cost of GSHP systems could be offset if site improvements such as site excavation or rainwater management are necessary. For instance, the site excavation may benefit the installation of horizontal-loop GSHPs, and a rainwater collector pond may benefit the installation of SWHPs. Although a part of the water-loop of GSHP system used in the KnowledgeWorks I and II buildings was installed in a rainwater collector pond, this study did not bring this benefit into the economic evaluations. In addition, the use of roof-top heat pumps may diminish the saving opportunities on maintenance and replacement costs from the use of indoor units. According to supporting evidence, the median life-time of roof-top units is about 15 years while the indoor units may last up to 20-25 years (ASHRAE 2003).

In summary, the energy performance of the geothermal heat pump system used in the KnowledgeWorks I and II buildings is evidently better than the performance of the buildings with the alternative HVAC scenarios. For the economy of the geothermal heat pump investment, it is unclear that the cost savings gained from the geothermal heat pump system could offset the system's initial investment costs, which are about two times more expensive than the alternative systems. There is a body of evidences indicating that the GSHP investment could be economically infeasible for this case. For more accurate results, it could be suggested that the coefficient of performance factors of the GSHP system should be monitored and the actual internal loads information should be collected. Moreover, further information about OM&R, water, residual, and replacement costs of the GSHP and the alternative systems would be the key to determining the economic benefits of the GSHP system over the alternative systems.

5.3 Suggestions for Future Research

In simulation and modeling research, the assumptions made on simulation variables usually have crucial influence on the simulation results. This characteristic of simulation and modeling research can link to a future study featuring the energy performance and economy of the geothermal heat pumps. One possible study could be a study on the variables that influence energy performance and economic benefits of GSHPs. Based on this thesis; it is interesting to understand more about how the changes of assumptions made on simulation variables affect the original research results summarized previously in this chapter. Therefore, two series of simulations based on the modifications made on simulation variables were performed to demonstrate one possible direction of the future research.

The first example is a modification made on the efficiency parameters of HVAC equipment. The intent of this modification is to understand how the differences between the efficiency parameters of water-air heat pumps used in the GSHP system and the efficiency parameters of VAV packages and air-air heat pumps used in the alternative systems affect the life-cycle cost saving results. In the modification, the efficiency parameters of the VAV packages and air-air heat pumps were reduced by 25% (x0.75), 50% (x0.50), and 75% (x0.25) from the values used in the original simulation models. Table 5.1 shows a series of variables used in this modification. In the Table, the x1 columns are the values used in the original simulation models.

Table 5.1 - Variables changed on the efficiency parameters of VAV with hot water coil heating and ASHP systems.

Unit Size	**GSHP**		**VAV**					**ASHP**				
				x1	**x0.75**	**x0.50**	**x0.25**		**x1**	**x0.75**	**x0.50**	**x0.25**
<65 kBtuh	**EER**	11.0528	**EER**	8.4200	6.3150	4.2100	2.105	**EER**	8.4200	6.3150	4.2100	2.105
or 5.4 tons	**COP**	3.7394	**Boiler Eff.**	80%	60%	40%	20%	**COP**	2.8100	2.1075	1.4050	0.703
90-135 kBtuh	**EER**	12.5703	**EER**	8.9000	6.6750	4.4500	2.225	**EER**	8.9000	6.6750	4.4500	2.225
or 7.5-11.25 tons	**COP**	3.8397	**Boiler Eff.**	80%	60%	40%	20%	**COP**	3.0000	2.2500	1.5000	0.750
135-240 kBtuh	**EER**	9.9198	**EER**	8.5000	6.3750	4.2500	2.125	**EER**	8.5000	6.3750	4.2500	2.125
or 11.25-20 tons	**COP**	3.7885	**Boiler Eff.**	80%	60%	40%	20%	**COP**	2.9000	2.1750	1.4500	0.725

The results in Figure 5.1 show that the differences of the efficiency parameters used in the simulations have remarkable influence on the life-cycle cost savings gained from using the GSHP system when compared to both VAV and ASHP cases. Figure 5.1 indicates that the investment in the GSHP system will break even when the efficiency parameters (EER and Boiler Efficiency) are 67.31% less than the values used in the original VAV model. The investment in the GSHP system will be economically feasible when compared to the VAV packages that have 74.64% less EER than the water-air heat pumps on average. In contrast, when compared to the ASHP alternative, the investment on the GSHP system could break even if the efficiency parameters of air-air heat pumps reach some point beyond 75% less efficient than the values used in the original ASHP model.

The second example is a modification made on utility rates. Use of modification is intended to understand how the increase in utility rates affects the life-cycle cost saving results of the original study. In the modification, the utility rates used in GSHP, VAV, and ASHP system scenarios were increased by 2 times (x2), 3 times (x3), and 4 times (x4) from the values used in the original simulation models. Table 5.2 shows a series of variables used in this modification. In the Table, the x1 columns are the values used in the original simulation models.

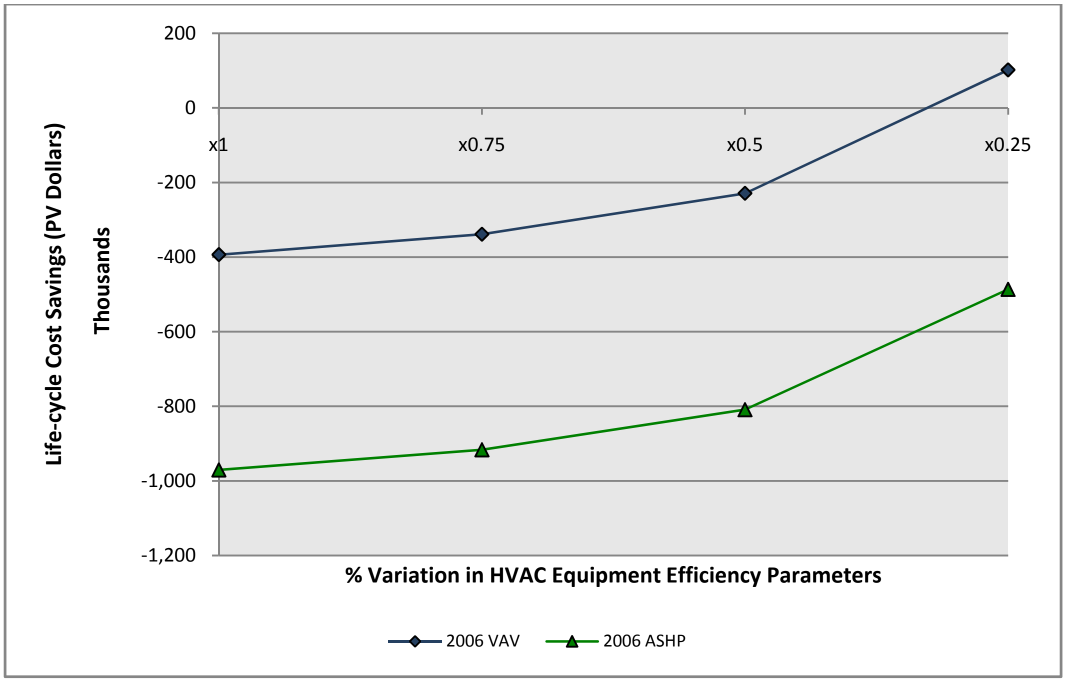


Figure 5.1 - Impacts of the modification made on HVAC equipment efficiency parameters

Table 5.2 - Variables change on the rates of electricity and natural gas

Electricity					
		x1	**x2**	**x3**	**x4**
Customer Charge ($/kWh)		18.00	36.00	54.00	72.00
Energy Blocks	**Blk Size**	**$/kWh**	**$/kWh**	**$/kWh**	**$/kWh**
kWh Block	673	0.055320	0.110640	0.165960	0.22128
kWh Block	900	0.044170	0.088340	0.132510	0.17668
kWh Block	2,500	0.044170	0.088340	0.132510	0.17668
kWh Block	50,000	0.043630	0.087260	0.130890	0.17452
kWh Block	999,999	0.043400	0.086800	0.130200	0.17360
Demand Blocks	**Blk Size**	**$/kW**	**$/kW**	**$/kW**	**$/kW**
kW Block	99,999	3.420	6.840	10.260	13.68
Natural Gas					
		x1	**x2**	**x3**	**x4**
Customer Charge ($/kWh)		14.50	29.00	43.50	58.00
Energy Blocks	**Blk Size**	**$/kWh**	**$/kWh**	**$/kWh**	**$/kWh**
Therm Block	96	0.206388	0.412776	0.619164	0.825552
Therm Block	513	0.128389	0.256778	0.385167	0.513556
Therm Block	999,999	0.109365	0.218730	0.328095	0.437460
Demand Blocks	**Blk Size**	**$/kW**	**$/kW**	**$/kW**	**$/kW**
Therm Block	99,999	0.000	0.000	0.000	0.000

Figure 5.2 illustrates that the changes made on utility rates have only little influence on the life-cycle cost savings results when compared to both the package VAV with hot water coil heating

and ASHP cases. The results show that the investment on the GSHP system is unlikely to be breakeven when compared to the VAV and ASHP alternatives even the utilities rates were increased by 4 times.

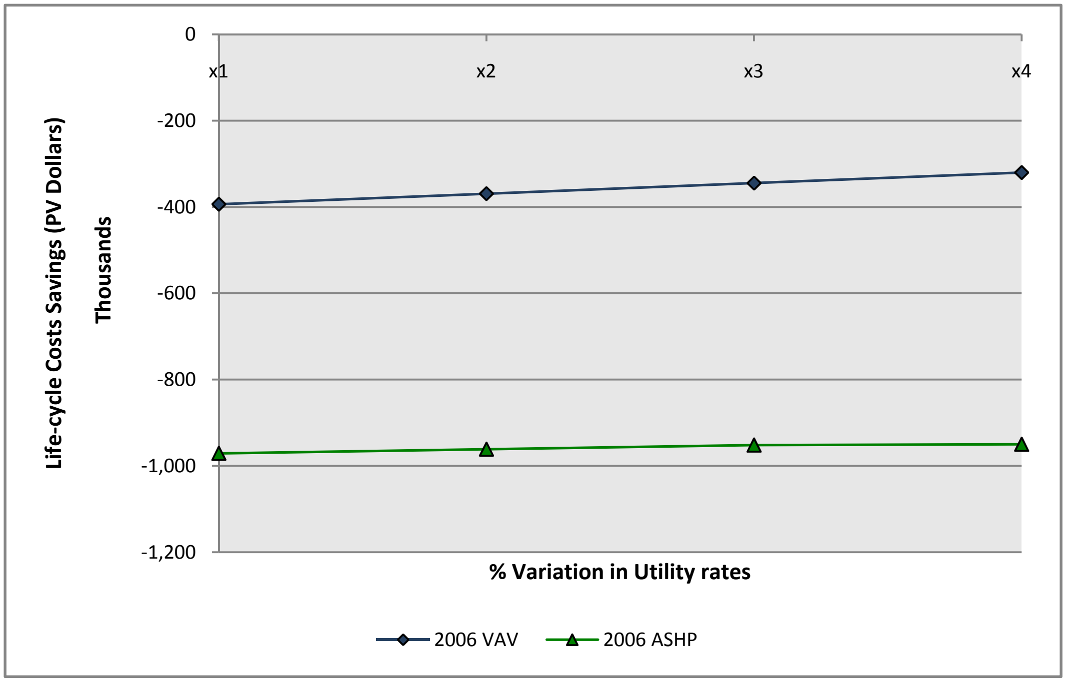


Figure 5.2 - Impacts of the modification made on utility rates

From the previous examples, the differences of HVAC equipment efficiency evidently have more impact on the life-cycle cost saving results than the increase in utility rates. In addition to these instance variables, there are a number of variables involve in a building energy performance simulation. Such variables can be categorized by their characteristics, including but not limited to, site/location, building, system, and economic variable categories. It could be suggested that a future study related to the changes of assumptions made on simulation variables in each category could be beneficial to address the variables that are critical to the energy performance and economic evaluation of GSHPs.

Another future research approach featuring the energy performance and economy of the geothermal heat pumps could be performed using the in-situ monitoring and statistical approaches. The in-situ monitoring can be used for collecting various samples from buildings with and without GSHPs based on site/location, building, system, and economics characteristics. The statistical procedures can be used for evaluating the energy performance and economic benefits of the geothermal heat pumps. This study can be performed as separate research or can be performed as a part of the simulation research to strengthen the accuracy of research outcomes.

One advantage of future studies on the variables that influence energy performance and economic benefits of GSHPs is that the studies may help develop a decision-making framework for architects or designers to determine the variables that are critical to design decisions related to the utilization of GSHPs in buildings. Another advantage is that the outcomes of the studies may support the manufacturer decisions on the directions of product developments and marketing strategies. Moreover, the outcomes of the studies may help government agencies develop public policies that encourage people to use GSHPs which are capable of reducing national energy consumption and corresponding emissions.

Appendix A

Refer to the image from the digital document eQUEST Training Workbook: Energy Simulation Training for Design & Construction Professionals by James J. Hirsch & Associates, 2004. pg 8 of 142. http://doe2.com/download/equest/eQuestTrainingWorkbook.pdf.

Refer to the image from the digital document eQUEST Training Workbook: Energy Simulation Training for Design & Construction Professionals by James J. Hirsch & Associates, 2004. pg 9 of 142. *http://doe2.com/download/equest/eQuestTrainingWorkbook.pdf.*

Appendix B

Table B.1 - Building site information and weather data

Building Location	Detail
Address:	2200 Kraft Drive
City, State Zip:	Blacksburg, VA 24060-6363
Building Owner	**Detail**
Name:	Virginia Tech Corporate Research Center
Address:	1872 Pratt Drive
City, State Zip:	Blacksburg, VA 24060-6363
Phone:	540-961-3600
Weather File (TMY2)	**Detail**
Location:	Roanoke, VA
Source:	http://doe2.com/Download/Weather/TMY2

Table B.2 - Building shell, structure, materials and shades

Building Exterior Constructions		
Roof surface	**Materials**	
Construction:	Metal frame, > 24 in. o.c.	
Exterior Finish/Color:	Gravel	Light' (abs = 0.4)
Exterior Insulation:	5 in. polystyrene (R-25)	
Additional Insulation:	- No batt or rad barrier -	
Interior Insulation:	- No board insulation -	
Above Grade Walls	**Materials**	
Construction:	Metal frame, 2x6, 24 in. o.c.	
Exterior Finish/Color:	Brick	, brown, medium light
Exterior Insulation:	1/2in. Fiber board sheathing (R-1.3)	
Additional Insulation:	R-19 batt	
Interior Insulation:	- No board insulation -	
Ground Floor	**Materials**	
Exposure:	Earth Contact	
Construction:	4in. Concrete	
Exterior Cavity/Insulation:	Vertical insulation board, R-5, 4ft deep	
Interior Finish:	Carpet with rubber pad	
Second Floor	**Materials**	
Exposure:	Over Condition Space (adiabatic)	
Construction:	6 in. Concrete	
Exterior Cavity/Insulation:	- No batt or board insulation -	
Interior Insulation:	- No board insulation -	
Cap & Finish:	- No concrete cap -	Carpet with rubber pad
Building Interior Constructions		
Ceilings	**Materials**	
Interior Finish:	Plaster Finish	
Batt Insulation:	R-11 batt	(2-1/2" sound batt insulation)
Vertical Walls:	Materials	
Wall Type:	Frame	

Batt Insulation:	R-11 batt	(2-1/2" sound batt insulation)
Exterior Doors/Windows		
Doors	**Materials**	
Type-01		
Dimension:	H8.0xW6.5 ft.	
Construction:	Double clr/tint	
Glass Category and Type:	Double bronze 1/4in, 1/2in Air (2204)	
Frame Type:	Aluminum with break	, W2.0 in
Type-02		
Dimension:	H7.0xW3.0 ft.	
Construction:	Double clr/tint	
Glass Category and Type:	Double Bronze 1/4in, 1/2in Air (2204)	
Frame Type:	Aluminum with break	, W2.0 in
Windows	**Materials**	
Type-01		
Dimension:	Window H6.0 ft.xW7.0ft., Sill 6.0 ft.	
Construction:	Double clr/tint	
Glass Category and Type:	Double Bronze 1/4in, 1/2in Air (2204)	
Frame Type:	Aluminum with break, ins. spacer (Assumed fixed), W2.0 in	
Exterior Window Shades and Blinds		
Exterior Window Shades:	Materials	
Overhangs:	None	
Fins:	None	
Window Blinds/Drapes:	Materials	
Type:	Horizontal blinds - light color	20% Occupied, 80% Unoccupied

Table B.3 - Building operation and scheduling

Description of Seasons:	Typical use throughout year
Number of seasons:	1
Season #1 Label:	Entire year, 1/1-12/31
Use:	High use

KnowledgeWorks I (RB XVIII)

Weekly Schedule	**Occupancy**		**RB18-1_RTU1-1,2,3,4**	
	Opens at	**Closed at**	**On at**	**Off at**
Mon	6:00 AM	6:00 PM	4:00 AM	4:00 PM
Tue	6:00 AM	6:00 PM	4:00 AM	4:00 PM
Wed	6:00 AM	6:00 PM	4:00 AM	4:00 PM
Thu	6:00 AM	6:00 PM	4:00 AM	4:00 PM
Fri	6:00 AM	6:00 PM	4:00 AM	4:00 PM
Sat	Closed	Closed	Off	Off
Sun	Closed	Closed	Off	Off
Holiday	Closed	Closed	Off	Off
Weekly Schedule	**RB18-1_WSHP1-1,2,3**		**RB18-1_RTU2-2,3,4**	
	Opens at	**Closed at**	**On at**	**Off at**
Mon	4:00 AM	4:00 PM	4:00 AM	4:00 PM
Tue	4:00 AM	4:00 PM	4:00 AM	4:00 PM
Wed	4:00 AM	4:00 PM	4:00 AM	4:00 PM
Thu	4:00 AM	4:00 PM	4:00 AM	4:00 PM
Fri	4:00 AM	4:00 PM	4:00 AM	4:00 PM
Sat	Off	Off	Off	Off

Sun	Off	Off	Off	Off
Holiday	Off	Off	Off	Off
Weekly Schedule	**RB18-1_WSHP2-1,2,3,4**		**RB18-1_RTU2-1,DX|EL**	
	On at	**Off at**	**On at**	**Off at**
Mon	4:00 AM	4:00 PM	On 24 hours	On 24 hours
Tue	4:00 AM	4:00 PM	On 24 hours	On 24 hours
Wed	4:00 AM	4:00 PM	On 24 hours	On 24 hours
Thu	4:00 AM	4:00 PM	On 24 hours	On 24 hours
Fri	4:00 AM	4:00 PM	On 24 hours	On 24 hours
Sat	Off	Off	On 24 hours	On 24 hours
Sun	Off	Off	On 24 hours	On 24 hours
Holiday	Off	Off	On 24 hours	On 24 hours
KnowledgeWorks I (RB XIX)				
Weekly Schedule	**Occupancy**		**RB18-2_RTU1,2,3,4**	
	Opens at	**Closed at**	**On at**	**Off at**
Mon	6:00 AM	10:00 PM	4:00 AM	8:00 PM
Tue	6:00 AM	10:00 PM	4:00 AM	8:00 PM
Wed	6:00 AM	10:00 PM	4:00 AM	8:00 PM
Thu	6:00 AM	10:00 PM	4:00 AM	8:00 PM
Fri	6:00 AM	10:00 PM	4:00 AM	8:00 PM
Sat	6:00 AM	10:00 PM	4:00 AM	8:00 PM
Sun	6:00 AM	10:00 PM	4:00 AM	8:00 PM
Holiday	Closed	Closed	Closed	Closed
Weekly Schedule	**RB18-2_RTU6,8**		**RB18-2_RTU5,7,9,10,CRACU**	
	On at	**Off at**	**On at**	**Off at**
Mon	4:00 AM	8:00 PM	On 24 hours	On 24 hours
Tue	4:00 AM	8:00 PM	On 24 hours	On 24 hours
Wed	4:00 AM	8:00 PM	On 24 hours	On 24 hours
Thu	4:00 AM	8:00 PM	On 24 hours	On 24 hours
Fri	4:00 AM	8:00 PM	On 24 hours	On 24 hours
Sat	4:00 AM	8:00 PM	On 24 hours	On 24 hours
Sun	4:00 AM	8:00 PM	On 24 hours	On 24 hours
Holiday	Closed	Closed	On 24 hours	On 24 hours
Holiday				
New Year' Day (celebrated)			Labor Day	
Martin Luther King Day			Columbus Day	
President's Day			Veteran's Day	
Memorial Day			Thanksgiving	
July 4th (celebrated)			Christmas Day (celebrated)	

Table B.4 - Internal loads

	Space Type	**Activity Area (ft^2)**	**Area Percent (%)**	**Design Max Occupancy (ft^2/person)**	**Design Ventilation (CFM/Per)**	**Lighting (W/ft^2)**	**Office Equipment (W/ft^2)**	**Kitchen Equipment (W/ft^2)**		**Misc. Equipment (W/ft^2)**	
								Load	**Sen.**	**Load**	**Sen.**
KnowledgeWorks I-1st Floor											
1.	Lobby (Main Entry and Assembly)	501	2.4%	100	20.00	1.77	-	-	-	0.25	1.00
2.	Office (General)	12,597	60.4%	200	20.00	1.24	2.00	-	-	1.50	1.00

3.	Office (Executive/ Private)	4,437	21.3%	200	20.00	1.49	1.50	-	-	1.50	1.00
4.	Conference Room	308	1.5%	50	20.00	0.92	1.00	-	-	0.20	1.00
5.	Restrooms	492	2.4%	300	50.00	0.77	-	-	-	0.10	1.00
6.	Mechanical	543	2.6%	2,000	100.00	0.81	-	-	-	0.10	1.00
7.	Storage	80	0.4%	500	75.00	1.19	-	-	-	-	1.00
8.	Corridor	1,893	9.1%	1,000	50.00	0.57	-	-	-	-	1.00
	Total	20,852	100.0%	100	0.103	x1.0					
KnowledgeWorks I-2nd Floor (Typical)											
1.	Lobby (Main Entry and Assembly)	436	2.8%	100	20.00	1.77	-	-	-	0.25	1.00
2.	Office (General)	4,802	30.4%	200	20.00	1.24	2.00	-	-	1.50	1.00
3.	Office (Executive/ Private)	1,672	10.6%	200	20.00	1.49	1.50	-	-	1.50	1.00
4.	High Tech/ Bio Tech Lab	6,066	38.5%	200	20.00	3.19	2.00	-	-	2.00	1.00
5.	Conference Room	220	1.4%	50	20.00	0.92	1.00	-	-	0.20	1.00
6.	Restrooms	492	3.1%	300	50.00	0.77	-	-	-	0.10	1.00
7.	Mechanical	439	2.8%	2,000	100.00	0.81	-	-	-	0.10	1.00
8.	Corridor	1,648	10.4%	1,000	50.00	0.57	-	-	-	-	1.00
	Total	15,775	100.0%	75	0.100	x1.0					
KnowledgeWorks I-2nd Floor (24/7)											
1.	High Tech/ Bio Tech Lab	4,698	100.0%	200	20.00	3.19	2.00	-	-	2.00	1.00
	Total	4,698	100.0%	23	0.095	x1.0					
KnowledgeWorks I-Above Breeze Way											
1.	Lobby (Main Entry and Assembly)	468	34.6%	100	20.00	1.77	-	-	-	0.25	1.00
2.	Conference Room	884	65.4%	50	20.00	0.92	1.50	-	-	0.20	1.00
	Total	1,352	100%	22	0.178	x1.0					
KnowledgeWorks II-1st Floor											
1.	Office (Open Plan)	4,572	21.1%	200	20.00	1.24	1.00	-	-	1.50	1.00
2.	Office (Executive/ Private)	5,562	25.6%	200	20.00	1.49	1.50	-	-	1.50	1.00
3.	High Tech/ Bio Tech Lab	3,227	14.9%	200	20.00	3.19	2.00	-	-	2.00	1.00
4.	Conference Room	3,663	16.9%	50	20.00	0.92	1.00	-	-	0.20	1.00
5.	Kitchen and Food Preparation	240	1.1%	200	15.00	1.19	-	2.00	1.00	1.00	1.00
6.	Restrooms	379	1.7%	300	50.00	0.77	-	-	-	0.10	1.00
7.	Mechanical	510	2.3%	2,000	100.00	0.81	-	-	-	0.10	1.00
8.	Corridor	3,543	16.3%	1,000	50.00	0.57	-	-	-	-	1.00

Total	21,694	100%	166	0.158	x1.0					
KnowledgeWorks II-2nd Floor (Typical)										
1. Office (Open Plan)	4,389	43.0%	200	20.00	1.24	2.00	-	-	1.50	1.00
2. Office (Executive/ Private)	2,525	24.7%	200	20.00	1.49	1.50	-	-	1.50	1.00
3. Conference Room	1,242	12.2%	50	20.00	0.92	1.00	-	-	0.20	1.00
4. Restrooms	364	3.6%	300	50.00	0.77	-	-	-	0.10	1.00
5. Mechanical	233	2.3%	2,000	100.00	0.81	-	-	-	0.10	1.00
6. Storage (Conditioned)	341	3.3%	500	75.00	1.19	-	-	-	-	1.00
7. Corridor	1,113	10.9%	1,000	50.00	0.57	-	-	-	-	1.00
Total	10,206	100%	186	0.187	x1.0					
KnowledgeWorks II-2nd Floor (24/7)										
1. Office (Open Plan)	5,869	57.5%	200	20.00	1.24	2.00	-	-	1.50	1.00
2. Office (Executive/ Private)	1,353	13.3%	200	20.00	1.49	1.50	-	-	1.50	1.00
3. Conference Room	739	7.2%	50	20.00	0.92	1.00	-	-	0.20	1.00
4. Computer Room (Mainframe/ Server)	1,611	15.8%	200	20.00	1.24	-	-	-	10.00	1.00
5. Mechanical	519	5.1%	2,000	100.00	0.81	-	-	-	-	1.00
6. Storage (Conditioned)	266	2.6%	500	75.00	1.19	-	-	-	-	1.00
7. Corridor	771	-1.5%	1,000	50.00	0.57	-	-	-	-	1.00
Total	11,129	100%	186	0.187	x1.0					

Domestic Hot Water	
Model:	DHW equipment with seasonal profile
Heater Type:	Electricity
Storage Capacity (gal):	50
Hot Water Usage (gal/person/day):	1
Supply Temperature (˚F):	135
Inlet Water Temp:	Equals Ground Temperature
Recirculation (%):	-
Input Rate (kW):	12.0
Tank Insulation R-Value (h-ft^2°F/Btu):	12.0
External End-Uses (not contributing to space loads)	
Exterior Lighting	
Model:	Connected intensity and seasonal profiles
Exterior Lighting Load (W/ft^2):	0.04

Table B.5 - HVAC equipment and performance

Table B.5.1 - Air-side system types

HVAC System Definition	
System Definition	**Detail**

System Type Name:	RB18-1 RTU1-1	RB18-1 RTU1-2	RB18-1 RTU1-3	RB18-1 RTU1-4
Cooling Source:	DX Coils	DX Coils	DX Coils	DX Coils
Heating Source:	DX HP	DX HP	DX HP	DX HP
Heat Pump Source:	Ground Loop	Ground Loop	Ground Loop	Ground Loop
System Type:	GSHP	GSHP	GSHP	GSHP
System per Area:	Zone	Zone	Zone	Zone
Return Air Path:	Ducted	Ducted	Ducted	Ducted
System Assignment to Thermal Zones	**Detail**			
Shell Component(s):	18-1 1st Fr.	18-1 1st Fr.	18-1 1st Fr.	18-1 1st Fr.
Description of Assigned Zones:	All Zones	All Zones	All Zones	All Zones
HVAC Zones: Temperatures and Air Flows				
Seasonal Thermostat Setpoints	**Detail**			
Occupied (°F)				
Cool:	76.0	76.0	76.0	76.0
Heat:	70.0	70.0	70.0	70.0
Unoccupied (°F)				
Cool:	85.0	85.0	85.0	85.0
Heat:	60.0	60.0	60.0	60.0
Design Temperatures	**Detail**			
Cooling Design Temp. (°F)				
Indoor:	75.0	75.0	75.0	75.0
Supply:	55.0	55.0	55.0	55.0
Heating Design Temp. (°F)				
Indoor:	72.0	72.0	72.0	72.0
Supply:	90.0	90.0	90.0	90.0
Air Flows (cfm/ft^2)				
Minimum Design Flow:	0.50	0.50	0.50	0.50
Packages HVAC Equipment	**Detail**			
Cooling				
Total Cooling Capacity (MBtu):	169.72	141.83	172.06	115.03
Overall Size (tons):	14.1433	11.8192	14.3383	9.5858
Typical Unit Size (tons):	15.00	12.50	15.00	10.00
Cooling Power (kW):	16.87	12.61	17.14	9.77
Efficiency (EER):	10.060	11.247	10.039	11.774
Crankcase Heating:	Allow	Allow	Allow	Allow
Heating				
Total Heating Capacity (MBtu):	149.64	122.24	151.87	108.60
Size (kBtu):	149,640.0	122,240.0	151,870.0	108,600.0
Heating Power (kW):	12.83	10.11	12.63	8.93
Efficiency (COP):	3.712	3.848	3.827	3.871
HVAC System Fans				
Supply Fans	**Detail**			
Power (BHP):	1.57	1.3	1.77	1.01
Mtr Eff:	High	High	High	High
Fan Flow (cfm):	5,000	4,120	5,600	3,200
OSA (%):	Auto	Auto	Auto	Auto
Return Fans	**Detail**			
Return Type:	None	None	None	None
Power (BHP):	-	-	-	-
Mtr Eff:	-	-	-	-
Fan Flow (cfm):	-	-	-	-
HVAC System Fan Schedules				
Operate Fans	**Detail**			
Hours before open:	2	2	2	2
Hours before close:	-2	-2	-2	-2

Cycle fan at night (Y/N): Mode (Con/Int):	N -	N -	N -	N -
HVAC Zone Heating, Vent and Economizers				
Zone Heat Sources & Capacities / Delta T	**Detail**			
Baseboards: Power (kW):	Electric -14.2	Electric -16.1	Electric -19.8	Electric -31.9
Economizer (s)	**Detail**			
Type: High Limit (˚F): Compressor:	DB Temp 55 Run with	DB Temp 55 Run with	DB Temp 55 Run with	DB Temp 55 Run with

Table B.5.1 - Air-side system types (continue)

HVAC System Definition				
System Definition	**Detail**			
System Type Name: Cooling Source: Heating Source: Heat Pump Source: System Type: System per Area: Return Air Path:	RB18-1 RTU2-1 DX Coils DX HP Ground Loop GSHP Zone Ducted	RB18-1 RTU2-2 DX Coils DX HP Ground Loop GSHP Zone Ducted	RB18-1 RTU2-3 DX Coils DX HP Ground Loop GSHP Zone Ducted	RB18-1 RTU2-4 DX Coils DX HP Ground Loop GSHP Zone Ducted
System Assignment to Thermal Zones	**Detail**			
Shell Component(s): Description of Assigned Zones:	18-1 2nd Fr. All Zones	18-1 2nd Fr. All Zones	18-1 2nd Fr. All Zones	18-1 2nd Fr. All Zones
HVAC Zones: Temperatures and Air Flows				
Seasonal Thermostat Setpoints	**Detail**			
Occupied (°F) Cool: Heat: **Unoccupied (°F)** Cool: Heat:	 76.0 70.0 85.0 60.0	 76.0 70.0 85.0 60.0	 76.0 70.0 85.0 60.0	 76.0 70.0 85.0 60.0
Design Temperatures	**Detail**			
Cooling Design Temp. (°F) Indoor: Supply: **Heating Design Temp. (°F)** Indoor: Supply: **Air Flows (cfm/ft^2)** Minimum Design Flow:	 75.0 55.0 72.0 90.0 0.50	 75.0 55.0 72.0 90.0 0.50	 75.0 55.0 72.0 90.0 0.50	 75.0 55.0 72.0 90.0 0.50
Packages HVAC Equipment	**Detail**			
Cooling Total Cooling Capacity (MBtu): Overall Size (tons): Typical Unit Size (tons): Cooling Power (kW): Efficiency (EER): Crankcase Heating: **Heating** Total Heating Capacity (MBtu): Size (kBtu): Heating Power (kW): Efficiency (COP):	 172.47 14.3725 15.00 17.17 10.045 Allow 152.29 152,290.0 12.60 3.847	 172.68 14.3900 15.00 16.93 10.200 Allow 149.71 149,710.0 12.83 3.714	 172.71 14.3925 15.00 16.92 10.207 Allow 149.74 149,740.0 12.83 3.715	 117.22 9.7683 10.00 9.83 11.925 Allow 108.85 108,850.0 8.91 3.888

HVAC System Fans				
Supply Fans	**Detail**			
Power (BHP):	1.79	1.57	1.57	1.03
Mtr Eff:	High	High	High	High
Fan Flow (cfm):	5,700	5,000	5,000	3,260
OSA (%):	Auto	Auto	Auto	Auto
Return Fans	**Detail**			
Return Type:	None	None	None	None
Power (BHP):	-	-	-	-
Mtr Eff:	-	-	-	-
Fan Flow (cfm):	-	-	-	-
HVAC System Fan Schedules				
Operate Fans	**Detail**			
Hours before open:	-	2	2	2
Hours before close:	-	-2	-2	-2
Cycle fan at night (Y/N):	Y	N	N	N
Mode (Con/Int):	Con.	-	-	-
HVAC Zone Heating, Vent and Economizers				
Zone Heat Sources & Capacities / Delta T	**Detail**			
Baseboards:	Electric	Electric	Electric	Electric
Power (kW):	-15.8	-16.8	-20.7	-11.7
Economizer (s)	**Detail**			
Type:	DB Temp	DB Temp	DB Temp	DB Temp
High Limit (˚F):	55	55	55	55
Compressor:	Run with	Run with	Run with	Run with

Table B.5.1 - Air-side system types (continue)

HVAC System Definition				
System Definition	**Detail**			
System Type Name:	RB18-1 WSHP1-1	RB18-1 WSHP1-2	RB18-1 WSHP1-3	RB18-1 DX|EL
Cooling Source:	DX Coils	DX Coils	DX Coils	DX Coils
Heating Source:	DX HP	DX HP	DX HP	Ele Res
Heat Pump Source:	Ground Loop	Ground Loop	Ground Loop	-
System Type:	GSHP	GSHP	GSHP	DX w/Ele
System per Area:	Zone	Zone	Zone	Zone
Return Air Path:	Ducted	Ducted	Ducted	Ducted
System Assignment to Thermal Zones	**Detail**			
Shell Component(s):	18-1 1st Fr.	18-1 1st Fr.	18-1 1st Fr.	18-2 2nd Fr.
Description of Assigned Zones:	All Zones	All Zones	All Zones	All Zones
HVAC Zones: Temperatures and Air Flows				
Seasonal Thermostat Setpoints	**Detail**			
Occupied (°F)				
Cool:	76.0	76.0	76.0	76.0
Heat:	70.0	70.0	70.0	70.0
Unoccupied (°F)				
Cool:	85.0	85.0	85.0	85.0
Heat:	60.0	60.0	60.0	60.0
Design Temperatures	**Detail**			
Cooling Design Temp. (°F)				
Indoor:	75.0	75.0	75.0	75.0
Supply:	55.0	55.0	55.0	55.0
Heating Design Temp. (°F)				
Indoor:	72.0	72.0	72.0	72.0

Supply: **Air Flows (cfm/ft^2)** Minimum Design Flow:	90.0 0.50	90.0 0.50	90.0 0.50	90.0 0.50
Packages HVAC Equipment	**Detail**			
Cooling				
Total Cooling Capacity (MBtu):	34.39	33.57	18.24	NA
Overall Size (tons):	2.8658	2.7975	1.5200	NA
Typical Unit Size (tons):	3.00	3.00	1.50	NA
Cooling Power (kW):	2.98	2.96	1.63	NA
Efficiency (EER):	11.540	11.341	11.190	8.5
Crankcase Heating:	Allow	Allow	Allow	NA
Heating				
Total Heating Capacity (MBtu):	29.62	30.14	16.20	NA
Size (kBtu):	29,620.0	30,140.0	16,200.0	Auto
Heating Power (kW):	2.54	2.54	1.35	NA
Efficiency (COP):	3.711	3.777	3.819	NA
HVAC System Fans				
Supply Fans	**Detail**			
Power (BHP):	0.34	0.34	0.2	NA
Mtr Eff:	High	High	High	NA
Fan Flow (cfm):	1,082	1,082	646	NA
OSA (%):	Auto	Auto	Auto	Auto
Return Fans	**Detail**			
Return Type:	None	None	None	None
Power (BHP):	-	-	-	-
Mtr Eff:	-	-	-	-
Fan Flow (cfm):	-	-	-	-
HVAC System Fan Schedules				
Operate Fans	**Detail**			
Hours before open:	2	2	2	-
Hours before close:	-2	-2	-2	-
Cycle fan at night (Y/N):	N	N	N	Y
Mode (Con/Int):	-	-	-	Con.
HVAC Zone Heating, Vent and Economizers				
Zone Heat Sources & Capacities / Delta T	**Detail**			
Baseboards:	Electric	Electric	Electric	Electric
Power (kW):	-2.7	-2.6	0.0	0.0
Economizer (s)	**Detail**			
Type:	DB Temp	DB Temp	DB Temp	DB Temp
High Limit (˚F):	55	55	55	55
Compressor:	Run with	Run with	Run with	Run with

Table B.5.1 - Air-side system types (continue)

HVAC System Definition				
System Definition	**Detail**			
System Type Name:	RB18-1 WSHP2-1	RB18-1 WSHP2-2	RB18-1 WSHP2-3	RB18-1 WSHP2-4
Cooling Source:	DX Coils	DX Coils	DX Coils	DX Coils
Heating Source:	DX HP	DX HP	DX HP	DX HP
Heat Pump Source:	Ground Loop	Ground Loop	Ground Loop	Ground Loop
System Type:	GSHP	GSHP	GSHP	GSHP
System per Area:	Zone	Zone	Zone	Zone
Return Air Path:	Ducted	Ducted	Ducted	Ducted
System Assignment to Thermal Zones	**Detail**			

Shell Component(s): Description of Assigned Zones:	18-1 2nd Fr. All Zones	18-1 2nd Fr. All Zones	18-1 2nd Fr. All Zones	18-1 2nd Fr. All Zones
HVAC Zones: Temperatures and Air Flows				
Seasonal Thermostat Setpoints	**Detail**			
Occupied (°F)				
Cool:	76.0	76.0	76.0	76.0
Heat:	70.0	70.0	70.0	70.0
Unoccupied (°F)				
Cool:	85.0	85.0	85.0	85.0
Heat:	60.0	60.0	60.0	60.0
Design Temperatures	**Detail**			
Cooling Design Temp. (°F)				
Indoor:	75.0	75.0	75.0	75.0
Supply:	55.0	55.0	55.0	55.0
Heating Design Temp. (°F)				
Indoor:	72.0	72.0	72.0	72.0
Supply:	90.0	90.0	90.0	90.0
Air Flows (cfm/ft^2)				
Minimum Design Flow:	0.50	0.50	0.50	0.50
Packages HVAC Equipment	**Detail**			
Cooling				
Total Cooling Capacity (MBtu):	18.35	47.03	18.35	46.38
Overall Size (tons):	1,5292	3.9192	1.5292	3.8650
Typical Unit Size (tons):	1.50	4.00	1.50	4.00
Cooling Power (kW):	1.62	4.52	1.62	4.53
Efficiency (EER):	11.327	10.405	11.327	10.238
Crankcase Heating:	Allow	Allow	Allow	Allow
Heating				
Total Heating Capacity (MBtu):	16.30	40.86	16.30	40.85
Size (kBtu):	16,300.0	40,860.0	16,300.0	40,850.0
Heating Power (kW):	1.37	3.57	1.37	3.56
Efficiency (COP):	3.787	3.643	3.787	3.652
HVAC System Fans				
Supply Fans	**Detail**			
Power (BHP):	0.19	0.55	0.19	0.56
Mtr Eff:	High	High	High	High
Fan Flow (cfm):	611	1,762	611	1,781
OSA (%):	Auto	Auto	Auto	Auto
Return Fans	**Detail**			
Return Type:	None	None	None	None
Power (BHP):	-	-	-	-
Mtr Eff:	-	-	-	-
Fan Flow (cfm):	-	-	-	-
HVAC System Fan Schedules				
Operate Fans	**Detail**			
Hours before open:	2	2	2	2
Hours before close:	-2	-2	-2	-2
Cycle fan at night (Y/N):	N	N	N	N
Mode (Con/Int):	-	-	-	-
HVAC Zone Heating, Vent and Economizers				
Zone Heat Sources & Capacities / Delta T	**Detail**			
Baseboards:	Electric	Electric	Electric	Electric
Power (kW):	-1.9	-13.2	-0.0	-7.8
Economizer (s)	**Detail**			
Type:	DB Temp	DB Temp	DB Temp	DB Temp

High Limit (˚F): Compressor:	55 Run with	55 Run with	55 Run with	55 Run with

Table B.5.1 - Air-side system types (continue)

HVAC System Definition (Continue)				
System Definition	**Detail**			
System Type Name:	RB18-2 RTU1	RB18-2 RTU2	RB18-2 RTU3	RB18-2 RTU4
Cooling Source:	DX Coils	DX Coils	DX Coils	DX Coils
Heating Source:	DX HP	DX HP	DX HP	DX HP
Heat Pump Source:	Ground Loop	Ground Loop	Ground Loop	Ground Loop
System Type:	GSHP	GSHP	GSHP	GSHP
System per Area:	Zone	Zone	Zone	Zone
Return Air Path:	Ducted	Ducted	Ducted	Ducted
System Assignment to Thermal Zones	**Detail**			
Shell Component(s):	18-2 1st Fr.	18-2 1st Fr.	18-2 1st Fr.	18-2 1st Fr.
Description of Assigned Zones:	All Zones	All Zones	All Zones	All Zones
HVAC Zones: Temperatures and Air Flows				
Seasonal Thermostat Setpoints	**Detail**			
Occupied (°F)				
Cool:	76.0	76.0	76.0	76.0
Heat:	70.0	70.0	70.0	70.0
Unoccupied (°F)				
Cool:	85.0	85.0	85.0	85.0
Heat:	60.0	60.0	60.0	60.0
Design Temperatures	**Detail**			
Cooling Design Temp. (°F)				
Indoor:	75.0	75.0	75.0	75.0
Supply:	55.0	55.0	55.0	55.0
Heating Design Temp. (°F)				
Indoor:	72.0	72.0	72.0	72.0
Supply:	90.0	90.0	90.0	90.0
Air Flows (cfm/ft^2)				
Minimum Design Flow:	0.50	0.50	0.50	0.50
Packages HVAC Equipment	**Detail**			
Cooling				
Total Cooling Capacity (MBtu):	149.00	153.00	144.00	120.80
Overall Size (tons):	12.4167	12.7500	12.0000	10.0667
Typical Unit Size (tons):	15.00	10.00	12.50	10.00
Cooling Power (kW):	17.70	10.30	13.30	10.30
Efficiency (EER):	8.418	14.854	10.827	11.728
Crankcase Heating:	Allow	Allow	Allow	Allow
Heating				
Total Heating Capacity (MBtu):	NA	NA	NA	NA
Size (kBtu):	NA	NA	NA	NA
Heating Power (kW):	NA	NA	NA	NA
Efficiency (COP):	3.800	3.800	3.800	3.800
HVAC System Fans				
Supply Fans	**Detail**			
Power (BHP):	1.44	1.26	1.55	1.24
Mtr Eff:	High	High	High	High
Fan Flow (cfm):	5,198	4,000	3,412	4,196
OSA (%):	Auto	Auto	Auto	Auto
Return Fans	**Detail**			
Return Type:	None	None	None	None

Power (BHP): Mtr Eff: Fan Flow (cfm):	- - -	- - -	- - -	- - -
HVAC System Fan Schedules				
Operate Fans	**Detail**			
Hours before open: Hours before close: Cycle fan at night (Y/N): Mode (Con/Int):	2 -2 N -	2 -2 N -	2 -2 N -	2 -2 N -
HVAC Zone Heating, Vent and Economizers				
Zone Heat Sources & Capacities / Delta T	**Detail**			
Baseboards: Power (kW):	Electric -15.8	Electric -16.8	Electric -20.7	Electric -11.7
Economizer (s)	**Detail**			
Type: High Limit (˚F): Compressor:	DB Temp 55 Run with	DB Temp 55 Run with	DB Temp 55 Run with	DB Temp 55 Run with

Table B.5.1 - Air-side system types (continue)

HVAC System Definition (Continue)				
System Definition	**Detail**			
System Type Name: Cooling Source: Heating Source: Heat Pump Source: System Type: System per Area: Return Air Path:	RB18-2 RTU5 DX Coils DX HP Ground Loop GSHP Zone Ducted	RB18-2 RTU6 DX Coils DX HP Ground Loop GSHP Zone Ducted	RB18-2 RTU7 DX Coils DX HP Ground Loop GSHP Zone Ducted	RB18-2 RTU8 DX Coils DX HP Ground Loop GSHP Zone Ducted
System Assignment to Thermal Zones	**Detail**			
Shell Component(s): Description of Assigned Zones:	18-2 2nd Fr. All Zones	18-2 2nd Fr. All Zones	18-2 2nd Fr. All Zones	18-2 2nd Fr. All Zones
HVAC Zones: Temperatures and Air Flows				
Seasonal Thermostat Setpoints	**Detail**			
Occupied (°F) Cool: Heat: **Unoccupied (°F)** Cool: Heat:	 76.0 70.0 85.0 60.0	 76.0 70.0 85.0 60.0	 76.0 70.0 85.0 60.0	 76.0 70.0 85.0 60.0
Design Temperatures	**Detail**			
Cooling Design Temp. (°F) Indoor: Supply: **Heating Design Temp. (°F)** Indoor: Supply: **Air Flows (cfm/ft^2)** Minimum Design Flow:	 75.0 55.0 72.0 90.0 0.50	 75.0 55.0 72.0 90.0 0.50	 75.0 55.0 72.0 90.0 0.50	 75.0 55.0 72.0 90.0 0.50
Packages HVAC Equipment	**Detail**			
Cooling Total Cooling Capacity (MBtu): Overall Size (tons): Typical Unit Size (tons): Cooling Power (kW):	 173.50 14.4583 15.00 17.70	 161.50 13.4583 12.50 13.30	 128.90 10.7417 15.00 17.70	 116.60 9.7167 15.00 13.30

Efficiency (EER):	9.802	12.143	7.282	8.767
Crankcase Heating:	Allow	Allow	Allow	Allow
Heating				
Total Heating Capacity (MBtu):	NA	NA	NA	NA
Size (kBtu):	NA	NA	NA	NA
Heating Power (kW):	NA	NA	NA	NA
Efficiency (COP):	3.800	3.800	3.800	3.800
HVAC System Fans				
Supply Fans	**Detail**			
Power (BHP):	1.75	1.42	1.54	1.34
Mtr Eff:	High	High	High	High
Fan Flow (cfm):	5,430	4,800	4,140	4,000
OSA (%):	Auto	Auto	Auto	Auto
Return Fans	**Detail**			
Return Type:	None	None	None	None
Power (BHP):	-	-	-	-
Mtr Eff:	-	-	-	-
Fan Flow (cfm):	-	-	-	-
HVAC System Fan Schedules				
Operate Fans	**Detail**			
Hours before open:	-	2	-	2
Hours before close:	-	-2	-	-2
Cycle fan at night (Y/N):	Y	N	Y	N
Mode (Con/Int):	Con.	-	Con.	-
HVAC Zone Heating, Vent and Economizers				
Zone Heat Sources & Capacities / Delta T	**Detail**			
Baseboards:	Electric	Electric	Electric	Electric
Power (kW):	-17.4	-19.7	-8.7	-13.6
Economizer (s)	**Detail**			
Type:	DB Temp	DB Temp	DB Temp	DB Temp
High Limit (˚F):	55	55	55	55
Compressor:	Run with	Run with	Run with	Run with

Table B.5.1 - Air-side system types (continue)

HVAC System Definition (Continue)				
System Definition	**Detail**			
System Type Name:	RB18-2 RTU9	RB18-2 RTU10	RB18-2 CRACU	-
Cooling Source:	DX Coils	DX Coils	DX Coils	-
Heating Source:	No Heating	No Heating	DX HP	-
Heat Pump Source:	-	-	Ground Loop	-
System Type:	DX	DX	GSHP	-
System per Area:	Zone	Zone	Zone	-
Return Air Path:	Ducted	Ducted	Ducted	-
System Assignment to Thermal Zones	**Detail**			
Shell Component(s):	18-2 2nd Fr.	18-2 2nd Fr.	18-2 2nd Fr.	-
Description of Assigned Zones:	All Zones	All Zones	All Zones	-
HVAC Zones: Temperatures and Air Flows				
Seasonal Thermostat Setpoints	**Detail**			
Occupied (°F)				
Cool:	76.0	76.0	76.0	-
Heat:	70.0	70.0	70.0	-
Unoccupied (°F)				
Cool:	85.0	85.0	85.0	-
Heat:	60.0	60.0	60.0	-

Design Temperatures	Detail			
Cooling Design Temp. (°F)				
Indoor:	75.0	75.0	75.0	-
Supply:	55.0	55.0	55.0	-
Heating Design Temp. (°F)				
Indoor:	72.0	72.0	72.0	-
Supply:	90.0	90.0	90.0	-
Air Flows (cfm/ft^2)				
Minimum Design Flow:	0.50	0.50	0.50	-
Packages HVAC Equipment	**Detail**			
Cooling				
Total Cooling Capacity (MBtu):	175.93	175.93	0.03	-
Overall Size (tons):	14.6608	14.6608	NA	-
Typical Unit Size (tons):	12 1/2-25	12 1/2-25	2.50	-
Cooling Power (kW):	19.12	19.12	NA	-
Efficiency (EER):	9.201	9.201	10.000	-
Crankcase Heating:	Allow	Allow	NA	-
Heating				
Total Heating Capacity (MBtu):	-	-	NA	-
Size (kBtu):	-	-	NA	-
Heating Power (kW):	-	-	NA	-
Efficiency (COP):	-	-	NA	-
HVAC System Fans				
Supply Fans	**Detail**			
Power (BHP):	1.54	1.34	NA	-
Mtr Eff:	High	High	NA	-
Fan Flow (cfm):	7,200	7,200	NA	-
OSA (%):	Auto	Auto	Auto	-
Return Fans	**Detail**			
Return Type:	None	None	None	-
Power (BHP):	-	-	-	-
Mtr Eff:	-	-	-	-
Fan Flow (cfm):	-	-	-	-
HVAC System Fan Schedules				
Operate Fans	**Detail**			
Hours before open:	-	-	-	-
Hours before close:	-	-	-	-
Cycle fan at night (Y/N):	Y	Y	Y	-
Mode (Con/Int):	Con.	Con.	Con.	-
HVAC Zone Heating, Vent and Economizers				
Zone Heat Sources & Capacities / Delta T	**Detail**			
Baseboards:	Electric	Electric	Electric	-
Power (kW):	0.0	0.0	0.0	-
Economizer (s)	**Detail**			
Type:	DB Temp	DB Temp	DB Temp	-
High Limit (˚F):	55	55	55	-
Compressor:	Run with	Run with	Run with	-

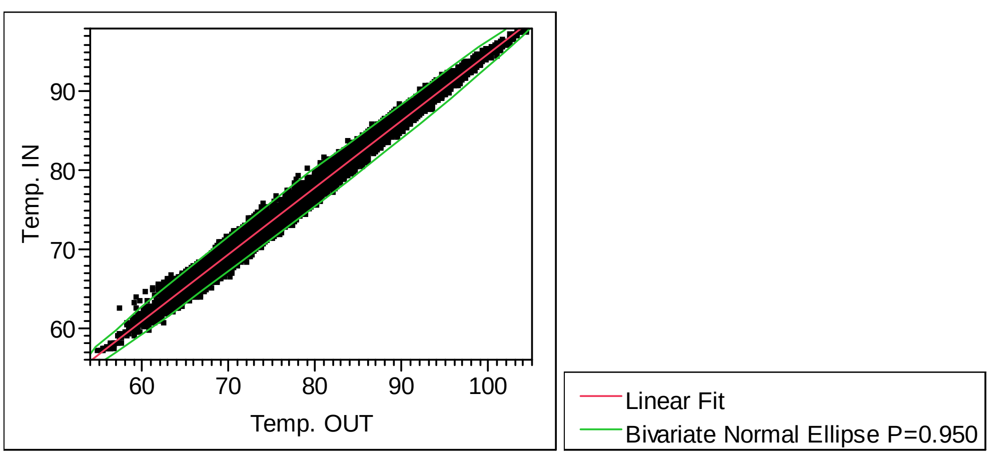
Table B.5.2 - Ground-source HP equipment

Water Loop Properties	
Pump Configuration:	Single Loop Pump (s) Only
Loop Flow:	Variable
Operation:	Standby

Loop Temp		
Min (˚F):	30	(Default)
Max (˚F):	100	(Default)
GHX Type:	Vertical Well Field	
Configuration:	2x4	
Number of Identical Well Field:	15	
Depth (ft):	500	
Spacing (ft):	20.0	
Borehole Diameter (in):	6.0	
Pipe Material:	Polyethylene	
Pipe Size (in):	1-1/4	
Rating:	SDR 11	
U-Tupe Leg Separation (in):	2.7	
GHX Pipe Head (ft):	Auto	
Loop Pump (s)		
Number:	3	
Head (ft.):	36.0	
Flow (gpm):	300	
Motor Eff:	High	
Control:	VSD	
Soil Thermal Properties & History		
Ground Temp (˚F):	Calculate via weather data	(Default)
Average (˚F):	5.0	(Default)
Years of Previous Operation (yrs):	0	(Default)
Ground:	Amphibolite	(Default)
Grout:	20% Bentonite -40% Quartzite	(Default)
Fluid Properties		
Fluid:	Propylene Glycol	
Anti-freeze Concen:	20.00%	(Default)

Table B.5.3 - Regression analysis of fluid temperature flowing to and from the underground loop (Feb 2007-Jan 2008) from JMP7 program

Bivariate Fit of Temp. IN By Temp. OUT



Linear Fit					
Temp. IN = 10.052837 + 0.8479862*Temp. OUT					
Summary of Fit					
RSquare	0.989594				
RSquare Adj	0.989592				
Root Mean Square Error	0.983137				
Mean of Response	77.3597				
Observations (or Sum Wgts)	7411				
Lack of Fit					
Source	**DF**	**Sum of Squares**	**Mean Square**	**F Ratio**	
Lack Of Fit	543	787.7541	1.45074	1.5629	
Pure Error	6866	6373.4760	0.92827	Prob > F	
Total Error	7409	7161.2301		<.0001	
				Max RSq	
				0.9907	
Analysis of Variance					
Source	**DF**	**Sum of Squares**	**Mean Square**	**F Ratio**	
Model	1	680995.99	680996	704557.6	
Error	7409	7161.23	1	Prob > F	
Parameter Estimates					
Term	**Estimate**	**Std Error**	**t Ratio**	**Prob>|t|**	
Intercept	10.052837	0.080996	124.12	0.0000	
Temp. OUT	0.8479862	0.00101	839.38	0.0000	
Correlation					
Variable	**Mean**	**Std Dev**	**Correlation**	**Signif. Prob**	**Number**
Temp. OUT	79.3726	11.3051	0.994783	0.0000	7411
Temp. IN	77.3597	9.636842			

Table B.6 - Utility rates

Table B.6-1 - Utility rates for eQUEST Inputs

Electric Utility Charges			**Fuel Utility Charges**		
Rate Name:	Custom Electric Rate		Rate Name:	Custom Gas Rate	
Type:	Block Charges		Type:	Block Charges	
Block Type:	Cumulative Blocks		Block Type:	Cumulative Blocks	
Entire Year			**Entire Year**		
Customer Charge ($/Month):	18.00		Customer Charge ($/Month):	14.50	
Uniform Charge			Uniform Charge		
Demand ($/kW):	0.00		Demand ($/Therm/hr):	0.00	
Energy ($/kWh):	0.00		Energy ($/Therm):	0.00	
Energy Blocks	**Blk Size**	**$/kWh**	**Energy Blocks**	**Blk Size**	**$/ Therm**
1. kWh Block	673	0.055320	1. Therm Block	96	0.206388
2. kWh Block	900	0.044170	2. Therm Block	513	0.128389
3. kWh Block	2,500	0.044170	3. Therm Block	999,999	0.109365
4. kWh Block	50,000	0.043630	- select another -		
5. kWh Block	999,999	0.043400			
Demand Blocks	**Blk Size**	**$/kW**	**Demand Blocks**	**Blk Size**	**$/ Therm/hr**
1. kW Block	99,999	3.420	1. Therm Block	99,999	0.0000

Table B.8.2 - Electricity Cost Estimation for eQUEST inputs

Customer Charge:	**$/Month**		**Uniform Charge:**	**$/kW**	**$kWh**
	$18.00			0.00	0.00
Energy Blocks	**Block Size (kWh)**	**Energy Price ($/kWh)**	**Consumption Tax ($kWh)**	**Town Tax ($)**	**Total ($/kWh)**
1. kWh Block	673	0.042690	0.001480	0.011150	0.055320
2. kWh Block	900	0.042690	0.001480		0.044170
3. kWh Block	2,500	0.042690	0.001480		0.044170
4. kWh Block	50,000	0.042690	0.009400		0.043630
5. kWh Block	999,999	0.042690	0.000710		0.043400
Demand Blocks	**Block Size (kW)**	**Energy Price ($/kW)**			**Total ($/kW)**
1. kWh Block	999,99	3.420			3.420
Electricity Cost (Actual)			**Electricity (Estimation)**		
03-06			**03-06**		
Energy (kWh):	91,680	$ 3,913.82	Energy (kWh):	91,680	$ 3,913.82
Demand (kW):	308.16	$ 1,053.91	Demand (kW):	308.16	$ 1,053.90
Customer Charges ($):		$ 18.00	Customer Charges ($):		$ 18.00
Consumption Tax ($):		$ 77.95	Consumption Tax ($):		$ 77.94
Town Tax ($):		$ 7.50	Town Tax ($):		$ 7.50
Current Charge ($):		$ 5,071.18	Current Charge ($):		$ 5,071.17

Table B.6.3 - Virginia Tech Electric's Tariff Structure[a] (Virginia Tech Electric Service 2008)

Type of Service	Minimum Monthly Service Charge	First 900 kWh Energy Charge per kWh	Over 900 kWh Energy Charge per kWh	Demand Charge per kW	Reactive Demand Charge per kVAR
Residential	$8.50	$0.067550	$0.067550	-	-
Sanctuary	$8.50	$0.069770	$0.069710	-	-
Small Gen	$11.50	$0.066570	$0.066570	-	-
Medium Gen	$18.50	$0.051000	$0.051000	$5.45	-
Large Gen	$18.92	$0.032770	$0.032770	$12.60	$0.75
a. The example tariff structure above is effective since July 1st, 2007. This structure is used as a reference for the 2006 electricity cost estimation.					

Table B.6.4 - State of Virginia's Electric Utility Consumption Tax (Commonwealth of Virginia 2008)

Monthly kWh Usage	State Consumption Tax Rate	Special Regulation Tax Rate	Local Consumption Tax Rate	Total Rate
0 to 2,500 kWh	$0.00102 per kWh	$0.00008 per kWh	$0.00038 per kWh	$0.00148 per kWh
2,501 to 50,000 kWh	$0.00065 per kWh	$0.00005 per kWh	$0.00024 per kWh	$0.00094 per kWh
Over 50,000 kWh	$0.00050 per kWh	$0.00003 per kWh	$0.00018 per kWh	$0.00071 per kWh

Table B.6.5 - Town of Blacksburg Electric Utility Consumers' Tax (Town of Blacksburg 2008)

Category	Delivered	Maximum
Residential	0.01135/kWh	maximum $2.25 per month
Commercial	0.01115/KWh	maximum $7.50 per month
Industrial	0.01200/kWh	maximum $7.50 per month

Table B.6.6 - ATMOS's Gas Tariff Structure for Small Commercial and Industrial Gas Service (Blose 2005)

Charge	Detail
Customer Charge:	A monthly customer charge of $14.50 is payable regardless of the usage of gas.

Monthly Rate:	All Consumption, Per Ccf $.1121
Activation Charge:	When a customer applies to initiate service, a charge of $40.00 will be assessed to cover the cost of activating service.

Table B.6.7 - State of Virginia’s Natural Gas Consumption Tax (Commonwealth of Virginia 2008)

Monthly CCF Usage	**State Consumption Tax Rate**	**Special Regulation Tax Rate**	**Local Consumption Tax Rate**	**Total Rate**
Not in excess of 500 CCF	$0.0135 per CCF	$0.004 per CCF	$0.002 per CCF	$0.0195 per CCF

Table B.6.8 - Town of Blacksburg Natural Gas Consumers’ Tax (Town of Blacksburg 2008)

Category	**Delivered**	**Maximum**
Residential	0.1891/CCF	maximum $2.25 per month
Commercial	0.07955/CCF	maximum $7.50 per month
Industrial	0.07955/CCF	maximum $7.50 per month

Table B.7 - Building Life-cycle cost analysis parameters

General LCC Data (this analysis)		
DOE/FEMP Fiscal Year:	2006	
Marginal Federal Income Tax Rate:	34.00%	
Marginal State Income Tax Rate:	0.00%	
Discount Rate Specification:	Real, After Tax	
General Inflation Rate:	1.75%	
Nominal, After Tax Discount Rate:	10.00%	
Number of Analysis Years:	25	
DOE Fuel Price Escalation Region:	3 (1-5, see listing below*; 6=User Defined)	
Analysis Sector:	2 (1=Residential; 2=Commercial; 3=Industrial)	
Second Fuel Type:	1 (0=None; 1=N.Gas; 2=LPG; 3=Dist Oil; 4=Resid Oil; 5=Coal)	
Federal Discount Rates		
	real	nominal
DOE/FEMP:	3.00 %	4.80 %
OMB 3-year:	1.60 %	3.40 %
5-year:	2.10 %	3.90 %
7-year:	2.40 %	4.20 %
10-year:	2.80 %	4.60 %
30-year:	3.50 %	5.30 %
Convert		
Nominal-to-Real:	4.80 % Nominal Discount Rate	
	1.75 % General Inflation Rate	
	3.00 % Real Discount Rate	
Real-to-Nominal:	3.00 % Real Discount Rate	
	1.75 % General Inflation Rate	
	4.80 % Nominal Discount Rate	
Computed Values	34.00 % Combined Marginal Federal & State Tax Rate	
	15.15 % Real, Pre-Tax Discount Rate	
	10.11 % Real, After Tax Discount Rate	
Uniform Electric Price Escalation Rate		(real: for DOE escalation rates, which vary by year, leave this entry empty)

Uniform Natural Gas Price Escalation Rate		(real: for DOE escalation rates, which vary by year, leave this entry empty)
*** DOE Fuel Price Escalation Regions:**		
1 Northeast (CT, MA, ME, NH, NJ, NY, PA, RI, VT) 2 Midwest (IA, IL, IN, KS, MI, MN, MO, ND, NE, OH, SD, WI) 3 South (AL, AR, DC, DE, FL, GA, KY, LA, MD, MS, NC, OK, SC, TN, TX, VA, WV) 4 West (AK, AZ, CA, CO, HI, ID, MT, NM, NV, OR, UT, WA, WY) 5 U.S. Average		

Appendix C

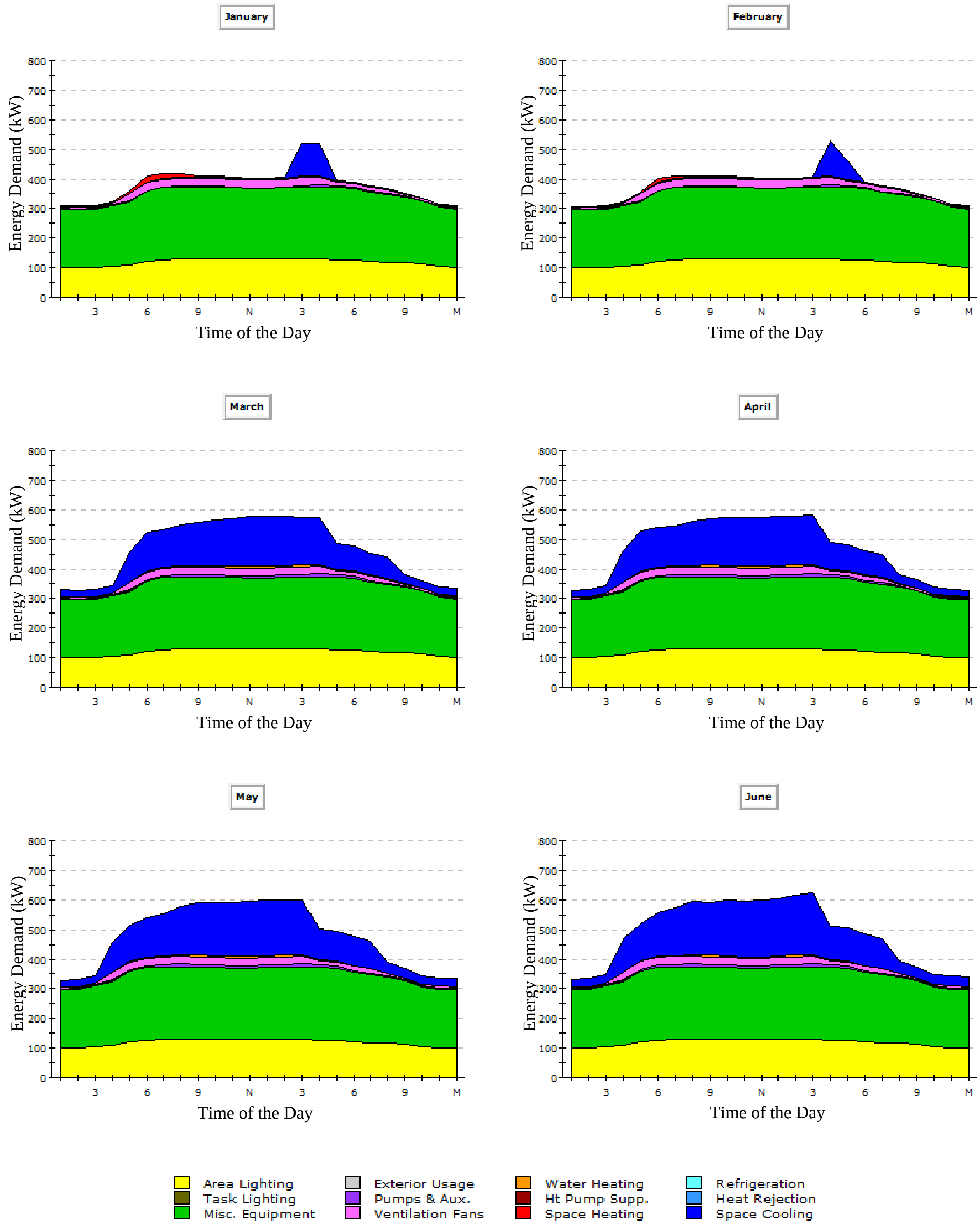


Figure C.1 - Monthly electric peak day load profiles (January-June, 2007)

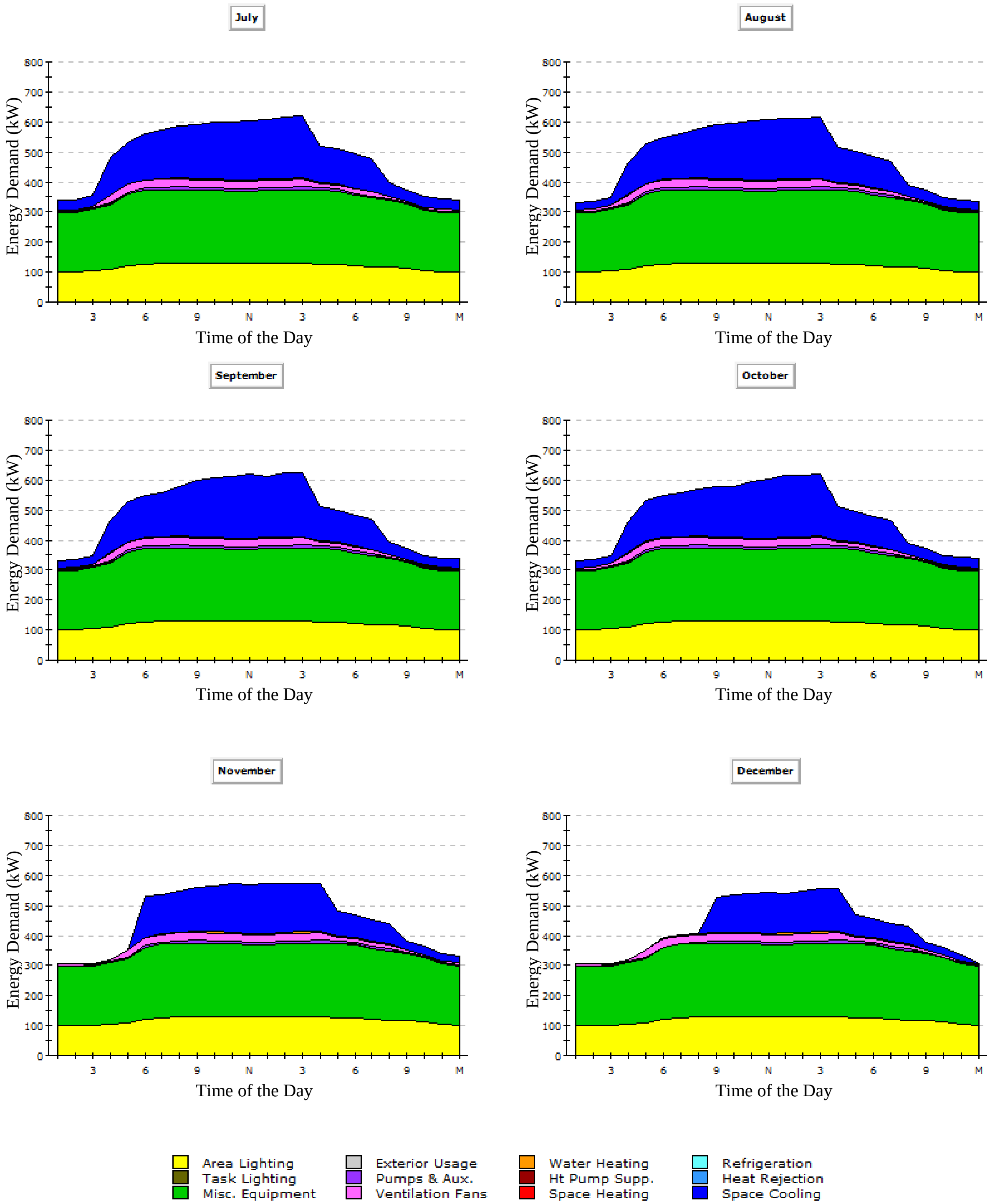


Figure C.1 (continue) - Monthly electric peak day load profiles (July-December, 2007)

Appendix D

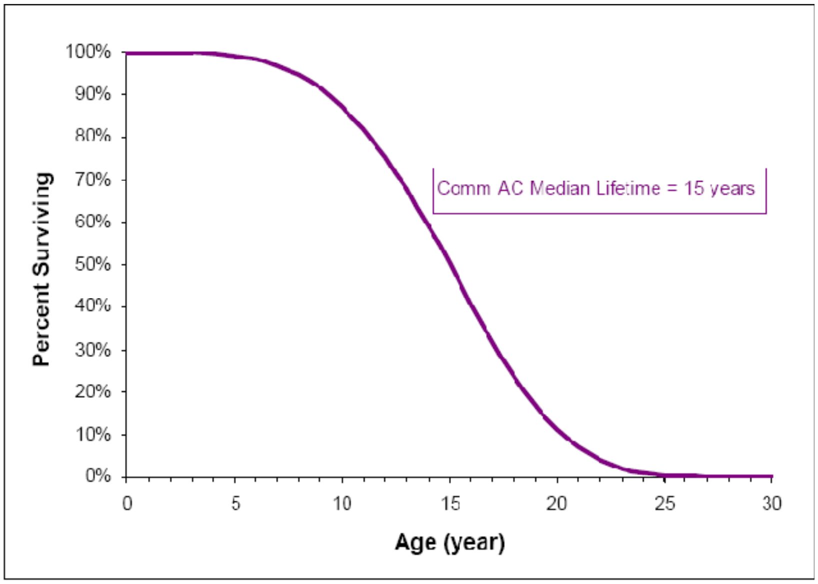


Figure D.1 - Equipment lifetime from (Lawrence Berkeley National Laboratory 2003).

In the figure, Median age of 15 years is based on 1999 ASHRAE HVAC Applications Handbook. The Survival function is based on Weibull probability distribution, in which 5% of population retired by 8th year and 90% of population retired by 21st year

Table 3 Estimates of Service Lives of Various System Components[a]

Equipment Item	Median Years	Equipment Item	Median Years	Equipment Item	Median Years
Air conditioners		Air terminals		Air-cooled condensers	20
Window unit	10	Diffusers, grilles, and registers	27	Evaporative condensers	20
Residential single or split package	15	Induction and fan-coil units	20	Insulation	
Commercial through-the-wall	15	VAV and double-duct boxes	20	Molded	20
Water-cooled package	15	Air washers	17	Blanket	24
Heat pumps		Ductwork	30	Pumps	
Residential air-to-air	15[b]	Dampers	20	Base-mounted	20
Commercial air-to-air	15	Fans		Pipe-mounted	10
Commercial water-to-air	19	Centrifugal	25	Sump and well	10
Roof-top air conditioners		Axial	20	Condensate	15
Single-zone	15	Propeller	15	Reciprocating engines	20
Multizone	15	Ventilating roof-mounted	20	Steam turbines	30
Boilers, hot water (steam)		Coils		Electric motors	18
Steel water-tube	24 (30)	DX, water, or steam	20	Motor starters	17
Steel fire-tube	25 (25)	Electric	15	Electric transformers	30
Cast iron	35 (30)	Heat exchangers		Controls	
Electric	15	Shell-and-tube	24	Pneumatic	20
Burners	21	Reciprocating compressors	20	Electric	16
Furnaces		Package chillers		Electronic	15
Gas- or oil-fired	18	Reciprocating	20	Valve actuators	
Unit heaters		Centrifugal	23	Hydraulic	15
Gas or electric	13	Absorption	23	Pneumatic	20
Hot water or steam	20	Cooling towers		Self-contained	10
Radiant heaters		Galvanized metal	20		
Electric	10	Wood	20		
Hot water or steam	25	Ceramic	34		

Notes: 1. ASHRAE makes no claims as to the *statistical* validity of any of the data presented in this table.
2. Table lists base values that should be adjusted for local conditions (see the section on Service Life).
3. For updated information on heat pump life, see Lovvorn and Hiller (2002).

Source: Data obtained from a survey of the United States by ASHRAE Technical Committee TC 1.8 (Akalin 1978).
[a] See Lovvorn and Hiller (1985) and Easton Consultants (1986) for further information.
[b] Data updated by TC 1.8 in 1986.

Figure D.2 - Estimates of service lives of various system components from Table 3, page 36.3 (ASHRAE 2003).

Table D.1 - Replacement and maintenance costs estimation for life-cycle cost analysis

Case	Description	Replacement (2006 $)		Maintenance[a,b] (2006 $/Year)
Baseline	GSHP	System replacement excluding underground loop (year 15th)	1,032,999.00	17,939.77
Alternative I	VAV w/ HW	System replacement (year 15th)	1,433,879.00	47,814.63
Alternative II	ASHP	System replacement (year 15th)	1,096,492.00	18,831.19

a. The maintenance cost per square foot can be calculated using the ASHRAE maintenance costs estimation procedure (ASHRAE 2003) as follow:

$$Maintenace\ Costs\ (\$/ft^2) = 0.3338 + 0.0018n + h + c + d \tag{D.1}$$

Where:
0.3338 = Base system maintenance costs
$0.0018n$ = Age Adjustment for year n
h = Heating system adjustment factor
c = Cooling system adjustment factor
d = Distribution system adjustment factor
Note: h, c, and d values are provided in Figure D.2

b. The 2006 costs can be calculated using historical cost index procedure (Means, 2006 Mechnical Cost Data 2006) as follow:

$$Cost\ in\ Year\ 2006 = \frac{Index\ for\ Year\ 2006}{Index\ for\ Price\ Year} \times \frac{City\ Index}{National\ Index} \times Cost\ in\ Price\ Year \tag{D.2}$$

Where:
$Index\ for\ Year\ 2006$ = 100
$Index\ for\ Year\ 1983$ = 51.4
$City\ Index$ = 74.9
$National\ Index$ = 100

Table 4 Annual HVAC Maintenance Cost Adjustment Factors (in dollars per square foot, 1983 U.S. dollars)

Age Adjustment	+0.0018n
Heating Equipment (h)	
Water tube boiler	+0.0077
Cast iron boiler	+0.0094
Electric boiler	−0.0267
Heat pump	−0.0969
Electric resistance	−0.133
Cooling Equipment (c)	
Reciprocating chiller	−0.04
Absorption chiller (single stage)*	+0.1925
Water-source heat pump	−0.0472
Distribution System (d)	
Single zone	+0.0829
Multizone	−0.0466
Dual duct	−0.0029
Constant volume	+0.0881
Two-pipe fan coil	−0.0277
Four-pipe fan coil	+0.0580
Induction	+0.0682

n = age, years
*These results pertain to buildings with older, single-stage absorption chillers. The data from the survey are not sufficient to draw inferences about the costs of HVAC maintenance in buildings equipped with new absorption chillers.

Figure D.3 - Annual HVAC Maintenance Cost Adjustment Factors (in dollars per square foot, 1983 U.S. dollars) from Table 4, page 36.6 (ASHRAE 2003)

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