Feasibility Study of Ground Coupled Heat Pump Systems

For Small Office Building Types in Phoenix, Arizona

by

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ABSTRACT

The need for alternative energy efficient building heating and cooling technologies has given rise to the development and widespread use of Ground Coupled Heat Pump (GCHP) systems. This dissertation looks at the feasibility of using GCHP systems as a viable economic alternative to traditional air source cooling systems (ASHP) for conditioning buildings in the hot, semi-arid climate of Phoenix, Arizona.

Despite high initial costs, GCHPs are gaining a foothold in northern climates where heating dominates, in large part due to government incentives. However, due to issues associated with low ground heat exchanger (GHE) efficiency and thermally-induced soil deformations, GCHPs are typically not considered a viable option in hot climates with deep groundwater and low permeability soil. To evaluate the energy performance and technical feasibility of GCHPs in Phoenix, the DOE 5,500 sq.ft small office, commercial building prototype was simulated in EnergyPlus to determine the cooling and heating loads. Next, a commercial software program, Ground Loop Design (GLD), was used to design and simulate the annual energy performance of both vertical (V-GCHPs) and horizontal GCHPs (H-GCHPs). Life cycle costs (LCC) were evaluated using realistic market costs both under dry, as well as fully saturated soil conditions (meant as an upper performance limit achievable by ground modification techniques). This analysis included performing several sensitivity analyses and also investigating the effect of financial rebates.

The range of annual energy savings from the GCHP system for space cooling and heating was around 38-40% compared to ASHPs for dry soil. Saturated soil condition significantly affects the length of the GHE. For V-GCHPs, there was about 26% decrease in the length of GHE, thereby reducing the initial cost by 18-19% and decreasing the payback period by 24-25%. Likewise, for H-GCHPs, the length of GHE was reduced by 25% resulting in 22% and 39-42 % reduction in the initial cost and payback period respectively. With federal incentives, H-GCHPs under saturated soil conditions have the least LCC and a good payback periods of 2.3-4.7 years. V-GCHPs systems were been found to have payback periods of over 25 years, making them unfeasible for Phoenix, AZ, for the type of building investigated.

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CHAPTER 1

INTRODUCTION

1.1 Energy Background:

According to the U.S. Department of Energy (DOE, 2010), buildings are the largest single sector of total U.S. energy consumption. Figure 1.1 illustrates that the building sector consumed around 41% percent of U.S. primary energy in 2010. Out of 29 quads consumed by the building sector, residential buildings consumed 54%, and commercial buildings consumed around 46% of the building sector energy. The building sector uses about one third more energy than either the industrial or the transportation sectors. The main energy source used by the U.S. buildings sector comes from fossil fuels, which accounted for 75% of the total, followed by nuclear generation at 16%, and 9% from the renewables. Figure 1.2 shows that the total primary energy consumption for US is expected to increase more than 45 quads by 2035, a 52% increase over 2010 levels. The use of coal is projected to increase by 11% over the same period, while natural gas consumption will increase by 17%. Use of non-hydroelectric renewable resources, including wind, solar, and biofuels, is expected to increase by 109%.

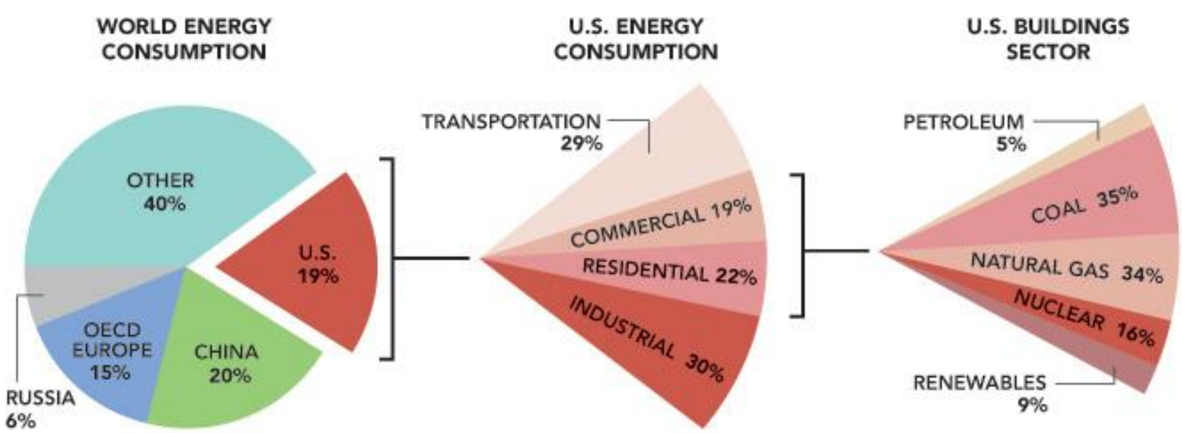


Figure 1.1: U.S. Energy Consumption by Sector (BEDB, U.S. DOE, 2010)

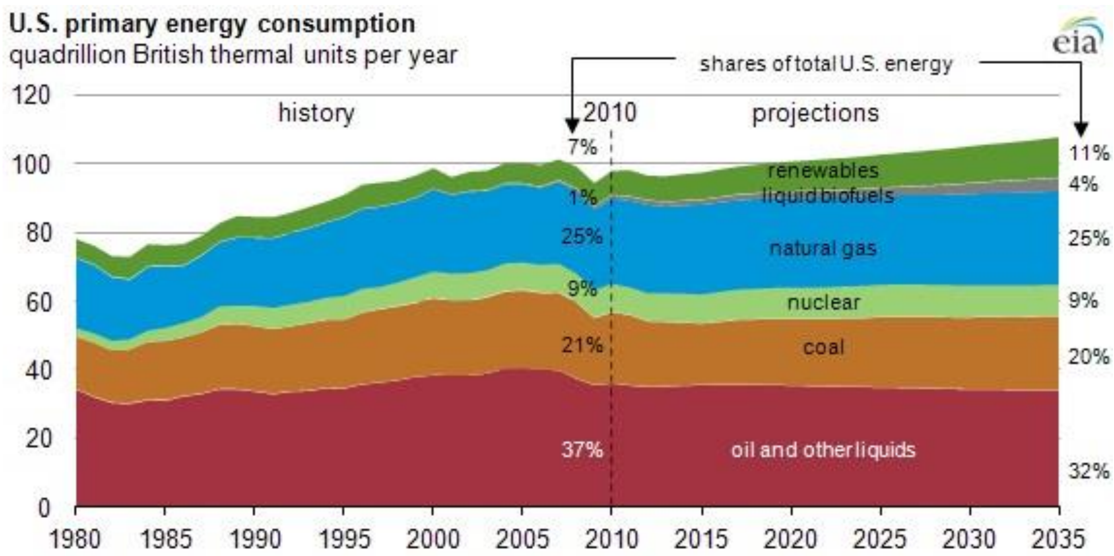


Figure 1.2: U.S. Primary Energy Consumption (EIA, 2010)

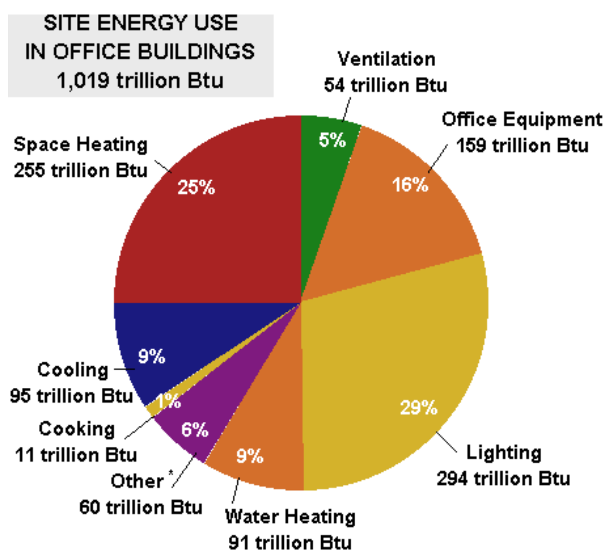


Figure 1.3: Site Energy Use in Office Buildings (BEDB, U.S. DOE, 2010)

From Figure 1.3, 48% of energy used in commercial buildings is for "thermal loads," which constitutes approximately 25% of all energy used in the nation is for space heating, cooling and water heating. Among all building systems, HVAC systems consume about 39% of operating energy in commercial buildings. . HVAC systems are also the main source of green-house gas (GHG) emissions in buildings, being responsible for about 40.6% of the total (DOE, 2010). According to the U.S. Environmental Protection Agency (EPA), GSHP systems have potential to reduce energy consumption and corresponding emissions by up to 44% compared to ASHPs, and up to 72% when compared to other conventional HVAC systems (DOE 2005). GSHP can also improve the indoor-air quality and humidity control in the conditioned area.

Based upon the above data, HVAC Ground source heat pumps can play a very important role in reducing the use of fossil fuels like petroleum, coal and gas for on-site heating and cooling applications. 70% of energy used in a GSHP is renewable energy from the ground (GHPC 2003). Furthermore, technologies like solar PV, wind, and hydro usually do very little to address thermal energy, which makes up roughly one third of our nation’s energy use.

1.2 Status of Ground Source Heat Pump Industry in the U.S.

In the United States, over the last several decades GSHP systems have improved gradually. GSHPs have now achieved a small but growing share in building heating, cooling, and water heating equipment markets. GSHP installations are steadily increasing over past 10 years in United States with an annual growth rate of about 12%. Here in the US, GSHPs are mainly designed based on the peak cooling loads. Therefore, excluding the northern parts of US, GSHPs are typically over-sized for heating loads and are estimated to average only 1,000 full-load heating hours per year. Most of this growth has occurred in the United States and Europe, followed by other countries such as Japan and Turkey. Table 1.1 lists the countries with the highest use of GSHPs. The heat pumps are rated in tonnage in the US. GSHPs in US are found predominantly in the mid-western and eastern states from North Dakota to Florida. As indicated in Figure 1.4 annual GSHP shipments in the US exceeded 1,00,000 units in 2008 and 2009, equaling more than 4,00,000 tons of capacity per year.

Table 1.1 Leading Countries Using GSHP technology (Lund et al., 2004)

Country Installed	MWt	GWh/yr	Number installed
Austria	275	370	23,000
Canada	435	600	36,000
Germany	640	930	46,400
Sweden	2,300	9,200	230,000
Switzerland	525	780	30,000
USA	6,300	6,300	600,000

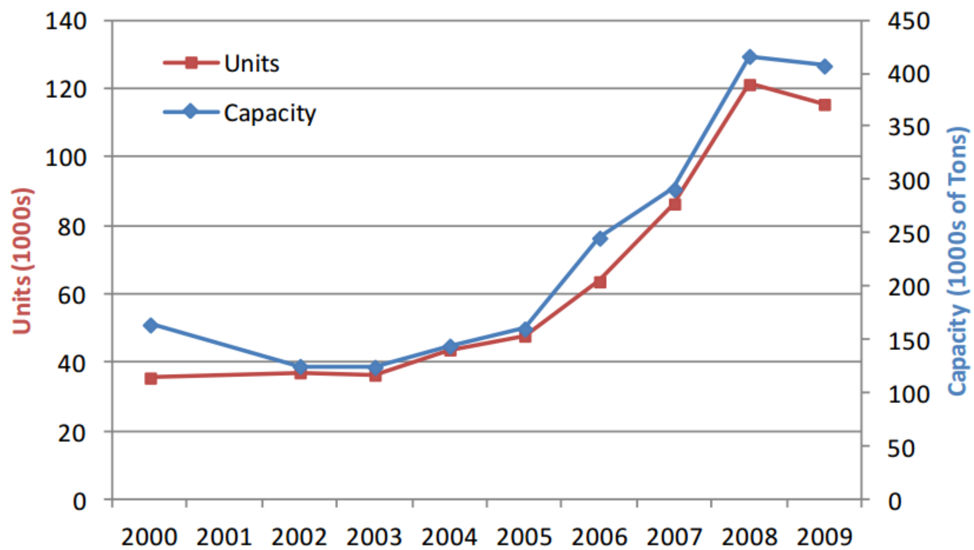


Figure 1.4: Annual GSHP Shipments (EIA, 2010)

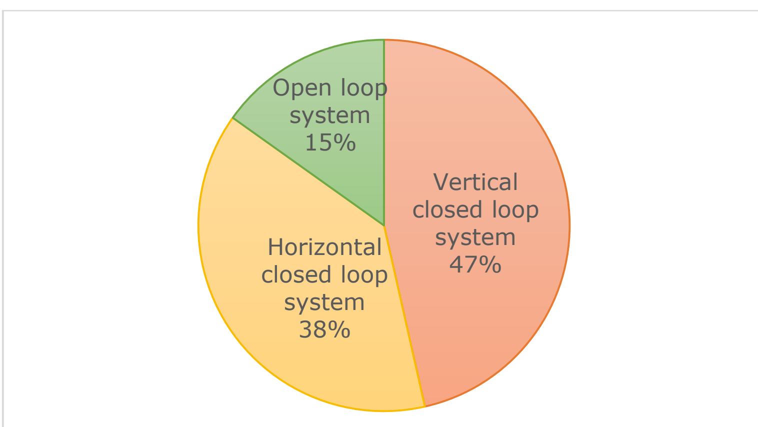


Figure 1.5: GSHP systems installed in the US (Lund et al., 2004)

As indicated in Figure 1.5, approximately 8,000 GSHP units are installed annually in US out of which 47% are vertical closed loop system, 38% are horizontal closed loops systems, and 15% are open loop systems.

Federal Benefits

The U.S. GSHPs industry has seen strong growth when compared to the broader economy. On 3rd October 2008, the federal Economic Stimulus Bill became law, providing a 10 percent investment tax credit to businesses that install GSHP systems. The bill extends these credits through 2016 and allows them to be used to offset the alternative minimum tax (AMT). The GSHP systems placed in service by businesses after October 3, 2008- will also get the benefit of a 5-year depreciation period. Moreover, the bill provides taxpayers a tax credit of 30 percent of the cost of a GSHP system for residential buildings.

Federal incentives for commercial sectors GSHP systems include:

a. Federal Income Tax Credit (ClimateMaster, 2010)

- 10% of total GHP system cost, with no cap
- Can be used to off-set AMT tax
- Can be used in combination with subsidized financing
- Can be used in more than one year

b. Accelerated Depreciation

- 5-year MACR depreciation for entire GHP system
- Eligible for bonus depreciation 50% write-off in first year

c. Eligibility

- Building must be located in the United States
- Original use begins with taxpayer
- Must be placed in service before 2017
- Can be used by regulated utilities

- Must be claimed by the owner of the property (effects non-taxable)

1.3 Overview of Ground Source Heat Pump Technologies

GSHP systems, also referred to as geothermal heat pump systems, earth energy systems, and GeoExchange systems. These terms describe a heat pump systems that uses the earth, ground water, or surface water as a heat source and/or sink. These systems extract heat from a heat source and transfer it to a heat sink using a mechanical system. This mechanical system cools a space by removing heat from it in summers and heats the space by supplying heat in the winter. Typically, a GSHP system has three major components: i. a heat pump; ii. a connection to the earth (borehole heat exchanger); and iii. an interior heating and cooling distribution system.

The concept behind the heat pump has been applied for many years in our day to day life. The majority of heat pumps work on a vapor compression cycle. The heat pump is a complete thermodynamic system whereby a liquid and/or gas medium is pumped through an assembly where it changes phases as a result of altering pressure. A GSHP with a closed-loop GHE offers a coefficient of performance (COP) between 3 and 5. Even though initial installation cost is high for heat pumps, the system provides a more energy efficient way to control temperatures and reuse existing heat energy.
GSHPs can be classified as ground coupled heat pump systems, ground water heat pump systems, surface water heat pump systems, and standing column well systems. Typically, GCHP consist of water-to-air or water-to-water heat pumps linked to a network of closed ground loop heat exchangers. Ground coupled heat pump systems use the stable earth temperature as a heat sink or source whereas groundwater heat pump systems use water from the ground or reservoir to extract or dump heat.

1.3.1 Ground Coupled Heat Pump Systems (GCHPs)

GCHPs uses the renewable storage capacity of the ground as a heat source or sink to provide space heating, cooling, and domestic hot water. GCHPs are subset of GSHPs and are also known as a closed-loop ground-source heat pumps. A GCHP consists of a reversible vapor compression cycle that is connected to a closed GHE buried in ground. GCHPs can be further classified according to the ground heat exchanger design.

1.3.2 Vertical Ground Loop Heat Exchanger Systems

Vertical ground loop heat exchanger or vertical ground coupled heat pump systems (V-GCHPs) consists of a borehole, or a group of boreholes, into which a loop of two straight legs with a thermally fused 'U' bend at the bottom is inserted. Since the ground temperature is generally much closer to room conditions than the ambient air temperatures over the whole year, GSHP systems are more efficient than conventional HVAC systems. As a result, worldwide applications of GSHP have been grew at an annual rate of 10% from 1995 to 2005 (Rybach, 2005). The term "Ground loop heat exchanger" used in this thesis refers to the vertical cylindrical hole with the U tube pipe, grout and the surrounding rock or soil (see Figure 1.7). High-density polyethylene (HDPE) pipes are mostly used due to HDPEs favorable physical and chemical properties. Pipe diameter typically ranges from ¾" to 1 ½" (Kavanaugh and Rafferty, 1997).

The U loop tube is secured by a borehole filling material called grout which has high thermal conductivity, enhancing the heat transfer between the U loop tube and the surrounding ground. The grout conductivity varies from .3 to .9 Btu/ft-hr-°F based on the grouting material (Murugappan, 2002). V-GCHPs are widely used as they are easy to install and requires less land area than H-GCHPs. Therefore, V-GCHPs are used mostly in commercial and institutional applications where the land area is restricted.

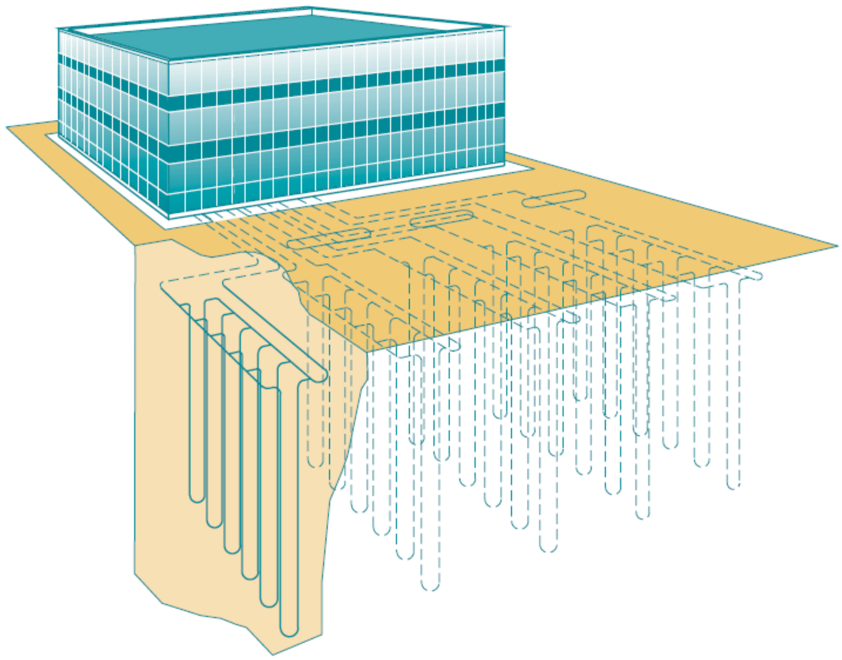
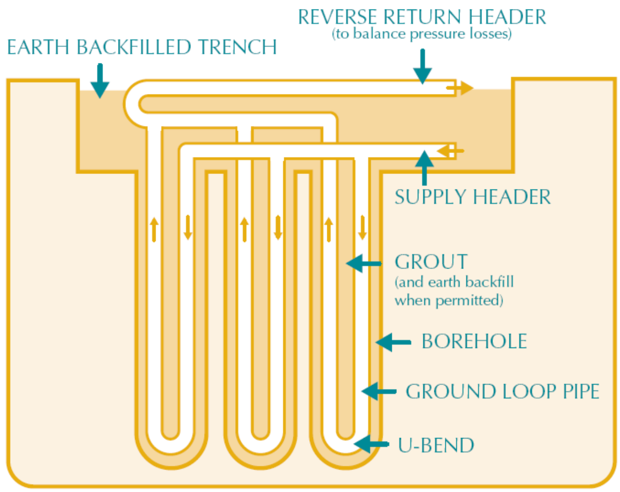
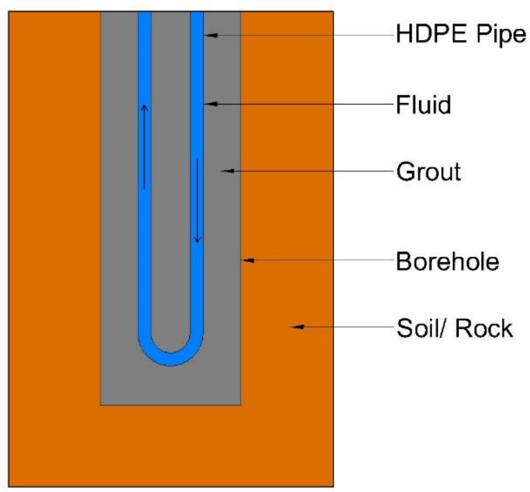


Figure 1.6: Vertical Ground Heat Exchanger System (RETscreen, 2005)

The other advantage of the V-GCHPs compared to H-GCHPs is that they are most efficient GCHPs configuration since they are in contact with soil that varies little in temperature and thermal properties, requiring least amount of pipe and pumping energy. The main disadvantage of V-GCHPs is their high initial installation cost, to some extent because of limited availability of appropriate drilling equipment and installation personnel. There may be a number of GHE or boreholes in a single loop system and the length of the GHE usually varies from 50 feet to 600 feet (Kavanaugh and Rafferty, 1997). Boreholes can be connected in parallel or series to form a borefield. The number and depth of the boreholes usually depends on the building loads and the thermal properties of the soil. The accurate sizing of the ground loop heat exchanger is very critical as the initial cost of installation and performance of V-GCHPS greatly depends on the GHE. Accurate sizing will help to reduce the initial cost and make GCHP more practical and sustainable.



(a)



(b)

Figure 1.7: (a) Vertical Ground Heat Exchanger (RETscreen, 2005) (b) vertical cross section of a typical Ground Heat Exchanger

1.3.3 Horizontal Ground Coupled Heat Pump Systems

Horizontal ground loop heat exchangers or horizontal ground coupled heat pump systems (H-GCHPs) can be divided into three sub-groups: single pipe, multiple pipe and coiled pipe (slinky). In single pipe horizontal GCHPs, the pipe is buried in narrow trench which is about 5 feet deep. This type of horizontal GCHP requires the largest land area (Figure 1.8). However, in the case of restricted land area, the trench length is reduced and the pipe length is increased. However, these systems are less efficient due to thermal interference with the adjacent pipes. In multiple pipe systems, two or four pipes are placed in a single trench, thus reducing the required land area. Contractors either use deep narrow trenches or wide trenches with pipes separate by 12" to 24" (Kavanaugh and Rafferty, 1997). The slinky design is perhaps the best example a restricted land area system, requiring the least ground area compared to the other two sub-categories.

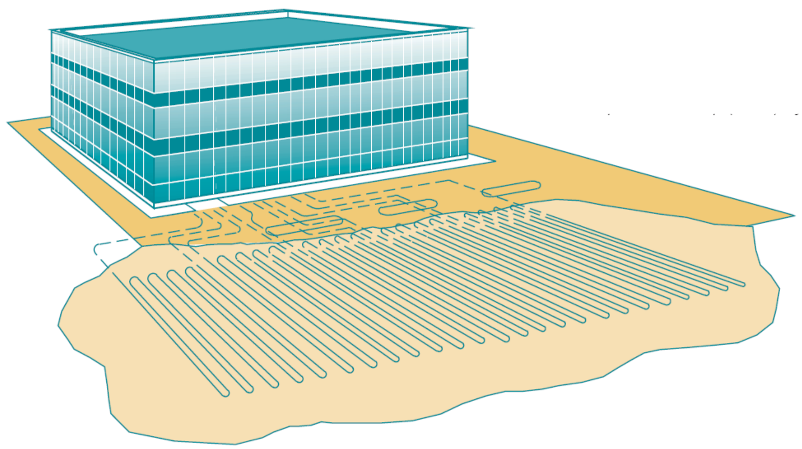
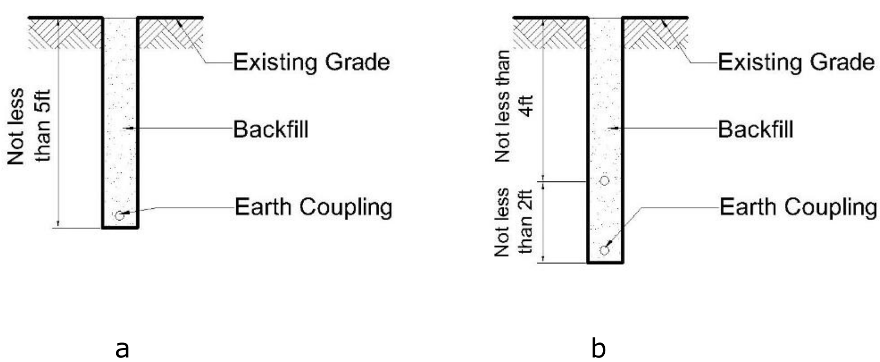
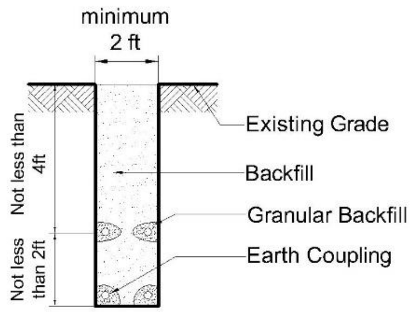


Figure 1.8: Horizontal Ground-Coupled Heat Pump System (RETscreen, 2005)





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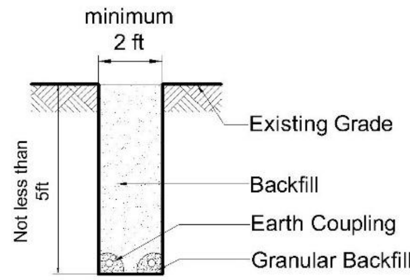


d

a. Single pipe -design guideline

b. Stacked two-pipe -design guideline (sand fill is required only if rocks larger than 5 cm across are present)

c. Stacked parallel four-pipe -design guideline

d. Parallel two-pipe -design guideline

Figure 1.9: Various configurations of horizontal GHE designed

The main advantage of horizontal H-GCHP systems is that they are less costly than V-GCHP systems. In residential applications, usually the required land for a H-GCHP system is available. Furthermore, trained equipment installers are more easily found for H-GCHPS. The major disadvantage of H-GCHPs is that there is greater variation in their performance (compared to V-GCHPs) due to fluctuations in ground temperature and thermal properties depending on the season, rainfall, and the burial depth. This results in slightly higher pumping energy requirement and lower system efficiency. Moreover, a H-GCHP system requires more ground area than a V-GCHP system.

1.3.4 Groundwater Heat Pump Systems (GWHPs)

A groundwater heat pump is a subset of GSHPs. GWHP systems were the most widely used GCHPs until the development of V- and H-GCHPs. GSHPs are, in contrast to V- and H-GCHPs, open loop systems that require a constant supply of groundwater as the heat transfer fluid (see Figure 1.10). Since V- and H-GCHPs requires low maintenance, many residential owners are more readily attracted to V- and H-GCHPs than GWHPs. In the commercial sector, however, GWHPs are more attractive as large quantities of water can be delivered from a relatively inexpensive well that requires much less ground area then even a V-GCHP. A properly designed groundwater loop and well developed water well requires no more maintenance than a conventional air and water central HVAC system.

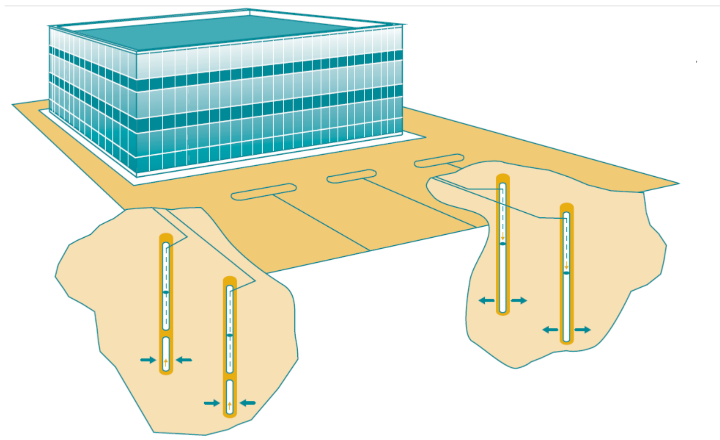


Figure 1.10: Ground-Water Heat Pump System (RETscreen, 2005)

There is a great variety of system configurations are possible with GWHPs. The most common GWHP system configuration used in a building is a central water to water heat exchanger between the groundwater and closed loop that is connected to water to air heat pumps. In smaller buildings, the water can be directly circulated through the heat pump. A third possibility is to circulate ground water through a central chiller or a heat pump to heat or cool the building with a pipe distribution system.

Advantages of GWHPs are that they have a lower installation cost compared to a GCHP systems. The water well required for a GWHP system is very compact, thus requiring less area than a GCHP, water well contractors are widely available, and the technology is very well developed and has been used for many years. The major drawbacks of a GWHP system is that local environmental regulations may preclude use of or injection of groundwater, it depends upon the availability of ground water, fouling precautions need to be taken if the well water is of poor quality, and the pumping energy may be excessive if the pump is over sized, poorly controlled or remote to the building.

1.3.5 Surface Water Heat Pump Systems (SWHPs)

A surface water heat pump system is another sub-set of ground source heat pumps. Surface water heat pump (SWHP) systems can be a closed loop or open loop systems similar to GWHPs. SWHPs can be water-to-air or water-to-water systems. SWHPs are usually installed in a building linked to a piping network placed in a lake, river or other open body of water. Figure 1.11 illustrates a typical SWHP closed loop system.

Figure 1.11: Surface Water Closed Loop (Mammoth, CES group)

In a SWHP system, a pump circulates an antifreeze-water mixture through the heat pump’s water to refrigerant coils, and the submerged piping loop transfers heat to or from the lake.

Fused high-density polyethylene tubes are highly recommended as they have excellent ultraviolet radiation resistance. Copper and polybutylene piping are also used, but polyvinyl chloride (PVC) should be avoided.

Advantages of closed-loop SWHPs are that they are less expensive compared to GCHP, having low pumping requirements, low maintenance, high reliability, and low operating cost. The major disadvantages are the possibility of coil damage in public lakes and wide temperature fluctuations with outdoor conditions if the lakes or ponds are small and shallow. This variation would affect the efficiency and capacity of the GWHPs.

1.4 Ground Source Heat Pumps Types

1.4.1 Passive and Forced Earth Coupled Duct System

Passive and forced earth coupled systems are generally self-installable where the designs can be obtained online. In this system, the air is forced from outside or recirculated from inside. The major advantages of this system are that they consume less energy, and are easy to install without a need of much training required.

The major disadvantage of passive and forced earth coupled systems includes the condensation on the inside surface of the underground pipes, which can lead to the growth of bacteria and fungus. Also, in this type of system there is no provision for mechanical refrigeration as this may create discomfort due to lack of adequate cool air during peak cooling loads in summer and fails to remove humidity to condition a space.

1.4.2 Water-To-Air Heat Pumps

Water – to – air heat pumps are the most common type of heat pumps used in buildings. In retrofit buildings, they can be installed where the systems air handling units are located or they can be placed in mechanical rooms, garage, attic etc. The water –to – refrigerant coil is linked to the outdoor water loop, and works as a condenser in cooling and as an evaporator in heating. The air –to refrigerant system is linked to the forced air system.

Two main precautions to be considered while installing this type of system are that the weight and size can vary as compared to the conventional system and, unlike standard air handling units, the air typically flows through the side and the out end of the unit.

1.4.3 Direct Expansion Ground Source Heat Pumps (DX-GSHPs)

Direct expansion ground source heat pump (DX-GSHP) systems are gaining popularity these days due to their high efficiency, easy installation and low maintenance. They work on similar principles to water- to air heat pumps but the DX GSHPs uses a buried copper piping network through which refrigerant, instead of water, is circulated. A refrigerant like R-410A is used which does not damage the ozone layer but has the potential to be a greenhouse gas which is much more potent than CO_2.

The major advantage of these systems is that less excavation is required to install the GHE. The systems are more efficient from a thermodynamic perspective, as one step in the heat exchange process is eliminated. They can be 30% more efficient than the GCHPs, achieving a COP of 4.5-5.0 for heating and a Seasonal Energy Efficiency Ratio (SEER) of up to 33 for cooling.

Unfortunately this type of system is not recommended for areas which have acidic soil properties as the acidic property can lead to the corrosion of tubes. Furthermore, this type of system uses more refrigerant than other systems, which increases the chance of refrigerant leakage.

1.4.4 Water-To-Water Heat Pumps

In water-to-water heat pumps, unlike a DX-GSHP system, the refrigerant system is inside the equipment and the heat is transferred to liquid such as water, glycol or brine. These systems are used for hydronic floor heating, domestic water heating, outdoor air preconditioning, hydronic heating and cooling, and refrigeration application.

1.4.5 Applications of Ground Source Heat Pumps

a. The ground coupled pool heaters use 75% less energy compared to conventional systems.

b. GSHPs can be efficiently used for heating domestic water.

c. GSHPs can be used for space heating and cooling. The biggest benefit of GSHPs is that they use 25-50% less electricity than conventional heating or cooling systems.

d. GSHPs can be used for process cooling and heating in factories with equipment such as plasma cutters, extrusion presses, etc.

e. The equipment can provide 100% fresh air and have significant energy saving compared with ASHPs or cooling towers. The equipment can be installed in buildings such as hospitals or hotels.

f. GSHPs can be used to keep livestock cool. This can increase the milk production, growth rates and birth rates.

1.5 Advantages of the Ground Source Heat Pump Technologies

a. High Efficiency and stable capacity.

When properly installed, GSHP systems show higher efficiency, requires less fan and pump energy, and hence are more cost effective than conventional HVAC systems. Further, the liquid in the GHE varies very little with the outdoor temperature, and hence the capacity of the system remains stable.

b. Better Air Quality and Comfort

GSHPs can achieve high efficiencies by reducing the ratio of compressor discharge and suction pressure without compromising on their latent cooling capacity. GSHPs can maintain humidity level very well, which make them suitable for public and office commercial applications without the need for additional dehumidification or a latent heat recovery system. GSHPs can effectively provide comfort both in summer and winter.

c. Simple control and Equipment.

Complex and expensive devices and distribution systems are not required resulting in a saving in the operating cost of a GSHP. GSHP systems can be configured to have heat pumps for individual zones that can be locally controlled as required for comfort. Thermal comfort and the system efficiency can be attained without using complex equipment.

d. Low Maintenance cost

Almost all the heat pump components for a GSHP can be installed indoors. This results in less maintenance as it reduces problems associated with corrosion and weathering.

e. No additional equipment required for auxiliary heating

Institutional and commercial buildings are mostly cooling dominated; therefore, the heating capacity of a GSHP is usually higher than the required heating capacity. This eliminates any requirement for auxiliary heating. The heating mode can be activated just by the reversing the valve and changing the thermostat controls.

f. Low cost water heating.

Commercial buildings are usually cooling dominated, as a lot of heat is generated from internal heat gains such as plug loads, lighting loads, and occupants. This unwanted heat can be used to generate hot water by installing heat recovery coils or with dedicated water-to-water heat pumps. This heat recovery system can also reduce the length of the GHE as most of the heat will be removed before entering the GHE.

g. Environment friendly –reducing carbon emissions

In the Unites States, one-fifth of total global greenhouse gas emissions are from heating cooling and electricity generation. According to the EPA, GSHPs produce the lowest CO_2 emissions and have the lowest overall environmental cost of all the other HVAC systems (Egg, 2013). Being more efficient systems, GSHPs consume less energy than other HVAC systems, resulting in less pollution.

i. Low demand characteristics

Typical demand reductions for GCHPS versus. conventional equipment in commercial buildings in the cooling mode are:

Rooftop unit vs. 0.5kW/ton

Multizone rooftop vs. 0.6kW/ton

Chiller (0.5 kW/ton) with VAV vs. 0.3kW/ton

Chiller (0.7 kW/ton) with VAV vs. 0.5kW/ton

j. Life cycle cost

In spite of high first or installation cost, GCHPs have lower operating cost, lower demand cost, lower maintenance cost, and extended life compared to conventional systems. All these factors can make the GSHPs very affordable and cost effective.

1.6 Drawbacks of the Ground Source Heat Pump Technologies

a. Higher initial cost

In commercial applications, the initial cost is very high, up to 40% higher than the conventional HVACs.

b. Performance depends on ground coil and equipment

The performance of the GSHPs not only depends on the heat pump, but also on the design and installation of the GHE. Often, when the cost of ground loop heat exchanger seems to be higher, the common practice is to install cheap, inferior quality heat pump equipment.

c. Limited number of qualified engineers and contractors

There are a limited number of qualified engineers working on GSHPS as engineers often hesitate to invest time learning new technology. Drilling contractors are also hesitant to accept GSHP drilling jobs since these are few, requiring travel to far off places and the work is often hard and dirty.

d. HVAC vendor profit is reduced

The simplicity of the GSHPs make engineers independent without relying on the manufactures to design the system. This reduces the overall profit per job for GSHP manufacturers, unlike other conventional HVAC systems.

CHAPTER 2

OBJECTIVE & SCOPE OF RESEARCH

2.1 Problem Statement

One of the prime factors towards the hindrance of adoption of GCHPs is higher installation cost, since the GHE accounts for about 34-40% of the installed cost. However, with the available federal incentives, GCHPs are gaining popularity in the northern colder parts of United States, and also in the hotter climates with high permeability soil and/or shallow groundwater. On the other hand, the soil present in Phoenix and much of the southwestern United States is dry, has poor thermal properties, and has a low permeability, which considerably affects the length and performance of the GHE. In fact, moisture content is the most influential factor determining the soil's thermal properties. An increase in the moisture content in the soil, significantly beneficially affects the soil's thermal properties. To improve the performance and design of the GHE, one needs to modify the soil so as to exhibit properties similar to a saturated soil. Moreover, buildings located in a dry climate like Phoenix consume a significant amounts of energy to cool the building. As a result, heat rejected to the ground over the years of operation of a GCHPs drying out the soil adjoining the GHE, thereby degrading the efficiency of the system, increasing cost and potentially leading to engineering failure. Therefore, a 20 year time period was considered when designing the V-GCHP and H-GCHP systems to take into account the effect of prolonged heat dissipation into the ground. Ground improvement can be used to improve the performance and design of the GHE in dry soil. But the performance of a GHE in saturated soil is taken as a liming condition on the beneficial effect of ground improvement. This study covered both V-GCHPs and H-GCHPs.

2.2 Objectives

The objectives of this research is to quantify the energy consumption and cost benefits of the ground coupled heat pump system as compared to conventional system, taken to be an air source heat pump (ASHPs). The scope is limited to commercial buildings in the hot, semi-arid climate of Phoenix, Arizona. This study uses building data based on ASHRAE 90.1 standards for the small office building prototype to develop the baseline model. The V-GCHP and H-GCHP systems are designed using commercial software able to perform simulations using hourly whole building energy consumption. The energy simulation is to be performed using the computer program EnergyPlus (EnergyPlus, 2013).

The intent of this thesis is also to study the effect of saturated soil conditions, and therefore the maximum potential benefit of ground improvement, on the required GHE length and energy performance of the both vertical and horizontal GHEs. Further, with the help of a market study, the costs of ASHP, V-GCHP and H-GCHP systems are to be estimated. The simulated energy consumption and estimated system costs are to be used for life-cycle costs analysis. Further, the associated greenhouse gas emissions were also to be evaluated.

2.3 Scope

The study focuses on two types of HVAC systems: ground coupled heat pump systems and air source heat pumps. The ASHP systems are further categorized into existing ASHPs and high efficiency ASHPs. Both horizontal and vertical GCHP systems are examined, as both the systems are suitable for small commercial buildings and residential buildings. H-GCHPs are usually not recommended for large commercial buildings as they require large areas.

This research is limited to hot and semi-arid climates, specifically the Phoenix. This type of climatic zone experiences a cooling dominated load which causes a thermal mismatch and may degrade the efficiency of a GCHP system. Moreover, the soils are usually dry in the Phoenix area and therefore has poor thermal properties. An increase in saturation of a soil beneficially improves its thermal properties, especially the thermal conductivity. Therefore, the effect of saturated soil on the performance and energy consumption of GCHPs is to be studied. A 20 year time period was considered for designing the V-GCHPs to take into account the thermal degradation due to the prolonged heat dissipation into the ground.

CHAPTER 3

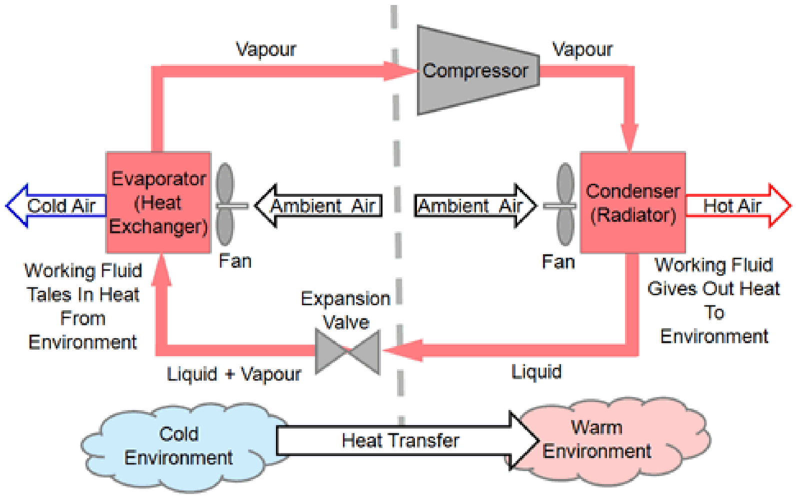
BACKGROUND AND LITERATURE REVIEW

3.1 Heat Pumps

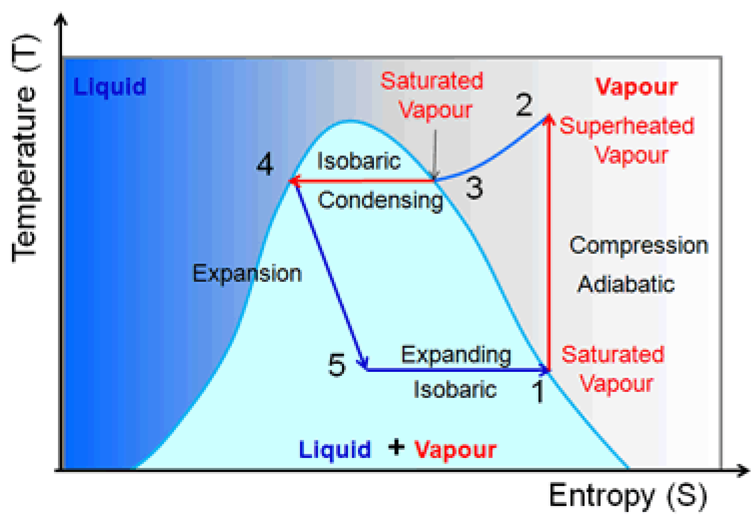
As mentioned in Chapter 1 of this dissertation, a heat pump works on a vapor compression cycle (see Figure 3.1). Mechanical heat pumps rely upon the physical properties of a volatile liquid known as the working fluid or refrigerant. The heat pump compresses this refrigerant to make it hotter on the side which is to be warmed and absorbs the heat by reducing the pressure on the side to be cooled.

A typical heat pump's vapor-compression refrigeration cycle has four components: i) condenser, ii) expansion valve, iii) evaporator, iv) compressor.

These components are connected to each other to form a closed circuit (see Figure 3.1). 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. As a result, refrigerant leaves the compressor and enters the condenser as a high pressure, high temperature and superheated vapor. In the condenser, the refrigerant 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 and low temperature vapor refrigerant then enters the evaporator, where the refrigerant evaporates by absorbing heat from the conditioned space. The cycle completes when the refrigerant leaves the evaporator and reenters the compressor. In the heating cycle, the refrigerant is made to circulate in the exactly reverse order. The pressure difference must be large enough for the fluid to condense at the hot side, and evaporate in the lower pressure region at the cold side.



(a)



(b)

Figure 3.1: Heat pump (a) Vapor Compression Cycle (b) Refrigeration Entropy Diagram (Electropedia, 2005)

For greater temperature difference between the evaporator and the condenser, greater pressure difference is required, and as a result more energy is needed to compress the fluid. Thus, the COP decreases with increasing temperature difference.

Table 3.1 Processes involved in vapor compression systems (Electropedia, 2005)

Change of State (see Fig 3.1)	Vapour Compression Heat Pump and Refrigerator Systems
1 to 2	The refrigerant in vapour state is compressed, raising its temperature and pressure.
2 to 3	The super-heated vapour is cooled to saturated vapour. Heat is removed from refrigerant at constant pressure and rejected to the environment.
3 to 4	The vapour condenses at constant temperature to a liquid releasing more heat.
4 to 5	The expansion valve (throttle) creates a sudden reduction of pressure which lowers the boiling point of the liquid, which flashes to liquid + vapour taking in heat from the medium surrounding the evaporator.
5 to 1	Liquid is evaporated and expands at constant pressure removing heat from the environment

3.2 Air Source Heat Pump Systems (ASHPs)

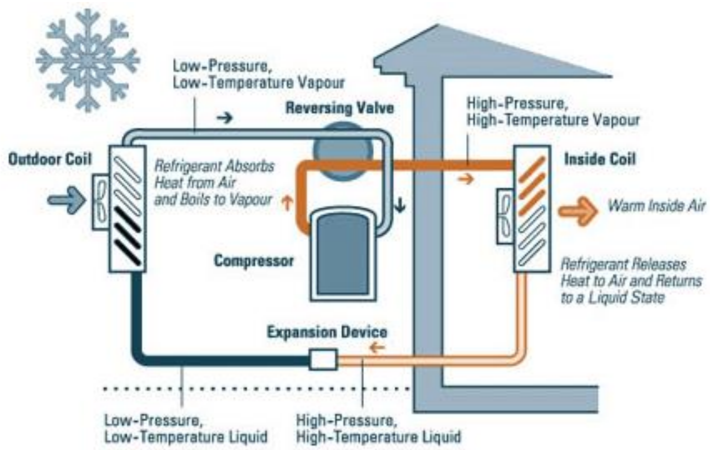
Air Source Heat Pump Systems transfer heat from inside to outside a building or vice versa. In the heating mode, ASHPs absorb heat from outside air and release it inside the space to be conditioned. When the same cycle is reversed, they can provide cooling in summers.

3.2.1 Main Components

In addition to the heat pump itself, the two main components are:

- An outdoor heat exchanger coil, to release or absorb heat from ambient air,
- An indoor heat exchanger coil, to transfer the absorbed or extracted heat to condition the space.

The components of an air-source heat pump are shown in Figure 3.2.



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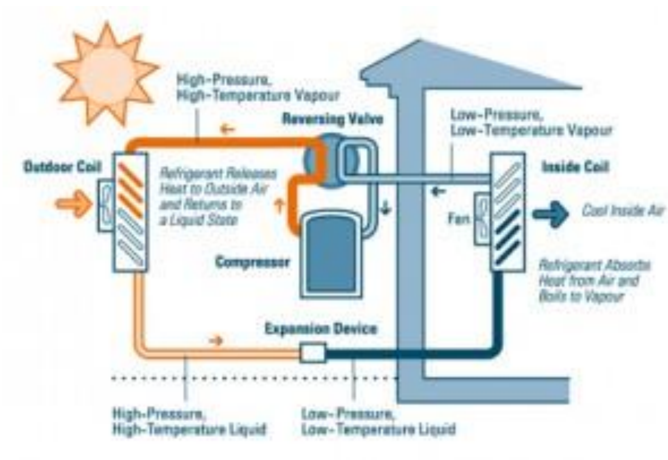


(b)

Figure 3.2 Components of an Air-source Heat Pump (a. Heating cycle, b.Cooling Cycle) (Paradise Air Inc.)

3.2.2 Applications of Air Source Heat Pumps

Like other heat pump systems, ASHPs are used to provide space heating and cooling in buildings, and can be used efficiently for water heating in milder climates.

Though the cost of installation is generally high for ASHPs, they are comparatively lower than the cost of GSHPs, in large part because ground excavation is not required.

Cold winter temperatures are the main limitation of all ASHPs. The heating cycle of an ASHPs becomes inefficient as outdoor air temperatures drops and approaches the freezing temperature. As a result, ASHPs are often paired with auxiliary heat systems to provide backup heat when outside temperatures are too low for the pump to work efficiently. Propane, natural gas, or oil furnaces can provide this supplementary heat.

3.3 Main Components of Ground Coupled Heat Pumps

The GCHPs have three basic components namely (see Figure 3.3) :

- The ground coupled heat exchanger,
- The building load which is the medium or object which needs to be heated or cooled.
- Mechanical refrigeration system or the heat pump

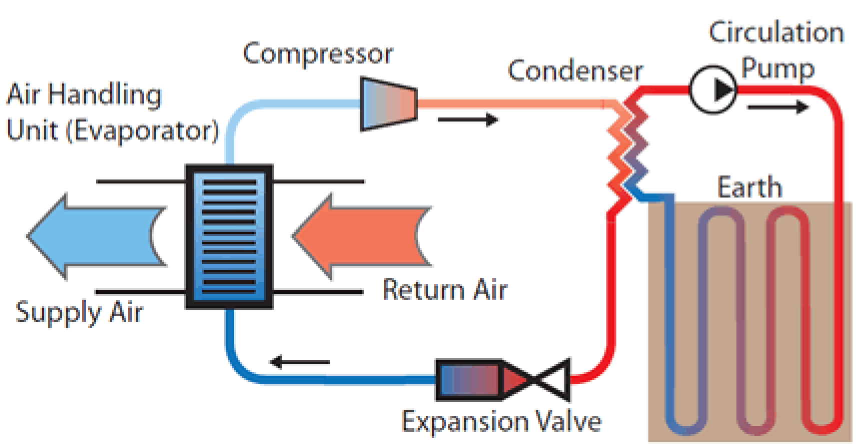


Figure 3.3: Typical GCHP unit (Energy Design Resources)

The ground coupled portion uses earth as source or sink according to the requirement. This consists of underground pipes which are made up of thermally conductive material such as plastic or metal.

3.4 Existing Models for Ground Loop Heat Exchangers

3.4.1 Overview

As discussed in the earlier section, the GCHP provides heating and cooling by drawing upon the thermal energy contained in the ground. If properly installed, the a ground coupled heat pump can provide high levels of comfort, efficiency and savings. However, improper installation of the GHE leads to under sizing or oversizing of the GHE, significantly affects the efficiency and the installation cost of the GCHPs.

Existing thermal models for designing GCHPs can be categorized into analytical models and numerical models. Some models are also based on combined analytical and numerical approach to simulate the behavior of the GHE. There are mainly two analytical methodologies available that are used to size and design the vertical GHE:

1. Kelvin's line source theory (1882),
2. Carslaw and Jaeger's cylinder source solution (1947).

The thermal conductivity of ground formation can be divided into steady state and transient methods based on the heat transfer applied to the ground sample. As the name suggests, in steady-state methods, the measurements are taken when the sample reaches steady state and does not change with time. On the other hand, transient methods are necessary when the temperature of the sample varies with time. The efficiency of the GHE depends mainly on two factors namely:

a. Its ability to reject or extract heat over long period of time,
b. Avoiding dumping or extracting excessive heat into or from the ground.

Therefore any model to design GHE has to be able to calculate the transient effects over a number of years. The methods based on analytical approach which have been developed over the years are based on various simplified assumptions. The analytical models are computationally very efficient but since pipes are not co-axial with the borehole, and there are various other materials involved, the analytical models are rather inefficient.

The equivalent diameter assumption is the most important simplified assumptions which considers the two legs of the U-tube to be a single pipe co-axial with the borehole so that the cylinder source solution (Carslaw and Jaeger, 1947) may be applied. The geometry can be further simplified assuming it to be an infinitely long line source (Kelvin 1882, Ingersoll 1948, 1954). The models described in the section 3.4.2 are referred and summarized in literature reviews by Yavuzturk (1999) and Murugappan (2002).

3.4.2 Brief Description of the Existing Models

Ingersoll (1948, 1954) approach is based on Kelvin's (1891) line source theory to model GHE. The main assumption of the Kelvin's line source theory is that an infinity long line source or sink with constant heat rate is buried in a medium initially at a uniform temperature. The heat source or sink is switched on at time zero. According to Ingersoll, this temperature variation can be given by the equation 1.

$$T-T_0 = \frac{Q'}{2\pi k} {}^{\infty}\int_x^{e-\beta 2} \frac{}{\beta} d\beta = \frac{Q'}{2\pi k} I(X) \quad \textit{Eqn. 1}$$

Where,

$$X = \frac{r}{2\sqrt{\alpha t}} \quad \textit{Eqn. 2}$$

T = Temperature of ground at any selected distance from the line source in [°F or °C] (Selecting a distance that is equal to the pipe radius represents the pipe surface temperature)

T_0 = Initial temperature of the ground in [°F or °C]

Q' = Heat transfer rate over the source in [BTU/(ft-hr) or W/m]

r = Distance from center line of pipe in [ft or m],

k = Thermal conductivity of the ground formation in [BTU/(ft-hr-°F or W/(m-°C)]

α = Thermal diffusivity of the ground formation defined to be K/°C,

ρ= Density of the ground formation in [lb/ft3 or kg/m3]

t = Time since the start of the operation in [hr]

β = Integration variable = r/ $2\sqrt{\alpha(t-t')}$

The values of I(X) can be found in Ingersoll et al. (1954).

The Equation 1 holds true only for a true line source, but according to Ingersoll this equation can be also applied after few hours of operation, to small pipes of 2" or less diameter. However, he pointed out that there will be a significant error associated with larger pipes and for periods of operation less than a few days. He also proposed a dimensionless term αt/r2 which must be greater than 20 to maintain an error that is small enough for practical applications. The Ingersoll approach provides very rough approximations of heat transfers.

Hart and Couvillion (1986) methodology is also based on line source theory. But they claim that the equation falsely predicts the temperature distribution of the surrounding ground once the line source is switched on. It is because Kelvin didn't consider any far field radius *r* beyond which the ground temperature remains at the undisturbed temperature. Therefore, they modeled the ground loop heat exchanger considering undisturbed far field temperature with the far field radius defined by the following equation:

$r_\infty = 4\sqrt{\alpha t}$ *Eqn. 3*

Where temperature can be given as,

$T-T_0 = \frac{Q'}{4\pi k} \int_y^\infty \frac{e^{-\lambda}}{\lambda} d\lambda$ *Eqn. 4*

And, $y = r^2/4\alpha t$ *Eqn. 5*

The solution to the integral in Equation 4 can be calculated from integral tables. This methodology is based on the assumption that the heat transfer takes place between the ground formation and the line source of radius r_∞, so the region beyond this radius is assumed to be at the undisturbed far field temperature. The value of far field radius depends on two factors, namely time and the thermal diffusivity of the ground. In cases where there are various boreholes and when the thermal interference becomes effective, the superposition technique is used to determine the ground temperature.

The model by Kavanaugh (1985) is built around the Carslaw and Jaeger (1947) cylindrical source approximation model. It determines the temperature distribution or heat transfer rate around a buried pipe by using the cylinder source solution. The model is based on the assumption that a single isolated pipe is surrounded by an infinite solid medium with constant thermo-physical properties. It also ignores the effect of both ground water movement and thermal interaction between adjacent GHE.

Kavanaugh applied this model at two test sites and provides the experimental data. According to Kavanaugh, the model works well if care is taken while choosing the properties of ground formation and initial entering water temperatures are not desired immediately after startup. Kavanaugh assumes a single U-tube pipe in his model which introduces some error into the solution due to thermal short-circuiting effects and pipe wall and contact resistances.

The simplifying assumptions in the Kavanaugh model may be negligible with respect to the long term performance of the GHE but may affect the short-term response (hours and weeks) of the borehole.

There are various models based on numerical solution technique. These models are able to calculate the complex phenomenon occurring around GHE, but Yavuzturk (1999) claims that they are computationally inefficient.

Eskilson's approach to estimate the temperature distribution around a borehole is based on a hybrid model combining analytical and numerical approaches. To determine the temperature of a multiple borehole GHE, the GHE field is converted to a set of non-dimensional temperature response factors, called g-functions. The numerical model employs a two-dimensional explicit finite difference equation in a radial-axial coordinate system for a single borehole in homogenous ground. In this model, the borehole has a finite length and diameter but the thermal properties of individual GHE materials such as the U-tube pipe and grout's resistances are neglected. The GHE thermal resistance is accounted for separately. Further this model is used to compute the response to a unit step function pulse. Using the spatial temperature response of a single borehole to the unit step function pulse. When the borehole outer wall temperature vs. time is non-dimensionalized, the resulting dimensionless temperature vs. dimensionless time curve is the *g*-function. When the individual response to a step function is known, the response to any arbitrary heat extraction/injection function can be calculated by devolving the heat extraction/injection into a series of unit step functions. The response factors of the GHE (the *g*-functions) to each unit step functions can be superimposed to determine the overall response.

Hellstrom (1989, 1991) developed a simulation model for vertical ground heat exchanger stores which uses densely packed GHEs for seasonal thermal energy storage (Yang et al., 2010). He subdivided the ground formation region with multiple GHE into two separate regions called the local region and the global problem. The latter region is concerned with the heat conduction problem between the bulk of the heat store volume and the far field. The local region is the volume that immediately surrounds the single borehole. The Hellstrom model represents the initial ground formation temperature as the superposition of three parts: a global temperature difference, a temperature difference from the local solution immediately around the individual borehole, and the temperature difference from the local steady-flux part. Hellstrom's model is a hybrid model, a numerical method is used for the local and global problems and an analytical approach is used to superimpose the solution from steady flux part with the local and global solutions. The limitation of this model is that it is not ideal for determining the short time response of the ground.

The Thornton et al. (1997) proposed a model based on Hellstrom's approach to modeling the GHE and was implemented in TRNSYS (Klein, et al. 1996). The model is a a detailed component model which was calibrated on monitored data from a family housing unit by adjusting the parameters like the far-field temperature and ground thermal properties. The model was able to match the measured data accurately (Yavuzturk, 1988).

Mei and Emerson (1985) developed a numerical model for a horizontal ground loop heat exchanger which is suitable for modeling the effects of frozen ground formation and pipes. Their model was based on three one-dimensional partial differential equations using a finite difference approach. Three one-dimensional conduction equations were used in this method with (1) one equation along the radial direction of the pipe, (2) one to the frozen ground formation and (3) one to the far-field region. These three one-dimensional equations were then coupled into one single partial differential equation forming a fourth quasi two dimensional equation. The model uses different time steps for various parts, i.e. it used a smaller time-step for the pipe wall and frozen ground and comparative very larger time-step for the unfrozen far-field region. The model has been experimentally verified based on a 448-day simulation period.

Yavuzturk and Spitler (1999) modeled the GHE to determine the short time step variation using response factors. The model is based on Eskilson's g-function algorithm to account for the effects of the thermal properties of the backfill material and thermal properties of the anti-freeze on the GHE performance. Yavuzturk and Spitler came up with a short time-step response factors using a transient, two-dimensional, implicit finite volume model on a polar grid and further adjusted the short time-step *g*-functions to match the long time-step *g* functions developed by Eskilson (1987). These short time step g-functions are implemented in EnergyPlus.

3.4.3 Theoretical Validation Method

In order to check the accuracy of GLD software, an appropriate theoretical solution was sought. The method mentioned in the ASHRAE handbook can be used to design the borehole heat exchanger for both residential and small scale commercial buildings, and was adopted to partially validate the GLD tool. This method is based upon the equation of the heat transfer from a cylinder buried in the ground by Carslaw and Jaeger. The equation and solution was suggested by the Ingersoll to design the GHE where the Kelvin's line source model fails for time steps less than 6 hours. Therefore, for the accurate predictions of the hourly time step, the cylindrical model is used. This method to determine the shorter time variation is based on the Ingersoll's method. This method uses simple steady state heat transfer equation.

$q = L(t_g - t_w)/R$ *Eqn. 6*

By rearranging the equation we can get the value of required borehole length. With the series of heat pulses, the steady state equation is changed to calculate the variable heat rate of a ground heat exchanger. Equations 7 and 8 consider three heat pulses to account for long term heat balances, (q_a) average annual heat rate during a year, (q_m) average monthly heat rate during design month and (q_d) maximum heat rate for short period of time during design day.

The required borehole length equation for cooling is given by:

$L_c = q_a R_{ga} + (q_{lc} - 3.41\ W_c)\ (R_b + PLF_m R_{gm} + R_{gd}\ F_{sc})\ /\ [t_g - ((t_{wi} + t_{wo})/2) - t_p]$ *Eqn. 7*

And the required borehole length equation for heating by:

$L_h = q_a R_{ga} + (q_{lh} - 3.41\ W_h)\ (R_b + PLF_m R_{gm} + R_{gd}\ F_{sc})\ /\ [t_g - ((t_{wi} + t_{wo})/2) - t_p]$ *Eqn. 8*

Where,

F_{sc} = short-circuit heat loss factor

L_c = required borehole length for cooling (ft)

L_h = required borehole length for heating (ft)

PLF_m = part load factor during designing month

q_a = net annual average heat transfer to the ground (Btu/h)

q_{lc} = building design cooling block load (Btu/h)

q_{lh} = building design heating block load (Btu/h)

R_{ga} = effective thermal resistance to the ground, annual pulse (h*ft*°F/Btu)

R_{gm} = effective thermal resistance to the ground, monthly pulse (h*ft*°F/Btu)

R_{gd} = effective thermal resistance to the ground, daily pulse (h*ft*°F/Btu)

R_b = thermal resistance of bore (h*ft*°F/Btu)

t_g = undisturbed ground temperature

t_p = temperature penalty for interference of adjacent boreholes

t_{wi} = liquid temperature at heat pump inlet (°F)

t_{wo} = liquid temperature at heat pump outlet (°F) (see Table 3.2)

W_c = power input at design cooling load (W)

W_h = power input at design heating load (W)

Heat transfer rates, building loads, and temperature penalties are positive for heating and negative for cooling. Table 3.2 and 3.3 gives the thermal resistance values of U-tube and thermal resistance adjustments for other borehole backfills or grouts.

Table 3.2: Liquid temperature change through GCHP units (Kavanaugh and Rafferty, 1997)

System flow (gpm/ton)	Temperature rise in Cooling (°F)	Temperature drop in heating (°F)
3.0	10	6
2.5	13	7-8
2.0	15	9

Table 3.3: Equivalent diameter and thermal resistances (R_b) for Polyethylene U-tubes (Kavanaugh and Rafferty, 1997)

U-tube Dia. (Eqv. Dia.)	SDR or Schedule	Pipe (bore) thermal resistance (h.ft.°F/Btu)			
		For water flows above 2.0 gpm	20% Prop. Glycol flow 3.0 gpm	20% prop. Glycol flow 5.0 gpm	20% prop glycol flow 10.0 gpm
¾ in. (0.15ft)	SDR 11	0.09	0.12	NR	NR
	SDR 9	0.11	0.15	NR	NR
	Sch 40	0.10	0.14	NR	NR
1.0 in.	SDR 11	0.09	0.14	0.10	NR
2.0 (0.18 ft)	SDR 9	0.11	0.16	0.12	NR
	Sch 40	0.10	0.15	0.11	NR
1 ¼ in. (0.22 ft)	SDR 11	0.09	0.15	0.12	0.09
	SDR 9	0.11	0.17	0.15	0.11
	Sch 40	0.09	0.15	0.12	0.09
1 ½ in. (0.25 ft)	SDR 11	0.09[1]	0.16	0.15	0.09
	SDR 9	0.11[1]	0.18	0.17	0.11
	Sch 40	0.08[1]	0.14	0.14	0.08

[1]Water flow must be at least 3.0 gpm to avoid laminar flow for these cases.

Table 3.4: Thermal resistance adjustments for other borehole backfills or grouts (Kavanaugh and Rafferty, 1997)

Natural Soil Cond	0.9 Btu/h.ft.°F		1.3 Btu/h.ft.°F			1.7 Btu/h.ft.°F	
Backfill or grout conductivity	0.5 Btu/h.ft.°F	2.0 Btu/h.ft.°F	0.5 Btu/h.ft.°F	1.0 Btu/h.ft.°F	2.0 Btu/h.ft.°F	0.5 Btu/h.ft.°F	1.0 Btu/h.ft.°F
4 in. bore							
¾ in. U-tube	0.11 (NR)	-0.05	0.14 (NR)	0.03	-0.02	0.17 (NR)	0.05
1 in. U-tube	0.07	-0.03	0.09	0.02	-0.02	0.13 (NR)	0.04
5 in. bore							
¾ in. U-tube	0.14(NR)	-0.06	0.18 (NR)	0.04	-0.04	0.21 (NR)	0.06
1 in. U-tube	0.11 (NR)	-0.04	0.14 (NR)	0.03	-0.02	0.16 (NR)	0.05
1 ¼ in. U-tube	0.06	-0.03	0.09	0.02	-0.02	0.12 (NR)	0.04
6 in. bore							
¾ in. U-tube	0.18(NR)	-0.07	0.21 (NR)	0.04	-0.05	0.24 (NR)	0.07
1 in. U-tube	0.14 (NR)	-0.06	0.17 (NR)	0.03	-0.04	0.21 (NR)	0.06
1 ¼ in. U-tube	0.09	-0.04	0.12 (NR)	0.03	-0.02	0.15 (NR)	0.05
1 ½ in. U-tube	0.07	-0.03	0.09	0.02	-0.02	0.11 (NR)	0.04

(NR) = Not Recommended – for low thermal conductivity grouts, use small bore
Negative values indicate a thermal enhancement and a lower net thermal resistance compared to natural backfills.

The thermal performance of the GHE depends upon the amount of heat extracted or rejected in the ground. (Claesson and Eskilson, 1987). In the case of multiple boreholes spaced closed to each other, the minimum and maximum temperatures may take up to several years to occur. Therefore, while designing the GHE, the performance of the system should be considered for an extended period of time.

The solution of Carslaw and Jaeger require that the time of operation, outside diameter and thermal diffusivity of the ground be related in the dimensionless Fourier Number (Fo)

$Fo = 4\alpha_g \tau/d^2$ *Eqn. 9*

This method is altered for varying heat pulses.

$\tau_l = 3650$, $\tau_2 = 3650+30 = 3680$,

$\tau_f = 3650+30+0.25 = 3685.25$ days,

Fourier number is calculated using following equations:

$F_{of} = 4\alpha_g \tau_f/d^2$, $F_{o1} = 4\alpha_g (\tau_f - \tau_1)/d^2$ and $F_{o2} = 4\alpha_g (\tau_f - \tau 2)/d^2$ *Eqn. 10*

The G factor for the respective Fourier numbers can be determined from Figure 3.4. The three equivalent thermal resistances during each heat pulse can be calculated by following equation:

$R_{ga} = (G_f - G_1)/ k_g$; $R_{gm} = (G_1 - G_2)/ k_g$; $R_{gd} = G_2/ k_g$ *Eqn. 11*

The following equation can be used to calculate the net average heat rate over an entire year:

$q_a = (\sum\ q_{lc} \times \frac{EER+3.41}{EER} \times h_c + = \sum\ q_{lh} \times \frac{EER+3.41}{EER} \times h_{hh}) / 8760/year$ *Eqn. 12*

Where q_{lc} and q_{lh} are peak block cooling and heating load of the system.

Ingersoll used a dimensionless term to relate soil thermal diffusivity, time of operations and distance from the heat source. The change in the ground formation temperature around a single U-tube bend can be determined by Equation 2.

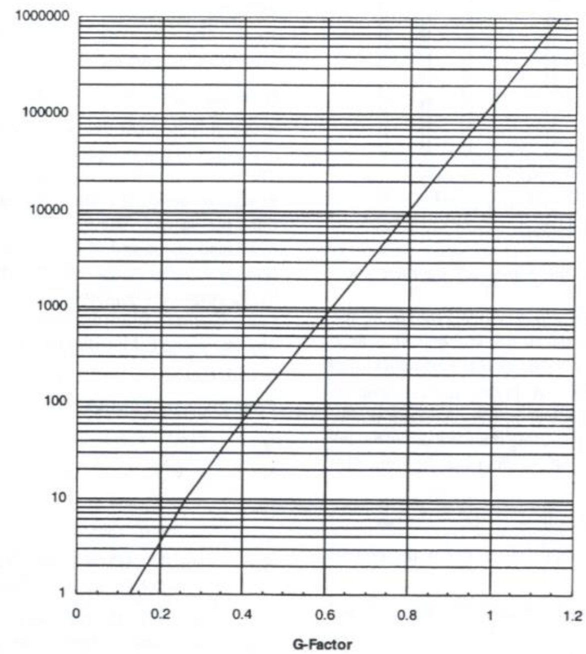


Figure 3.4: Fourier/G-factor graph for ground thermal resistance (Kavanaugh and Rafferty, 1997)

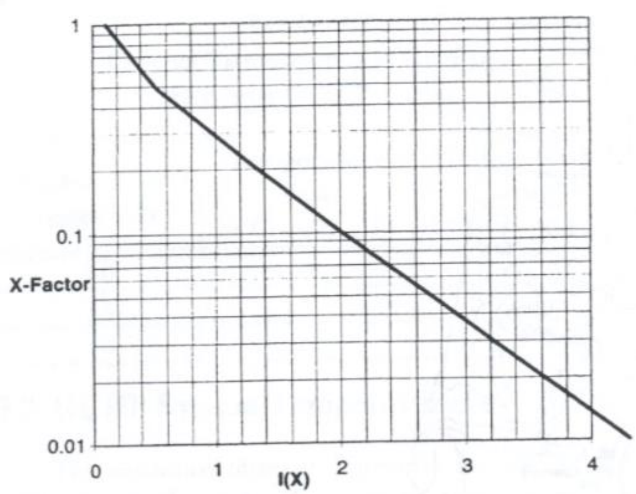


Figure 3.5: Chart for determining I(X) for temperature penalty (Kavanaugh and Rafferty, 1997)

The difference between the undisturbed ground formation temperature and the temperature at a distance r from the vertical ground coil can be given by

$\Delta t_r = q_a I(X_a) / 2\pi k_g L$. Where the values of the I(X) can be determined from Figure 3.5. The field temperature penalty is based on the number of adjacent boreholes and given by the equation

$t_p = (N_4 + 0.5N_3 + 0.25N_2 + 0.1N_1) \times t_{pl}$ / total number of boreholes *Eqn. 13*

Where,

t_{pl} = penalty for a bore surrounded on all four sides

N = number of boreholes surrounded by one, two, three or four adjacent bores.

3.5 Soil Properties

For vertical or horizontal closed loop systems, heat exchange between the fluid and the ground depends upon the thermal properties of the material in the borehole and of the surrounding ground. The borehole can be backfilled with soil or grout material. Therefore, the thermal conductivity soil and grout are critical factors to be considered while determining the length and cost of the ground loop. The term soil includes uncemented or partially cemented inorganic and organic material found in the ground. Soil Classification systems classify soil into two major divisions depending on the grain size: fine grained and coarse grained soil. Moist soils are a combination of fine and coarse grains. Coarse grained soils are soils where less than 50 percent of the soil (by weight) pass through a U.S. Standard Series No. 200 sieve (a sieve with openings of 0.074mm). Fine grained soils are soils where more than 50 percent (b weight) pass through the No. 200 sieve. Fine grained soils can be further classified as silt and clay according to their plasticity (their ability to deform without cracking). The thermal properties of the soil are functions of the soil type (coarse or fine grained), mineral content, dry density and degree of saturation. Saturation is defined by the ratio of the volume of moisture contained in the soil to the volume of the pore space.

The dry density refers to the mass of soil particles per unit volume. Among all parameters, the moisture content is the most influential factor while determining the soil's thermal conductivity. An increase in moisture content (or saturation) will result in an increase in its conductivity.

Table 3.5: Soil Thermal Conductivity and Diffusivity of Sand and Clay Soils*
(Kavanaugh and Rafferty, 1997)

Soil Type	Dry Density (lb/ft^3)	5% Moist		10% Moist		15% Moist		20% Moist	
		k Btu/h.ft.°F	α ft^2/day	k Btu/h.ft.°F	α ft^2/day	k Btu/h.ft.°F	α ft^2/day	k Btu/h.ft.°F	α ft^2/day
Coarse 100% Sand	120	1.2-1.9	0.96-1.5	1.4-2.0	0.93-1.3	1.6-2.2	0.91-1.2	-	-
	100	0.8-1.4	0.77-1.3	1.2-1.5	0.96-1.2	1.3-1.6	0.89-1.1	1.4-1.7	0.84-1.0
	80	0.5-1.1	0.60-1.3	0.6-1.1	0.60-1.1	0.6-1.2	0.51-1.0	0.7-1.2	0.52-0.90
Fine Grain 100% Clay	120	0.6-0.8	0.48-0.64	0.6-0.8	0.4-0.53	0.8-1.1	0.46-0.63	-	-
	100	0.5-0.6	0.48-0.58	0.5-0.6	0.4-0.48	0.6-0.7	0.37-0.48	0.6-0.8	0.41-0.55
	80	0.3-0.5	0.36-0.6	0.35-0.5	0.35-0.5	0.4-0.55	0.34-0.47	0.4-0.6	0.30-0.45

*Values indicate ranges predicted by five independent methods. (Evaluation of Methods for Calculating Soil Thermal Conductivity, O.T. Farouki, 1982)
k = Thermal Conductivity, α= Thermal Diffusivity

The thermal conductivity of the soil varies from 0.2 – 2.0 Btu/ hr*ft*°F (Mitchell, 1976). Porosity is an important factor governing thermal conductivity. Porosity is largely determined by the origin and nature of the soil (and rock). Most rocks are formed under high temperature and pressure, and therefore they usually have lower porosity as compared to the soil. The lower porosity of rock provides higher thermal contact area and, hence, rocks usually possess better thermal conductivities than soil, regardless of the mineral content. Table 3.5 provides the thermal properties of sand (coarse) and clay (fine) soils. Table 3.6 lists the thermal properties of rocks. The table is based upon a large number of samples in the United States.

Table 3.6: Thermal Properties of Soil and Rocks at 77 °F (Kavanaugh and Rafferty, 1997)

Rock Type	% Occurrence in Earth's Crust*	k - All** *Thermal Conductivity* Btu/h.ft.°F	k - 80%*** *Thermal Conductivity* Btu/h.ft.°F	c p *Specific Heat* Btu/lb.°F	ρ *Density* (lb/ft^3)	α (k/ρc_p) *Thermal Diffusivity* (ft^2/day
Dense Rock	--	2.00	--	0.20	200	1.20
Average Rock	--	1.40	--	0.20	175	0.96
Dense Concrete	--	1.00	--	0.20	150	0.79
Heavy Soil, Saturated	--	1.40	--	0.20	200	0.84
Solid masonry	--	0.75	--	0.21	143	0.60
Heavy Soil, Damp	--	0.75	--	0.23	131	0.60
Heavy Soil, Dry	--	0.50	--	0.20	125	0.48
Light Soil, Damp	--	0.50	--	0.25	100	0.48
Light Soil, Dry	--	0.20	--	0.20	90	0.26
Granite (10% Quartz)	10.4	1.1-3.0	1.3-1.9	0.21	165	0.9-1.3
Granite (25% Quartz)			1.5-2.1			1.0-1.4
Amphibolite	42.8	1.1-2.7	1.5-2.2		175-195	
Andesite		0.8-2.8	0.9-1.4	0.12	160	1.1-1.7
Basalt		1.2-1.4		0.17-0.21	180	0.7-0.9
Gabbro (US Cen. Plains)		0.9-1.6		0.18	185	0.65-1.15
Gabbro (US Rocky Mtns)		1.2-2.1				0.85-1.5
Diorites	11.2	1.2-1.9	1.2-1.7	0.22	180	0.7-1.0
Grandiorites		1.2-2.0		0.21	170	0.8-1.3
Claystone		1.1-1.7				
Dolomite		0.9-3.6	1.6-3.6	0.21	170-175	1.1-2.3
Limestone		0.8-3.6	1.4-2.2	0.22	150-175	1.0-1.4

Rock Salt		3.7		0.20	130-135	
Sandstone	1.7	1.2-2.0		0.24	160-170	0.7-1.2
Siltstone		0.8-1.4				
Wet Shale (25% Quartz)	4.2	0.6-2.3	1.0-1.8	0.21	130-165	0.9-1.2
Wet Shale (No Quartz)			0.6-0.9			0.5-0.6
Dry Shale (25% Quartz)			0.8-1.4			0.7-1.0
Dry Shale (No Quartz)			0.5-0.8			0.45-0.55
Gneiss	21.4	1.0-3.3	1.3-2.0	0.22	160-175	0.9-1.2
Marble	0.9	1.2-3.2	1.2-1.9	0.22	170	0.8-1.2
Quartzite		3.0-4.0		0.20	160	2.2-3.0
Schist	5.1	1.2-2.6	1.4-2.2		170-200	
Slate		0.9-1.5		0.22	170-175	0.6-0.9

* Percentage of sedimentary rocks is higher near the surface.

** "All" represents the conductivity range of all samples tested.

*** "80%" represents the mid-range for samples of rock.

3.6 Backfill Grout Materials for Ground Coupled Heat Pumps

In V-GCHPs, grout is used to fill the borehole and secure the U loop tube. The grout typically used in GCHPs is usually a thermally enhanced grout to enables efficient heat transfer between the soil and the GHE pipes by providing a better surface contact between them. The grout also provides a water resistant seal around the U-tube to guard against migration of contaminants into the groundwater system. Table 3.7 gives the thermal properties of typical grouts and backfills used in vertical boreholes.

Table 3.7: Thermal Conductivities of Typical Grouts and Backfills (Kavanaugh and Rafferty, 1997)

Grouts without Additives	k (Btu/h·ft·°F)	Thermally Enhanced Grouts	k (Btu/h·ft·°F)
20% Bentonite	0.42	20% Bentonite—40% Quartzite	0.85
30% Bentonite	0.43	30% Bentonite—30% Quartzite	0.70–0.75
Cement Mortar	0.40–0.45	30% Bentonite—30% Iron Ore	0.45
Concrete @ 130/150 lb/ft^3	0.60/0.80	60% Quartzite – Flowable Fill (Cement + Fly Ash + Sand)	1.07
Concrete (50% quartz sand)	1.1–1.7		

3.6.1 Traditional Materials

The most common grout used in geotechnical engineering is employs a clay based substance called 'bentonite'. Bentonite grout consists of water and powdered bentonite mixed into slurry and hen supplemented with additives (including, sometimes, Portland cement) and specialized grout chemical. The bentonite solid content reduces the permeability of the grout. It also helps prevent boreholes from being flooded and from general debris getting between the pipes and the rock surface. Standard bentonite grout has a thermal conductivity that is lower than most soils or geologic materials (0.43 BTU/ hr*ft*°F v/s 0.8 to 1.8 BTU/ hr*ft*°F), thus it acts as an insulator around the heat exchange pipes (Smith and Perry, 1999).

The thermal conductivity of the backfill material is very important due to high heat flux rates that normally occur near the U tube. Poor thermal conductivity of backfill material will impede heat transfer and result in longer loop/borehole length requirements. For this reason, thermally enhanced grout (bentonite, silica sand, cement, super-plastisizer, iron particles and water) is a very viable and popular option.

3.6.2 Thermally Enhanced Materials

To enhance the thermal conductivity of grouts, they are mixed with silica sand and iron particles, and at times other materials such as cement and super-plasticizer. Thermally enhanced bentonite grouts have been developed and have thermal conductivities of 0.85 to 1.4 BTU/ hr*ft*°F (Rafferty, 2003) and retain low hydraulic conductivity (<0.33-0.23 ft/sec), based on technical data from manufacturers.

3.7 Economics of Ground Coupled Heat Pumps

GCHPs, like many other energy conserving technologies, usually require high initial investments, and such costs are claimed to be offset by the energy savings. There are several economic measures available to determine the feasibilities of investments on building projects such as return on investment (ROI) and simple payback period. These measures are very beneficial but limited to compare investment alternatives. The major drawbacks of these methods are that they fail to compare the cost benefits of energy efficiency investments. These measures do not consider either the future costs appearing after the initial investment or the value of money over a project life-time. Therefore, investments in energy efficiency technologies require economic evaluation procedures that consider future costs including operating costs, maintenance costs, and repair costs (OM&R). Moreover, the methodology should also consider inflation and opportunity costs. All the above parameters are essential to determine how lower future costs can compensate for the higher initial investment or first costs of energy saving alternatives. 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) have developed standards for the economic evaluation for energy efficiency investments. In this thesis two methods are explained; the simple payback method and the life cycle costing (present value) analysis method.

3.7.1 Simple Payback Method

The simple payback method considers the initial investment cost and the resulting annual cash flow. The payback period is the time required to recover the initial investment. This method is usually used to compare the various HVAC system choices. The major drawback of this method is that it fails to consider the savings that may continue from a project after the initial investment is paid back from the profits of the project. If the annual cash flow over the years are equal, then the payback period can be determined by dividing the initial investment by the annual savings. For an annual cash flow which differs year to year, the payback period is calculated when the accrued cash savings equals to the initial investment cost, or in other words, when the cumulative cash flow balance is equal to zero.

For HVAC systems, the payback can be termed as the total time taken for an annual utility and maintenance cost savings to offset an initial difference in cost between two systems. The simple payback period can be summarized in the following equation:

$$Y_{PB} = \frac{K_{2-}K_1}{(E+M)_1-(E+M)_2} \qquad \textit{Eqn. 14}$$

Where,

Y_{PB} = payback time, years

K = capital investment

E = annual energy cost

M = annual maintenance cost

$_1$ = system under consideration

$_2$ = alternative system

There are two major disadvantages of this method. Firstly, the energy benefits and operating cost savings after the payback period are not taken into account. Second, the time value money or the increasing cost of the maintenance and energy cost over time is ignored.

3.7.2 Life-Cycle Cost Analysis (LCCA)

LCCA is one of the most popular and widely used procedures for assessing the total cost of facility ownership. In addition to LCCA, many economic measures have been developed to include the future costs and time value of money. In order to account for the increased utility cost over the years, escalation rates are taken into consideration. The future costs are estimated by applying a discount rate and are expressed in present day dollars. LCCA takes into account all costs related to building, operating, and maintaining a project over a defined period of time. LCCA considers the overall cost effectiveness between alternatives by comparing their life-cycle costs. The alternative which provides the lowest lifecycle cost is the most cost-effective. Life Cycle Costing (LCC) is an important economic analysis which compares the initial investment alternatives and identifies the least cost alternative for a twenty five year period. LCC consists of two cost categories: investment related costs and operating costs. The investment cost include initial investment costs like land acquisition, construction or installation costs, replacement costs, and residual values. The operating costs include utility costs like energy, water costs, and the operation, maintenance, and repair costs (OM&R). The finance charges including loan interest payments, and contract costs such as for the energy saving performance contract, or utility energy services contract, and the non-monetary benefits can be also included in LCCA. The costs considered as future costs are discounted to their present values (PV). The summary of LCC can be expressed by equation 15.

Life-cycle costs (LCC) = Initial investment cost[a] (I)

+ Present value (PV) capital replacement costs (Repl)

- PV residual values (Res)

+ PV of energy costs (E)

+ PV of water costs (W)

+ PV OM&R

+ PV of other costs (O) *Eqn. 15*

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

According to the FEMP/NIST method, the present values can be divided into two categories of costs: goods or services, and energy. The method for goods and services considers the same inflation rate unlike energy costs. The first step in estimating the PVs of goods or services is to identify their future costs. The future cost of a particular element can be calculated using Equation 16. The FEMP/NIST method limits the maximum study period to 25 years.

$F_t = P_0 \times (1 + i)^t$ *Eqn. 16*

Where,

F_t = Future cost

P_0 = present value of goods or services

i = assumed rate of inflation

t = assumed future year

After estimating the initial cost, the discount rate is determined for PV calculations. The discount rates can be the nominal discount rate or the real discount rate. The nominal discount rate can be defined as the minimum rate of return on an alternative investment for borrowed capital such as the loan rate (Addison 1999). The nominal discount rate is based on market interest rates, and is considered sensible for an investment if it is greater than the inflation rate. In the FEMP/NIST method, the discount rate considered should be real rather than nominal. The real discount rate can be calculated by Equation 17.

$d = \frac{1-D}{1-i} - 1$ *Eqn. 17*

Where,

d = real discount rate,

D = nominal discount rate

i = inflation rate

The present value (PV) of items can be derived from Equation 16 and 17 and given by the Equation 18,

$$PV = F_t \times \frac{1}{(1+d)^t} \qquad \textit{Eqn. 18}$$

PV = present value

F_t = future cost

d = real discount

t = study period

The real discount rates are annually revised by FEMP. The FEMP/NIST LCCA method uses the annual energy price index to determine the PV of energy cost. The energy price index is revised yearly and available through the U.S. Department of Energy. The present value of the energy cost can be given by the equation 19.

PV energy cost = Energy price (on the specific date) x the energy price index *Eqn. 19*

The present value of the energy cost is constant as the energy escalation rates are real and can be used in LCCA and other supplemental economic measures that considers the energy cost.

3.8 Overview of Ground Couple Heat Pump Sizing Software

3.8.1 GLHE-Pro:

GLHEPro is a design tool mainly used for commercial building applications. It is based on the design methodology developed by Eskilson and was developed by Jeffrey D. Spitler of Oklahoma State University. The programing languages used to develop the software are VBA and Fortran. GLHE-pro can size the GHE, taking into account both monthly and peak loads simultaneously. The cost of the software ranges from $525-$725 for single licenses.

In GLHE-pro, the g-functions are calculated using a finite difference model, which limits the user to limited predetermined configurations. There is a list of approximately 307 different borehole configurations available in the software, each with a respective g-function, to determine the depth of the borehole(s). GHLE-pro requires input of monthly heat and cooling loads, monthly peak loads (optional), with number of peak hours, and heat pump system specifications such as entering water temperature and performance map. There is also a built in catalogue of existing U.S. heat pump system models available from several manufacturers. Further, the set of inputs include the thermal properties of the ground such as thermal conductivity, volumetric heat capacity and undisturbed ground temperature. Working fluid properties such as volumetric heat capacity, density and flow rate are also required. There is a library of standard characteristics available for different fluid types, different ground types and a map of undisturbed ground temperature for different areas in the USA. Configurations of GHEs include single borehole, lines, L-shape or grids of borehole arrays can be specified in the software. GHE details which can be specified include borehole diameter, u-tube diameter/material, and grout thermal properties.

The sizing calculation can be performed for a time period of 25 years to give the minimal depth required to meet specified minimum and maximum temperatures of the heat transfer fluid entering the heat pump. The results output file consists of the recommended borehole/boreholes length and maximum/minimum average ground temperature. If peak data is entered, the maximum and minimum temperatures under peak conditions will also be included. An output file can also be obtained from a simulation, for a specified depth, with heat rejection rate, power, maximum and minimum entering water temperatures for each month of the simulation. The major limitation of the software is that it is capable of sizing vertical systems only.

3.8.2 Earth Energy Designer (EED)

EED was first released in 1995 with promising early validation when compared with actual installations. Version 2.0 of EED has been developed as a joint project by:

• Dr. Thomas Blomberg, Building Technology Group, Massachusetts Institute of Technology, USA

• Prof. Johan Claesson, Dept. of Building Physics, Chalmers University of Technology, Sweden

• Dr. Per Eskilson, Dept. of Mathematical Physics, Lund University, Sweden

• Dr. Göran Hellström, Dept. of Mathematical Physics, Lund University, Sweden

• Dr. Burkhard Sanner, Justus-Leibig University, Germany

The software is available for $540 and uses the g- function for calculation methodology. The programming language used for the software is Delphi.

Like GHLEPro, the EED software requires input ground properties such as thermal conductivity, volumetric heat capacity, surface temperature and geothermal heat flux. A table of recommended values for different ground types is given, as are temperatures and heat flux's for selected locations in Germany, Italy, Sweden and Switzerland.

The pipe arrangements which can be considered in EED are:

• Coaxial (one tube inside another).

• Single, double or triple u-pipe(s) per borehole. For these, shank spacing that is the spacing between the two U pipes can be specified. U-pipe properties are specified in a standard library with information on diameter, thickness and thermal conductivity for a range of different types.

The borehole pattern may be chosen in EED from a database of more than 300 basic configurations including lines, L-shapes, U-shapes and rectangles. If a multiple borehole model is being used the spacing between each should be specified. The thermal resistance of the borehole is calculated through taking account of borehole geometry, grouting material and pipe material properties. Heat carrier (antifreeze / brine) thermal properties are also available in a library of seven fluids at different concentrations.

In EED calculation of fluid temperatures is estimated for defined monthly heating and cooling loads. The building heating / cooling load can either be entered as an annual figure or preferably monthly breakdown, with expected performance factor. There is also an option to specify peak heat / cool power and associated number of hours for each month for simultaneous sizing based on peak and average values. The seasonal performance factor of the heat pump is also requested.

The simulation period selected in EED and can be up to 25 years. The starting month is also selected. The output file from an EED simulation includes:

• Design data entered.

• Required length of borehole(s).

• Average monthly specific heat extraction rate

• End of month mean fluid temperatures for years 1, 2, 5, 10 and 2064.

Minimum and maximum average fluid temperature with month of occurrence for the final year of simulation can be also calculated with the EED software.

The software can design only vertical systems.

3.8.3 GCHPcalc

This programme has been developed by Kavanaugh and Rafferty at the University of Alabama. GCHPcalc has a purchase cost of $400. The program requires the user to have an understanding of the fundamentals of heat pump technology in order to utilize it fully. GCHPcalc is based on Kavanaugh's cylindrical source method. Variation in load (and the switching on/off of the heat pump) is represented by four (four hour, daily, monthly and annual) cyclic pulses of heat, representing the load of the building.

The program considers the building in terms of design thermal comfort zones. There is also an option for entering hot water requirements if these are required to be met by the heat pump system. Heat losses and gains then need to be entered for the time periods 8am - 12pm, 12pm - 4pm, 4pm –8pm, 8pm – 8am as well as the equivalent full load heating and cooling hours.

The program specifies the minimum entering water temperature allowable to the heat pump for heating (maximum for cooling). The following key inputs are required for the software: design inlet heat pump heat / cool temperatures and flow, undisturbed ground temperature, thermal conductivity and diffusivity of ground and grout, borehole diameter, number of boreholes, distance between boreholes and borehole arrangement (grid only), tube properties i.e. diameter, flow regime (laminar, transient turbulent etc), spacing within pipe (for u-pipe) and data on the heat pump which will be used. A list of standard values is available as is the option to produce average results for multi-layer ground profiles. A table of typical values for soil and rock types and thermal conductivity of fill/grout and other typical materials are also available. Heat pumps can only be selected from a standard list of fifteen North American systems.

The program output gives required bore lengths for minimal or a high rate of groundwater movement alongside a summary of the inputdesign data. Units are only in English with no option to switch to SI.

In GCHPcalc there is no means to import the data on the heat pump to be simulated. In addition, another drawback is that the units are only in English. Finally, it will size only vertical heat exchangers.

3.8.5 Ground Loop Design (GLD)

Ground Loop Design can be used to determine the lengths of borehole or pipe required for both residential and commercial projects. It can calculate the annual and lifetime energy/operating/emissions costs associated with the design.

GLD is a modular program which makes it simple to use, permits flexibility in the designing process, and customization based on designer preferences. It is capable of designing vertical GHEs, horizontal GHEs, and surface water closed loop systems. The units can be specified both in English and metric systems, and the software is available in many languages. The cost of GLD ranges from $800–$3500 (U.S), depending on the version. The programing language used in GLD are VB, C ++, and ASP.net.

The calculation methodology is based on cylindrical source model by Carslaw and Jaeger and the G-function.

The various modules in GLD are as follows:

a. GHE Modules: They have a user interface for each GHE type. Vertical Module is capable of performing peak, monthly and hourly design simulations. Both the horizontal and vertical GHE modules have featured Hybrid controls.

b. Load Modules: There are two load modules available in GLD, Average Block Load Module and Zone Manager Module. GLD has the ability to module various zones with different loads and equipment. GLD library has more than 1,000 pre-loaded heat pump which are fully modeled with automatic performance curves, operational data. The average block module is capable of monthly and 8760 hourly simulations.

c. Productivity Modules: The Fluid Dynamics module which allows to design the GHE module piping systems automatically. The GSA module is the world's most comprehensive life cycle costing tool for ground source heat pump systems. The GridBuilder allows to create a loopfiled design for buildings. The thermal conductivity module can be used for thermal conductivity testing.

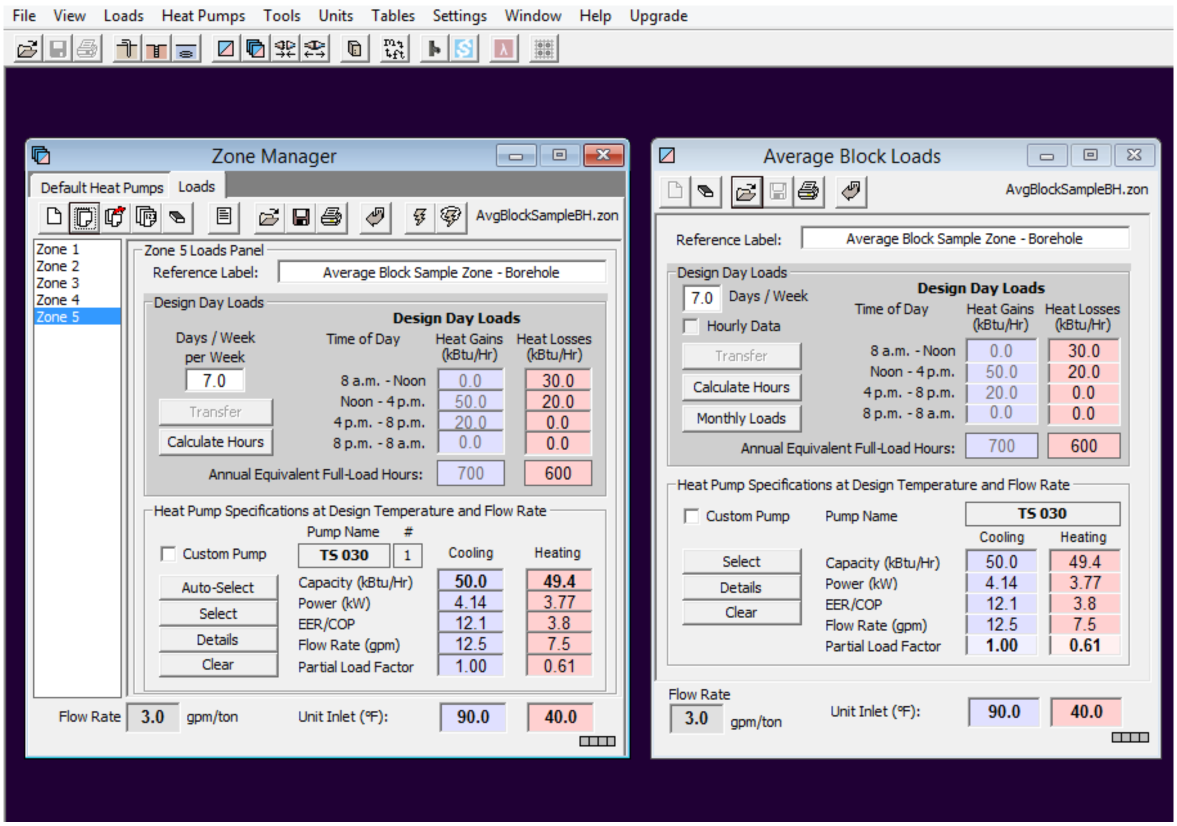


Figure 3.6 Screenshot of GLD user interface – Zone Manager Module and Average Block Module

Users can import monthly or 8,760 hourly heating and cooling HVAC loads profiles from csv files or a range of proprietary program formats. Several tables are included with GLD. These includes, Fluid Properties, Soil Properties, Pipe Properties and Conversions. Figures 3.6, 3.7 and 3.8 shows the GLD user interface screen shorts.

The output includes the length of the GHE, heat transferred to or from the ground, heat pump power consumption, borehole temperature, average temperature of fluid in the borehole, average exiting water temperature, average entering water temperature, and maximum entering water temperatures. The main limitation of GLD is that it cannot be used for open loop or DX geothermal systems.

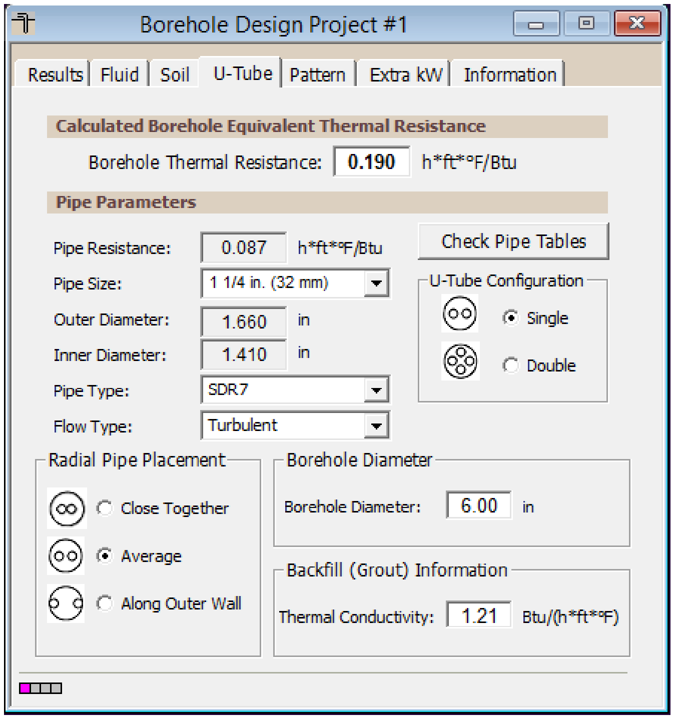


Figure 3.7 Screenshot of GLD user interface showing Borehole Design Module

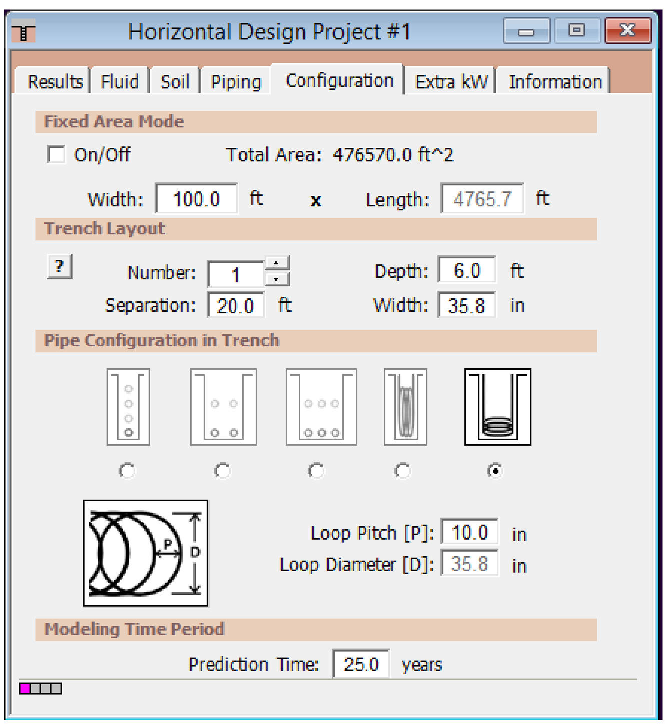


Figure 3.8 Screenshot of GLD user interface showing Horizontal Design Module

3.9 Energy Simulation Software – EnergyPlus

EnergyPlus is an energy analysis and thermal load simulation program which contains a number of innovative features, including sub hourly time steps, user-configurable modular HVAC systems that are integrated with a heat and mass balance-based zone simulation, and as input and output data structures that can facilitate third party module and interface development.

EnergyPlus is built upon the best features of DOE- 2 2 and BLAST and adds new modeling features beyond these two programs. With DOE-2's limitations in modeling emerging technologies, more people in industry are using EnergyPlus for their simulation needs. EnergyPlus is capable of performing sub-hourly calculations and integrates the load and system dynamic performance into the whole building energy. It is capable of more accurate simulation results and performs simulation as would the real building.

EnergyPlus program uses models for a water source heat pump with a ground loop heat exchanger in the whole-building GCHP annual energy simulation. The parameter estimation water-to-water heat pump model in EnergyPlus was developed by Jin and Spitler (2002). One of the major limitations of the EnergyPlus is that the heat pump model is implemented as two component models, one each for cooling and heating. The ground loop heat exchanger model is based on the short time step g-functions for a vertical borehole field model developed by Yavuzturk and Spitler (1999). Yavuzturk and Spitler extended Eskilson's g-function model to include short time steps of less than an hour. The operation of this model was verified by comparing results to analytical values (Fisher & Rees, 2005).

There are four different types of Ground Heat Exchangers which can be implemented in Energy Plus software.

1. Vertical - Ground Heat Exchanger

2. Pond - Ground Heat Exchanger

3. Surface - Ground Heat Exchanger

4. Horizontal Trench - Ground Heat Exchanger

Limitations of EnergyPlus

a. The heat pump has to be installed as two component models (for heating and cooling).A schedule is used to select the required models.

b. No integrated heat pump models available

There are no heat pump models available in the database of the EnergyPlus. The input object specifications for a heat pump must be filled in from available manufacture's catalogue data.

c. Precautions needs to be taken for calculating the g functions accurately.

To estimate the g-functions for every time-step current simulation, time is required by the model. Also, when the length and mass flow rates of the GHE are changed, the g-functions also changes. Therefore, for every simulation run, the g-functions must be recalculated. This can be achieved by using the g-function calculation dll of GLHEPro, or the GLD GLHE sizing tool.

d. Need professional Ground loop sizing tool to design ground loop heat exchanger.

EnergyPlus doesn't have design or sizing capabilities of ground heat exchangers and require third party tool such as GLHEPro and GLD.

CHAPTER 4

RESEARCH METHODOLOGY

This research compares three different types of HVAC systems, an air source heat pump (a conventional HVAC system) and a ground coupled heat pump, operating in a hot and semi-arid climatic zone for Phoenix, Arizona (see Figure 4.1). The GCHP was further subdivided into horizontal and vertical closed-loop ground source heat pump systems. EnergyPlus software was used to perform the simulation of a baseline building model that accurately represents the thermodynamics and energy performance of actual buildings. Ground Loop Design (GLD) was used to design the ground loop heat exchangers. The small DOE office (commercial) building prototype of about 5,500 sq.ft. was selected for the study. The office building consisted of 5 zones with a core zone and four perimeter zones. The office building considered for this study is smaller in size compared to the average office buildings but represents a large subset of commercial buildings.

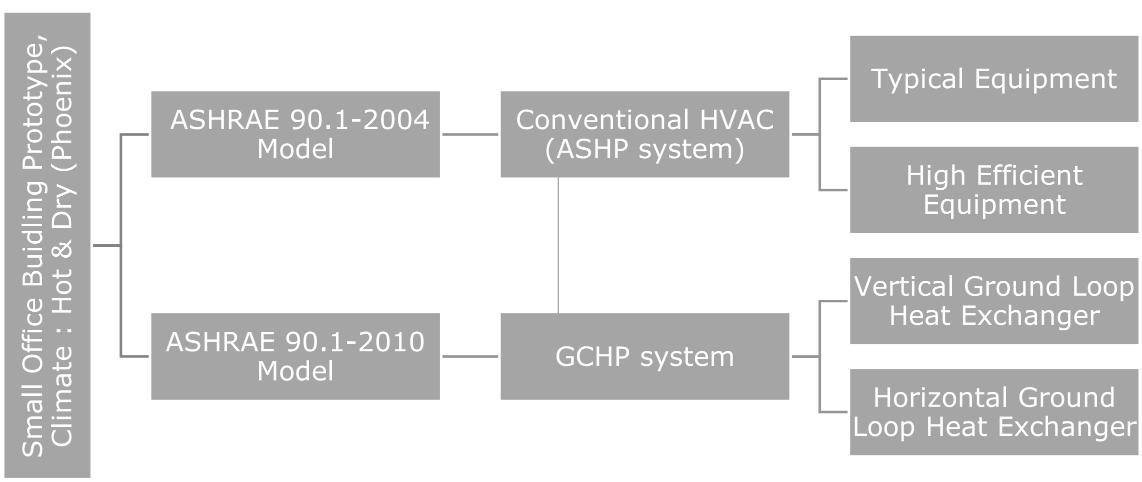


Figure 4.1: Methodology Overview of Different HVAC systems

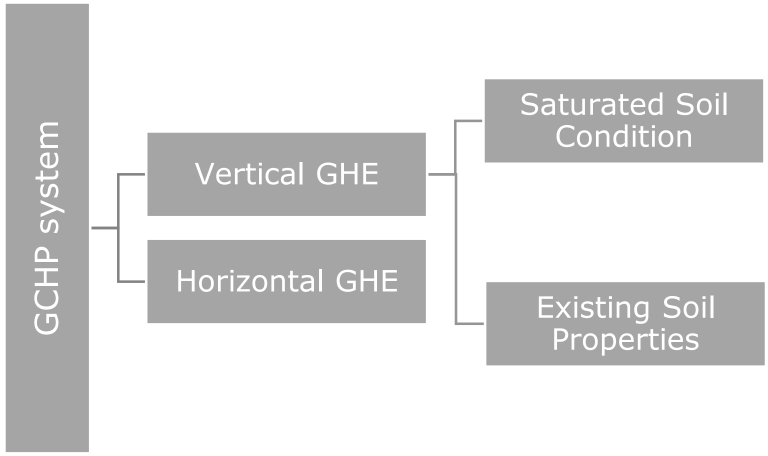


Figure 4.2: Types of parametric analysis to be conducted related to GCHPs

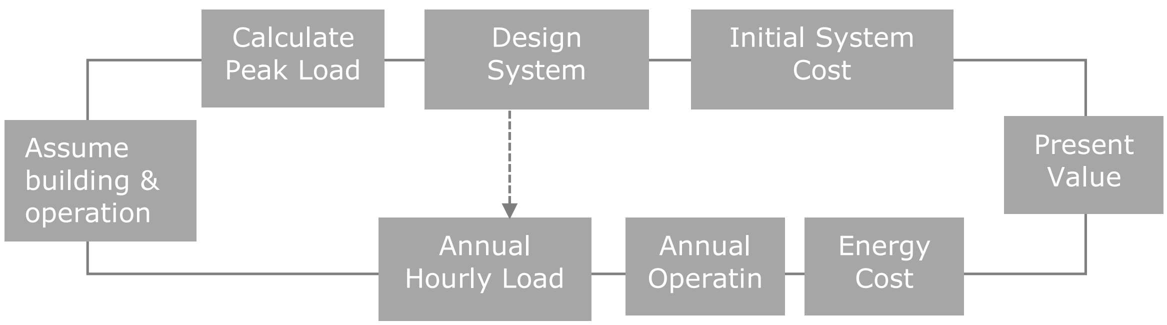


Figure 4.3: Analysis steps to be performed for each of the different system types showing in figure 4.1 & figure 4.2

The methodology considered for the research study was as follows:

1. Calculation of peak and annual hourly loads using EnergyPlus.

The EnergyPlus energy modeling tool (version 8.0.0 using DOE2.2 engine) was used to run hourly simulations covering the whole year to study the energy use. Two cases, the ASHRAE 90.1-2004 building model and the ASHRAE 90.1-2010 building model, were selected for the research. The principal features of the baseline building models were compliance with ASHRAE 90.1-2004 and 90.1-2010 standards. The peak and annual hourly loads were then simulated using EnergyPlus.

2. Simulation of annual performance and energy consumption of conventional HVAC systems using EnergyPlus.

The baseline model having a conventional HVAC system Air Source Heat Pump was simulated to calculate the annual energy consumption using EnergyPlus. This was used to compare the annual performances of the ASHP with a GCHP.

3. Designing Vertical and Horizontal Ground Loop Heat Exchanger Using Ground Loop Design (GLD) software.

The annual hourly load calculated in EnergyPlus was fed as input in the GLD software. Using the average block load module and the horizontal and vertical GHE module, the GHE for both horizontal and vertical systems were designed and sized (see Figure 4.3). The best configuration in terms of energy saving was employed for the horizontal ground loop heat exchangers. Further, the heat pump energy consumption was simulated and the GHE was designed using GLD software. The monthly and hourly energy consumption could be calculated only for V-GCHPs as there are no models available in the software to do the H-GCHPs calculations on a monthly or hourly basis. The parametric runs were performed considering existing soil conditions and saturated soil conditions (see Figure 4.2). The thermal properties of existing soil conditions in the Phoenix area were established according to soil testing performed by Geothermal Resource Technology Inc. (GRTI) at the Desert View Elementary School at 8621North 3rd Street in Phoenix, Arizona. Three years (2008, 2009 and 2010) of test reports were available and average values of the three tests were used for the analysis (see Table 4.1). The calculated average of the thermal soil properties were; formation thermal conductivity = 0.89 Btu/hr-ft-°F and formation thermal diffusivity = 0.61 ft^2/day

Table 4.1 Thermal Soil Properties at the Desert View Elementary School in Phoenix (GRTI, 2010)

Test Month & Year	Thermal Conductivity (Btu/hr-ft-°F)	Thermal Diffusivity (ft^2/day)
April 2009	0.79	0.54
April 2010	1.05	0.73
January 2012	0.82	0.55
Average	0.89	0.61

Table 4.2: Drill log from the Desert View Elementary School in Phoenix (GRTI,2010)

Description	Depth
Decomposed granite w/clay	0'-210'
Decomposed granite	210'-230'
Clay	230'-240'
Decomposed granite w/clay	240'-260'
Decomposed granite	260'-300'
Granite	300'- 402'

The saturated soil thermal properties considered in the analysis are thermal conductivity (k) = 1.8 BTU/ft/hr/°F and diffusivity = 1.23 ft^2/day (Mitchell, 1976)

4. Comparison and analysis of energy consumption of the HVAC systems

5. Analysis of the GCHPs performance due to the saturated soil conditions

6. Life Cycle Cost Analysis

Analysis of life cycle costs was conducted for both the conventional and GCHP systems. The spreadsheet developed by M.S. Addison and Associates, Tempe, AZ was used for the LCC analysis. The calculation procedure used in the spreadsheet was based on the methodology recommended by DOE's Federal Energy Management

Program (FEMP) and the National Institute of Standards and Technology (NIST). Both the alternatives, with and without incentives for ASHRAE 90.1-2004 and ASHRAE 90.1-2010 models were considered for life cycle cost. The installation cost for vertical GCHPs and ASHPs were obtained from contractors Mr. Jay Egg, (owner of Egg systems) and Mr. Benjamin Carbonell (owner and operator of Grace Air LCC) respectively. The first cost for H-GCHPs were not available and were estimated assuming $4200/ton for a slinky configuration (Rafferty, 2008). The inflation rates were calculated using the US Inflation Calculator (see Figure 4.4). To estimate the initial cost of the H-GCHP under saturated conditions, the first cost for a H-GCHP was calculated for existing soil conditions. Then the cost of heat pump was deducted from the first cost to give the horizontal slinky installation and pipe cost. Further, the horizontal slinky installation and pipe cost per sq.ft was determined to calculate the initial cost of the H-GCHPs in saturated soil condition.

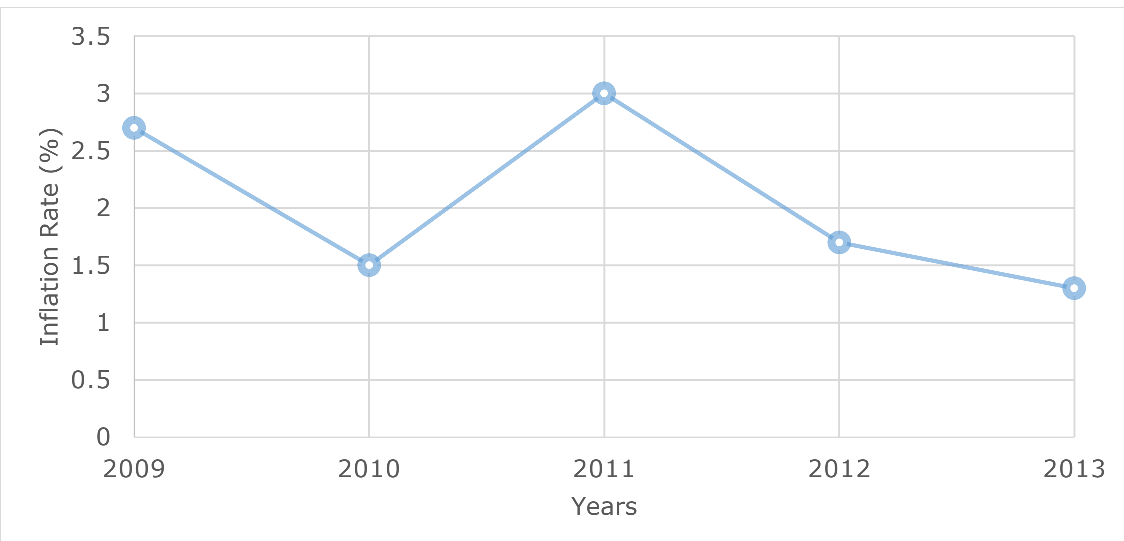


Table 4.4: Annual Inflation Rates (2009-2013) (US Inflation Calculator)

Utility rate schedules were obtained from Arizona Public Service (APS) for year 2013 in order to estimate annual energy costs. Commercial electric rates from APS were used in the cost analysis included a $0.672 daily customer charge, in addition to winter and summer energy charges. In winter, these charges are $0.08718/kWh for the first 200 kWh per month and $0.4638/kWh for all additional use. In summer, the energy charges are $0.10337/kWh and $0.06257 for all additional use. Summer rates apply from May 1 through October 31.

7. Greenhouse Gas Analysis

Greenhouse gas analysis was conducted to estimate the possible reduction in greenhouse gas emissions by using geothermal heat pump. RetScreen clean energy project analysis software (NRC, 2012) was used to estimate the reduction in greenhouse gas emissions such as carbon dioxide (CO2), methane (CH4), and nitrous oxides (N2O) through the use of a GCHP system.

8. Sensitivity analysis

The main purpose of the sensitivity analysis was to quantify the uncertainty in the V-GCHPs cost estimates and explore which assumptions were critical. The installation costs, annual utility costs, and periodic costs of the GCHP system were varied from -80% to +80% for V-GCHPS and 40% to -40% for H-GCHPs in ASHRAE 90.1-2010 model.

9. Calculation of maximum remediation cost

The maximum remediation cost is calculated to determine the maximum amount to be spent on grouting material or saturated soil treatment to enhance the performance of a GCHP installed under existing conditions. The remediation cost $ per cubic feet can be estimated by taking the difference between the installation costs of GCHPs in existing soil condition and saturated soil condition and dividing by the length of GHE in latter case.

CHAPTER 5

MODELLING SPECIFICATIONS & DATA COLLECTION

5.1 Location and Climate

Phoenix (33°27' N, 112°4' W), the capital of Arizona, is located in the Salt River Valley on the northeastern reaches of the Sonoran Desert and lies at a mean elevation of about 1,100 feet. The topography of the city is mostly flat with the mountains in and around the city. Phoenix has a dry, semi-arid desert climate and the temperature ranges from extremely hot summers to mild winters. The water supply is partly from reservoirs on Salt and Verde Rivers, and partly from underground water table. Figure 5.1 illustrates the monthly dry bulb temperature and relative humidity in Phoenix.

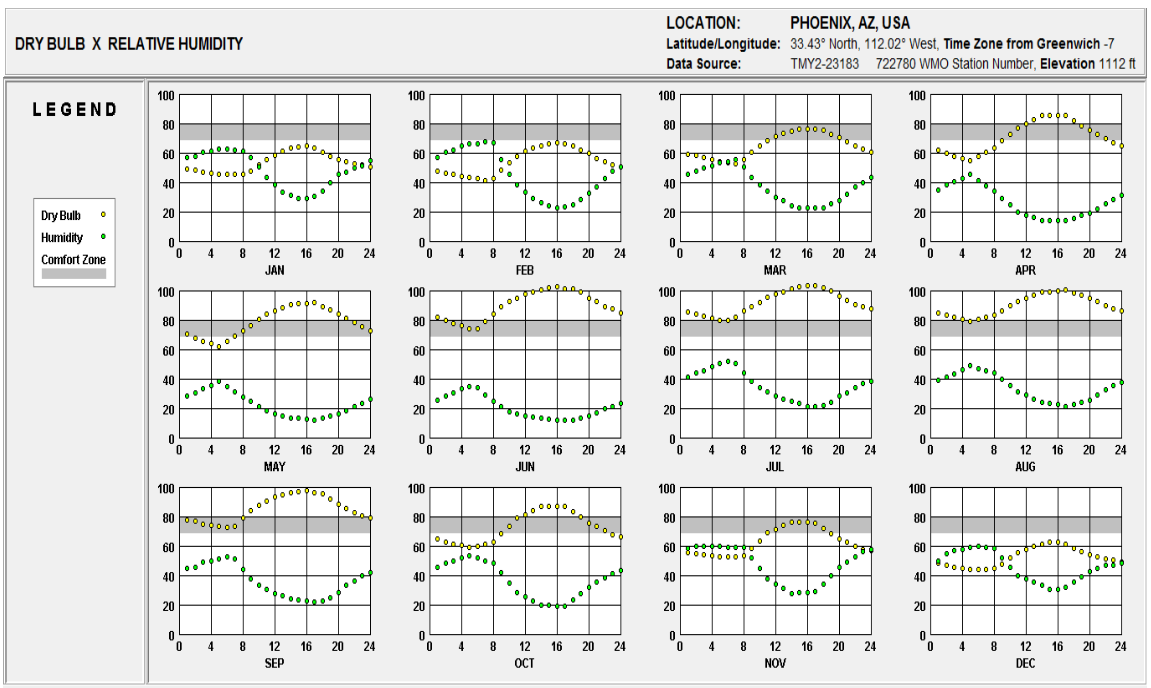


Figure 5.1: Monthly dry bulb temperature X relative humidity in Phoenix (Climate Consultant)

Many winter days in Phoenix reach over 70 °F and typical high temperatures in the middle of the winter are in the 60-70°F. The normal high temperature in summers is over 90°F from early May through early October, and over 100°F from early June through early September. Many days each summer will exceed 110°F in the afternoon.

The humidity levels in Phoenix are very low in summer, giving minimum thermal comfort. Annual precipitation is only about 7 inches, and afternoon humidity range from about 30 % in winter to only about 10 % in June.

Outdoor temperature varies significantly over the length of the day and influences the amount of heat transferred or rejected to the environment. The outdoor temperature directly affects the performance of an ASHP.

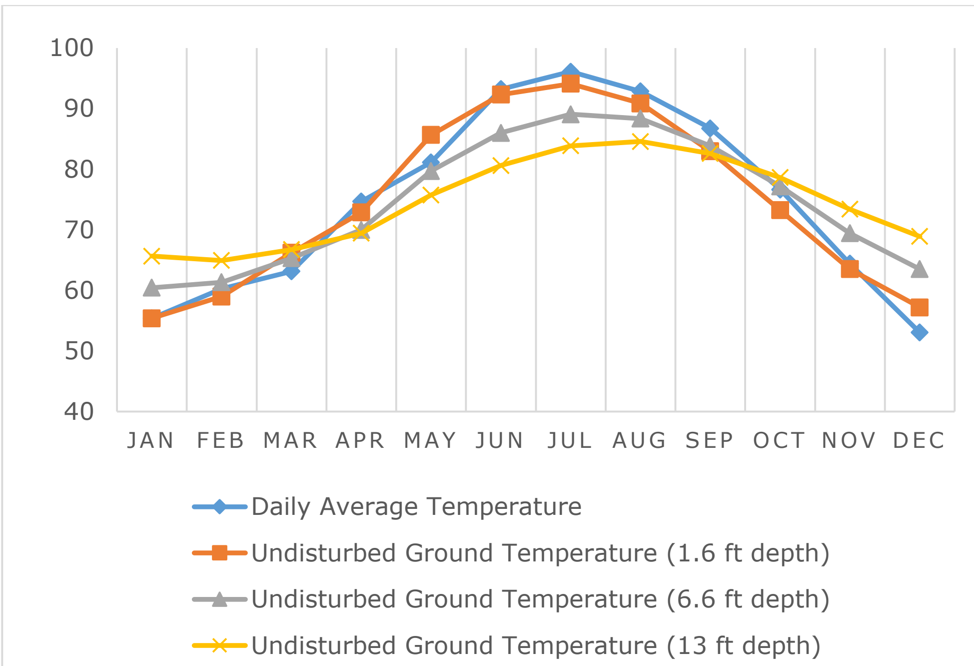


Figure 5.2: Typical Average Monthly Temperature in Phoenix

A GCHP system is more efficient then an ASHP because there is relatively small change in the ground temperature, contrary to the air temperature in Phoenix. Figure 5.2 illustrates that the ground temperature becomes more stable with depth. This is because a few meters of surface soil insulate the earth and groundwater below, minimizing the amplitude of the variation in soil temperature in comparison with the temperature in the air above the ground.

The temperature at which the controlled indoor space needs to be maintained throughout seasons is known as the indoor design temperature. It was set to a default value of 70°F in winter and around 75°F in summer for this study to estimate the annual energy consumption. For this location, EnergyPlus automatically obtains the Phoenix TMY2 weather file which is based on data from the Phoenix Sky Harbor International Airport Weather Station.

5.2 Baseline Model Specifications

As mentioned in Chapter 4, the ASHRAE Standard 90.1-2004 and 90.1-2010 prototype building models developed by Pacific Northwest National Laboratory were considered for this research. The total air-conditioned area is about 5,500 sq.ft, and it is distributed among 5 zones: the core zone and 4 perimeter zones. It is assumed that there is no structure or landscape element casting any shadow on the building or changing the micro climate of the site. Appendix A specifies the details of the ASHRAE 90.1-2004 & ASHRAE 90.1-2010 Baseline model. The conventional system that is implemented in the building prototype consisted of packaged single zone ASHPs with a gas furnace. The gas furnace in both the Baseline Cases is changed into supplementary electric heating.

Table 5.1: Characteristics of Small Building Prototype (DOE 2009)

Item	Description
Construction	Small Office 1 Story ~5,500 sq. ft. Conditioned Space
Occupancy	31 Occupants

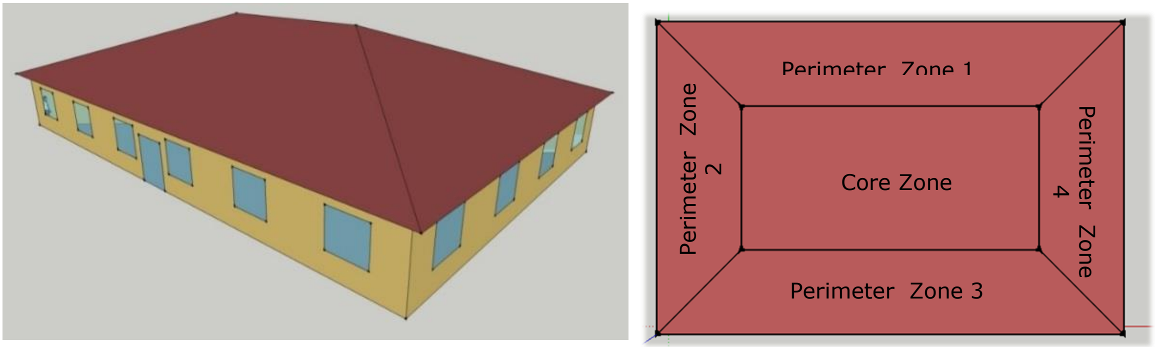


Figure 5.3 – Baseline model ASHRAE 90.1-2004 & ASHRAE 90.1-2010 on Sketch-Up representing (a) building geometry (b) zones (DOE, 2010)

Table 5.2: Competing Residential Space-Conditioning Technologies (EIA, 2010)

Technology	Rated Cooling Efficiencies	Rated Heating Efficiencies	Typical Installation Cost ($/ton)
Gas-Fired Furnace	--	Typical: 80% AFUE; 430 kWh/yr ENERGY STAR®: 90% AFUE; 371 kWh/yr 2010 Best Available: 98% AFUE; 340 kWh/yr	2000-3000 2500-3000 3500-4000
Central A/C (Air Source)	Typical: 13.7 SEER ENERGY STAR®: 14.5 SEER Best Available: 24 SEER	--	2550- 4750 3000- 5200 7000- 14500
Central Heat Pump (Air Source)	Typical: 14 SEER ENERGY STAR®: 14.5 SEER Best Available: 22 SEER	Typical: 8 HSPF ENERGY STAR®: 8.2 HSPF 2010 Best Available: 10.7 HSPFb	4400- 5500 5900- 7000 7400- 8500
GSHP	Typical: 13.4 EER ENERGY STAR®: 16.1 EER Best Available: 30 EER	Typical: 3.1 COP ENERGY STAR®: 3.5 COP 2007 Best Available: 5.0 COP	9000- 11000 10000- 12000 12000- 14000

Table 5.3: Competing Commercial Space-Conditioning Technologies (EIA, 2010)

Technology	Rated Cooling Efficiencies	Rated Heating Efficiencies	Typical Installation Cost ($/ton)
Gas-Fired Furnac ce	--	Typical: 80% thermal High Efficiency: 82% thermal	3050- 3275 3275- 3625
Roof-Top Air Conditioner	Typical: 11.2 EER Mid-range: 12 EER High Efficiency: 13.9 EER	--	7500- 8500 9000- 10000 22000- 24000
Roof Top Heat Pump	Typical: 11 EER High Efficiency: 12 EER	Typical: 3.3 COP High Efficiency: 3.4 COP	6500- 7300 7900- 9500
GSHP	Typical: 14 EER High Efficiency: 27.8 EER	Typical: 3.5 COP High Efficiency: 4.9 COP	14000- 15000 17000- 20000

Table 5.4 EER/COP of the Heat Pumps

Case	Description	EER/COP
ASHRAE 90.1-2004:Baseline model	Existing ASHP	10.7/3.00
Alternative Case 1	High Efficiency ASHP	16.5/3.6
Alternative Case 2/3/4/5	V-GCHPs – Existing Soil	17.1/3.6
Alternative Case 6/7/8/9	V-GCHPs – Saturated Soil	17.1/3.6
ASHRAE 90.1-2010: Baseline model	Existing ASHP	11.01/3.3
Alternative Case 1	High Efficiency ASHP	16.5/3.6
Alternative Case 2/3/4/5	V-GCHPs – Existing Soil	17.1/3.6
Alternative Case 6/7/8/9	V-GCHPs – Saturated Soil	17.1/3.6

The peak load values for ASHRAE 90.1-2004 model in cooling and heating modes were sized at 87.7kBtu/hr and 39.6 kBtu/hr respectively and for ASHRAE 90.1-2010 model were 82.1 kBtu/hr and 48 kBtu/hr respectively. Tables 5.2 & 5.3 illustrates the residential and commercial space-conditioning technologies. Table 5.4 shows the EER/COP of the heat pumps used in the alternative cases. The maximum efficiency of an available ASHPs equipment for commercial application is 11.7 EER and for residential application is 16.5 EER (Grace Aire, LLC). Since small office buildings have cooling and heating loads similar to large residential buildings, 16.5 EER was considered for high efficient ASHP equipment.

5.3 Alternative Models Description

To understand the benefits associated specifically with coupling a heat pump to the ground, the potential impacts of GCHPs were compared to those with typical and high efficient ASHPs. The two baseline cases were compared with three other systems namely Advanced ASHPs, Vertical GCHPS and Horizontal

GCHPs. The following proposed cases were compared with the two ASHEAE 90.1-2004 and ASHRAE 90.1-2010 baseline models:

1. Alternative Case 1 – Advanced ASHP
2. Alternative Case 2 – Vertical GCHPs without federal incentives
3. Alternative Case 3 – Vertical GCHPs with federal incentives
4. Alternative Case 4 – Vertical GCHPs saturated soil condition without federal incentives
5. Alternative Case 5 – Vertical GCHPs saturated soil condition with federal incentives
6. Alternative Case 6 – Horizontal GCHPs without federal incentives
7. Alternative Case 7 – Horizontal GCHPs with federal incentives
8. Alternative Case 8 – Horizontal GCHPs saturated soil condition without federal incentives
9. Alternative Case 9 – Horizontal GCHPs saturated soil condition with federal incentives.

5.3.1 Alternative Case: High Efficiency ASHP system

The alternative case 1 model was developed by replacing the ASHP of of 10.7/3 EER/COP (in ASHRAE 90.1-2004 model) and 11.1/3.3 EER/COP (in ASHRAE 90.1-2010 model) with high efficient equipment of 16.5/3.6 EER/COP keeping same rest of the building properties. Table 5.4 shows the EER/COP of the heat pump equipment used in this study.

5.3.2 Alternative Case: Vertical GCHP System

The vertical GHE system designed consists of 16 boreholes. Table 5.5 illustrates the specifications of the vertical GHE for the two cases. The depth of the borehole varied depending upon the space cooling and heating loads of ASHRAE 90.1-2004 &

ASHRAE 90.1-2010 models. A single 1 ¼ inch diameter HDPE U shaped return pipe coupled at the bottom were proposed for vertical GHE with the borehole diameter of about 6 inches. The GHEs were designed 20 feet apart, covering about 3,600 sq.ft of land. These boreholes were proposed to be installed in the parking lot area.

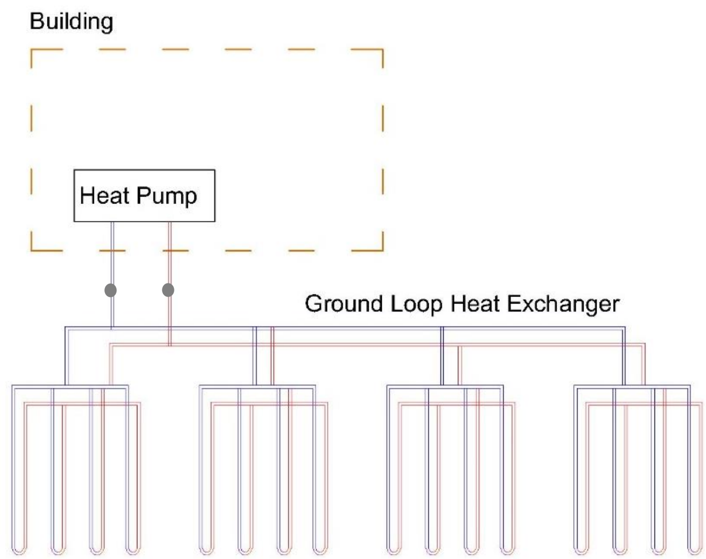


Figure 5.4 Schematic diagram of Vertical GCHPs

To secure and seal the U tube the borehole was considered to be filled with concrete and 50% quartz sand grout having a relatively high thermal conductivity of 1.1-1.7 BTU/ hr ft °F (see Table 3.6). Figure 5.4 shows the schematic diagram of the vertical GCHPs. In the heating mode the GHE will serve as a heat source and in the cooling mode as a heat sink.

Table 5.5: Description of Vertical Ground Coupled Heat Pump System

	Description	ASHRAE90.1-2004	ASHRAE90.1-2010
1.	Fluid		
1.1	design heat pump inlet fluid temperature – cooling	93 °F	93 °F
1.2	design heat pump inlet fluid temperature – heating	63 °F	63 °F
1.3	design system flow rate	3 gpm/ton	3 gpm/ton

	1.4	Fluid type	Pure water	Pure water
	1.5	specific heat of the fluid	1 Btu/(°F*lbm)	1 Btu/(°F*lbm)
	1.6	Density	62 lb/ft^3	62 lb/ft^3
2.		Soil		
	2.1	undisturbed ground	73 °F	73 °F
	2.2	soil thermal conductivity	0.89 Btu/hr*ft*°F	0.89 Btu/hr*ft*°F
	2.3	soil thermal diffusivity	0.61 ft^2/day	0.61 ft^2/day
	2.4	Saturated Soil conductivity	1.8 Btu/hr*ft*°F	1.8 Btu/hr*ft*°F
	2.5	Saturated Soil Thermal Diffusivity	1.23 ft^2/day	1.23 ft^2/day
	2.6	modelling time period	25 years	25 years
3.		U Tube		
	3.1	Borehole thermal resistance	Automatically calculated	Automatically calculated
		Pipe parameters		
	3.2	Pipe resistance	0.087	0.087
	3.3	Pipe size	1 ¼ "	1 ¼ "
	3.4	Pipe type	SDR 7	SDR 7
	3.5	Pipe flow	Turbulent	Turbulent
	3.6	U tube configuration	Single	Single
	3.7	Radial pipe placement	Average Placed	Average Placed
	3.8	Borehole diameter	6”	6”
	3.9	Backfill grout thermal conductivity	1.4 Btu/h*ft*°F (Concrete w 50% quartz sand)	1.4 Btu/h*ft*°F (Concrete w 50% quartz sand)
4.		Pattern		
		Vertical grid arrangement		
	4.1	Rows across	4	4
	4.2	Rows down	4	4
	4.3	Borehole separation	20 ft	20 ft
	4.4	Boreholes per parallel circuit	3	3
	4.5	Borehole length	185 ft	171 ft
	4.6	Total borehole length	2954.1 ft	2729.5 ft

Following is the brief description of the ASHRAE Borehole Design guidelines (Kavanaugh and Rafferty, 1997) used to design GCHPs:

a. For pure water, the minimum liquid flow of 2gpm for ¾ inch – 1 ¼ inch.

b. There are short circult heat losses between the upward and downward flow of fluid in the U tube, which is about 4% when the liquid flow rate is 3 gpm per ton.

c. Selecting the EWT is very critical. Values near the undisturbed ground temperature (UGT) will have higher system efficiency. However such EWT values will increase the required bore length and the first cost. On the other hand, assuming a EWT value far away from the UGT reduces the system performance. A rule of thumb is that the entering water temperature (t_{wi}) should be 20°F to 30°F above UGT (t_g) and 10 ° F - 20°F below for heating.

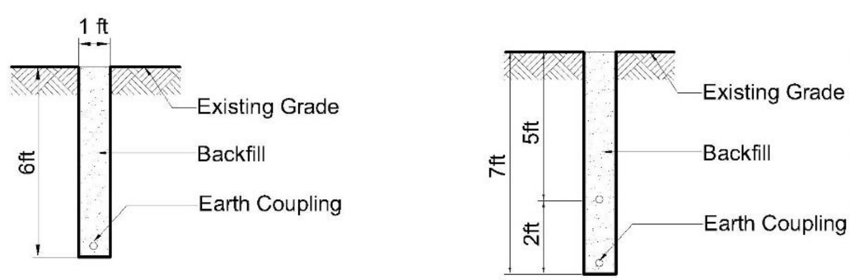
- $T_{wi,min} = T_{g,min}$-15°F (minimum design entering water temperature)
- $T_{wi,max}$ = min($T_{g,max}$+20°F, 110°F) (maximum design entering water temperature) (RETscreen, 2005)

d. Larger diameter tubes such as 1 ½ inch pipes have various advantages over smaller ones (e.g. ¾ inch). The larger tubes have lower pressure drops. Larger tubes also require shorter borehole lengths, saving in drilling cost and land, and they are more durable. Unfortunatey not many contractors are equipped for such installation, and hence installing larger diameter tubes can be expensive. This makes smaller diameter pipes easy to install and cheaper. For smaller diameter pipes, about 15% longer borehole depths are required than the 1 ½ " pipe. The boreholes can be maximum of 250 feet due to the head losses. The 1 inch and 1 ¼ inch diameter pipes are available as comprises and the latter pipe diameter was used to design V-GCHPs.

5.3.3 Alternative Case – Horizontal GCHP System

The horizontal GHE were designed following guidelines similar to those for the vertical GHE. To study the various horizontal GHE configurations, the horizontal GCHPs were designed considering the ASHRAE 90.1-2004 model. Figure 5.4 illustrates the six different horizontal pipe configurations in trench which were designed. In alternative 1, a single pipe is laid in a 1 foot wide and 6 feet deep trench. In alternative 2, two pipes 2 feet apart are laid vertically in a trench of width 1 foot and 7 feet deep, whereas alternative 3 has 2 pipes placed in a single trench but horizontally 2 feet apart. Alternative 4 has 4 pipes in a single trench of width 2 feet by 7 feet (see figure 5.5). The last two alternatives 5 & 6 are coiled or slinky pipe configurations placed vertically and horizontally respectively in 6 ft deep trench.

Table 5.7 illustrates the sizes of the six horizontal GHEs. The total trench length required by a single pipe configuration is the largest, at about 4,654.7 feet and the horizontal slinky pipe configuration has the shortest trench length of about 828 feet, which is about 82 % less than single pipe. From Table 5.6 it can be observed that as the number of pipes increases in the trench the total trench length decreases from Alt 1: single pipe (4,654.7 ft) to Alt 2: two pipes (2,697 ft or 42% less than the former) to Alt 4: four pipe (1,688.4 ft or 37.4% less than the former) to Alt 5: vertical and Alt 6: horizontal slinky pipe configuration which are about 922 ft (or 45.4% less than the previous pipe configuration) & (851 ft or 49.6% less than the four pipe configuration) respectively.



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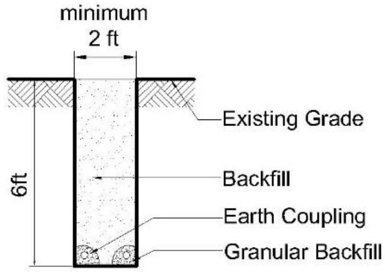

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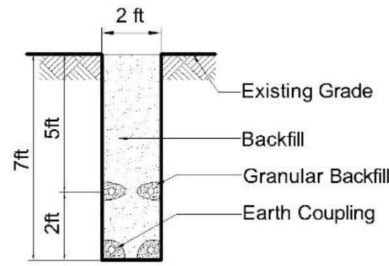

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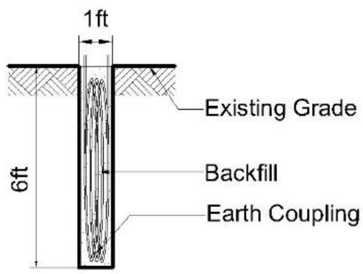

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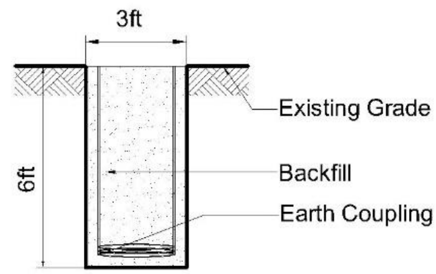

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a. Single pipe -designed

b. Stacked two-pipe -designed

c. Parallel two-pipe –designed

d. Stacked parallel four-pipe –designed

e. Coiled pipe laid vertically in a narrow trench – designed

f. Coiled pipe laid horizontally in a wide trench

Figure 5.4: Various configurations considered for horizontal GHE design

For the LCCA study, Alt 6: horizontal slinky pipe was considered as it requires less area as compared to rest of the configurations. While this reduces the amount of land used, it requires more pipe, which results in additional costs. The total area required for the horizontal GHE was about 13,248 sq.ft. The trench was considered to be backfilled with the existing soil.

Table 5.6: Description of Horizontal Ground Coupled Heat Pump System Configuration

Sr no	Parameters	Alt-1: Pipe/ trench	Alt 2: 2 pipe/ trench (Vertical)	Alt 3: 2pipe /trench (Horizontal)	Alt 4: 4pipe /trench	Alt 5: Vertical slinky	Alt 6: Horizontal Slinky	Units
1.	Total Trench Length	4654.7	2697	2708.1	1688.4	922	851	ft
2.	Total Width	80	80	80	80	80	80	ft
3.	Length	931	339.4	541.6	337.7	190	165.6	ft
	Trench Layout							
4.	Number	5	5	5	5	5	5	
5.	Depth	6	7	6	7	6	6	ft
6.	Separation	20	20	20	20	20	20	ft
7.	Width	12	12	24	24	8	35.8	in
	Pipe configuration in trench							
8.	No of pipes	1	2	2	4	-	-	
9.	Vertical separation	-	24	-	24	-	-	in
10.	Horizontal separation	-	-	24	24	-	-	in
11	Loop Pitch	-	-	-	-	9.8	10	in
12	Loop Dia	-	-	-	-	35.8	35.8	in

5.4 Validation of Vertical GHE Design

The vertical GHE designed using GLD was partially validated using the theoretical methodology explained in chapter 4, section 3.4.2. Both ASHRAE 90.1-2004 and ASHRAE 90.1-2010 vertical GHE (existing soil properties) were validated (see Appendix B). For ASHRAE 90.1-2004 case, the GLD software estimated the vertical GHE length for cooling to be 2954.1 feet whereas the length calculated by the theoretical method was 3020.35 feet (2.23%). Similarly for ASHRAE 90.1-2010 case, GLD estimated 2,729.5 feet length for cooling and the length determined by the theoretical method was 2,804.25 feet (2.73%).

The designed vertical GHE shows good agreement with the theoretical method within ±3.00% and with minor discrepancies due to approximations, such as borehole resistance, three heat pulses, and considering only maximum block cooling and heating loads in the theoretical methodology. Rest all the parameters used in the theoretical method were the same as used for designing GHE with GLD software. Appendix B gives the detailed description of the calculations and parameters used for theoretical validation.

CHAPTER 6

ANALYSIS OF SIMULATION RESULTS

6.1 Comparison of Energy Performance between Conventional system and GCHP systems

6.1.1 Analysis Based on ASHRAE 90.1-2004 Model

According to the ASHRAE 90.1-2004 model, simulation results using EnergyPlus shows that the baseline model consumes 68,333 kWh energy annually. The total energy consumed can be divided into end-use categories of equipment, area lighting, and space heating and cooling. Figure 6.1 and Table 6.1 show that the lighting dominates (26,849 kWh or 40%) for the major share in the building's energy consumption, followed by space cooling (14,808 kWh or 22%), equipment (14,666 kWh or 22%) and ventilation fans and pumps (8,457 kWh or 12%). This indicate that internal loads dominates, which is common for this building type. Since the building is in cooling dominated area like Phoenix, space heating accounts only for 1% of the building's total energy consumption.

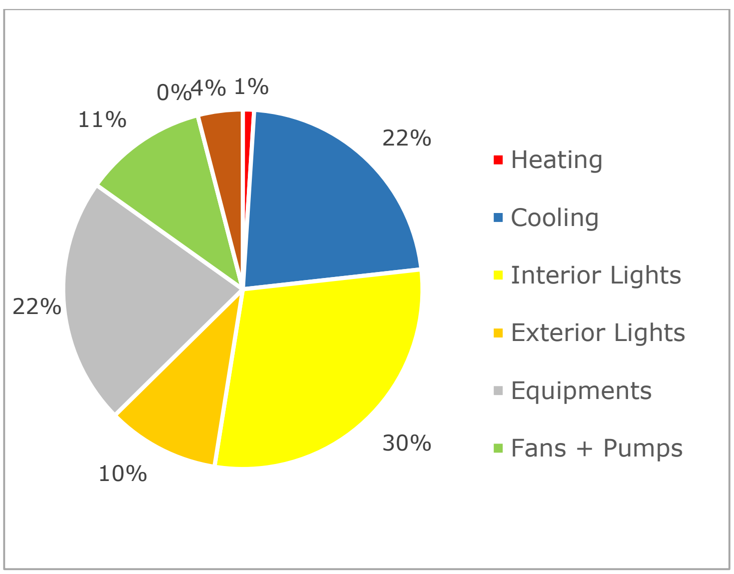

Figure 6.1 ASHRAE 90.1-2004: Baseline model predicted Annual Energy Consumption by End Use

The HVAC system including space cooling, heating, ventilation fans and pumps consumes 24,159 kWh or 34 % of the building's total energy consumption. The majority of the HVAC electricity consumption is space cooling accounting for 61% of the total HVAC consumption. Other components include pumps and ventilation fans, and space heating that consume 35 % and 4 % respectively. Further, Figure 6.2 illustrates that the HVAC system consumes less energy in the heating months as compared to the cooling months.

Table 6.1 ASHRAE 90.1-2004: Baseline model monthly energy consumption by end use categories in kWh

Cate-gory	End Use	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Lighting	Internal Lighting	1,657	1,498	1,744	1,589	1,700	1,677	1,613	1,744	1,589	1,657	1,633	1,613	19,712
	External Lighting	706	602	617	546	520	484	507	544	575	646	671	718	7137
	Total Lighting	2,362	2,100	2,361	2,135	2,221	2,160	2,120	2,288	2,164	2,303	2,304	2,331	26,849
	% of Total	47%	46%	45%	42%	39%	32%	31%	32%	35%	42%	45%	47%	40%
Equip-ment	Equipment	1,232	1,114	1,300	1,181	1,266	1,249	1,198	1,300	1,181	1,232	1,215	1,198	14,666
	% of Total	24%	24%	24%	23%	22%	19%	17%	18%	19%	23%	24%	24%	22%
HVAC	HVAC Cooling	277	393	522	887	1264	2323	2606	2616	1878	1195	621	224	14,808
	HVAC Heating	247	146	96	29	0	0	0	0	0	1	36	338	894
	HVAC Fan & Pump	695	629	761	662	731	751	706	785	679	702	695	662	8,457
	HVAC	1,219	1,169	1,379	1,577	1,996	3,074	3,313	3,401	2,558	1,898	1,353	1,224	24,159
	% of Total	24%	25%	26%	31%	35%	46%	48%	47%	41%	35%	26%	24%	34%
Others	Others	251	225	250	220	218	202	197	206	202	216	231	241	2,659
	% of Total	5%	5%	5%	4%	4%	3%	3%	3%	3%	4%	5%	5%	4%
Total		5,064	4,608	5,290	5,114	5,701	6,685	6,827	7,196	6,105	5,448	5,103	4,994	68,333

The highest energy consumption takes place in the month of August whereas February consumes the least energy. The end use categories such as equipment, lighting are relatively constant and vary very little throughout the year. The energy variation in these categories differs based on change in occupancy schedule whereas the space heating and space cooling are influenced by the weather conditions.

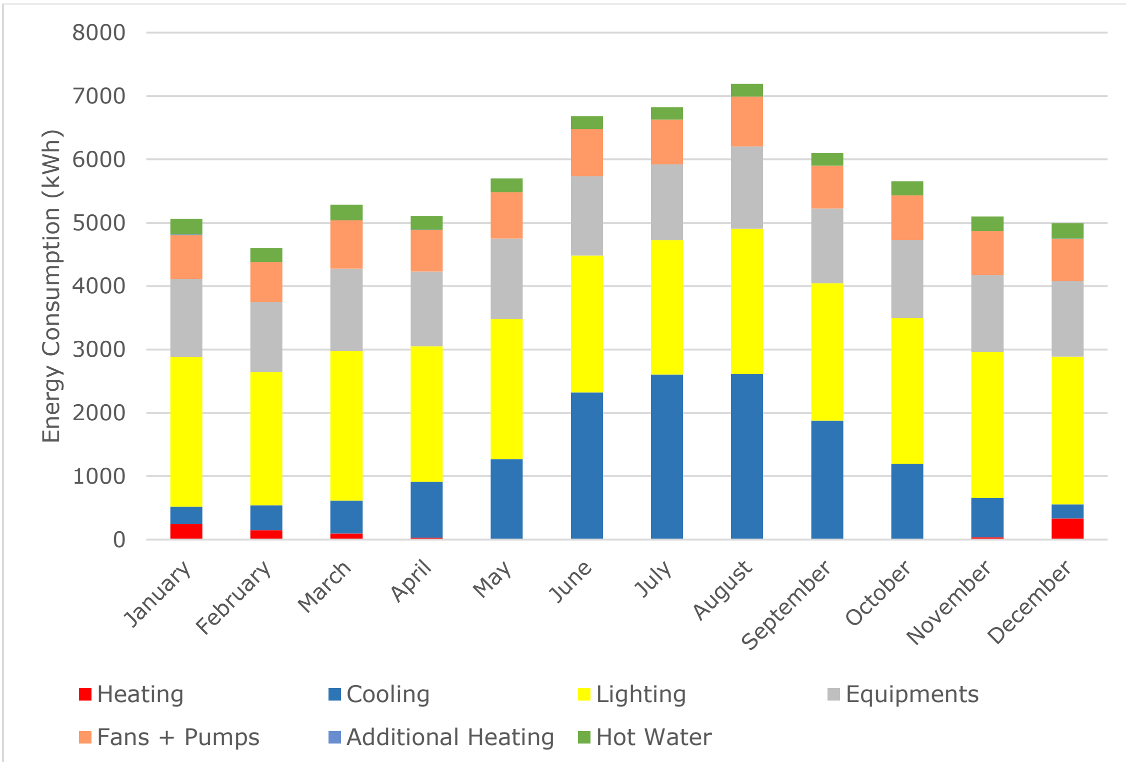

Figure 6.2 ASHRAE 90.1-2004: Baseline Monthly Energy Consumption by End-use category

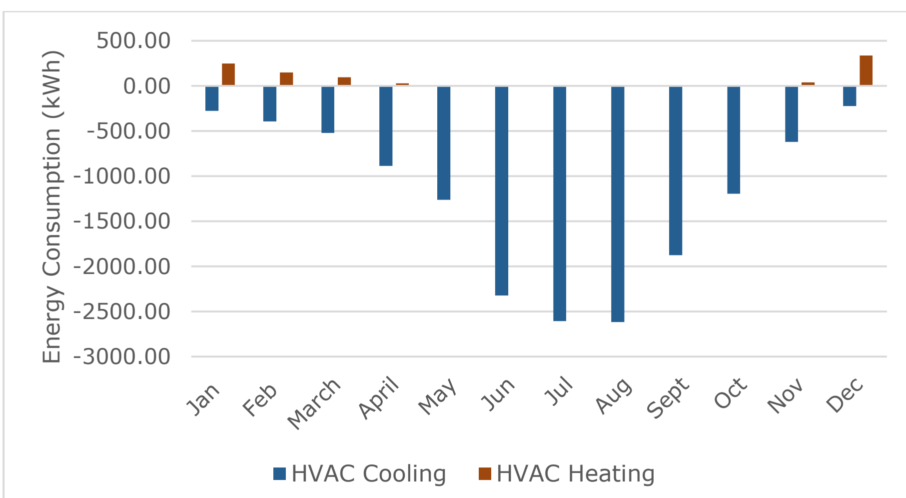

Figure 6.3 ASHRAE 90.1-2004-Baseline: Monthly Energy Consumption by space cooling and space heating

6.1.2 ASHRAE 90.1-2004 - Alternative Models: High Efficiency ASHPs and V-GCHPs

Like ASHPs, GCHPs also use electricity as system energy source.

Table 6.2: ASHRAE 90.1-2004 Model-Monthly energy consumption in kW from baseline model and alternative models –Efficient ASHP & Vertical GCHPs

Category		Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Lighting	Baseline	2,362	2,100	2,361	2,135	2,221	2,160	2,120	2,288	2,164	2,303	2,304	2,331	26849
	High Eff ASHP	2,362	2,100	2,361	2,135	2,221	2,160	2,120	2,288	2,164	2,303	2,304	2,331	26849
	Savings	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0
		0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
	V GCHPs	2,362	2,100	2,361	2,135	2,221	2,160	2,120	2,288	2,164	2,303	2,304	2,331	26849
	Savings	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
		0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
Equipment	Baseline	1,232	1,114	1,300	1,181	1,266	1,249	1,198	1,300	1,181	1,232	1,215	1,198	14666
	High Eff ASHP	1,232	1,114	1,300	1,181	1,266	1,249	1,198	1,300	1,181	1,232	1,215	1,198	14666
	Savings	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
		0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
	V GCHPs	1,232	1,114	1,300	1,181	1,266	1,249	1,198	1,300	1,181	1,232	1,215	1,198	14666
	Savings	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00		0.00	0.00
		0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
HVAC	Baseline	1,219	1,169	1,379	1,577	1,996	3,074	3,313	3,401	2,558	1,898	1,353	1,224	24159
	High Eff ASHP	1,097	1,014	1,179	1,254	1,540	2,236	2,373	2,458	1,880	1,466	1,127	1,118	18741
	Savings	122	155	200	323	456	838	940	944	678	431	226	106	5418
		10%	13%	15%	20%	23%	27%	28%	28%	26%	23%	17%	9%	22%
	V GCHPs	881	837	1,039	1,016	1,206	1,583	1,652	1,770	1,456	1,217	1,015	818	14490
	Savings	338	332	340	562	790	1491	1661	1631	1101	681	337	405	9669
		28%	28%	25%	36%	40%	48%	50%	48%	43%	36%	25%	33%	40%
Others	Baseline	251	225	250	220	218	202	197	206	202	216	231	241	2,659
	High Eff ASHP	251	225	250	220	218	202	197	206	202	216	231	241	2,659
	Savings	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
		0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
	Proposed V GCHPs	251	225	250	220	218	202	197	206	202	216	231	241	2,659
	Savings	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
		0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%

From Table 6.2, we can see the high efficient ASHPs can save up-to 5,418 kWh or 22% and V-GCHPs can save 9,669 kWh or 40 % of the energy consumed by the existing ASHP and indicate that all energy savings comes from the HVAC end-use category. It is noticed that as the cooling load increases in the months of June, July and August, the energy consumption savings increase for both vertical GCHPs (by 48-50%) and high efficient ASHPs (by 26-28%).

6.1.3 Analysis Based on ASHRAE 90.1-2010 Model

The ASHRAE 90.1-2010 model is based on ASHRAE 90.1 2010 standards. The baseline model consumes 57,367 kWh energy annually that is about 16 % less than ASHRAE 90.1-2004: Baseline model. This shows that the ASHRAE 90.1 2010 model has greater energy efficiency than the ASHRAE 90.1 2004 model as the former model was updated with more efficient technologies including, but not limited to, better heat pump equipment, more efficient lighting, equipment, better building envelope. Figure 6.4 illustrates that the lighting consumes 19,200 kWh (or 33%) and accounts for the major share in the building's energy consumption, followed by equipment consuming 13,607 kWh (or 24%), space cooling 13,501 kWh (or 23.7 %), and ventilation fans and pumps 7,603 kWh (or 13%).

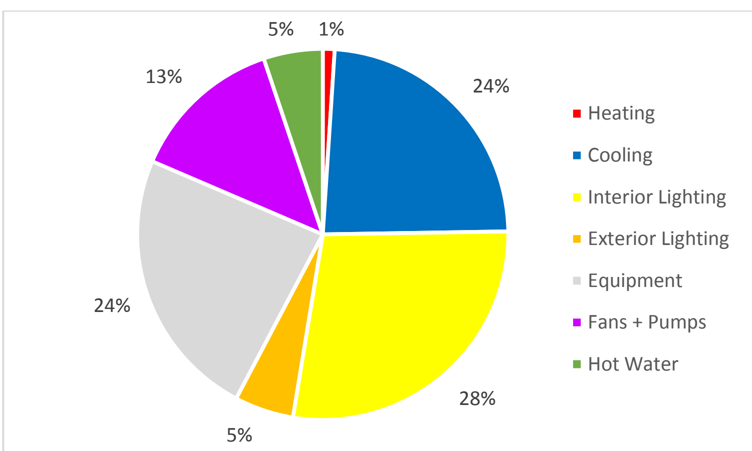

Figure 6.4 ASHRAE 90.1-2010: Baseline model predicted Annual Energy Consumption by End Use

Table 6.3 ASHRAE 90.1-2010: Baseline model monthly energy consumption by end use categories in kWh

Category	End Use	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sept	Oct	Nov	Dec	Total
Lighting	Internal Lighting	1,388	1,254	1,466	1,339	1,434	1,414	1,359	1,469	1,339	1,392	1,366	1,352	16,571
	Ext Lighting	280	233	229	193	174	157	166	188	209	246	265	288	2,628
	Total Lighting	1,668	1,487	1,695	1,532	1,608	1,571	1,526	1,656	1,548	1,638	1,631	1,640	19,200
	% of Total	38%	37%	37%	35%	34%	29%	28%	28%	31%	35%	37%	38%	33%
Equip-ment	Equipm ent	1,142	1,033	1,208	1,095	1,175	1,161	1,109	1,208	1,095	1,142	1,128	1,109	13,607
	% of Total	26%	26%	26%	25%	25%	21%	20%	21%	22%	24%	26%	26%	24%
HVAC	HVAC Cooling	473	524	645	868	1,134	1,878	2,062	2,106	1,581	1,106	701	426	13,501
	HVAC Heating	228	136	85	25	0	0	0	0	0	0	35	316	826
	HVAC Fan & Pump	626	567	686	597	659	672	627	703	610	633	627	596	7603
	HVAC	1,327	1,227	1,416	1,490	1,793	2,550	2,689	2,809	2,191	1,739	1,363	1,338	21,930
	% of Total	30%	31%	31%	34%	37%	47%	49%	48%	44%	37%	31%	31%	38%
Others	Others	248	222	247	217	215	199	194	203	199	219	228	238	2,630
	% of Total	6%	6%	5%	5%	4%	4%	4%	3%	4%	5%	5%	6%	5%
Total		4,385	3,969	4,566	4,334	4,791	5,481	5,518	5,876	5,033	4,738	4,350	4,325	57,376

Similar to the ASHRAE 90.1-2004: Baseline model results, Figures 6.4 and 6.5 show that building is cooling dominated and that space heating accounts only for 1% of the building's total energy consumption. Figure 6.5 reveals that the month of August consumes the highest energy whereas February consumes the least energy. Like the ASHRAE 90.1-2004: baseline model, the end use categories such as equipment, lighting, and hot water vary very little throughout the year. Table 6.2 shows the energy consumption by various endues categories.

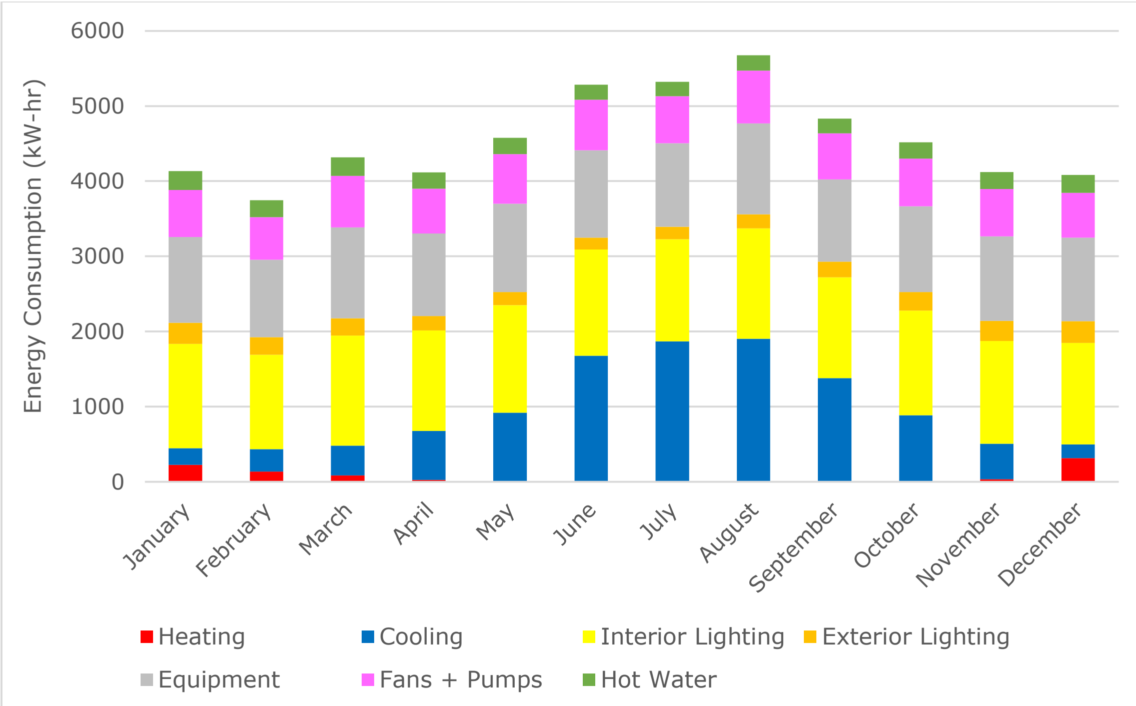

Figure 6.5 ASHRAE 90.1-2010: Baseline Model Monthly Energy Consumption by End-use category

6.1.4 ASHRAE 90.1 2010- Alternative Model: High Efficiency ASHPs and V-GCHPs

From Table 6.4, we can infer that the high efficiency ASHPs can save up-to 4,910 kWh or 22.6% and vertical GCHPs can save 8,304 kWh or 38 % of the energy consumed by the ASHP, and all energy savings are due to HVAC end-use category. The lighting and equipment end-use categories are relatively constant for both the alternative systems. Similar pattern is seen in the energy consumption as compared to ASHRAE 90.1-2004 baseline model, where the energy savings by the V-GCHPs increase (40-42%) with the increase in cooling loads during the months of June, July and August.

Table 6.4 ASHRAE 90.1-2010: Baseline model monthly energy consumption by end use categories in kWh

Category		Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Lighting	Baseline	1,668	1,487	1,695	1,532	1,608	1,571	1,526	1,656	1,548	1,638	1,631	1,640	19,200
	High Eff	1,668	1,487	1,695	1,532	1,608	1,571	1,526	1,656	1,548	1,638	1,631	1,640	19,200
	Savings	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%
	V GCHPs	1,668	1,487	1,695	1,532	1,608	1,571	1,526	1,656	1,548	1,638	1,631	1,640	19,200
	Savings	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%
Equipment	Baseline	1,142	1,033	1,208	1,095	1,175	1,161	1,109	1,208	1,095	1,142	1,128	1,109	13,607
	High Eff	1,142	1,033	1,208	1,095	1,175	1,161	1,109	1,208	1,095	1,142	1,128	1,109	13,607
	Savings	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%
	V GCHPs	1,142	1,033	1,208	1,095	1,175	1,161	1,109	1,208	1,095	1,142	1,128	1,109	13,607
	Savings	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	1,327	1,227	1,416	1,490	1,793	2,550	2,689	2,809	2,191	1,739	1,363	1,338	21,930
	High Eff	1,019	935	1,081	1,137	1,389	2,006	2,111	2,215	1,708	1,337	1,037	1,046	17,020
	Savings	308	292	335	353	404	544	578	594	483	402	326	292	4,912
		19%	18%	18%	15%	22%	21%	21%	21%	22%	13%	17%	18%	23%
	V GCHPs	808	772	960	949	1133	1503	1570	1685	1385	1147	948	767	13,626
	Savings	519	455	456	541	660	1047	1119	1124	806	592	416	571	8304
		39%	37%	32%	36%	37%	41%	42%	40%	37%	34%	30%	43%	38%
Others	Baseline	248	222	247	217	215	199	194	203	199	219	228	238	2,630
	High Eff	248	222	247	217	215	199	194	203	199	219	228	238	2,630
	Savings	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%
	V GCHPs	248	222	247	217	215	199	194	203	199	219	228	238	2,630
	Savings	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%

Figure 6.6 and Figure 6.7 provide the time variation for the heat pump entering fluid temperature and the power consumption (space heating and space cooling) by the V-GCHPs over the time of 25 years.

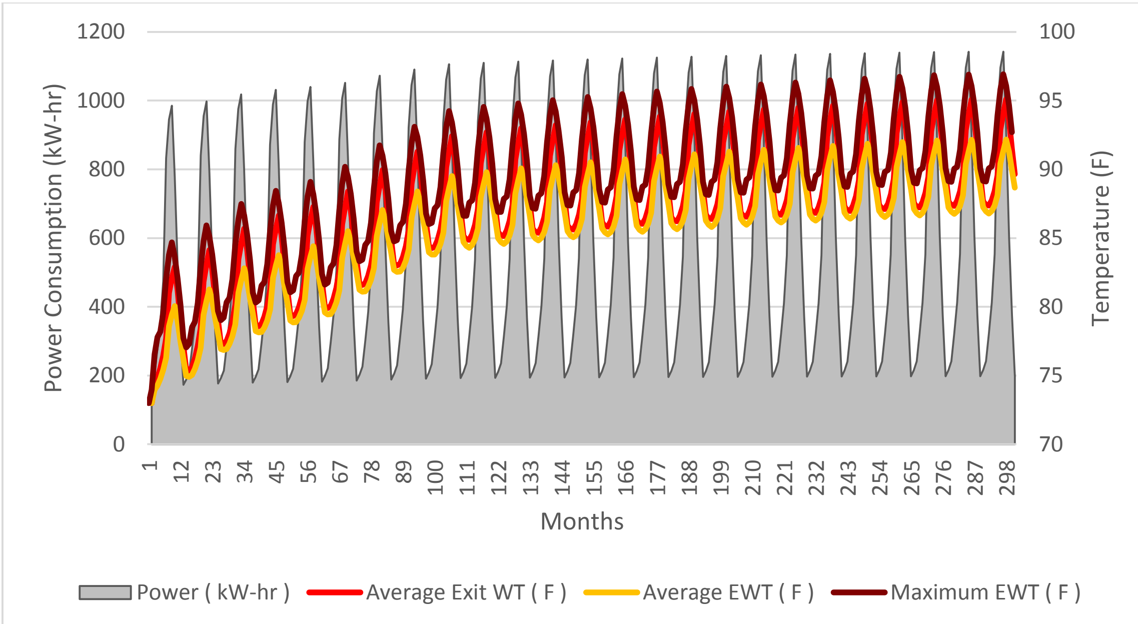

Figure 6.6: ASHRAE 90.1-2004 Model -The Entering water temperature, exit water temperature and power consumption by the vertical GCHPs over the time of 25 years of operation.

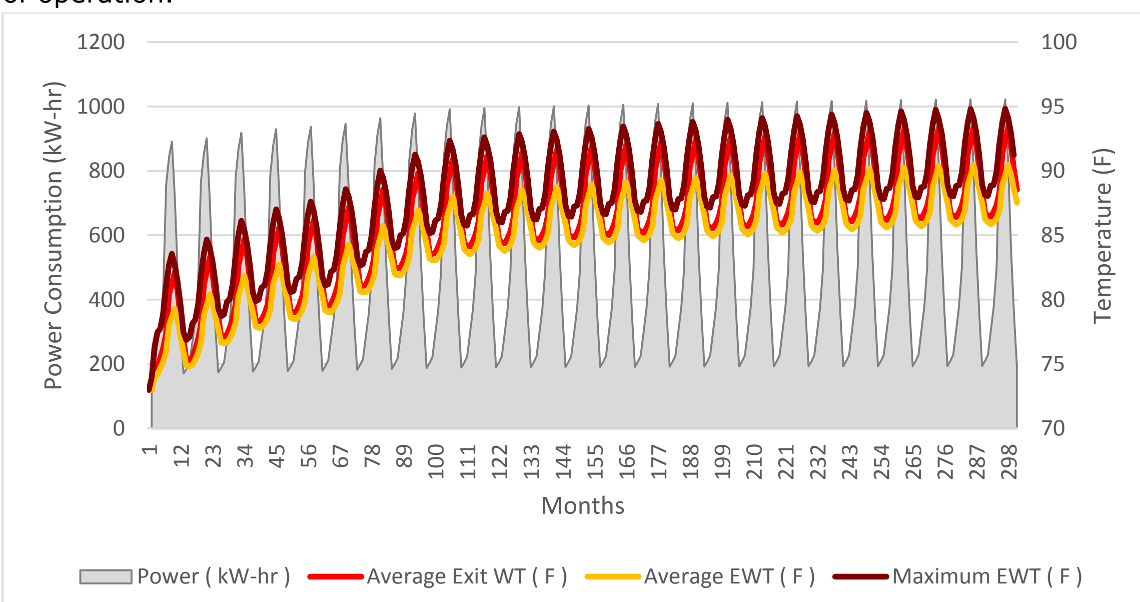

Figure 6.7: ASHRAE 90.1-2010 Model-The Entering water temperature, exit water temperature and power consumption by the vertical GCHPs over the time of 25 years of operation.

The values for the maximum and minimum GHE outlet temperatures have a fairly limited range of acceptable values. Extended heat pump range will usually have a 20°F recommended $T_{wi,min}$ and 110°F recommended as $T_{wi,max}$ (section 5.3.2). Figure 6.5 and figure 6.6 indicate that the ground loop design was a good one since the EWT values were below 110°F. The EWT gradually rises over the years as the soil temperature rises due to prolonged heat dissipation into the ground reducing the efficiency of the heat pump, and hence increasing the power consumption. For the first 8 years of operation of V-GCHPs, the EWT values rise significantly increasing the power consumption from 6,046 kWh to 6,705 kWh (10.9 % increase) for ASHRAE 90.1-2004 model and 5,570 kWh to 6,219 kWh (11.7% increase) for ASHRAE 90.1-2010 model. In the later years of operation, there was not significant rise of EWT values and this stabilized the power consumption. In ASHRAE 90.1- 2004 model, the power consumption after 25 years rises to 7,029 kWh or increases by 16.2 % from the 1st year (4.6 % after 8th year) and in ASHRAE 90.1 2010 model, the power consumption rises to 6410 kWh or 15 % increase in power consumption from 1st year (3 % after 8th year).

The maximum predicted EWT to the heat pump after two years was about 85.9 °F for ASHRAE 90.1-2004 model and 84.7 °F for ASHRAE 90.1-2010 model rising to a maximum of 96.92°F (32.8 % increase) and 94.82°F (29.9 % increase) after 25 years of simulation.

6.2 Analysis of the GCHPs performance due to the Saturated Soil conditions

In this analysis, the existing soil for both V-GCHPs and H-GCHPs was replaced with the saturated soil. The thermal properties of both soil types (existing and saturated) are provided in Chapter 4. From Table 6.5, it can be inferred that there is 26% decrease in the length of vertical GHE for saturated conditions compared to existing conditions whereas there is little difference in the annual energy consumption. In both the cases, the annual cooling and heating energy consumption show 2-3%

decrease from the existing soil condition case, which may be due to decrease in the vertical GHE length reducing the pumping power required.

Table 6.5 Effect of Saturated soil condition on V-GCHPs.

Cases	Vertical GCHP						
	Existing Soil Condition			Saturated Soil Condition			% Decrease in Length
	Cooling (kWh)	Heating (kWh)	Length (ft)	Cooling (kWh)	Heating (kWh)	Length (ft)	
ASHRAE 90.1-2004 Case	6,475	61	2954.1 (185)	6,283	60	2,180.8 (136)	26.2
ASHRAE 90.1-2010 Case	5,994	53	2729.5 (171)	5,806	52	2,015.7 (126)	26.1

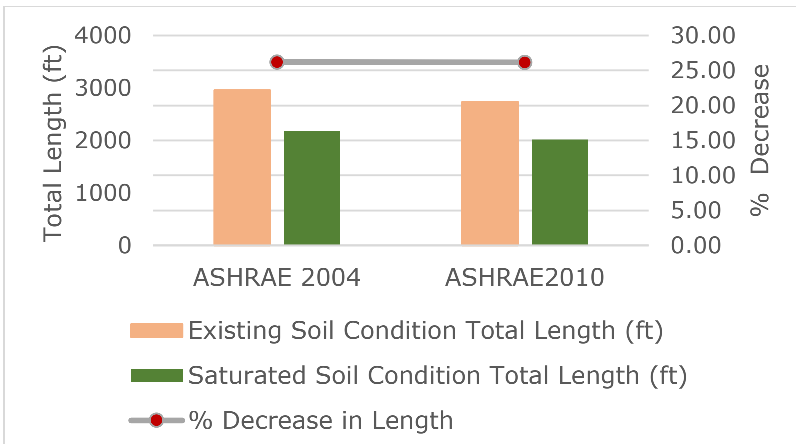

Figure 6.8: Effect Of Saturated Soil Condition on Vertical borehole lengths

Similar results are seen in both the ASHRAE 90.1-2004 and ASHRAE 90.1-2010 models. In the ASHRAE 90.1-2010 case, the length of the vertical GHE has decreased from 2729.5 feet to 2015.7 feet which is about 26% reduction of the GHE length due to saturated soil condition (see Table 6.5). There is no significant change in the energy consumption, only about 3.12% reduction in the annual cooling and heating energy consumption for this case.

Table 6.6 Effect of Saturated soil condition on H-GCHPs.

Case	Description	Existing Soil Condition	Saturated Soil Condition	% Decrease in the Length
		Trench Length (ft)	Trench Length (ft)	
ASHRAE 90.1-2004	Horizontal Slinky	922.9	689.4	25.30
	Vertical Slinky	1,058	782	26.09
	4 pipes/trench	1,889	1,428	24.40
	2 pipes/trench	3,021	2,284	24.38
ASHRAE 90.1-2010 model	Horizontal Slinky	851	636.3	25.23
	Vertical Slinky	977	734.4	24.83
	4 pipes/trench	1,743	1,308	24.96
	2 pipes/trench	2,787	2,094	24.87

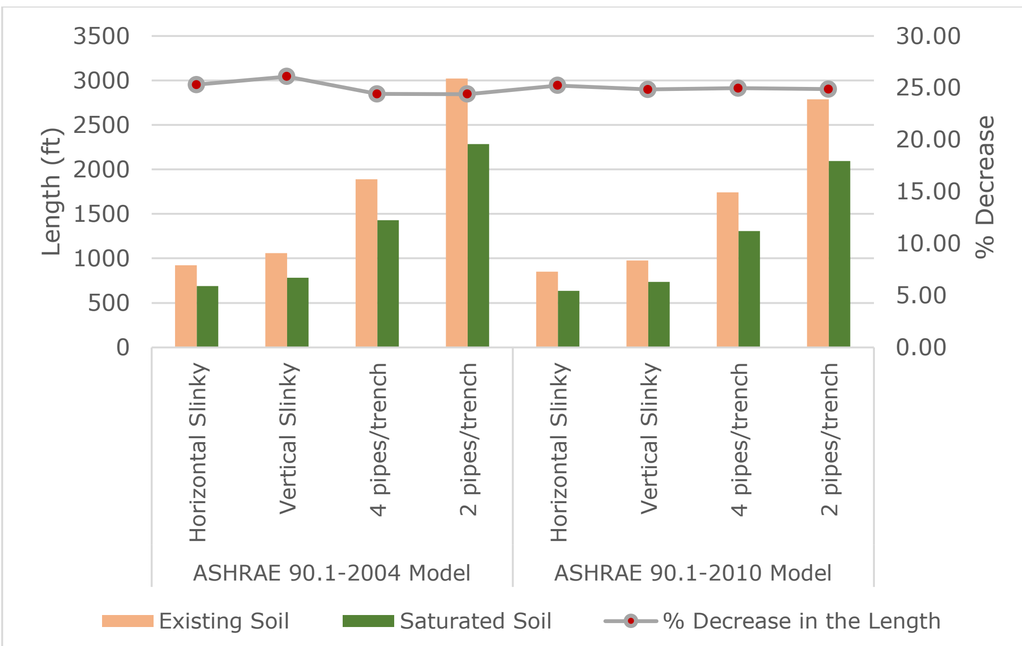

Figure 6.9: Effect Of Saturated Soil Condition on Horizontal Trench Length

Similarly for H-GCHPs, from Table 6.6 and Figure 6.9 we can infer that there is a 22- 26% decrease in the trench length for saturated soil conditions. The software fails to calculate the monthly and hourly energy consumption by the H-GCHPs as the software lacks the capability. Hence the effect of the saturated soil condition on the annual energy consumption could not be evaluated but, based on the results for the vertical system should not change much from the existing soil case to the saturated soil case.

6.3 Life Cycle Cost Analysis and Greenhouse Gas Analysis

The lifecycle costs was used for economic evaluation of V-GCHPs and H-GCHPs when compared with existing ASHPs and high efficient ASHPs. The estimated installation cost for all the four systems types are mentioned in table 6.7.

Table 6.7: First cost of ASHPs, Vertical GCHPs and Horizontal GCHPs

Sr no	System Types	First Cost ($)	
		ASHRAE 90.1-2004	ASHRAE 90.1-2010
Base	ASHP	$15,000	$16,000
Alt 1	ASHP - Inc Efficiency	$17,000	$17,000
Alt 2	V GCHP- Exs Soil (w/o Incentives)	$76,196	$71,981
Alt 3	V GCHP - Exs Soil (Incentives)	$49,032	$46,319
Alt 4	V GCHP - Sat Soil w/o Incentives	$61,629	$58,542
Alt 5	V GCHP - Sat Soil (Incentives)	$39,658	$37,672
Alt 6	H GCHP (w/o incentives)	$34,947	$32,582
Alt 7	H GCHP (Incentives)	$22,488	$20,966
Alt 8	H GCHP –Sat Soil (w/o incentives)	$27,269	$25,523
Alt 9	H GCHP -Sat Soil (Incentives)	$17,547	$17,024

6.3.1 ASHRAE 90.1-2004: LCC Analysis

The annual energy costs of the different alternatives were estimated as:

Baseline Case, rooftop air-source heat pump units: $ 1,481

Alterative Case 1, High Efficient (ASHP): $ 1,161

Alterative Case 2/3, Vertical GCHPs – Existing Soil: $ 912

Alterative Case 4/5, Vertical GCHPs – Saturated Soil : $ 910

Alternative Case6/7, Horizontal GCHPs – Existing Soil: $944

Alternative Case8/9, Horizontal GCHPs – Saturated Soil: $942

Table 6.8 summarizes the result from the LCC spreadsheet. The results show that the GCHP system can save $2900 or 38% on the replacement costs.

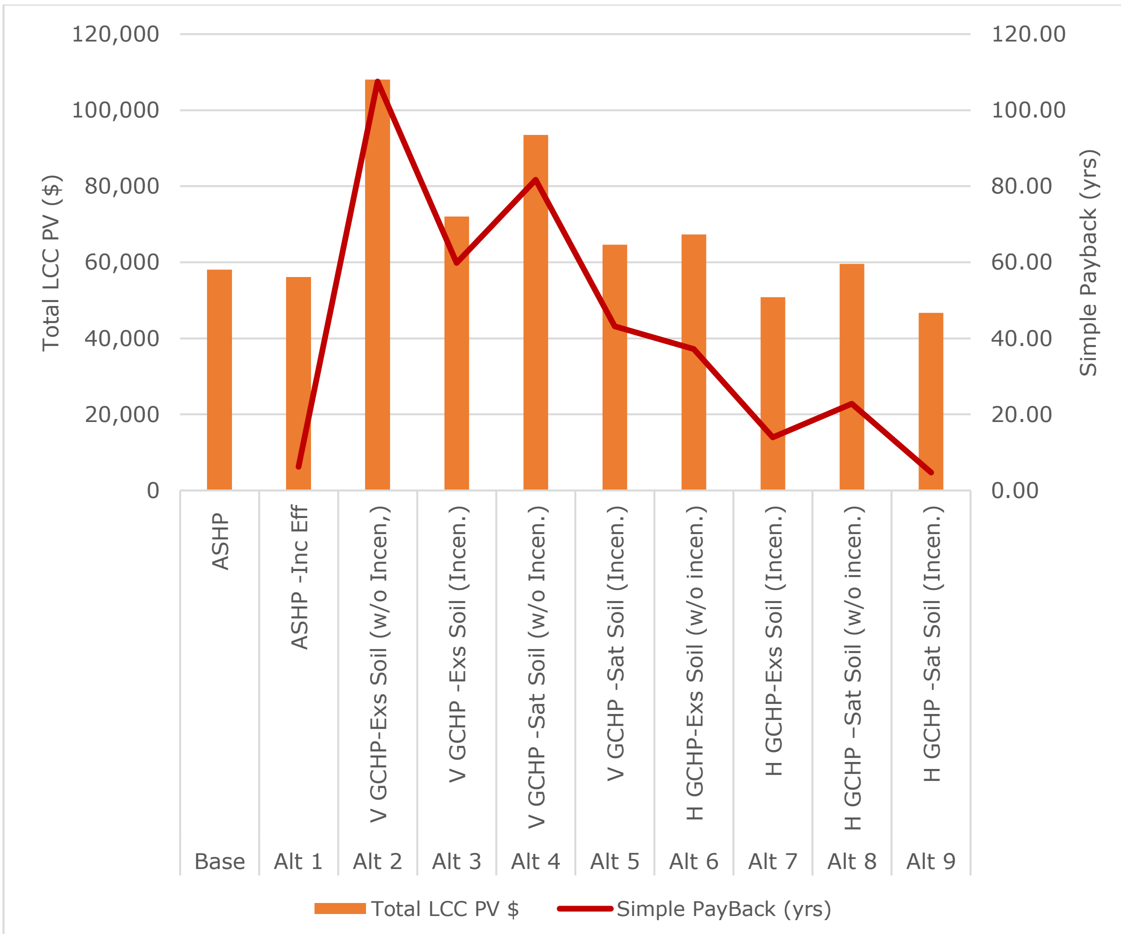

Figure 6.10 ASHRAE 90.1-2004: Total Life-cycle costs and payback period of the alternative models

The V-GCHP system have the lowest utility bills and can save $569 or 38.4 % and $249 or 21.4 % on annual utility bills when compared to the ASHPs and high efficient ASHPs respectively. Likewise, horizontal GCHPs can save $ 537 or 36.2 % as compared with the existing ASHPs and $217 or 18.7% as compared with high efficient ASHPs.

Table 6.8 ASHRAE 90.1-2004: Life-cycle costs analysis summary of baseline model vs. various other alternative models

Case	Description	One-Time Costs		Total Utility		Maintenance		Total	Simple	Disc nt'd	Invest
		1st year	LCC	1st year	LCC	1st year	LCC	LCC	Payback	Payback	Ratio
		$	PV $	$	PV $	$	PV $	PV $	yrs	yrs	SIR
Life-Cycle COSTS											
Base	ASHP	15,000	19,814	1,481	24,302	800	13,931	58,046	n/a	n/a	n/a
Alt 1	ASHP -Inc Efficiency	17,000	23,098	1,161	19,051	800	13,931	56,079	n/a	n/a	n/a
Alt 2	V GCHP-Exs Soil (w/o Incentives)	76,196	79,149	912	14,965	800	13,931	108,044	n/a	n/a	n/a
Alt 3	V GCHP -Exs Soil (Incentives)	49,032	43,157	912	14,965	800	13,931	72,053	n/a	n/a	n/a
Alt 9	V GCHP -Sat Soil (w/o Incentives)	61,629	64,582	910	14,932	800	13,931	93,444	n/a	n/a	n/a
Alt 4	V GCHP -Sat Soil (Incentives)	39,658	35,741	910	14,932	800	13,931	64,604	n/a	n/a	n/a
Alt 5	H GCHP-Exs Soil (w/o incentives)	34,947	37,900	944	15,490	800	13,931	67,320	n/a	n/a	n/a
Alt 6	H GCHP-Exs Soil (Incentives)	22,488	21,390	944	15,490	800	13,931	50,811	n/a	n/a	n/a
Alt 7	H GCHP –Sat Soil (w/o incentives)	27,269	30,222	942	15,457	800	13,931	59,610	n/a	n/a	n/a
Alt 8	H GCHP -Sat Soil (Incentives)***	17,547	17,340	942	15,457	800	13,931	46,728	n/a	n/a	n/a
Life-Cycle SAVINGS (negative entries indicate increased costs)											
Alt 1	ASHP -Inc Efficiency	-2,000	-3,284	320	5,251	0	0	1,967	6.25	7.2	1.60
Alt 2	V GCHP-Exs Soil (w/o Incentives)	-61,196	-59,335	569	9,337	0	0	-49,998	107.5	0	0.16
Alt 3	V GCHP -Exs Soil (Incentives)	-34,032	-23,343	569	9,337	0	0	-14,006	59.81	0	0.40
Alt 4	V GCHP -Sat Soil (w/o Incentives)	-46,629	-44,768	571	9,370	0	0	-35,398	81.66	0	0.21
Alt 5	V GCHP -Sat Soil (Incentives)	-24,658	-15,927	571	9,370	0	0	-6,558	43.18	0	0.59
Alt 6	H GCHP-Exs Soil (w/o incentives)	-19,947	-18,086	537	8,812	0	0	-9,274	37.15	0	0.49
Alt 7	H GCHP-Exs Soil (Incentives)	-7,488	-1,577	537	8,812	0	0	7,235	13.94	7.4	5.59
Alt 8	H GCHP –Sat Soil (w/o incentives)	-12,269	-10,408	539	8,845	0	0	-1,563	22.76	0	0.85
Alt 9	H GCHP -Sat Soil (Incentives)***	-2,547	2,474	539	8,845	0	0	11,318	4.73	1.5	-3.58
* LCC Choice ** Simple Payback choice											

The analysis shows that the one-time cost of V GCHPs is $76,196 or 80 % more than Baseline case, but with the available federal incentives the cost is reduced to $49,032. With the effect of saturated soil condition, the cost of V GCHPs is further reduced by $14,567 or 19%. The GHG emissions by ASHRAE 90.1-2004 baseline model is approximately 20.2 tons of CO_2, high efficiency ASHPs is 15.7 tons and V-GCHPs is 12.1 tons of CO_2 annually. The GCHP system can save approximately 8.1 tons (40%) of carbon dioxide emission per year as compared to ASHPs.

The alternative case 9 that is horizontal GCHPs with saturated soil conditions and incentives has the least LCC of $46,728 as compared to the base case and all the other alternative cases. The LCC for alternative case 9 is $ 11,318 or 19.5 % less than the base case, $9,351 less or 16.8 % less than the alternative case 2 or the high efficient ASHPs. Further, the payback period for this alternative is just 4.7 years and discounted payback period of around 1.5 years. The next best alternative, considering the payback period, is alternative case 1 having payback period of 6.25 years. The next best economical viable option is alternative 7 or horizontal GCHPs with federal incentives. The total LCC for this alternative is $50,811 and is less than 12.5 % than the base case and 9.4 % than the alternative case 1 or high efficient ASHPs. Figure 6.10 indicates that the simple payback is over 25 years, and the SIR is close to zero for Alternative cases 2,3,4,5, 6 & 8. Therefore, the LCCA results lead to the conclusion that vertical GCHP system is not economically feasible when compared to the ASHP. Whereas the horizontal system is economical, particularly when federal incentives are considered.

6.3.2 ASHRAE 90.1-2010 Model: LCC Analysis

The annual energy costs of the ASHPs, vertical GCHPs and horizontal GCHPs were estimated as:

Baseline Case, rooftop air-source heat pump units: $ 1,337

Alterative Case 1, High Efficient (ASHP): $ 1,175

Alterative Case 2/3, Vertical GCHPs – Existing Soil: $ 866

Alterative Case 4/5, Vertical GCHPs – Saturated Soil: $ 862

Alternative Case6/7, Horizontal GCHPs – Existing Soil: $891

Alternative Case8/9, Horizontal GCHPs – Saturated Soil: $889

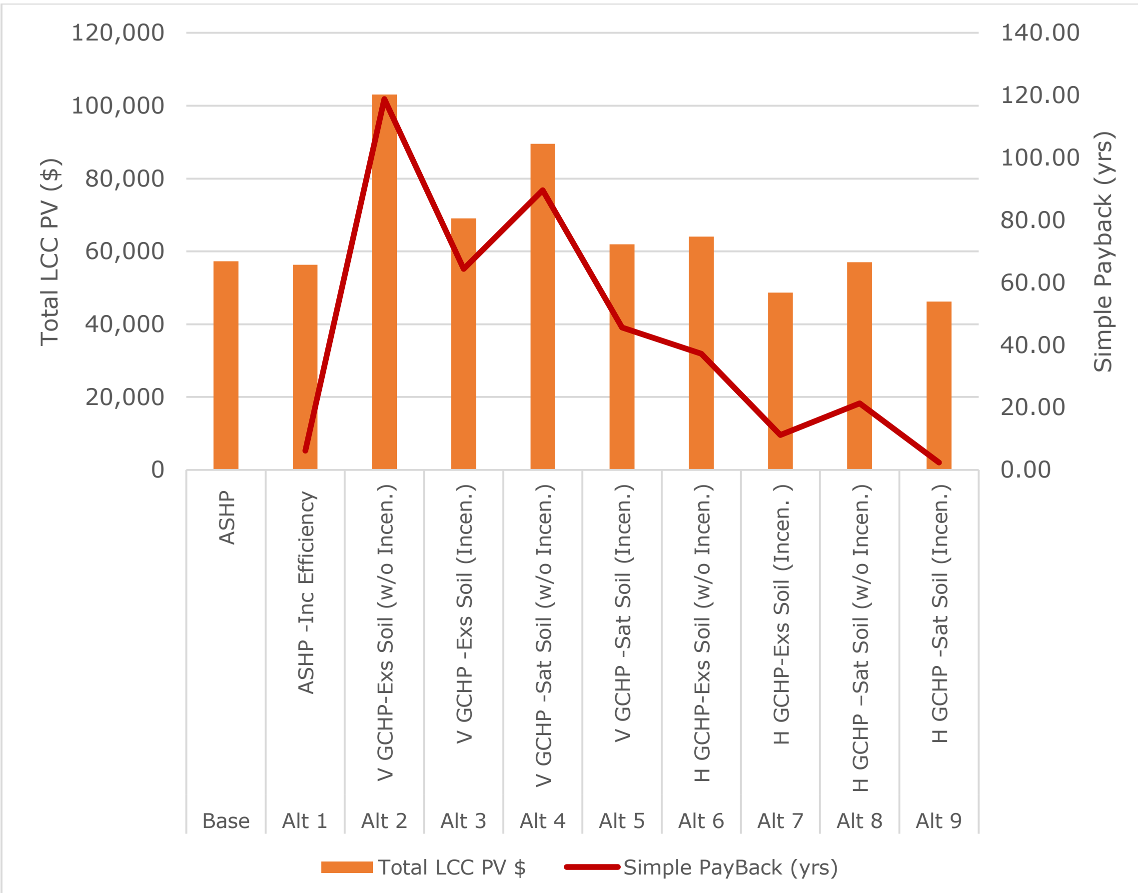

Figure 6.11 ASHRAE 90.1-2010 Model: Total Life-cycle costs and payback period of the alternative models.

The LCC results for ASHRAE 90.1-2010 model is similar to ASHRAE 90.1-2004 model except for the slight changes in the total LCC and payback years mainly due to improved ASHP equipment in former case.

Table 6.9 ASHRAE 90.1-2010 Model: Life-cycle costs analysis summary of baseline model vs. various other alternative models

Case	Description	One-Time Costs		Total Utility		Maintenance		Total	Simple	Discnt'd	Saving-to-
		1st year	LCC	1st year	LCC	1st yr	LCC	LCC	Payback	Payback	Ratio
		$	PV $	$	PV $	$	PV $	PV $	yrs	yrs	SIR
Life-Cycle COSTS											
Base	ASHP	16,000	21,456	1,337	21,939	800	13,931	57,325	n/a	n/a	n/a
Alt 1	ASHP -Inc Efficiency	17,000	23,098	1,175	19,281	800	13,931	56,309	n/a	n/a	n/a
Alt 2	V GCHP-Exs Soil (w/o Incentives)	71,981	74,934	866	14,210	800	13,931	103,074	n/a	n/a	n/a
Alt 3	V GCHP -Exs Soil (Incentives)	46,319	40,932	866	14,210	800	13,931	69,073	n/a	n/a	n/a
Alt 4	V GCHP -Sat Soil (w/o Incentives)	58,542	61,495	862	14,145	800	13,931	89,570	n/a	n/a	n/a
Alt 5	V GCHP -Sat Soil (Incentives)	37,672	33,842	862	14,145	800	13,931	61,917	n/a	n/a	n/a
Alt 6	H GCHP-Exs Soil (w/o incentives)	32,582	35,535	891	14,621	800	13,931	64,086	n/a	n/a	n/a
Alt 7	H GCHP-Exs Soil (Incentives)	20,966	20,142	891	14,621	800	13,931	48,694	n/a	n/a	n/a
Alt 8	H GCHP –Sat Soil (w/o incentives)	25,523	28,476	889	14,588	800	13,931	56,994	n/a	n/a	n/a
Alt 9	H GCHP -Sat Soil (Incentives)***	17,024	17,649	889	14,588	800	13,931	46,167	n/a	n/a	n/a
Life-Cycle Savings (negative entries indicate increased costs)											
Alt 1	ASHP -Inc Efficiency	-1,000	-1,642	162	2,658	0	0	1,016	6.17	7	2
Alt 2	V GCHP-Exs Soil (w/o Incentives)	-55,981	-53,478	471	7,729	0	0	-45,749	118.9	>0.03	0
Alt 3	V GCHP -Exs Soil (Incentives)	-30,319	-19,476	471	7,729	0	0	-11,747	64.37	>0.03	0
Alt 4	V GCHP -Sat Soil (w/o Incentives)	-42,542	-40,039	475	7,794	0	0	-32,244	89.56	>0.03	0
Alt 5	V GCHP -Sat Soil (Incentives)	-21,672	-12,386	475	7,794	0	0	-4,592	45.63	>0.03	1
Alt 6	H GCHP-Exs Soil (w/o incentives)	-16,582	-14,079	446	7,318	0	0	-6,760	37.18	>0.03	1
Alt 7	H GCHP-Exs Soil (Incentives)	-4,966	1,313	446	7,318	0	0	8,632	11.13	4	-6
Alt 8	H GCHP –Sat Soil (w/o incentives)	-9,523	-7,020	448	7,351	0	0	332	21.26	23	1
Alt 9	H GCHP -Sat Soil (Incentives)***	-1,024	3,807	448	7,351	0	0	11,158	2.29	1	-2
* LCC Choice ** Simple Payback choice											

Table 6.9 show that the vertical GCHP system can save $569 or 38.42% and $309 or 26.3% and horizontal GCHPs can save $ 446 or 33.35% and $284 or 24.17% on annual utility bills when compared with the ASHPs and high efficient ASHPs

respectively. With the effect of saturated soil condition, the cost of V GCHPs is further reduced by $14,567 or 19%.

The GHG emissions by baseline model is approximately 18.3 tons of CO_2, high efficiency ASHPs is 14.21 tons and V-GCHPs is 11.4 tons of CO_2 annually. This shows that GCHP system can save approximately 7 tons (37.7%) of carbon dioxide emission per year as compared to ASHPs. Like the ASHRAE 90.1-2004 model, alternative case 9, i.e. H-GCHP with saturated soil conditions and incentives, has the least LCC of $46,167 as compared to all the alternative cases. The LCC for alternative case 9 is $ 11,158 or 19.5 % less than the base case, $10,142 less or 18 % less than alternative case 2 or the high efficiency ASHPs. Alternative case 9 gives the best payback period of 2.3 years. The next best alternative considering the payback period is alternative case 1 having payback period of 6.2 years. Alternative option 7, H-GCHPs with federal incentives, is also economically feasible with total LCC of $48,694 and payback period of 11.1 years.

Figure 6.11 show that for Alternative cases 2,3,4,5, 6 & 8 the simple payback year is over 25 years, and the SIR is close to zero. Therefore the LCCA results lead to the conclusion that a V-GCHP system is not economically feasible when compared to the ASHP.

6.4 Sensitivity Analysis

The main objective of the sensitivity analysis was to quantify the uncertainty in the V-GCHPs and H-GCHPs cost estimates and investigate which assumptions are critical. The installation costs, annual utility costs, and periodic costs of the GCHP system were varied from - -80% to +80% for V-GCHPS and 40% to -40% for H-GCHPs for ASHRAE 90.1-2010 model. The results of the sensitivity analysis for V-GCHPs are shown in Figures 6.12, 6.13, 6.14 & 6.14 and for H-GCHPs 6.15, 6.16, 6.17 & 6.18. From the all the above mentioned figures, we can conclude that the most sensitive cost element for both H-GCHPs and V-GCHPs (except for the Alt Case9: H-GCHPs with incentives and saturated soil condition), is the initial cost, followed by the utility cost, maintenance cost, and then lastly the periodic costs. For the Alt Case 9, H-GCHPs with saturated soil condition and incentives, the utility cost is most sensitive closely followed by installation cost, maintenance cost and then the periodic cost. The sensitivity analysis for various GCHPs alternative were as follows.

a. Alt Case 2- V-GCHPs- existing soil and without incentive

Figure 6.12 indicates that the capital cost should be decrease by 64% to approach the LCC PV of ASHPs. A 30 % increase in cost of each element will result in a 21.2% increase of LCC PV for installation cost, around a 4 % increase for utility and maintenance cost and an insignificant increase in the LCC PV of 0.82 % for periodic cost. This shows that in this case the installation cost is most sensitive to LCC PV among all the other cost elements.

b. Alt Case 3- existing soil and with incentive

For this case, the capital cost should be decreased by 35% to meet the LCC PV of ASHPs. Figure 6.12 shows that a 30 % increase in cost items will increase the LCC PV by 16.7 % for installation cost, by approximately 6% for utility and maintenance cost, and by merely 1.2% in the LCC PV for the periodic cost.

c. Alt Case 4- V-GCHPs- saturated soil condition:

For this case the results were very similar to Alt Case 1. From figure 6.13, a 30% increase in the installation cost increases the LCC PV by 20% whereas the LCC PV for utility and maintenance increases by 5%. There is just a 1% increase in LCC PV for a 30% increase in periodic cost. The capital cost would need to increase by 59% above the base case assumptions in order for the PV of its life-cycle cost to approach that of the ASHPs.

d. Alt Case 5- V-GCHPs- saturated soil and with federal incentives:

Figure 6.15 implies that a 30% increase in the individual cost items results in the annual energy cost increasing its LCC PV by about 7 % whereas the percentage increase in initial cost would increase the LCC PV by 15.5 %. The installation cost of the V-GCHPs should be increased by 19% to match the LCC PV of ASHPs.

e. Alt Case 6- H-GCHPs- existing soil and without incentive:

Figure 6.15 indicates that the installation cost would need to increase by 36 % above the base case assumptions in order for its LCC PV to approach that of the ASHPs. The most sensitive cost item like all previous cases is the installation cost followed by utility cost, maintenance cost and then the periodic cost.

f. Alt Case 7- H-GCHPs - existing soil and with incentive

For this case, since the LCC PV value is more than the baseline case, the installation cost needs to be decreased by 39% to match the values of baseline case (see Figure 6.16). For this case, the results are different compared to the previous cases. The most sensitive cost item of this case is the installation cost, closely followed by the energy cost, maintenance cost, and then the periodic costs. A 30% increase in cost of each element would result in an increase in LCC PV by 11% for capital cost, 9% for utility cost, 8% maintenance and 1.4% for periodic cost.

g. Alt Case 8- H-GCHPs- saturated soil condition without incentives:

Figure 6.17 indicates that the installation cost would need to increase by 6 % above the base case assumptions in order for its LCC PV to approach that of the ASHPs. The most sensitive cost item, like all previous cases, is the installation cost followed by utility cost, maintenance cost and then the periodic cost.

h. Alt Case 9- H-GCHPs- saturated soil and with federal incentives

Particularly for this alternative the most sensitive cost item is the utility cost closely followed by installation cost and maintenance cost, and then the periodic costs. With a 30 % increase in the various cost items, the LCC PV is increased by 9.92% for utility cost, 9.24 % for installation cost, 9 % for maintenance cost and finally 1.92% for periodic cost.

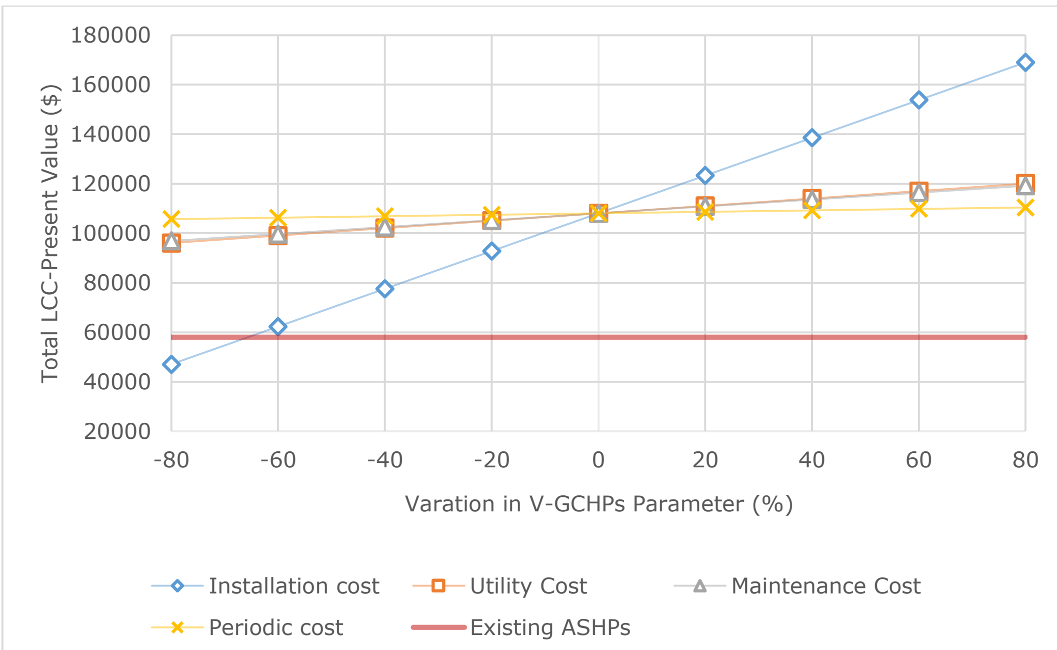

Figure 6.12: Alt Case 2: Sensitivity analysis of V-GCHPs (existing soil condition, w/o incentives) cost items on the system's net present value of 25-year life cycle cost.

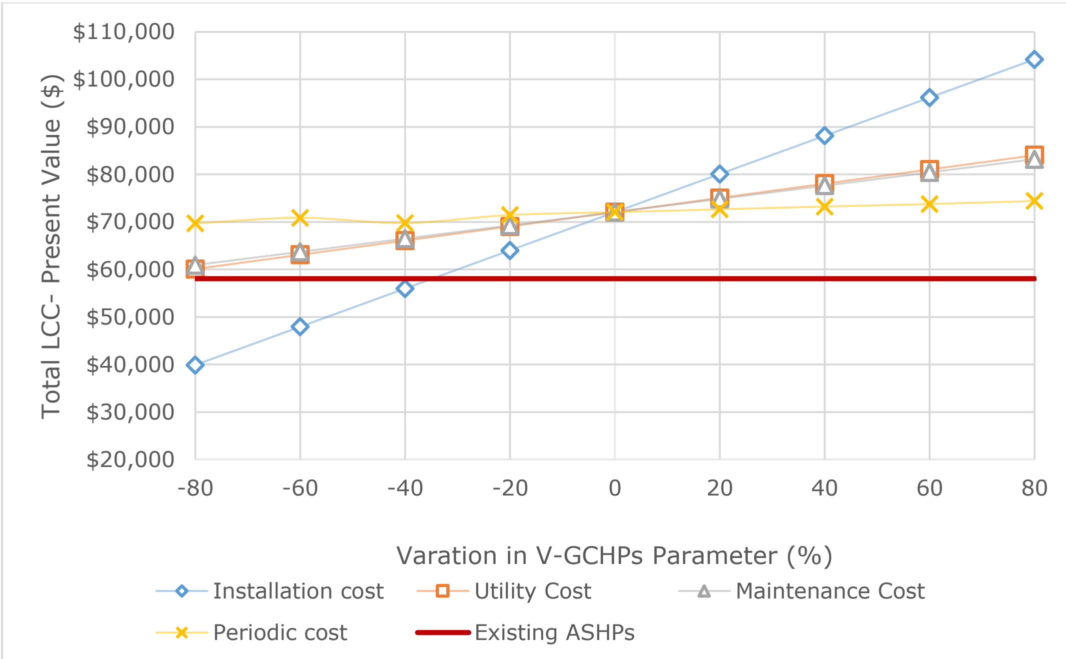

Figure 6.13: Alt Case 3: Sensitivity analysis of V-GCHPs (existing soil condition, with incentives) cost items on the system's PV of 25-year life cycle cost

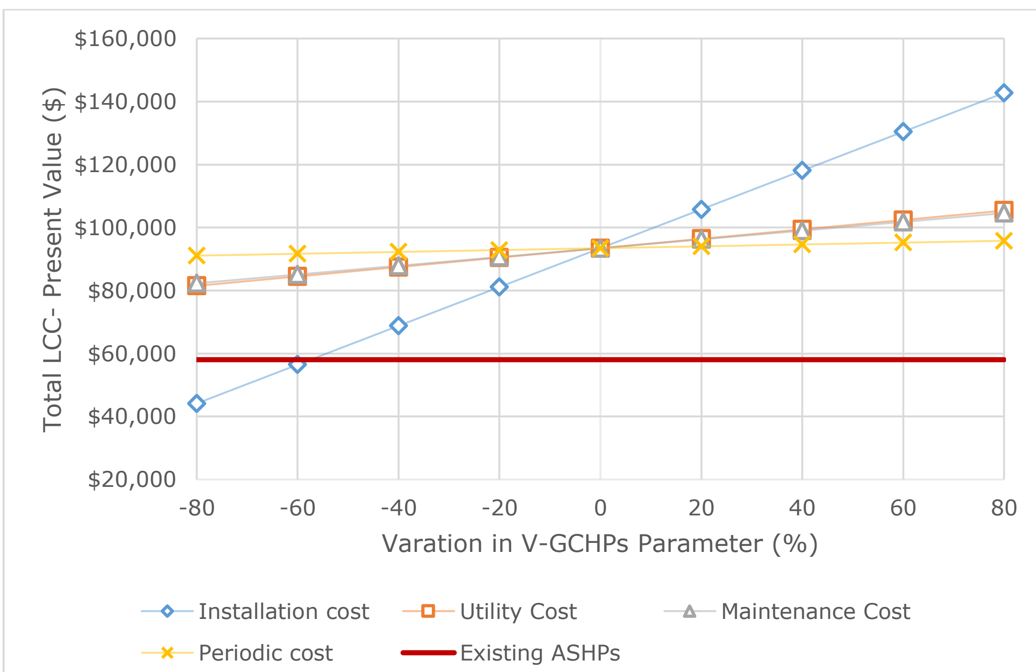

Figure 6.14: Alt Case 4: Sensitivity analysis of V-GCHPs (saturated soil condition, w/o incentives) cost items on the system's PV of 25-year life cycle cost

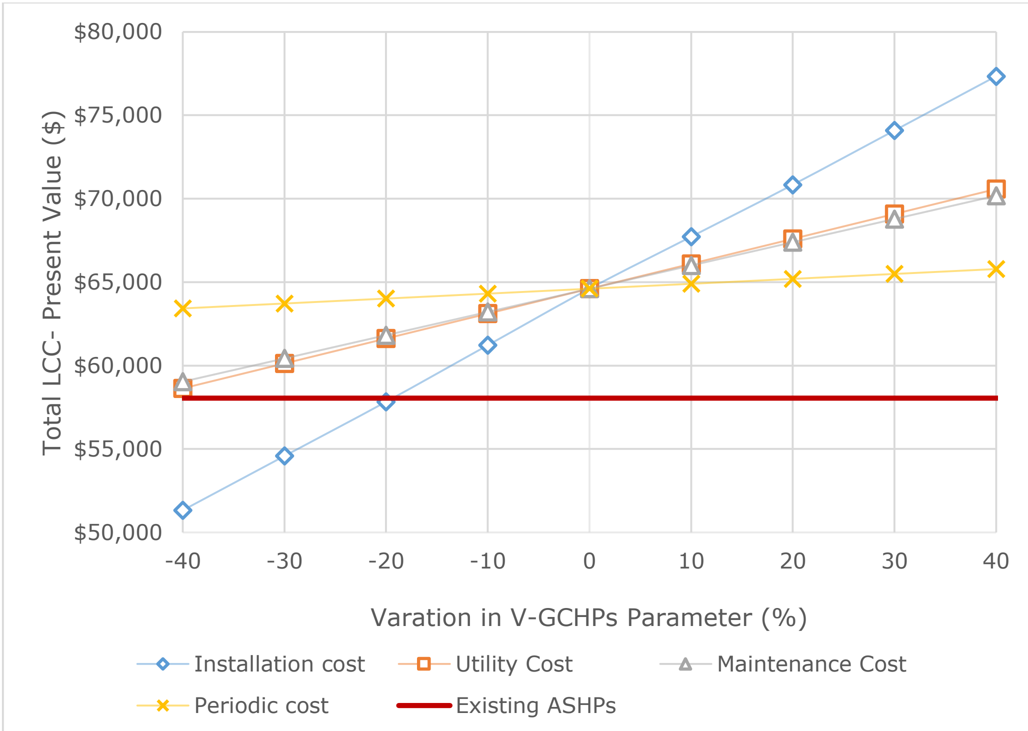

Figure 6.15: Alt Case 5: Sensitivity analysis of V-GCHPs (saturated soil condition, with incentives) cost items on the system's PV of 25-year life cycle cost

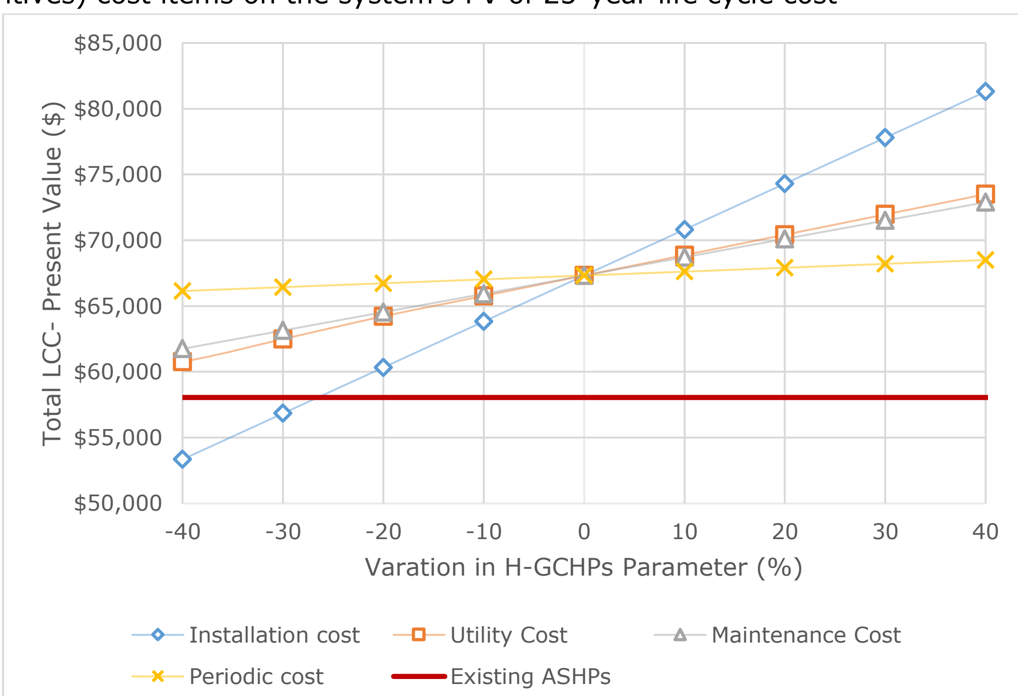

Figure 6.16: Alt Case 6: Sensitivity analysis of H-GCHPs (existing soil condition, w/o incentives) cost items on the system's PV of 25-year life cycle cost

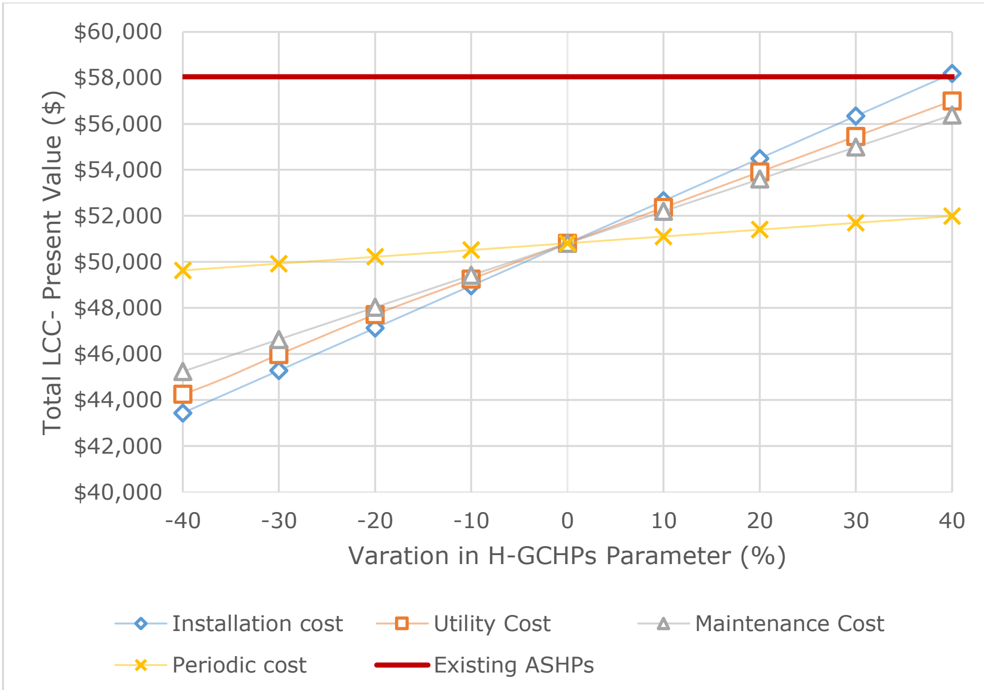

Figure 6.17: Alt Case 7: Sensitivity analysis of H-GCHPs (existing soil condition, with incentives) cost items on the system's PV of 25-year life cycle cost

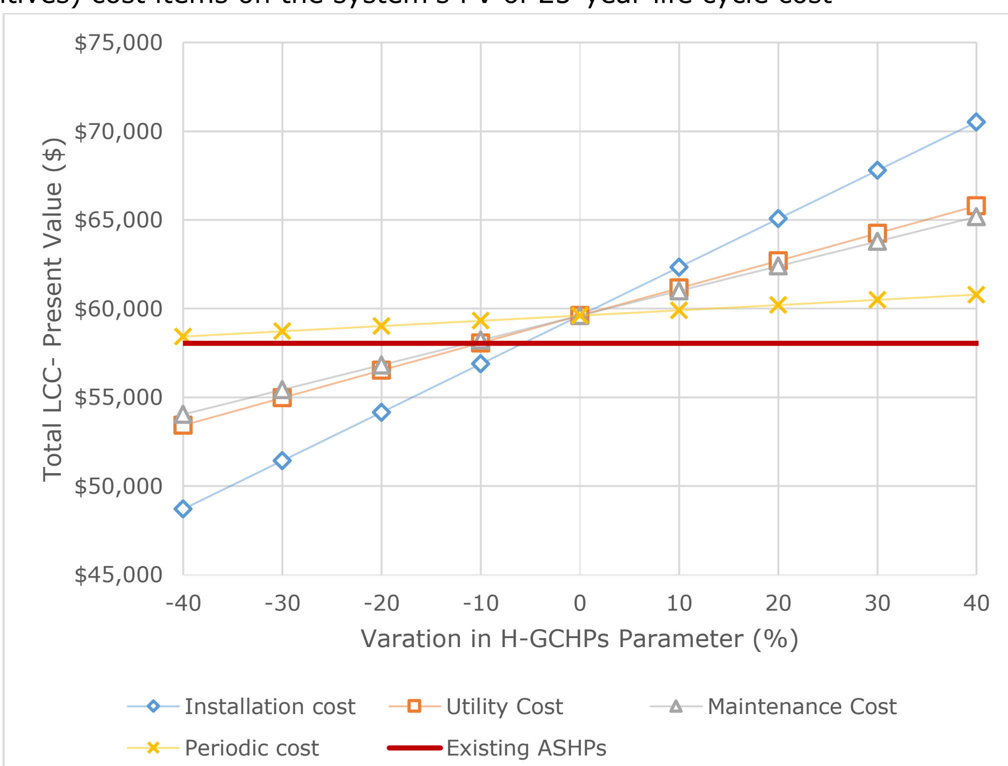

Figure 6.18: Alt Case 8: Sensitivity analysis of H-GCHPs (saturated soil condition, w/o incentives) cost items on the system's PV of 25-year life cycle cost

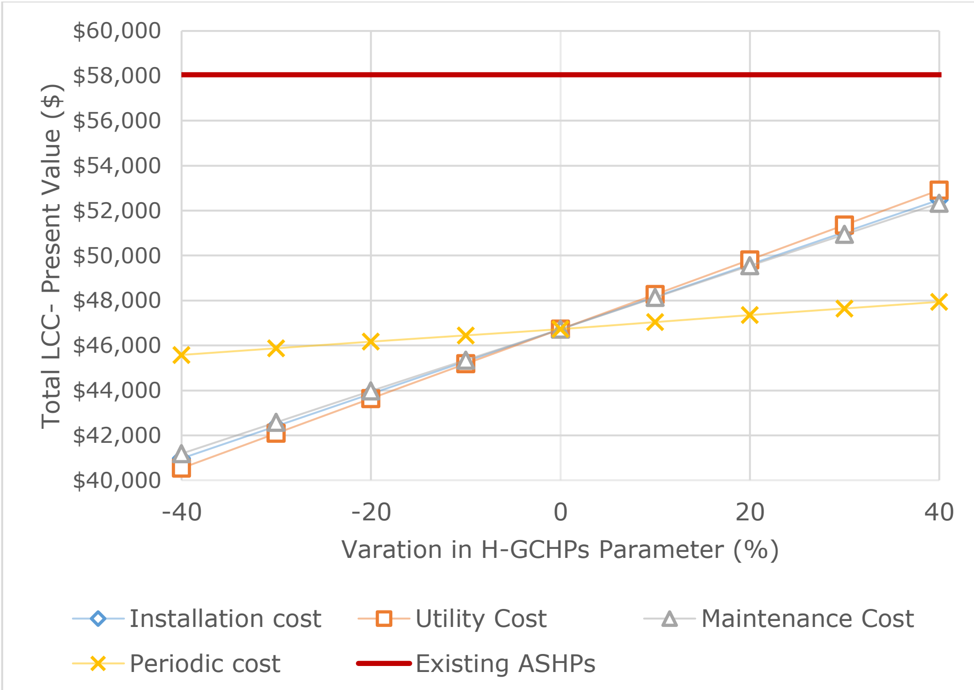

Figure 6.19: Alt Case 9: Sensitivity analysis of H-GCHPs (saturated soil condition, with incentives) cost items on the system's PV of 25-year life cycle cost

Variations on the periodic cost have the least effect on the NPV of life-cycle cost of the GCHP system for both H-GCHPs and V-GCHPs.

6.5 Remediation cost:

Remediation cost can make the decision process of choosing a grout material easier. The remediation cost was calculated considering the ASHRAE 90.1 2010 model. The maximum remediation cost calculated for V-GCHPs was $64/cu.ft. That is a maximum of $64/cu.ft can be invested in grouting material to make the system more cost effective than a system using existing ground conditions. However, to match the assumed saturated soil condition, a significant volume of soil beyond 8" diameter will have to be treated which would be very cost consuming. Similarly, the remediation cost determined for H-GCHPs was 0.60 $/cu.ft. The remediation cost for horizontal systems is very low and not very encouraging, as it would be very difficult to achieve such low cost. Therefore, it would seem that it is more economical to install the H-GCHPs in the existing dry soil condition without any soil treatment.

CHAPTER 7

CLOSURE

7.1 Summary & Conclusions

The main objective of the research was to study the viability of GCHPs for the small DOE commercial prototype buildings in a hot and semi-arid climatic zone such as Phoenix, Az. Further, the overall intent of the research study was to evaluate the effect of saturated soil conditions on sizing and thermal performance of both vertical and horizontal GHE. The energy and economic performance of GCHPs were evaluated and compared with ASHPs. In this study, EnergyPlus software was used for energy performance simulations for ASHPs and GLD software was used to design and determine the energy performance of GCHPs. The lifecycle cost analysis evaluated the economic aspect of vertical and horizontal GCHPs with and without saturated soil conditions as compared with conventional ASHPs and high efficient ASHPs. The available federal incentives were also considered for LCC.

From the analysis we can conclude that, for the ASHRAE 90.1-2004 model, vertical GCHPs can save $569 (38.42%) and $249 (21.4%) on utility bills annually when compared to conventional ASHPs and high efficient ASHPs respectively. This energy savings would result in a reduction of 6.3 metric tons of carbon dioxide emissions each year. Similar results are seen with the ASHRAE 90.1-2010 model. According to it, the V-GCHPs can save $569 or 38.4 % and $309 or 26.3% and horizontal GCHPs can save $ 446 or 33.35% and $284 or 24.2 % on annual utility bills when compared with conventional ASHPs and high efficient ASHPs respectively. Savings in energy and utility cost for the ASHRAE 90.1-2004 model is greater than the savings predicted by the ASHRAE 90.1-2010 baseline model as the latter model, due to it being more energy efficient having better ASHPs equipment than the former case. From the results it is seen that the building energy consumption in both the baseline cases is dominated by the space cooling subcategory. The GCHPs have a higher efficiency for space heating than space cooling and this may have reduced the benefit of GCHPs. Moreover, the efficiency of GCHPs decreases over time due to the rise in the soil temperature because of prolonged heat dissipation into the ground.

Referring to the installation cost of the V-GCHPs and H-GCHPs, it is interesting to note that V-GCHPs are about 35-36% more costly than H-GCHPs. Moreover, the GHE is the single most expensive component in GCHPs accounting for about 34-40% of the installed cost.

Saturated soil conditions decreases the length of the vertical and horizontal GHE by 26% and 25% respectively. This decrease results in a decrease in the installation cost of approximately 19% in the case of vertical GCHPs and 15 % for horizontal GCHPs.

For the ASHRAE 90.1-2004 model, the results of the economic analysis indicates that the LCC of Alt case 9 (H-GCHPs with saturated soil conditions and federal incentives) is the least of all alternatives at $ 46,728 with a payback period of 4.8 years followed by Alt 7 (H-GCHPs with existing soil conditions and federal benefits) with a LCC of $50,811 and a payback of 14 years as compared with ASHP system to the LCC of $58,046. The vertical GCHPs has a greater LCC than ASHPs with a payback period of more than 25 years, and therefore is not considered an economically viable option.

The ASHRAE 90.1-2010 model displays similar results as compared to ASHRAE 90.1-2004 model. H-GCHPs with saturated soil condition and federal incentives were found to have the lowest LCC PV with a life-cycle cost of $46,167 (or 19% less than baseline case) and with a 2.3 year payback period, followed by horizontal GCHPs in existing soil condition with federal incentives at $48,694 LCC (or 15% less than baseline case) with a 11.1 years payback period and discounted payback of 4 years.

The high efficiency ASHPs have a LCC of $56,309 with 6.2 years of payback. The H-GCHPs with saturated soil condition and without incentives have comparatively lower LCC compared to the baseline case, with a LCC of about $ 56,994 or $331 less than the baseline case, but has payback period of about 21.3 years.

The results lead to the conclusion that the LCC of the vertical & horizontal GCHP systems without incentives and vertical systems with incentives and saturated soil condition would be greater than the ASHP system. Also, the SIR for these system investment would be less than 1 and so the direct payback period would exceed 25 years when compared to the ASHP system. Thus, the results show that the cost saving from both horizontal and vertical GCHPs without incentives are unlikely to overcome the first cost investment between the ASHPs and GCHPs. This might be due to the high cost required for the installation. With saturated soil conditions, the payback period for the ASHRAE 90.1-2004 case decreases from 107.5 years to 81.6 years (24% decrease) for V-GCHPs and from 42 years to 22 years (45% decrease) for horizontal GCHPs. Likewise, for the ASHRAE 90.1-2010 model the payback period is decreased from 119 years to 94 years (21 % decrease) for vertical GCHPs and from 42.7 years to 25.5 years (40% decrease) in the case of H-GCHPs. In terms of simple annual cash flows, the horizontal GCHPs with saturated soil and incentives has a simple payback period of 4.8 years and 3.6 years with respect to ASHPs in cases 1 and 2.

A sensitivity analysis on the vertical and horizontal GCHPs (except for H-GCHPs with incentives and saturated soil condition) cost items has shown that the sensitivity of the LCC PV to the installation cost is very significant followed by energy savings and maintenance cost, while being almost insensitive to the periodic cost. For the H-GCHPs with saturated soil condition and federal incentives, the most sensitive cost item was the installation cost, closely followed by the energy cost, maintenance cost, and then the periodic costs.

In terms of Energy Use Intensities (EUI), for the ASHRAE 90.1-2004 model, baseline model is more energy intensive at value 42.4kBtu/ft^2 as compared with higher efficiency ASHP 39.03kBtu/ft^2 and V-GCHPs 36.4kBtu/ft^2. For the ASHRAE 90.1-2010 model, the EUI for baseline case is 35.5kBtu/ft^2, higher efficiency ASHPs is 32.5kBtu/ft2 and for V-GCHPs is 30.3kBtu/ft^2

A greenhouse gas analysis has shown that use of a GCHPs system can reduce annual greenhouse gas emissions by 8.1 tons (40 %) of CO2 equivalent over ASHPs if the ASHRAE 90.1-2004 model is assumed, and by 7 tons (37.7 %) of CO2 equivalent over the use of ASHPs for the ASHRAE 90.1-2010 model.

It can be concluded that the energy performance of both horizontal and vertical GCHP systems for small office buildings in hot and dry climatic location like Phoenix is better than the performance of ASHPs. But with respect to the economic benefit of vertical and horizontal GCHPs to be economical (without incentives), it is unclear that the energy cost savings gained from the GCHPs could offset the system's initial investment costs, which are about 2-4.5 times more expensive than the baseline ASHPs. The body of evidences indicates that the V-GSHP investment is economically infeasible for the hot and semi-arid climate of Phoenix. But with the federal incentives and saturated soil conditions, H-GCHPs have excellent LCC PV values and payback periods.

The maximum remediation cost calculated for V-GCHPs was 64 $/cu.ft. whereas for H-GCHPs was 0.60 $/cu.ft. The remediation cost for H-GCHPs was significantly low and would likely prove uneconomical for any kind of soil enhancement treatments.

7.2 Future Work

The economic and energy performance of the GCHPs in this thesis was studied only for a small office building prototype. The study could possibly be extended to the feasibility of GCHPs for different building types, including residential buildings, mid to large office buildings, and retail and school buildings. Also, it would be interesting to study the decrease in the GHE length and hence the reduction in the first cost of GCHPs in school buildings where the building is mostly unoccupied during peak load season. The GCHPS installed in cooling dominated area like Phoenix reject more heat to the ground than they extract over number of years of operation which results in the gradual increase of the ground temperature. This imbalance could result in the larger GHE length increasing the first cost. Therefore a hybrid systems can possibly be used for supplemental heat rejection and would reduce the installation cost, especially for the vertical GCHPs. The hybrid system could be a cooling tower or pond cooling which could be operated during peak load conditions. Moreover, the climate in Phoenix is extremely hot and dry. The performance of GCHPs could be studied with respect to other locations, which experience less temperature variation and more balanced type of climatic like that of New Mexico.

Further research work could entail installing a prototype GCHP and performing in-situ monitoring followed by statistical approaches to evaluate the performance of GHE under various grouting materials. Also, this study was conducted on a hypothetical building with the energy performance evaluation using software rather than monitoring the performance data in an existing building. The Desert School in Phoenix has installed a V-GCHPS, and could be a good case study to further validate performance of GCHPs. There is also a need for a more sophisticated model to calculate the length and the energy use by the horizontal GCHPs on a monthly or hourly basis.

It would be useful to obtain the installation quotes from several contractors and to investigate the effect on the economic analysis of the systems. This will help us acquire more accurate installation costs and perform life cycle costing of the GCHPs more realistically.

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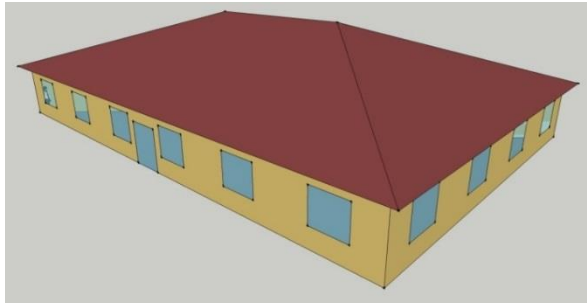
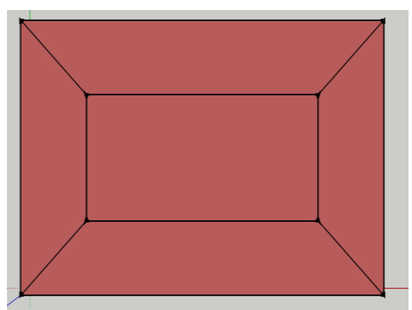
APPENDIX A

REFERENCE BUILDING SPECIFICATIONS

Table A-1 Reference Building Zone Internal Loads (IP units) (DOE, 2009)

Item	Descriptions			Data Source
Program				
Vintage	NEW CONSTRUCTION			
Location (Representing 8 Climate Zones)	Zone 1A: Miami (very hot, humid) Zone 1B: Riyadh, Saudi Arabia (very hot, dry) Zone 2A: Houston (hot, humid) Zone 2B: Phoenix (hot, dry) Zone 3A: Memphis (warm, humid) Zone 3B: El Paso (warm, dry) Zone 3C: San Francisco (warm,marine)	Zone 4A: Baltimore (mild, humid) Zone 4B: Albuquerque (mild, dry) Zone 4C: Salem (mild, marine) Zone 5A: Chicago (cold, humid) Zone 5B: Boise (cold, dry) Zone 5C: Vancouver, BC (cold, marine)	Zone 6A: Burlington (cold, humid) Zone 6B: Helena (cold, dry) Zone 7: Duluth (very cold) Zone 8: Fairbanks (subarctic)	Selection of representative climates based on Briggs' paper. See Reference.
Available fuel types	gas, electricity			
Building Type (Principal Building Function)	OFFICE			
Building Prototype	Small Office			
Form				
Total Floor Area (sq feet)	5500 (90.8 ft x 60.5ft)			
Building shape				



Aspect Ratio	1.5	2003 CBECS Data and PNNL's CBECS Study 2007.
Number of Floors	1	
Window Fraction (Window-to-Wall Ratio)	24.4% for South and 19.8% for the other three orientations (Window Dimensions: 6.0 ft x 5.0 ft punch windows for all façades)	
Window Locations	evenly distributed along four façades	
Shading Geometry	none	
Azimuth	non-directional	
Thermal Zoning	Perimeter zone depth: 16.4 ft. Four perimeter zones, one core zone and an attic zone. Percentages of floor area: Perimeter 70%, Core 30% 	
Floor to floor height (feet)	10	

Floor to ceiling height (feet)	10	
Glazing sill height (feet)	3 (top of the window is 8 ft high with 5 ft high glass)	
Architecture		
Exterior walls		
Construction	Wood-Frame Walls (2X4 16IN OC) 1in. Stucco + 5/8 in. gypsum board + wall Insulation+ 5/8 in. gypsum board	Construction type: 2003 CBECS Data and PNNL's CBECS Study 2007. Exterior wall layers: default 90.1 layering
U-factor (Btu / h * ft^2 * °F) and/or R-value (h * ft^2 * °F / Btu)	ASHRAE 90.1 Requirements Nonresidential; Walls, Above-Grade, Wood-Framed	ASHRAE 90.1
Dimensions	based on floor area and aspect ratio	
Tilts and orientations	vertical	
Roof		
Construction	Attic Roof with Wood Joist: Roof insulation + 5/8 in. gypsum board	Construction type: 2003 CBECS Data and PNNL's CBECS Study 2007. Roof layers: default 90.1 layering
U-factor (Btu / h * ft^2 * °F) and/or R-value (h * ft^2 * °F / Btu)	ASHRAE 90.1 Requirements Nonresidential; Roofs, Attic	ASHRAE 90.1
Dimensions	based on floor area and aspect ratio	
Tilts and orientations	Hipped roof: 10.76 ft attic ridge height, 2 ft overhang-soffit	
Window		
Dimensions	punch window, each 5 ft high by 6 ft wide	
Glass-Type and frame	Hypothetical window with the exact U-factor and SHGC shown below	
U-factor (Btu / h * ft^2 * °F)	ASHRAE 90.1 Requirements Nonresidential; Vertical Glazing, 20-30%, U_fixed	ASHRAE 90.1

SHGC (all)		
Visible transmittance	Hypothetical window with the exact U-factor and SHGC shown above	
Operable area	0	Ducker Fenestration Market Data provided by the 90.1 envelope subcommittee
Skylight		
Dimensions	Not Modeled	
Glass-Type and frame		
U-factor (Btu / h * ft^2 * °F)	NA	
SHGC (all)		
Visible transmittance		
Foundation		
Foundation Type	Slab-on-grade floors (unheated)	
Construction	8" concrete slab poured directly on to the earth	
Thermal properties for ground level floor U-factor (Btu / h * ft2 * °F) and/or R-value (h * ft2 * °F / Btu)	ASHRAE 90.1 Requirements Nonresidential; Slab-on-Grade Floors, unheated	ASHRAE 90.1
Thermal properties for basement walls	NA	
Dimensions	based on floor area and aspect ratio	
Interior Partitions		
Construction	2 x 4 uninsulated stud wall	
Dimensions	based on floor plan and floor-to-floor height	
Internal Mass	6 inches standard wood (16.6 lb/ft²)	
Air Barrier System		
Infiltration	Peak: 0.2016 cfm/sf of above grade exterior wall surface area (when fans turn off) Off Peak: 25% of peak infiltration rate (when fans turn on)	Reference: PNNL-18898: Infiltration Modeling Guidelines for Commercial Building Energy Analysis.
HVAC		
System Type		
Heating type	Air-source heat pump with gas furnace as back up	2003 CBECS Data, PNNL's CBECS Study
Cooling type	Air-source heat pump	

Distribution and terminal units	Single zone, constant air volume air distribution, one unit per occupied thermal zone	2006, and 90.1 Mechanical Subcommittee input.
HVAC Sizing		
Air Conditioning	autosized to design day	
Heating	autosized to design day	
HVAC Efficiency		
Air Conditioning	Various by climate location and design cooling capacity ASHRAE 90.1 Requirements Minimum equipment efficiency for Packaged Heat Pumps	ASHRAE 90.1
Heating	Various by climate location and design heating capacity ASHRAE 90.1 Requirements Minimum equipment efficiency for Packaged Heat Pumps and Warm Air Furnaces	ASHRAE 90.1
HVAC Control		
Thermostat Setpoint	75°F Cooling/70°F Heating	
Thermostat Setback	85°F Cooling/60°F Heating	
Supply air temperature	Maximum 104F, Minimum 55F	
Chilled water supply temperatures	NA	
Hot water supply temperatures	NA	
Economizers	Various by climate location and cooling capacity Control type: differential dry bulb	ASHRAE 90.1
Ventilation	ASHRAE Ventilation Standard 62.1 See under Outdoor Air.	ASHRAE Ventilation Standard 62.1
Demand Control Ventilation	ASHRAE 90.1 Requirements	ASHRAE 90.1
Energy Recovery	ASHRAE 90.1 Requirements	ASHRAE 90.1
Supply Fan		
Fan schedules	See under Schedules	ASHRAE 90.1 requirements for motor efficiency and fan power limitation
Supply Fan Total Efficiency (%)	Depending on the fan motor size	
Supply Fan Pressure Drop	Various depending on the fan supply air cfm	
Pump		
Pump Type	NA	

Rated Pump Head	NA	
Pump Power	autosized	
Cooling Tower		
Cooling Tower Type	NA	
Cooling Tower Efficiency	NA	
Service Water Heating		
SWH type	Storage Tank	
Fuel type	Natural Gas	
Thermal efficiency (%)	ASHRAE 90.1 Requirements Water Heating Equipment, Gas storage water heaters, >75,000 Btu/h input	ASHRAE 90.1
Tank Volume (gal)	40	
Water temperature setpoint	120F	
Water consumption	See under Schedules	
Internal Loads & Schedules		
Lighting		
Average power density (W/ft^2)	ASHRAE 90.1 Lighting Power Densities Using the Building Area Method	ASHRAE 90.1
Schedule	See under Schedules	
Daylighting Controls	ASHRAE 90.1 Requirements	
Occupancy Sensors	ASHRAE 90.1 Requirements	
Plug load		
Average power density (W/ft^2)	See under Zone Summary	User's Manual for ASHRAE Standard 90.1-2004 (Appendix G)
Schedule	See under Schedules	
Occupancy		
Average people	See under Zone Summary	User's Manual for ASHRAE Standard 90.1-2004 (Appendix G)
Schedule	See under Schedules	
Misc.		
Elevator		

Quantity	NA	Reference: DOE Commercial Reference Building Models of the National Building Stock
Motor type	NA	
Peak Motor Power (W/elevator)	NA	
Heat Gain to Building	NA	
Peak Fan/lights Power (W/elevator)	NA	90.1 Mechanical Subcommittee, Elevator Working Group
Motor and fan/lights Schedules	NA	DOE Commercial Reference Building TSD and models (V1.3_5.0) and Addendum DF to 90.1-2007
Exterior Lighting		
Peak Power (W)	1,634	ASHRAE 90.1
Schedule	See under Schedules	
Parking Lot Area	8,910 ft2	

Table A-2 Reference Building Zone Internal Loads (IP units)

Building Type/Zone	Area ft^2	Vol. ft^3	ft^2/ person	1989 Lights W/ft^2	2004 Lights W/ft^2	Elec. Proc. W/ft^2	Gas Proc. W/ft^2	Vent. cfm	Exhst cfm	Infil. ACH	SWH gal/h
Total	5,502	55,056									
Core_ZN	1,611	16,120	200.0	1.81	1.0	0.8	0.0	161.1	0.0	0.00	3.0
Perimeter_ZN_1	1,221	12,220	200.0	1.81	1.0	0.8	0.0	122.1	0.0	0.62	0.0
Perimeter_ZN_2	724	7,249	200.0	1.81	1.0	0.8	0.0	72.4	0.0	0.66	0.0
Perimeter_ZN_3	1,221	12,220	200.0	1.81	1.0	0.8	0.0	122.1	0.0	0.62	0.0
Perimeter_ZN_4	724	7,249	200.0	1.81	1.0	0.8	0.0	72.4	0.0	0.66	0.0
Attic	6,114	25,433	0.0	0.00	0.0	0.0	0.0	0.0	0.0	1.00	0.0

Table A-3 Minimum outdoor air requirements

Zone	Area (ft^2)	Assumed Space Type	Total Occupants 62.1-2004	Total OSA Ventilation (cfm/zone) 90.1-2004 (62-1999)	Total OSA Ventilation (cfm/zone) 90.1-2010 (62.1-2007)	Total OSA Ventilation (cfm/ft2) 90.1-2004 (62-1999)	Total OSA Ventilation (cfm/ft2) 90.1-2010 (62.1-2007)
Core_zn	1611	Office space	8	161	137	0.100	0.085
Perimeter_zn_1	1221	Office space	6	122	104	0.100	0.085
Perimeter_zn_2	724	Office space	4	72	62	0.100	0.085
Perimeter_zn_3	1221	Office space	6	122	104	0.100	0.085
Perimeter_zn_4	724	Office space	4	72	62	0.100	0.085
TOTAL	5,503		28	550	468	0.100	0.085

Table A-4 Zone Summary

Zone	Area [ft^2]	Conditioned [Y/N]	Volume [ft^3]	Gross Wall Area [ft^2]	Window Glass Area [ft^2]	Lighting [W/ft^2]	People [ft^2/person]	No. of People	Plug and Process [W/ft^2]
Core_zn	1,611	Yes	16,122	0	0	1.00	179	9	0.63
Perimeter_zn_1	1,221	Yes	12,221	909	222	1.00	179	7	0.63
Perimeter_zn_2	724	Yes	7,250	606	120	1.00	179	4	0.63
Perimeter_zn_3	1,221	Yes	12,221	909	180	1.00	179	7	0.63
Perimeter_zn_4	724	Yes	7,250	606	120	1.00	179	4	0.63
Attic	6,114	No	25,437	0	0	0.00	-	0	0.00
Total	5,503		80,502	3,030	643			31	
Area weighted average						1	179		0.63

APPENDIX B

VERTICAL GHE SIZING SPECIFICATIONS

Table B-1 ASHRAE 90.1-2004 Baseline Model: Input Parameters considered for designing V-GHE

Parameters		Values	Units	
Building Design Cooling Block Load	Q_{lc}	-87.70	kBtu/h	
Building Design Heating Block Load	Q_{lh}	39.60	kBtu/h	
Clg Load Tons		-9,928.69	Tons	
Htg Load Tons		111.75	Tons	
Energy Efficiency Ratio	EER	17.10		
Coefficient of Performance	COP	3.60		
Short Circuit Heat Loss Factor	F_{sc}	1.01	(For 3gpm/ton)	
Thermal Resistance OF Bore	R_b	0.17	h.ft.°F/Btu	Table 3.3 and Table 3.4
Days of Operation	τ_1	9,125.00	days	(25 years)
Peak Summer Month	τ_2	9,155.00	days	(9155+30 days)
Peak Four Hr Period	τ_f	9,155.17	days	(9125+30+0.167 days)
Thermal Conductivity of soil	k_g	0.89	Btu/h.°F.ft	
Thermal Diffusivity of soil	α_g	0.61	ft^2/day	
Equivalent Diameter	d_{egv}	0.22		(For 1 1/4" pipe)
Undisturbed Ground Temp	t_g	73	°F	
Liquid Temp At Heat Pump Inlet (Clg)	t_wi	93	°F	
Liquid Temp At Heat Pump Outlet (Clg)	t_{wo}	103	°F	(From Table 3.7 for 3gpm/ton)
Liquid Temp At Heat Pump Inlet (Htg)	t_{wi}	63	°F	
Liquid Temp At Heat Pump Outlet (Htg)	t_{wo}	57	°F	(From Table 3.7 for 3gpm/ton)
Temperature penalty for interference of adjacent borehole	t_p	-1.5	°F	Eqn. 13
Power Input at Design Clg Load	W_c	4.12	W	
Power Input at Design Htg Load	W_h	4.12	W	
Net annual average heat transfer to ground	q_a	-1.35	Btu/h	Eqn. 12
Monthly Part Load Factor	PLF_m	0.35		
Fourier Number for 10 yrs (3650dys) pulse	Fo_f	4,61,541.47		Eqn. 10

G Factor for 10 yrs pulse	G_f	1.28		(From Figure 3.4)
Fourier No for 1month (30dys) pulse	Fo_1	1,520.81		Eqn. 10
G factor for 1 month pulse	G_1	0.7		(From Figure 3.4)
Fourier No for 4hrs (0.167dys) pulse	Fo_f	8.41		Eqn. 10
G factor for 4 hours pulse	G_2	0.35		(From Figure 3.4)
Effective Thermal Resistance of Ground, annual pulse	R_{ga}	0.65	h.ft.°F/Btu	Eqn. 11
Effective Thermal Resistance of Ground, monthly pulse	R_{gm}	0.39	h.ft.°F/Btu	Eqn. 11
Effective Thermal Resistance of Ground, daily pulse	R_{gd}	0.39	h.ft.°F/Btu	Eqn. 11

Table B-2 ASHRAE 90.1-2010 Baseline Model: Input Parameters considered for designing V-GHE

Parameters		Values	Units	
Building Design Cooling Block Load	Q_{lc}	-82.10	kBtu/h	
Building Design Heating Block Load	Q_{lh}	48.00	kBtu/h	
Clg Load Tons		-8756.38	Tons	
Htg Load Tons		151.39	Tons	
Energy Efficiency Ratio	EER	17.10		
Coefficient of Performance	COP	3.60		
Short Circuit Heat Loss Factor	F_{sc}	1.01	(For 3gpm/ton)	
Thermal Resistance OF Bore	R_b	0.17	h.ft.°F/Btu	Table 3.3 and Table 3.4
Days of Operation	τ_1	9125.00	days	(25 years)
Peak Summer Month	τ_2	9155.00	days	(9155+30 days)
Peak Four Hr Period	τ_f	9155.17	days	(9125+30+0.167 days)
Thermal Conductivity of soil	k_g	0.89	Btu/h.°F.ft	
Thermal Diffusivity of soil	α_g	0.61	ft^2/day	
Equivalent Diameter	d_{egv}	0.22		(For 1 1/4" pipe)
Undisturbed Ground Temp	t_g	73	°F	Eqn. 13

Liquid Temp At Heat Pump Inlet (Clg)	$t_{w}i$	93	°F	
Liquid Temp At Heat Pump Outlet (Clg)	t_{wo}	103	°F	(From Table 3.7 for 3gpm/ton)
Liquid Temp At Heat Pump Inlet (Htg)	t_{wi}	63	°F	
Liquid Temp At Heat Pump Outlet (Htg)	t_{wo}	57	°F	(From Table 3.7 for 3gpm/ton)
Temperature penalty for interference of adjacent borehole	t_{p}	1.5	°F	
Power Input at Design Clg Load	W_{c}	4	W	
Power Input at Design Htg Load	W_{h}	4	W	
Net annual average heat transfer to ground	q_{a}	-1.19	Btu/h	Eqn. 12
Monthly Part Load Factor	PLF_{m}	0.35		
Fourier Number for 10 yrs (3650dys) pulse	Fo_{f}	461541.48		Eqn. 10
G Factor for 10 yrs pulse	G_{f}	1.25		(From Figure 3.4)
Fourier No for 1month (30dys) pulse	Fo_{1}	1520.82		Eqn.10
G factor for 1 month pulse	G_{1}	0.75		(From Figure 3.4)
Fourier No for 4hrs (0.167dys) pulse	Fo_{f}	8.42		Eqn. 10
G factor for 4 hours pulse	G_{2}	0.3		(From Figure 3.4)
Effective Thermal Resistance of Ground, annual pulse	R_{ga}	0.56	h.ft.°F/Btu	Eqn. 11
Effective Thermal Resistance of Ground, monthly pulse	R_{gm}	0.51	h.ft.°F/Btu	Eqn. 11
Effective Thermal Resistance of Ground, daily pulse	R_{gd}	0.34	h.ft.°F/Btu	Eqn. 11