INVESTIGATION OF PUMPING AND PIPING SYSTEM DESIGN FOR COMMERCIAL GROUND SOURCE HEAT PUMPS

By

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NOMENCLATURE

GSHP – Ground Source Heat Pump
LCC – Life Cycle Cost
HVAC – Heating, Ventilating, and Air Conditioning
GHX – Ground Heat Exchanger
IEA – Internal Energy Agency
AOP – Actual Operating Point
ECM – Electronically Commutated Motor
VFD – Variable Frequency Drive
WTW – Wire to Water Efficiency
IM – Induction Motor
AC – Alternating Current
DC – Direct Current
AFLV – Automatic Flow Limiting Valve
HDPE – High Density Poly Ethylene
DP – Differential Pressure (sensor)
EFT – Entering Fluid Temperature
COP – Coefficient of Performance
q – Heat Transfer Rate; W (Btu/hr)
$\dot{m}$ – Mass flow rate; kg/s (lbm/s)
Cp – Specific Heat; kj/kg--°C (Btu/lb--°F)
T – Temperature; °C (°F)
ΔT – Change in Temperature; -°C (°F)
R – Thermal Resistance; °C /W (hr-°F/Btu)
L – Length; m (ft)
D – Diameter; m (ft)
Z – Depth; m (ft)
k – Thermal Conductivity; W/m°C (Btu/hr°F-ft)
r – Radius; m (ft)
ε – Roughness; mm (in)
h – Head Loss; m (ft)
ρ – Density; kg/m^3 (lb/ft^3)
V – Velocity; m/s (ft/s)
g – Acceleration of Gravity; m/s^2 (ft/s^2)
K – Fitting Loss Coefficient
Q – Volumetric Flow Rate; L/s (GPM)
η – Efficiency; - (-)
E – Energy; kW
SG – Specific Gravity; - (-)

CHAPTER I

INTRODUCTION

On 5 October 2009, President Barrack Obama signed Executive Order 13514 (EO 13514) which set sustainability goals for federal government buildings and to increase energy efficiency. Starting in 2020, all federal buildings entering the design phase are required to be designed to meet "Net Zero" energy consumption. This in layman's terms means that the building must be able to produce as much energy with renewable sources (solar, wind, hydro) as it consumes over a one year time frame. According to the United States Department of Energy, the Department of Defense (DoD) is responsible for 54% of all federal building primary energy consumption. This is not surprising as the DoD also accounts for 63% of the floor space.

A Ground Source Heat Pump (GSHP) takes advantage of the earth's ground temperature remaining constant. Below a location-specific depth, the temperature of the earth is unaffected by the daily temperature swings and remains a constant temperature year round. By taking advantage of this fact, a buildings heating and cooling system will consume up to 75% less energy by installing a GSHP system rather than a conventional heating and cooling system (Omer 2008). Despite the probability for reduced energy consumption, if not properly designed, GSHP systems can operate with higher energy consumption than a conventional system (Kavanaugh and McInerny 2001). The decrease in energy consumption comes at a cost. Compared to a rooftop unitary system, which is commonly used in commercial applications, the first cost of a GSHP is 20-40% higher (Kavanaugh and Rafferty 1997). Many consumers tend to shy away from GSHP's due to the higher first cost even though the overall life cycle cost (LCC) is lower.

In its basic form, a closed loop GSHP system is broken into four parts, one or more heat pumps, one or more circulation pumps, the fluid distribution system, more commonly known as the piping system, and the valves used to control the flow in the piping system. The focus of this thesis is the investigation of optimal pumping and piping designs of a commercial closed loop GSHP to ensure optimal performance of the entire system.

1.1 BACKGROUND GSHP SYSTEMS

When a designer has determined to specify a GSHP system for a commercial heating, ventilating, and air conditioning (HVAC) system, there are many project-specific factors that have to be taken into account in addition to the normal HVAC requirements. In a GSHP, heat is either absorbed by or dissipated from a facility via a water source heat pump. The water source heat pump provides heating and cooling by absorbing or dissipating heat from a supply of source water and transferring it to the air supplied to the facility to maintain a desired indoor temperature. The heat gained or lost by the water is then transferred to another median through the ground heat exchanger (GHX). Figure 1.1 shows the four different categories of GSHP's and some basic configurations of each. This thesis will focus on the vertical closed loop configuration of the GHX. In the vertical closed loop GHX, the fluid is pumped into the earth to a depth sufficient enough to transfer enough heat as required by the HVAC system. This is normally in the range of 50 m (164 ft) to 150 m (492 ft) (Bernier 2006). The hole going into the ground is called a borehole. Commercial systems usually require multiple boreholes run in parallel to obtain sufficient heat transfer; together they are called a borefield. We will discuss borefields more in depth later in the discussion of piping design.

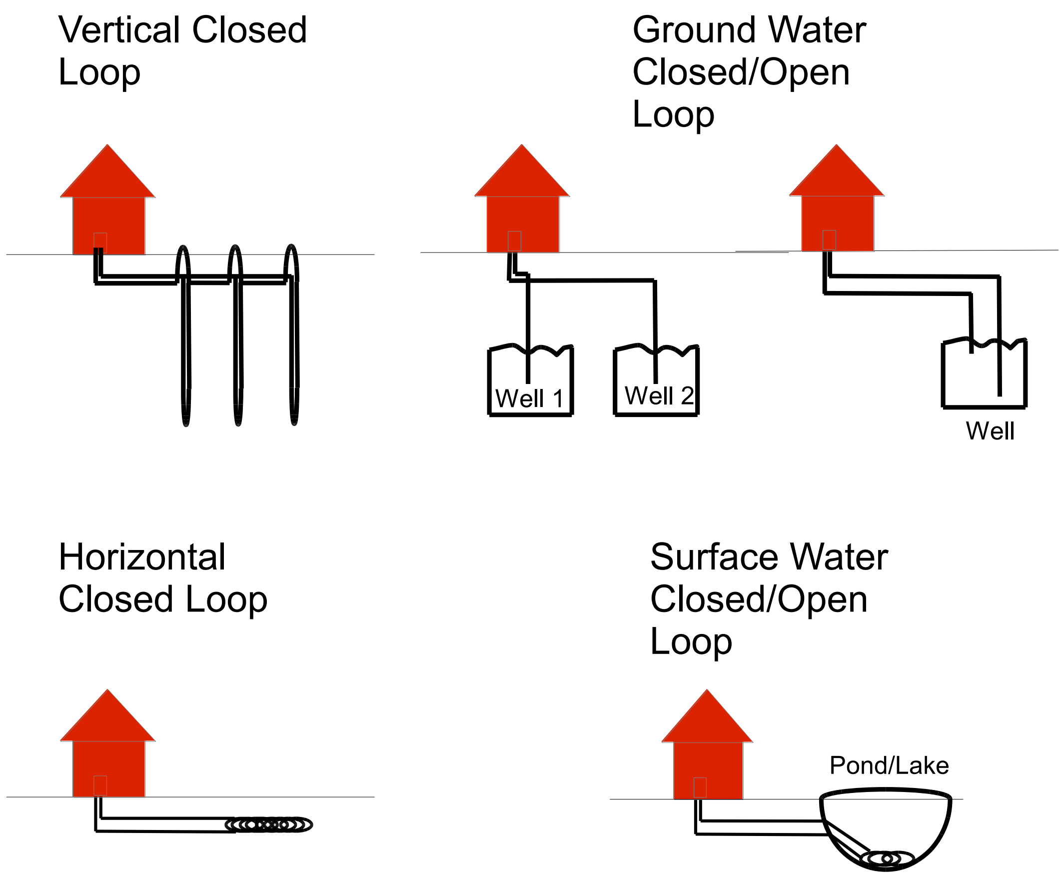

Figure 1.1 Types of GSHPs

The fluid is pumped through the borefield and back into the facility through a network of pipes. Though the pipes themselves do not directly consume energy, they have a large effect on the performance of the whole system. Since we are focusing on a closed loop system, the system does not have any static head loss. The only head losses in the system are the frictional losses of the fluid flowing through the pipes and the friction head associated with the fittings, valves, and the heat pump itself. There are many ways to configure the piping system both inside and outside the facility. The most common ones will be discussed later.

GSHP are gaining popularity due to the reduced energy consumption as compared to the traditional HVAC system. If no supplemental systems are used (such as a dedicated outdoor air system), and assuming that control system energy consumption is negligible, there are only two sub-systems in a basic GSHP that consume energy, the heat pump and the circulation pump. With high efficiency heat pumps the circulation pump can account for as much as 45% of the total consumption (Kavanaugh and Rafferty 1997). If the circulation pump efficiencies are improved and with proper design of the whole system, the circulation pump is estimated to be able to be reduced to below 5% of the total system consumption (Kavanaugh et al. 2003).

1.2. BACKGROUND PIPING SYSTEMS

When designing a GSHP system, great care should be taken in designing the hydronic piping loop. Standard convention in pipe sizing calls for designing with a maximum head loss or maximum fluid velocity. Proper design of the distribution system should minimize the number of valves and fittings. Significant energy savings can be realized by simply converting a constant speed system to a variable speed system and removing the balancing valves used (Rishel 2005).

1.2.1 BACKGROUND PUMPS

Proper pump selection is critical to the efficiency of a GSHP system. According to an independent study reported by the International Energy Agency (IEA), energy efficiency savings of 80% to 90% could be achieved in heating system circulator pumps if all available technologies were used (Waide and Brunner 2011). Part of these savings comes from selecting a pump that is properly sized for the system in which it is pumping for. Shown in ASHRAE research project 1217, oversizing the pump is common and can result in pumping costs that amount to more than the compressor and fans combined (Kavanaugh et al. 2003). With a 33% change in flow rate, there is only a 2% change in heating/cooling capacity for the newer extended range high capacity heat pumps (Kavanaugh and Rafferty 1997). This statement can be verified with manufacturer's data and was found to be the case with an arbitrarily selected ClimateMaster Tranquility series 3 ton unit (ClimateMaster 2012) . What is not mentioned by Kavanaugh and Rafferty, for the selected heat pump, is that the 33% change in flow rate also has a 4% change in the energy efficiency ratio (EER), which directly affects the energy consumption.

In percentage numbers, supply temperatures to the heat pumps can have a much bigger effect on the efficiency of the system. Figure 1.2.1.1 is data taken from a Climate Master Tranquility heat pump as an example. Each data point represents the EER for a specific flow rate at each given fluid temperature entering the heat pump. For each degree of temperature change, there is a 1-2 percent change in the performance of the heat pump. There are three data points for each fluid temperature to signify the performance at the low, mid, and high fluid flow rates.

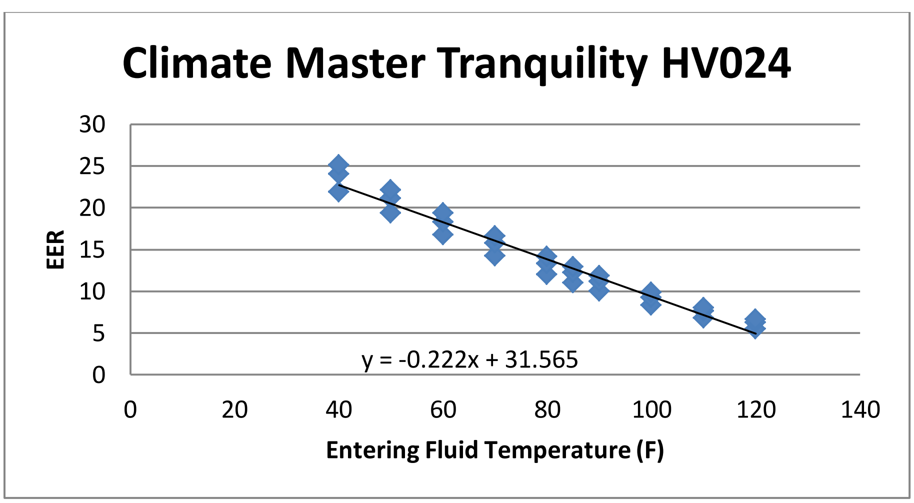

Figure 1.2.1.1 Performance Data From a Climate Master Heat Pump(ClimateMaster 2012)

Once a heat pump has been selected, the flow rate of a GSHP system is determined by how many heat pumps are installed and what the manufacturer specifies as a required flow rate for each of the installed heat pumps. Since the manufacturer gives a wide range of flow rates rather than dictating a specific flow rate, the designer must select a flow rate within that range. This range can vary according to heat pump capacity, model number, and manufacturer. The pressure drop across a system will be discussed later in this thesis, but is a function of the size and length of the pipes installed and the number and types of fittings, valves, and heat pumps installed. Together these two characteristics (flow rate and pressure drop) make up the design operating point. A pump is selected by its ability to match the design operating point and the power required for the pump to do so. In a typical pump curve, the pump manufacturer will provide performance curves that show how the pump will perform at various speeds and/or impeller diameters. The power required for a pump to match the design operating point is determined by the pump efficiency and motor efficiency at that point. Ideally, the pump and motor should be operating at their respective maximum efficiency point at or near the design operating point. This both decreases energy consumption and increases the life of the pump (Rishel et al. 2006). The most basic way to operate a pump in a GSHP system is to install a constant speed pump that runs continuously. Figure 1.2.1.2 shows a pump curve and a system curve. When turned on, a pump will "ride" or operate on its curve increasing the flow rate or decreasing the flow rate until equilibrium is reached. This is when the pressure rise and the flow rate produced by the pump at its operating speed is matched by the pressure drop across the system at the produced flow rate. When the pump is properly sized for the system, the equilibrium point, known as the actual operating point (AOP) will be close to the design operating point and the maximum efficiency point of the pump.

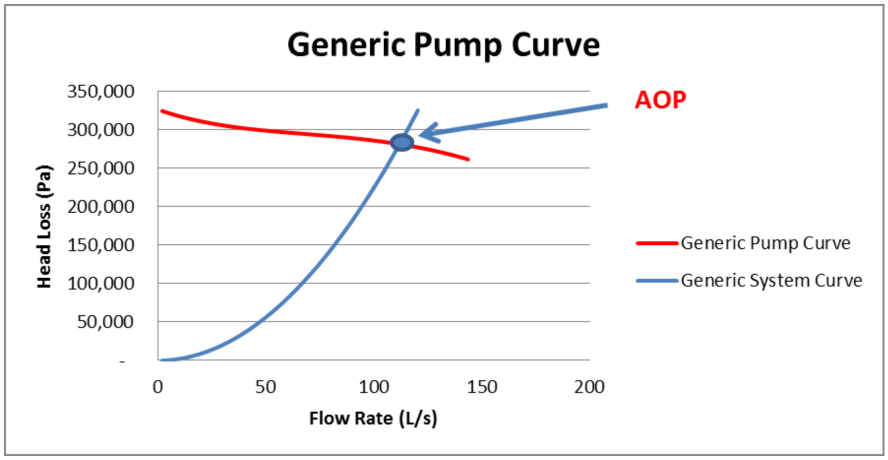

Figure 1.2.1.2 Generic Pump/System Curve

Commercial buildings rarely have only a single zone or just a single heat pump. Systems with multiple zones may fall under section 6.5.4.4.1 of ASHRAE standard 90.1 which requires two-position automatic valves be installed that are interlocked with the compressor shutting off flow to the heat pump when the heat pump is turned off. ASHRAE standard 90.1 section 6.5.4.1 dictates that "hydronic heat pumps that have a total pumping power of 5 HP or greater, must have an ability to decrease the pump motor wattage to below 30% of the design wattage at 50% of the design fluid flow rate". According to the ASHRAE Standard 90.1 User's Manual, this can be achieved by using multiple smaller pumps that turn off when not needed, two speed motors, a VFD, or an electronically commutated motor (ECM). If using a technology or method to meet this requirement other than a VFD, data demonstrating that the performance of the system meets the required energy reduction is required to be submitted to the Authority Having Jurisdiction. Most commonly this is achieved using a VFD.

Energy savings can be significant if there is a significant amount of operation below the design load and means to turn the VFD off when no heat pumps are running. As a general rule of thumb, the savings from a VFD can be shown by the pump affinity laws. This states that the energy savings are proportional to the speed cubed. This is an approximation at best. As shown by Bernier and Bourret (1999), this would be true if the pump, motor, and VFD efficiencies were constant at all speeds. Since all of the efficiencies vary with speed, using this approximation can lead to significant over estimation of the energy savings. Oversizing the pump for the system will only compound the effects. From measured results, the energy savings by installing a VFD versus using a constant speed pump ranged from 36% to 88% (Henderson et al. 2000). When multiple heat pumps are operated in parallel, each time a heat pump turns on or off the system curve changes, which means this approximation will not apply for this type of system.

There are multiple opinions on what the lowest speed allowable is on a VFD. Of course this may vary by size and manufacturer, but peer reviewed literature has shown ranges from 10% to 25% (Underwood 2003; Hamstra 2012). Rishel (2006) says there is no minimum speed for a variable frequency drive, "that there has been misinformation leading to an artificial limit". To obtain the wire to water (WTW) efficiency for a pumping system, the efficiencies of the pump, the motor, and the VFD are multiplied together to form an equivalent efficiency. Bernier and Bourret (1999) gave an argument showing that the WTW efficiency decreases with motor speed in a non-proportional manner, and decreases faster than the theoretical power required at speeds below 30% to 40% of the design speed. In the scenario provided in the article, the efficiency decreased so much below that speed that it actually required more power input to pump at 20% of the design speed than at 30%.

Multiple authors (Gao 2002; Rishel 2003; Rishel 2010; Nailen 2012) have discussed the effects of a VFD on motor efficiencies and that the pump efficiency can decrease up to 5%. Nailen explains that the effects of the VFD on the performance of the motor can have up to 10% error on calculations. All authors state that the effects vary depending on the VFD that is installed. The motor is heavily dependent on the switching speed of the converter. Gao states that the efficiency of the motor decreases with increasing switching frequency. Currently most or all energy calculations assume the motor efficiency is not affected by this.

Designers often run pumps in parallel for system redundancy and as a way to save in energy costs. When running two constant speed pumps in parallel, you can graph the output of the two pumps by doubling the flow rate for a given pump head. This does not by any means double the flow rate in the system as shown in Figure 1.2.1.3. In this example a single pump would provide slightly more than 5 kg/s (80 GPM) of flow to the system at full speed shown as point A. If two pumps were run in parallel, the frictional losses of the system would greatly increase for a small increase in flow rate. Because of the increase in head loss in the system, the flow rate through each individual pump would be less than if only one was operating. The actual operating point with two pumps running is shown as point B. The actual output of each individual pump in this case dropped and only slightly increased the total flow rate in the system. The amount of the increase is heavily dependent on the pump selected for the system and the nature of the system curve, but this example shows the flow in the system does not double.

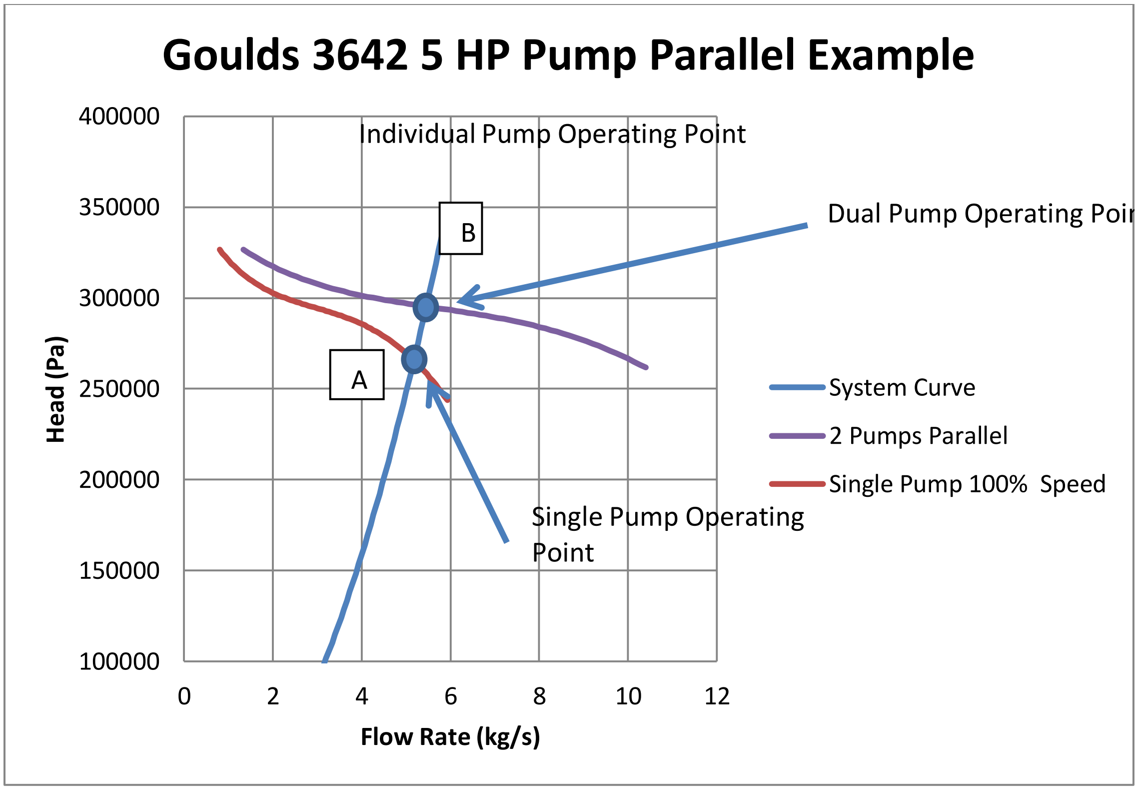

Figure 1.2.1.3 Parallel Pumping Example

When operated in parallel, you should always have two pumps that are similar in size. If one pump is much larger than the other, it will force the smaller pump to operate at the shut off head condition causing it to overheat and wear out. Rishel (2006) recommend that the smaller pump should not be sized less than 62% capacity of the larger pump. If the pumps are installed with a VFD, the two pumps should be run at the same speed or within a couple percent speed of each other (Tillack and Rishel 1998; Rishel et al. 2006). The reason for both of these can be described using Figure 1.2.1.4 and basic pump principles. When one pump is operated at 100% speed and the other pump starts reducing the speed, the combined pump curve starts shifting down as shown in the “Second Pump 95% Speed to 80% Speed” curves. Each time you decrease the speed of the pump, that reduced pump will operate along the new pump curve to match the pump head of the full speed pump. As the curve shifts down more and more the pump will operate well outside it's best efficiency range and will eventually operate at the shut off head condition, shown in this figure as the black line, and damage the pump and motor. From this figure, the pump that is being slowed down will reach the shut off head somewhere between 80% to 85% of the design speed. If the pump is operated any slower than this speed, it will not be able to match the pressure rise of the pump running at full speed.

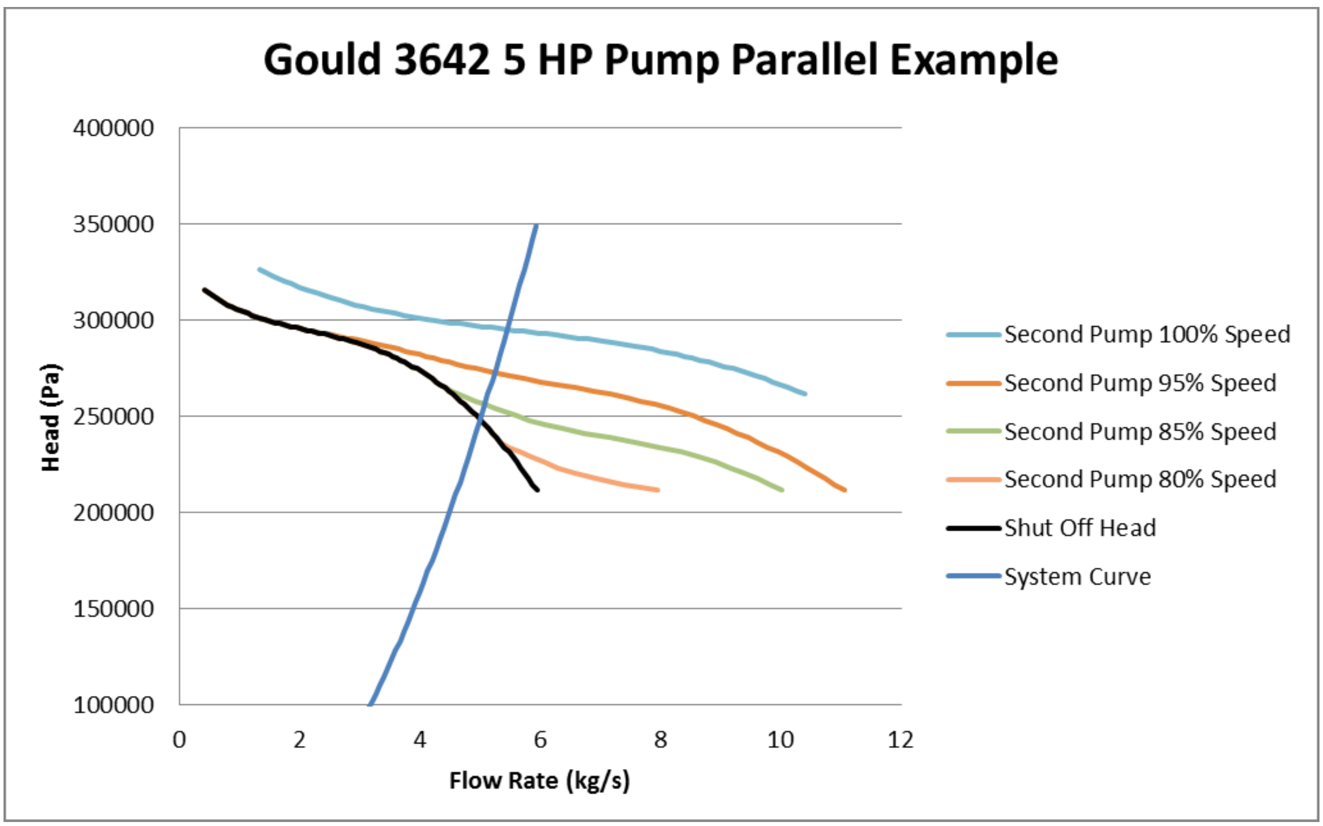

Figure 1.2.1.4 Parallel Pumps at Different Speeds

A common problem with installed systems using a VFD is that there are no provision to turn off the pump when no zones are operating (Kavanaugh 2011). It is recommended by the 2007 ASHRAE handbook-HVAC Applications that for buildings with occupancies less than 60 hours a week, there should be means to turn the main pump off and supply critical zones with an alternate smaller pump. Some heat pumps are either equipped with relays that can be used to open/close motorized valves or turn on/off individual circulator pumps or have the option for them to be installed. But when using a distributed heat pump design, provision needs to be made to detect if one or more heat pumps are operating. A multi zone relay center can be used to control the operation of individual circulator pumps or opening and closing zone valves, sensing if the heat pump is operating. These types of relays also can be used to control the primary pump to sense if any one of the zones is operating.

VFDs are installed to control induction motors (IM) that operate on alternating current (AC) power. One of the downfalls of this type of motor is that it has poor efficiencies in smaller motors and at decreased speeds (Kavanaugh et al. 2003). Direct current (DC) motors are common among small motors, but have short life spans and are more expensive (Jones 2011; Waide and Brunner 2011). The reason for this is that the DC motor requires a brush to commutate the motor. The brush rubs against the commutator and requires routine maintenance to keep the motor running optimally. Because of the frictional losses from the brush to the commutator, these motors normally get warm and efficiency decreases over time. To fix this problem, some DC motors are now brushless commutated motors or commonly referred to as electronically commutated motors (ECMs). ECMs essentially replaced the brushes with a Hall effect sensor which changes its output voltage based on the magnetic field that it senses. Since this allows the motor to commutate without the friction of a brush the motor operates more efficiently and has a longer more reliable lifespan (AAON n.d.). Much like the traditional DC motor, the ECM can vary the speed of the motor by changing the voltage supplied and does not need a separate VFD to be installed. The ECM had proven to have much higher efficiencies than their equivalent sized IM that have controlled the small pump market. Markusson (2011) performed tests of a couple readily available circulator pumps on the market, three with ECMs and one with an IM. The pumps with ECM motor obtained about a 50% max efficiency, where the pumps with IM only reached a maximum of 20%. She also references tests performed on small circulators by the Swedish Energy Agency which gave circulator efficiencies from 10% to 25%. In the US, these pumps are becoming more common in industry (Mescher 2012). Manufacturers such as Grundfos, Wilo, Bell and Gossett, and Taco have introduced pumping systems that have integrated controls and use ECM technology.

Operating modes for some common pumps with ECMs can include maintaining a constant pressure differential, linearly increasing the pressure differential as flow increases, increasing pressure differential with increasing temperature or decreasing pressure differential with increasing temperature. These pumps have many control algorithms that make the circulation pump demand driven. This expands the variable flow capabilities for smaller systems to become even more efficient. Combining a multiple speed circulation pump with a multiple speed heat pump could produce significant savings.

1.2.2 VALVES AND FITTINGS

Hydronics in HVAC is evolving. There is a trend that shows that installing multiple valves to ensure proper flow to each circuit is not only unnecessary, but it is wasteful in terms of energy efficiency (Taylor and Stein 2002; Rishel 2005). Rishel states that with today's hydronic designs, there should be pumps, piping, isolation valves, and coils with control valves. Now this is specified with the traditional hydronic HVAC system in mind, but the same could be said about GSHP designs. With VFDs installed on a two-pipe system, the designer can control the flow to each heat pump by balancing the flow at full capacity and at a pressure set point. In a laboratory study, Lambert and Kavanaugh (2004) studied the flow rates in parallel circuits when varying the flow rate through the system. They did this with both constant speed circulators and with a VFD. Instead of heat pumps though, they used water coils that acted as a "load" in the system. The result in this case, showed that this type of system with a VFD maintains a constant flow rate across each coil regardless of how many coils are in operation. If the VFD maintains the pressure set point, the flow automatically adjusts each time a heat pump turns on or off opening or closing its respective valve. As a part of this study, they showed that using a spring check valve has larger pressure drops than using a swing check valve. Since valves and fittings can represent a significant portion of the friction loss in a HVAC piping system, care should be taken to minimize the effects (Rishel et al. 2006). Rishel also states that calculating frictional losses to fittings is important but is imprecise. He says that variances in fitting loss coefficients can be as much as 50% depending on the fitting. Even though there are inaccuracies, this should not be overlooked. When specifying valves, a designer should take into account the loss of the fitting. For example, swing check valves have a lower pressure drop then spring check valves while accomplishing the same job (Lambert and Kavanaugh 2004). 45 degree elbows have lower pressure drops than 90 degree elbows but are not always appropriate. Ball valves have a lower pressure drop than both globe valves and gate valves(Rishel et al. 2006). Some valves are not suitable for every application and dependability does play a factor, but a designer should try and minimize the effects of fittings and valves. According to the pressure loss charts in Rishel's Pump Handbook (2006), installing an extended radius 90 degree bend can decrease the pressure loss in the elbow by half when compared to a standard 90 degree bend. Further analyzing the pressure loss tables shows for other cases, screw in fittings typically have twice the pressure drop as a flanged fitting.

Isolation valves are required when there is more than one heat pump in a system to allow other zones to continue operating while any one zone is being serviced. An isolation valve is any type of valve that can be manually operated to stop flow to a branch. This can be done with a ball valve or a gate valve. These two types of valves are ideal for this application as they provide a low pressure drop when they are fully open. A globe valve has a much higher pressure drop across it and should not be used for isolation (ClimateMaster 2011). The globe valve is designed to throttle flow through a pipe by creating a larger pressure drop. As discussed above, the use of throttling flow is discouraged unless absolutely necessary as it wastes energy in a pumping system.

For any type of closed hydronic system, a thermal expansion tank is commonly installed to alleviate pressure buildup from changes in temperature. As the temperature of a circulating fluid rises, the static pressure of a system also rises. Expansion tanks also help reduce the effects of water hammer on hydronic systems. Since most expansion tanks are also a closed tank, a pressure relief valve is also good idea and in most cases required by code. The 2012 International Plumbing Code section 504 requires a valve be installed when a boiler is installed (Ching and Winkel 2012). In the case of a malfunction, a pressure relief valve will prevent excess pressure from causing a blow out in one of the pipes or possibly damaging equipment in the case that these pressures become extremely high.

As recommended earlier by Rishel (2005) and required by ASHRAE standard 90.1, isolation valves are also used to stop flow to a heat pump when it is not operating. This is done with an automatic two way zone valve which is a type of flow control valve. These types of valves are rated on the closing power or amount of flow they can close against and the speed at which they close. Valves can close in as little as a couple seconds up to a couple of minutes. The valves are wired in a normally closed fashion and wired to the compressor of the heat pump such that when the compressor turns on, the valve opens and when the compressor shuts off, the valve returns to the closed position. The pressure drop across these valves is typically much greater than that of an isolation valve.

Another type of flow control valve commonly used in GSHP system is the automatic flow limiting valve (AFLV). This valve also creates a large pressure drop in a pipe system, but is used when there is a significant difference in required flow rates through parallel heat pumps. This type of valve is given a minimum and maximum pressure set point to which it will perform. As long as the minimum and maximum pressures are satisfied, this valve will allow a set flow rate to pass through. A designer should do an economic comparison to see if it would be more cost effective to install an AFLV or a booster pump to compensate for the higher flow rate required.

A standard header design that distributes the flow to each individual borehole consists of tees, elbows, reducers, and expanders. An example of this is shown in Figure 1.2.2.1 A) where a step down header supplies three boreholes. Each fitting has a small pressure drop associated with it. As the flow rate decreases going down the length of the header, the pipe size is decreased. This has been the accepted way to design a header (ASHRAE 1995). In the design of a three borehole header, the installer has eight joints that need to be thermally fused together. This can be labor intensive and will increase the installation cost. Companies such as GeoSolar have come out with fittings that can help reduce the total pressure drop across the header and create parallel loops with equal pressure drops. Figure B) shows an example of an equivalent header feeding three boreholes. This design has a couple of advantages. For the installer there only four joints that need to be fused together which cuts the number of joints in half. A crew of two to three people can install five thermal fusions per hour (Group 2013). In this simple example, the consolidated header saves half of the time for this one header in addition to equalizing the pressure drop across the three boreholes.

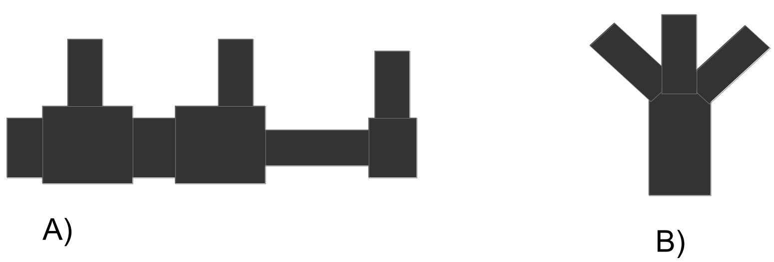

Figure 1.2.2.1 Header Designs

This fitting would work well for small systems with multiple boreholes in a line, L-shape, or an open rectangle configuration. This may not work well for larger borefields, in which the standard step down header design would be recommended.

1.2.3 PIPING

Pipe sizing can be done using various rules of thumb. One commonly accepted rule of thumb is for pipes two inches and smaller, the maximum flow velocity should be no more than 1.22 m/s (4 ft/s). For pipes larger than two inches, a maximum head loss of 1.2 m (4 ft) per 30.5 m (100 ft) of pipe is used (ASHRAE 1995; McQuiston et al. 2010). This was originally the case as small pipes were more frequently found near occupied areas and were given a velocity limitation to ensure minimal noise from fluid circulating in the pipes. The result from this was excessively high head loss in small pipes. Figure 1.2.3.1 shows the head loss per 30.5 m (100 feet) of pipe for various nominal diameters of SDR-11 HDPE pipe ranging from three quarters of an inch to two inches. In this figure, the head loss for three quarters of an inch pipe is over double the four feet of head loss for the larger diameter pipes when using the 1.22 m/s (4 ft/s) design criteria. Since GSHP systems can have a significant portion of the piping below two inches, this rule of thumb (maximum flow rate = 1.22 m/s (4 ft/s)) will result in piping networks with a large head loss.

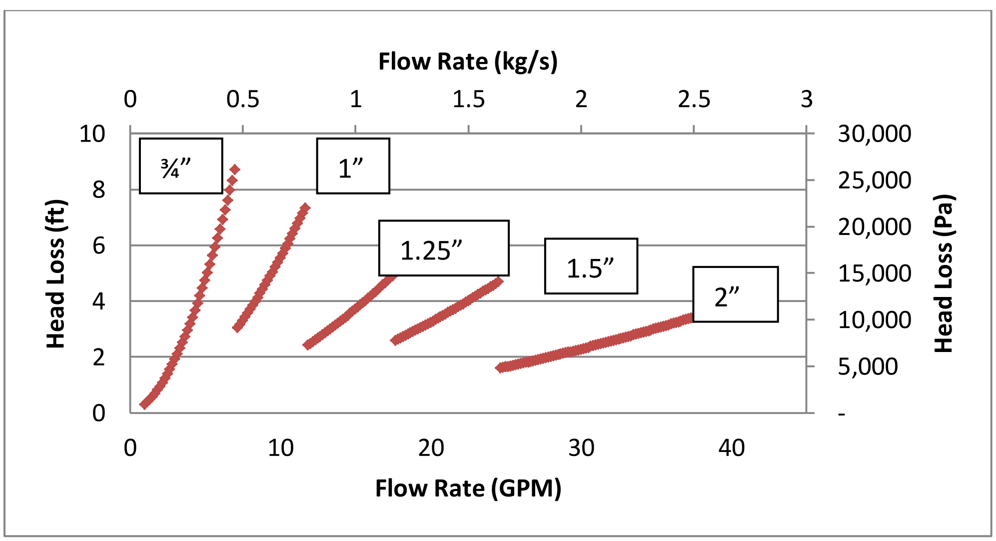

Figure 1.2.3.1 Head Loss per 100 Feet of Pipe

Piping for a GSHP can be broken up into two main sections; the ground loop and the building loop. The ground loop typically consists of high density polyethylene pipe (HDPE) that is inserted into a borehole that can range from 15 m (50 ft) to 183 m (600 ft) in depth and has a thermally fused U-bend at the bottom of the hole (McQuay 2002). HDPE pipe has a lifespan of at least 50 years in the ground. Since HDPE pipe is thermally fused it is less susceptible to leaks.

The building loop commonly has a different type of pipe installed. Some of the most common are PVC, black steel, and galvanized steel (ASHRAE 1995). Where allowed by local codes, PVC is frequently used due to the ease of installation and low cost. PVC is lighter to hang, but is not as stiff as steel and requires more support hangers. For smaller systems, HDPE can be used for the building loop also, but in large buildings HDPEs susceptibility to thermal expansion may cause issues.

As a general guideline, to maximize heat transfer between the fluid and the ground, the designer should ensure a minimum flow velocity that maintains turbulent flow in all conditions. Recent research suggests that this may not be as important as once thought. According to Kavanaugh, high fluid velocities can result in high pumping power with little thermal advantage over non-laminar flow during part load conditions (2011). He explains that though this does enhance heat transfer from the fluid to the ground, designers shouldn't assume that turbulent flow must be maintained and should do an economic analysis on this strategy. He goes on to explain that the most important consideration in designing the ground loop is the thermal resistance of the ground. To help promote heat transfer with the ground, the borehole is filled with a thermally enhanced grout. Kavanaugh et al. (2003) report that if design guides (Kavanaugh and Rafferty 1997) are used, the pressure drop in a single borefield can be as low as 1.83 m (6 ft) of head loss.

According to Kirk Mescher, a properly designed borefield should be kept to less than 7.6 m (25 ft) of head loss with a total system pressure drop of 15.2 m (50 ft) (ASHRAE 2011). As a result, the designer can minimize the size of the pump required to circulate the fluid as well as minimize the total pumping energy consumption.

1.3 PIPE CONFIGURATIONS

Vertical closed loop GSHP systems transfer heat from a facility to the ground or vice versa. This is done through a fluid that is pumped through a network of pipes into boreholes and then back to the facility. Most commercial vertical GSHP systems will require more than one borehole and should be connected in parallel. When connecting boreholes in parallel, pipes are run in direct return or reverse return. Examples of each are shown in Figure 1.3.1

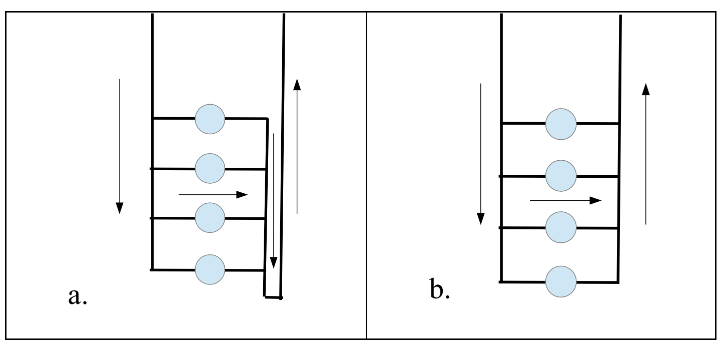

Figure 1.3.1 Parallel Pipe Configurations

Figure 1.3.1 (a) shows a reverse return configuration. The advantage of this configuration is that regardless of which borehole the fluid flows through, it sees the same length of pipe and therefore has relatively equal pressure drop. As a result, each borehole will also see approximately the same flow rate. Picture (b) shows a direct return configuration. This requires less pipe than the reverse return configuration which decreases installation costs, but does not provide equal pressure drop across boreholes. A direct return system can be used if flow rates can be kept within 15% of design flows, otherwise it is recommended to use reverse return (McQuay 2002). A detailed analysis should be done for each situation to determine which configuration is appropriate.

1.3.1 BOREFIELD CONFIGURATIONS

There are many different ways to arrange a borefield. Figure 1.3.1.1 shows sample configurations from GLHEPro V4.0, a design software for designing borefields. Arrangement of the borefield is limited only by the imagination of the designer and the site conditions.

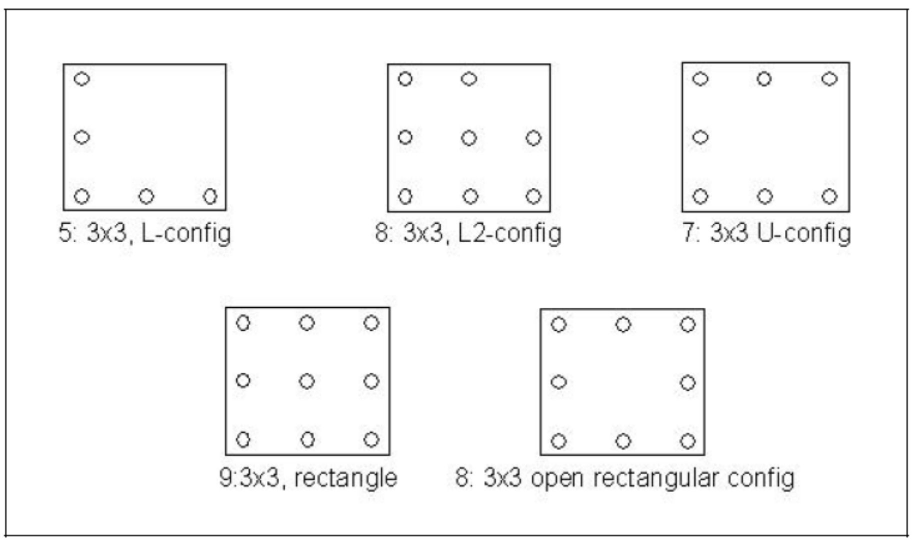

Figure 1.3.1.1 Common Borehole Arrangements (GLHEPRO_V4.0)

Typical spacing of boreholes in these configurations can range from 6.1 m (20 ft) to 9.1 m (30 ft) (Rafferty 2000; ASHRAE 2007 ; Kavanaugh 2008; Spitler and Bernier 2011). Overall, the design of the headers can have a significant impact on the system (Kavanaugh et al. 2003). If designed properly, the boreholes should not need any valves with the exception of an isolation valves put on the primary or sub header, used for purging the system of air during commissioning. This can also be used to isolate a line of boreholes in the case that a pipe starts to leak while allowing the remaining functional boreholes to be used.

1.3.2 BUILDING PIPING CONFIGURATIONS

Interior building distribution piping design, much like ground distribution pipe design is situational. Despite this, there are three different ways to configure the piping, a unitary system, a two-pipe system and a one-pipe system.

The unitary system is the easiest way to design the system and is recommended for smaller commercial systems, less than 4,645 square meters (<50,000 sq ft.) (Kavanaugh et al. 2003), where feasible. The energy performance for these systems has shown to be very good compared to other types of configurations (Kavanaugh and Kavanaugh 2012a) In this type of system, shown in Figure 1.3.2.1, each zone or heat pump is connected to an individual bore field. The pump is operated on/off with the heat pump and is usually a small constant speed circulator.

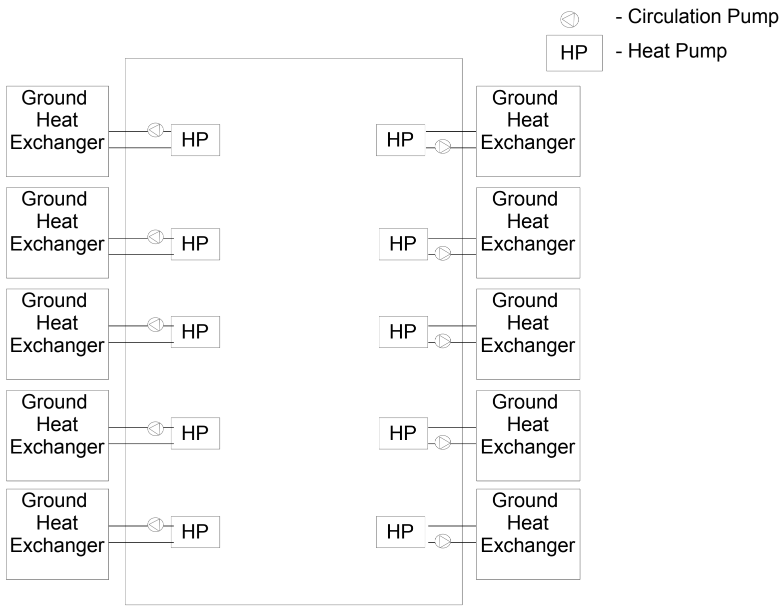

Figure 1.3.2.1 Unitary Distribution Loop

In addition to being a simple design, advantages of this system is that each pump is sized for each individual system. This eliminates the need for control valves and complex headers. Also this reduces the amount of piping needed inside the facility and hence reduces costs. The total size of the borefield could be larger with a unitary design over a shared system, because it can not take advantage of diversity in loads. However, higher drilling costs may be partially or fully offset by reduced costs for valves and controls.

The sub-central loop configuration divides a building into smaller loops, shown in Figure 1.3.1.2. This takes multiple heat pumps from a common area and ties them together hydraulically. It then uses a common borefield to perform the heat transfer.

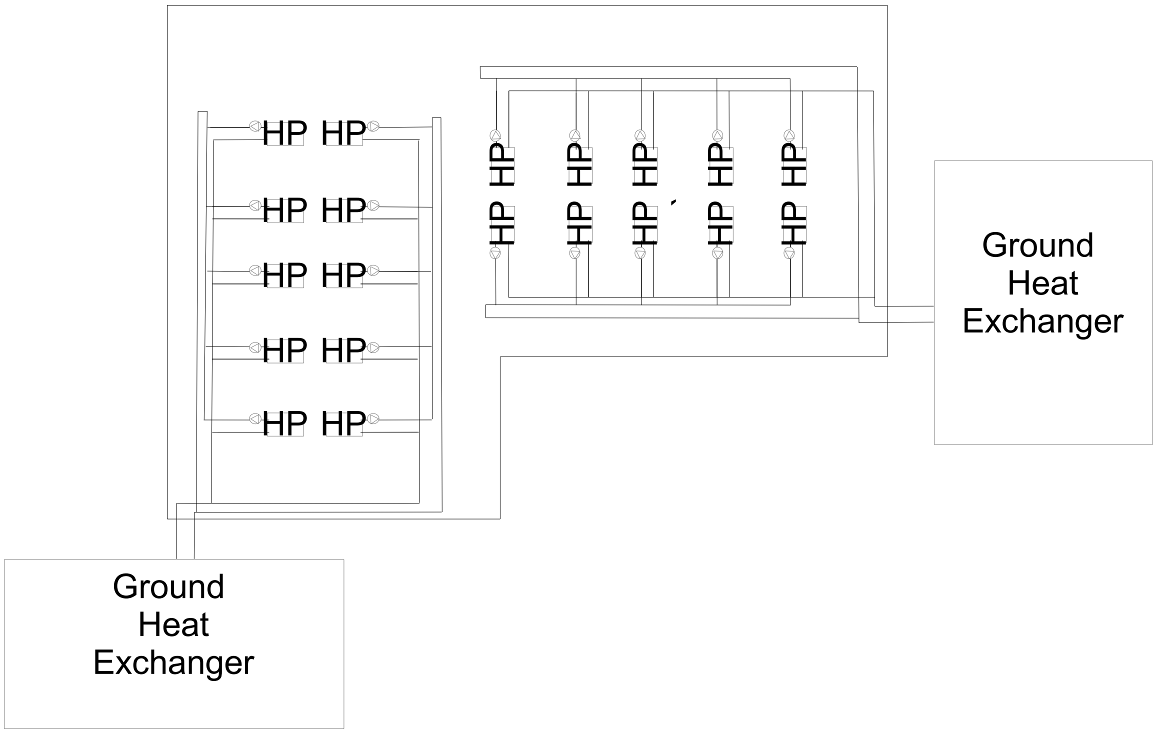

Figure 1.3.2.2. Sub-Central Loop

With large buildings that have significant load diversity it is recommended that a central loop to be installed. Figure 1.3.1.3 shows a central loop configuration. With this style, the borefield is located in a central location. The primary header can be in the mechanical room or in a vault installed near the borefield.

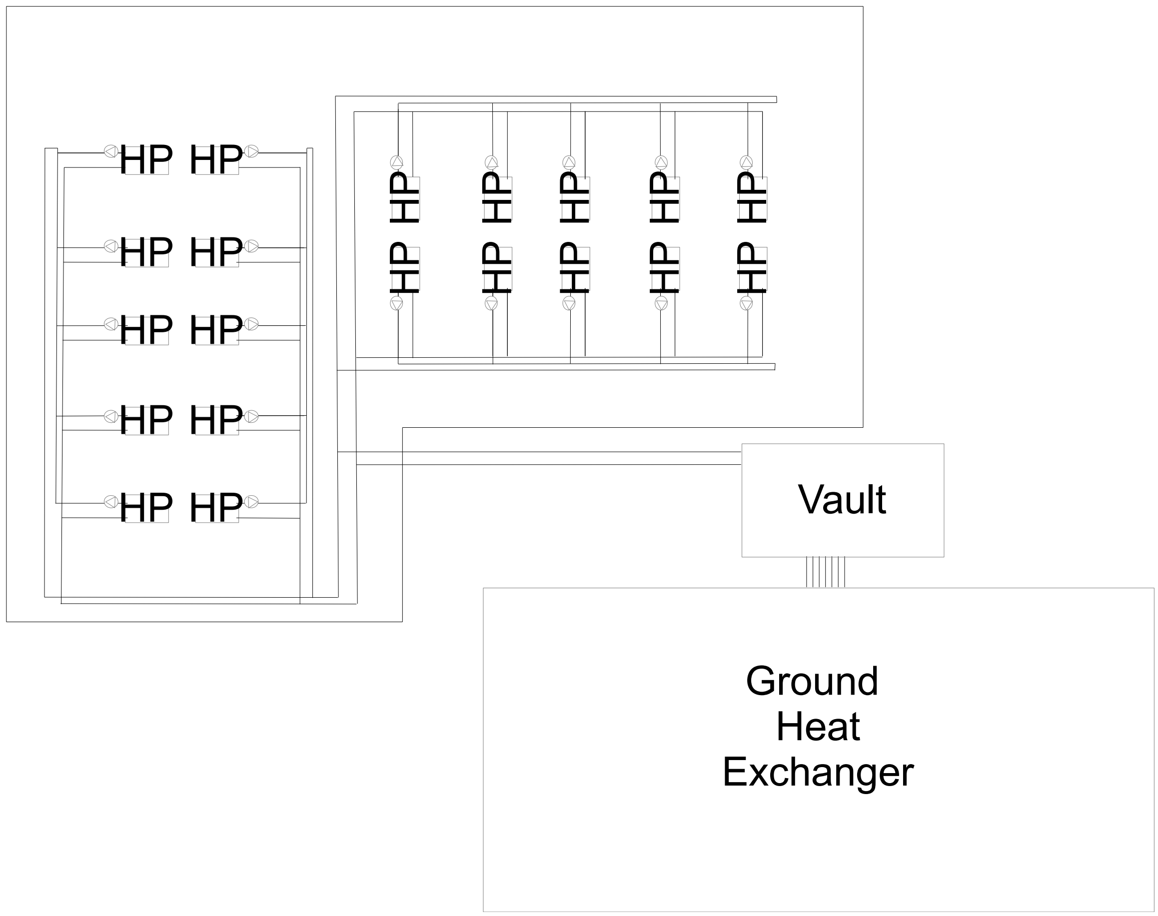

Figure 1.3.2.3 Central Loop

1.3.3 DISTRIBUTION PIPE DESIGN AND PUMP STRATEGY

If using the sub-central loop or a central loop there are many ways to distribute the fluid to each heat pump. Some of the more common distribution methods will be discussed below. Distribution methods can be divided into two types, the two-pipe system and the one-pipe system. The two-pipe system, shown in Figure 1.3.2.1, is the standard approach to designing a shared distribution system (Mescher 2009). In this configuration, fluid is piped through a common supply pipe that feeds multiple heat pumps in either direct or reverse return.

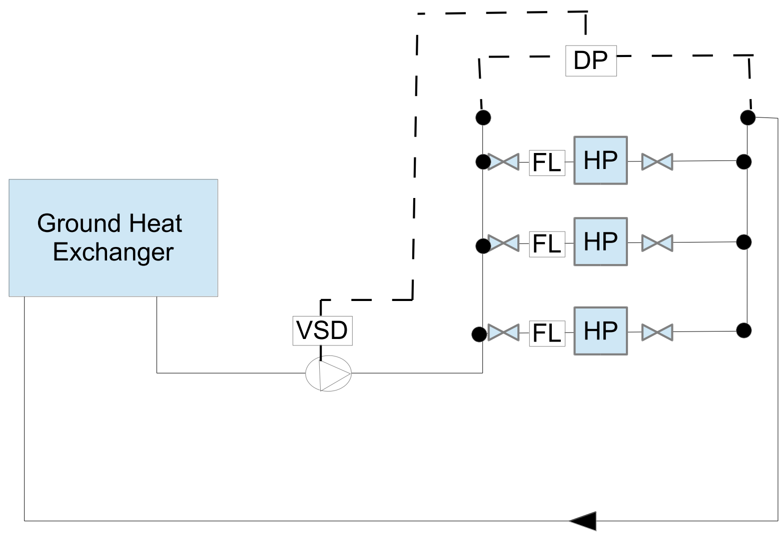

Figure 1.3.3.1 Two-Pipe Reverse Return System Central Pumping

Once through the heat pump, the fluid flows through a common return pipe and through the ground heat exchanger. In Figure 1.3.3.1 the two-pipe system is shown with a central pump controlled with a variable speed drive (VSD). To regulate flow, each heat pump is controlled by a motorized valve that opens when the heat pump turns on. Since all of the heat pumps run in parallel, flow control valves may be needed to assure proper flow rates through each heat pump and that each leg has an equal pressure drop. As heat pumps turn off, the pressure will rise and the pump will slow down until the pressure is back to the set point. The advantage of this system is sharing the capacity of other zones, therefore being able to decrease the total size of the borefield. In exchange, the install cost inside the building increases to connect all of the heat pumps together and to add valves. Another side effect of the installed costs of installing valves is the wasted energy associated with their added pressure drop to the system. If used in conjunction with a variable frequency drive (VFD), performance of this system can be very efficient compared to others when the facility is used for more than 60 hours per week (Kavanaugh et al. 2003; Lambert and Kavanaugh 2004). Kavanaugh (Kavanaugh 2009; Kavanaugh and Kavanaugh 2012a) more recently found that the actual performance of installed systems is surprisingly poor. In multiple surveys, on average, they have been shown to not operate as efficiently as expected. One cause of this was found that many of the installed systems did not have a means to turn the pump off when no heat pumps were running.

Using the same pipe distribution layout, another option with the two-pipe distribution system is to have distributed pumping shown in Figure 1.3.3.2. With this pumping configuration, each heat pump has a constant speed circulator that is turns on and off with the heat pump. All heat pumps will have to be equipped with a one way valve to prevent reverse circulation through a heat pump.

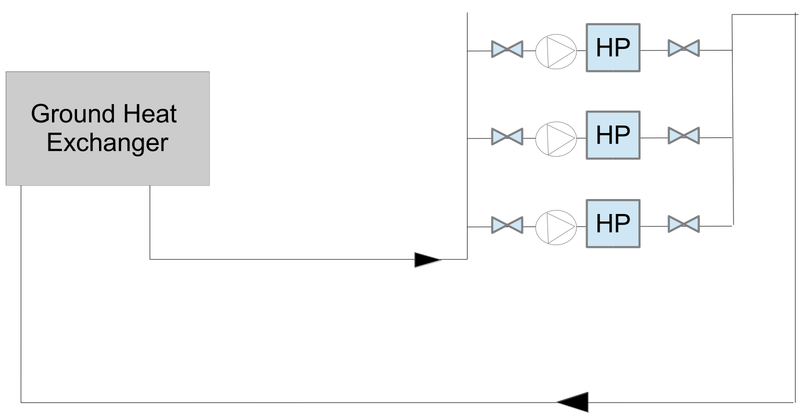

Figure 1.3.3.2 Two-pipe Distribution Distributed Pumping

Lambert and Kavanaugh (2004) showed in laboratory tests that this is a feasible strategy if the number of heat pumps is small such as a sub-central loop. Each pump has to be sized to provide the flow rate required for its heat pump and the pressure drop across that plus the pressure drop through the rest of the system at a flow rate where all heat pumps are running. This can cause the installed horsepower/100 ton to be rather high. This doesn't necessarily mean this system will be inefficient, since it is still demand oriented. As Lamber and Kavanaugh showed in the tests, with ten heat pumps, as heat pumps turned off the flow rate through each heat pump increased from 50% to 100%. This variety of pressure and flow rates will force the pump to operate well away from its best efficiency point causing extra wear on the pump. If equipped with a variable speed ECM pump, this could be a very efficient system that can adjust the flow rate according to the pressure.

Another option with the two-pipe system is to combine the central pump and distributed pumping in a hydraulically separated pipe system shown in Figure 1.3.3.3.

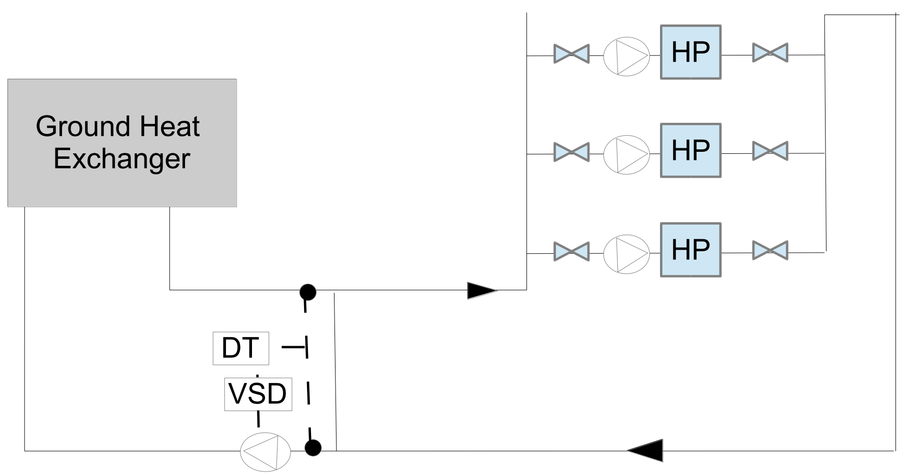

Figure 1.3.3.3 Two-pipe Central and Distributed Pumping

In this system, the individual heat pumps have a circulator that operates on/off with the heat pump. In low load situations where the temperature increase of the circulating fluid may be minimal, the main pump will not operate. The circulators would then only pump the fluid through the building. A bypass line with a flow check valve would allow the fluid to skip the borefield until the fluid reaches a designated temperature. At that point, the primary pump would kick on and run until the load decreases. This would decrease the size of the circulators compared to a distributed pumping scheme as they would not have to pump through the borefield and allow them to operate more consistently. The disadvantage is that it adds another pump to make up for that difference and is a more complex control strategy. Hart and Price (2000) compared four different pumping schemes including central pumping, central pumping with a decoupled ground loop pump, distributed pumping, and distributed pumping with a decoupled ground loop pump, each of these were given multiple control strategies. At the end of the comparison, the combination of distributed pumping with a decoupled ground loop pump setup was considered to have the best overall value when controlled properly.

The one-pipe system, shown in Figure 1.3.3.4, has shown to be a well performing system that has so far had limited traction (Mescher 2009; Kavanaugh and Kavanaugh 2012a). Like the two-pipe system, this is a central system that supplies the fluid to each heat pump through a common supply pipe. Instead of having a second return pipe, the fluid is pumped back into the supply pipe. With this system, each heat pump has a small circulator pump that operates only when the heat pump is operating. This reduces the need for a motorized valve but will still require isolation valves. This type of system also takes advantage of sharing the capacity and reducing the bore field size like the two-pipe system. In addition to that, there is less distribution piping than the two-pipe system which allows for a lower installed cost. If there is diversity in the load of the facility, the one-pipe system can take advantage of that since each zone sees the heat extraction and rejection from the previous zone. This is also a disadvantage of the one-pipe system. If all zones are heating or cooling at the same time, the efficiency of the last heat pump could be significantly lower than the first zone.

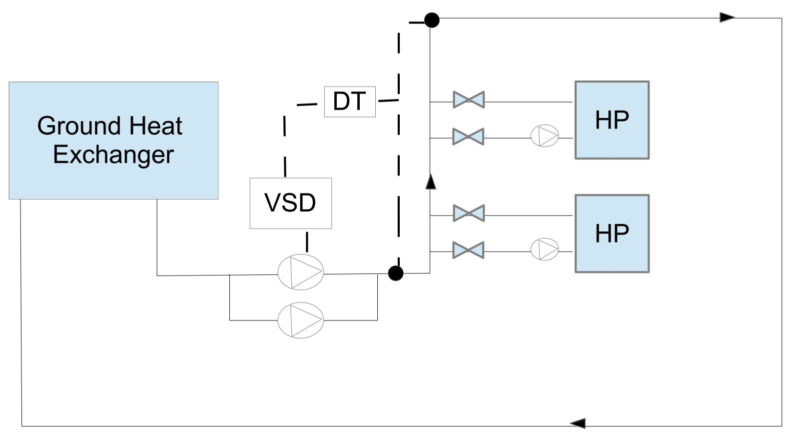

Figure 1.3.3.4 One-pipe System Variable Speed Pump

In Figure 1.3.3.4 the main pump is controlled by VFD that controls the speed of the pump according to the change in temperature across the building loop. This can be done with a single variable speed pump or two pumps in parallel. Another option is shown in Figure 1.3.3.5 where two parallel constant speed pump are used.

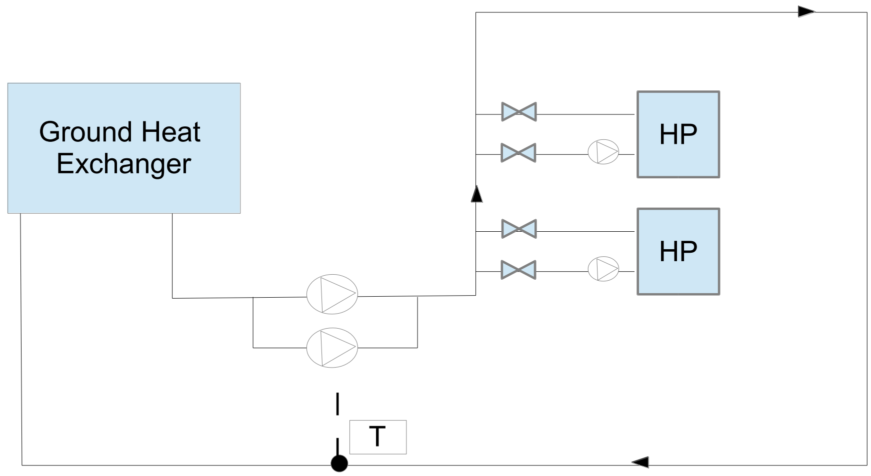

Figure 1.3.3.5 One-pipe System Parallel Constant Speed Pumps

Instead of using the change in temperature across the building loop, this strategy uses the actual temperature of the fluid. A single pump will provide a constant flow of fluid in partial loads. When the temperature rises above or below a set point, the second pump would turn on. This helps provide redundancy in pumps.

1.3.4 HEADER CONFIGURATIONS

A manifold is defined as a pipe or chamber branching into several openings. For this thesis, the term "manifold" will be used to describe a branching of pipes within a few pipe diameters of each other or a central location designated to distribute flow. A "header" will relate to distribution pipes that cover a larger length such as the width of a borefield or the length of a building and distributes flow to multiple pipes or boreholes. Headers that distribute flow to the borefield can have four basic configurations. Most of the difference in them is the location of the first or primary header. The first two covered here are better suited for smaller borefields. The first is when the primary manifold is in the equipment room as shown in Figure 1.3.4.1. This type of configuration should only be used if the borefield is close to the building otherwise there would be an unnecessarily large amount of pipe required in the ground and a high pressure drop associated with the borefield.

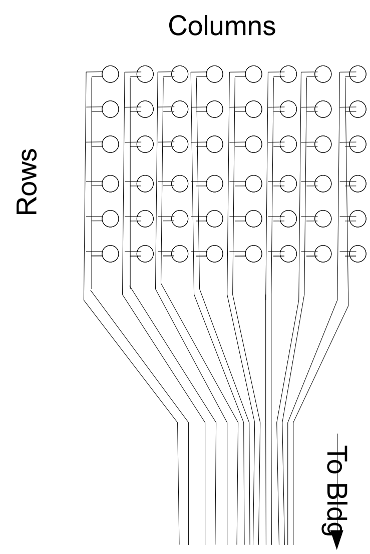

Figure 1.3.4.1 Manifold in Equipment Room

Another option for smaller borefields would be to run the supply and return lines, referred to here as runouts, to the borefield with the primary header feeding each column of bore holes as shown in Figure 1.3.4.2. This type of header is not suitable for large borefields as it would be difficult to purge the system of air during installation and if one bore hole has a leak, the entire borefield would have to be shut down to repair it or isolate it.

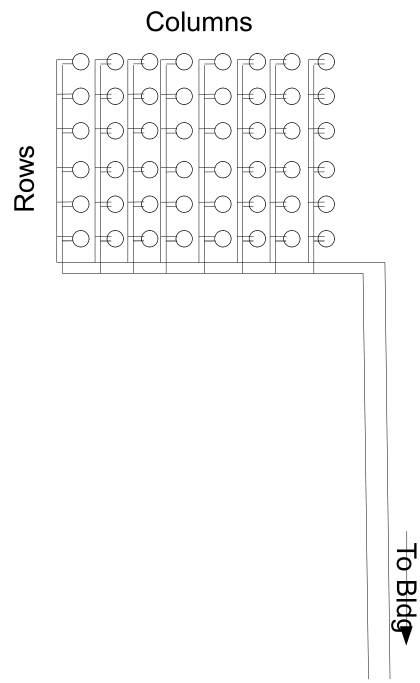

Figure 1.3.4.2 Primary Header Feeding Sub Headers

For larger commercial systems or in cases where the borefield is further away from the building it may be more cost effective to install a vault with a header that is close to the borefield and is fed by one supply and one return runout. Vaults with manifolds are commercially available or they can be designed for each case. Figure 1.3.4.3 shows this configuration with the manifold distributing flow to headers for each column of boreholes.

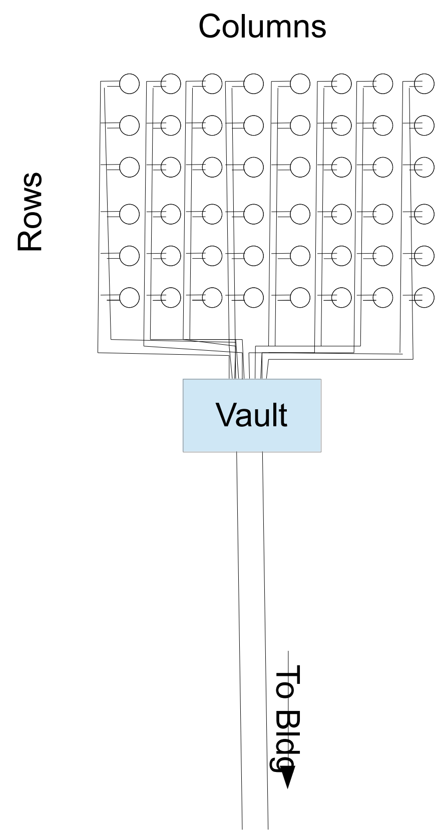

Figure 1.3.4.3 Vault Manifold Feeding Columns

The fourth and final configuration covered by this thesis is a combination of the second and third configurations. This also has a manifold in a vault installed near the borefield, but instead of each branch of the manifold feeding a single column of boreholes, each branch of the manifold would supply a section of the borefield with a primary header that distributes flow to each column and a secondary header that distributes flow to each borehole, shown in Figure 1.3.4.4.

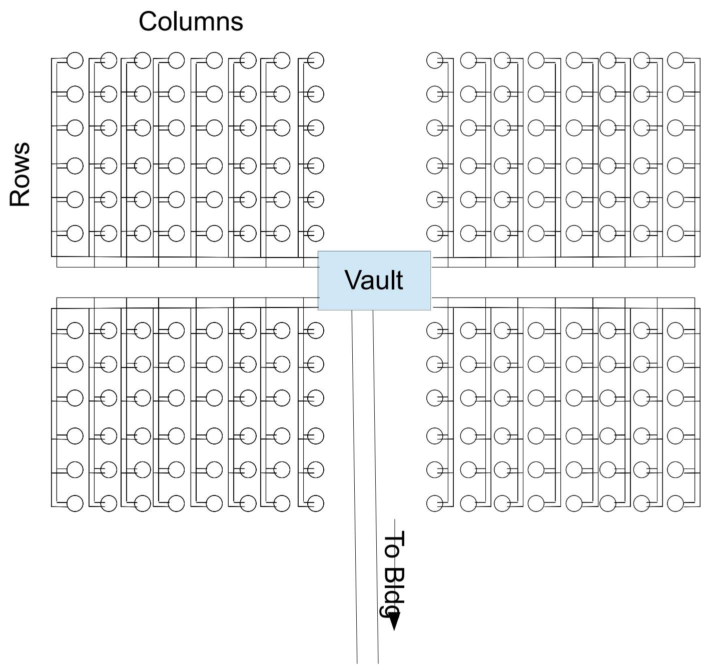

Figure 1.3.4.4 Vault with Primary and Secondary Headers

While this list of configuration is not an exhaustive list, it is intended to cover what the author believes to be the most logical ways to configure the headers.

1.3.5 CONTROL STRATEGIES

Current guidance in minimizing pumping energy gives an A through F grading scale based on pumping horsepower per 100 tons of cooling shown in Table 1.3.5.1(Kavanaugh and Rafferty 1997).

Table 1.3.5.1. Pumping Power Grading Scale (Kavanaugh and Rafferty 1997)

Pumping Power/Cooling Capacity (W/Ton)	Pumping Power/Cooling Capacity (HP /100 Ton)	Grade

50 or less	5 or less	A - Excellent
50 - 75	5 - 7.5	B - Good
75 - 100	7.5 - 10	C - Mediocre
100 - 150	10 - 15	D - Poor
More than 150	More than 15	F - Bad

Hamstra (2012) mentions this as a good place to start your design efforts, but it may not be a good grading scheme for pumping energy. Multiple authors have provided studies that show that while designing the system to require a small pump is important, careful control of the pump has been crucial to lowering operating costs and a straight horse power per 100 ton grade can misrepresent system performance (Hart and Price 2000; Henderson et al. 2000; Kavanaugh 2011; Hamstra 2012). In ASHRAE research project 1217 it was recommended that for buildings with a high occupancy rating (>60 hours per week) to design a central building distribution piping system with a variable speed pump (Kavanaugh et al. 2003) and to use on/off circulator pumps for buildings with unitary or small (<6 heat pumps) sub central loops. A central loop has advantages such as less pumps to maintain and a central location. This usually requires larger pumps to be installed.

In a traditional two-pipe system the most common control strategy for a VFD is a differential pressure (DP) sensor (ASHRAE 1995; McQuiston et al. 2010). With this strategy, the speed of the pump, linearly related to the flow rate of the pump, is increased or decreased as the pressure changes as a result of heat pumps turning on and off. Recommended location for the DP sensor is at the most remote heat pump on the system or the heat exchanger requiring the greatest pressure drop (Tillack and Rishel 1998; Rishel et al. 2006; ASHRAE 2010) . The pressure set point for this control strategy can also have significant effects on the performance of the pumping system. Henderson et al. (2000) showed case studies from multiple systems installed in buildings that showed significant savings with proper pump operation. In a simulation to show savings potential, they took the observed energy consumption at a DP sensor set point of 124 kPa (18 psi). Compared to a constant speed pump, the energy savings was about 54%. If the set point was reduced to a more reasonable value, of which the author did not specify, they estimated the savings could have been up to 79%. This is only one example of how big of an effect that this set point could have.

As previously discussed, the one-pipe system can operate with multiple control strategies; temperature sensor, differential temperature sensor, or some combination of all of the above. No matter which method is used, some means of turning the pump off when no requirement exists should be incorporated into the design.

Circulator pumps on unitary distribution systems typically are driven by demand. A relay on the heat pump itself sends an on/off signal to the pump or in some cases to open/close a motorized valve. These types of systems have been shown to be some of the best performing systems in practice (Kavanaugh and Kavanaugh 2012b). Since most heat pumps run at partial load a majority of the time, less flow is required to transfer the required amount of heat. A way to accommodate this is to use a two stage heat pump. Two stage heat pumps are favored in the higher end models of water source heat pumps in the US (Staffell et al. 2012) Figure 1.3.5.1 shows a connection diagram for a two stage heat pump. At peak conditions, the stage two motorized valve would open only when stage two is operating. The stage one motorized valve would be open whether the unit is operating in stage one or stage two.

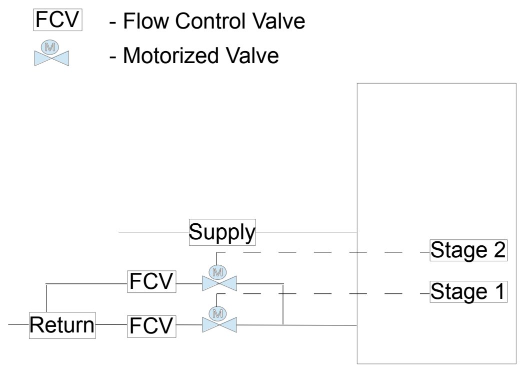

Figure 1.3.5.1 Two Stage Heat Pump Piping Diagram

1.4 OBJECTIVES

The objective of this research was to investigate piping and pumping design for commercial GSHP systems. This was done by investigating pipe sizing criteria; various pump control strategies, and piping configurations. As a part of this, a pumping analysis program was developed in Excel and was used to calculate total annual energy consumption for multiple pumping configurations and strategies for four locations. Chapter two of this thesis describes the concepts and equations that were used to create the program for this research. Chapter three summarizes some design assumptions and simplifications that were used to simplify the calculations and to make the program more efficient. Chapter four describes the program that was developed to use the models and how to use the program. Chapter five discusses the results that were found. Finally chapter six provides a conclusion to the research and results and recommends areas of further research.

CHAPTER II

MODELING METHODOLOGY

To compare pumping and piping design, first a building must be selected. Designing a building and creating realistic load profiles can be a difficult task. For that reason a sample building available for Energy Plus – a strip shopping mall – was selected. In this program, an hourly annual simulation was run to get zone hourly loads that could be used to size equipment and calculate annual system energy consumption. Many researchers have come up with ways to approximate pump energy consumption based off compiled data sets and equation fits to that data, from which a set of equations was found and implemented in this research. Actual manufacturer's data was used to estimate likely performance of a heat pump with calculated conditions. This data also allowed for calculating the change in temperature of the fluid across the building. In order to calculate the heat transferred between the fluid and the ground, a borehole model was created in HVACSIM+ and linked with Excel to calculate fluid temperatures returning from the borefield.

2.1 BUILDING LOADS

EnergyPlus (DOE 2012) is an open source whole building simulation program developed by the department of energy that can be used to simulate the thermal loads of a building and calculate the overall energy consumption including lighting and water usage. This program can use pre-loaded building designs or a building created by the user. The facility chosen for this thesis is a reference strip mall building created by DOE/NREL/PNNL/LBNL for use in

EnergyPlus. The building is a ten zone strip mall with independent loads. An individual heat pump was required to be installed in each zone. A sample of how this could be configured is shown using Google SketchUp 8 in Figure 2.1.1.

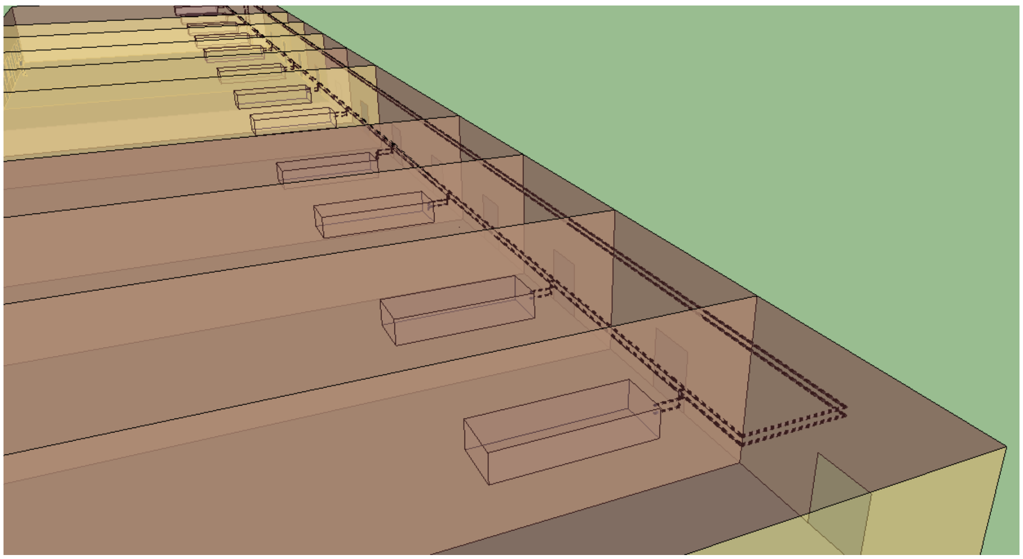

Figure 2.1.1 Building Design

EnergyPlus has weather data for a typical meteorological year for many areas in the country. To get a variety of weather profiles and runtime conditions, this building was analyzed in 4 locations; Chicago, Illinois; Tampa, Florida; Tulsa, Oklahoma; and Sterling, Virginia. In the EnergyPlus program a building simulation was run for one year. From this simulation the hourly HVAC loads and equipment sizing summary for each zone were exported to a Microsoft Excel spreadsheet. The sizing criteria as laid out in Unified Facilities Criteria (UFC) 3-410-01FA, which is the HVAC design guideline document for the Department of Defense, is for comfort cooling, to design to the 1% outdoor dry bulb temperatures . Building interior temperature set points were set to 21°C (70°F) for heating and 24°C (75°F) for cooling. The difference in sizing for the four weather locations ranged in cooling from 13.1 to 37.2 kW (3.7 to 10.6 Ton) in cooling and 10.6 to 42.40 kW (36,000 to 144,000 Btu/hr) in heating. These calculated loads are shown for each location in Tables 2.1.1 through 2.1.4. The proper heat pump was selected to ensure that the heat pump can satisfy both the calculated heating and cooling loads. It was assumed for all locations that the cooling coils were sufficiently capable of handling the latent loads and equipment selection was only dependent on total cooling and total heating loads.

For Chicago, being a heating dominated location, the heating load was the driving factor for equipment selection.

Table 2.1.1 Chicago Equipment Sizing Details

Chicago, Illinois Sizing Criteria	Calculated Design Cooling Load [kW]	Calculated Design Heating Load [kW]	Calculated Design Cooling Load [Ton]	Calculated Design Heating Load [Btu/hr]
Zone 1	31.9	42.4	9.1	144,736
Zone 2	14.5	19.4	4.1	66,217
Zone 3	14.6	19.3	4.2	65,998
Zone 4	14.4	19.3	4.1	65,988
Zone 5	14.4	19.4	4.1	66,030
Zone 6	25.2	38.4	7.2	131,180
Zone 7	13.1	19.4	3.7	66,031
Zone 8	13.1	19.3	3.7	66,007
Zone 9	13.1	19.4	3.7	66,309
Zone 10	15.0	23.3	4.3	79,402

Each unit has performance capacities based on the fluid temperatures entering the heat pump and the fluid flow rates through the heat pump. Since the temperature of the fluid returning from the borefield varies throughout the year and day, it would make sense to ensure that the heat pump performance meets the required capacities when the fluid temperatures are at the extreme operating ranges, 0°C (32°F) for heating and 37°C (100°F). Using these as guidelines for the system selection, the ClimateMaster Tranquility High Efficiency Series model 060 was selected for all of the zones in Chicago with the exception of zones 1 and 6. For zones 1 and 6, one 060 model would not be sufficient to meet the load. At the extreme operating ranges, this heat pump had a capacity of 14.56 kW (49,700 Btu/hr) in heating and 15.89 kW (54,200 Btu/hr) in cooling. It was assumed that for heating, normal operating temperatures would be 10°C (50°F) where the capacity of the heat pump is 19.31 kW (65,900 Btu/hr). For ease of simulation, rather than having 2 heat pumps that had different performance characteristics, on both efficiency for energy consumption and flow rate differences, two of the 060 models were selected to satisfy the requirement. This created a building with 12 total identical heat pumps and allowed a simplified simulation. These loads were then used as the loads to compare different pumping strategies and piping configurations to calculate the total pumping and heat pump energy consumptions of each.

As expected when analyzing the sizing details in Tulsa, the cooling load was the driving factor. The details are shown in Table 2.1.2, the cooling load ranged from 14.1 – 37.2 kW (4.0 – 10.6 Ton). Same as Chicago, this drove the selection of the ClimateMaster Tranquility High Efficiency Series model 060 for all but the two larger zones which again required approximately twice the capacity and two 060 models were selected.

Table 2.1.2 Tulsa Equipment Sizing Details

Tulsa, Oklahoma Sizing Criteria	Calculated Design Cooling Load [kW]	Calculated Design Heating Load [kW]	Calculated Design Cooling Load [Ton]	Calculated Design Heating Load [Btu/hr]
Zone 1	37.2	32.3	10.6	110,049
Zone 2	18.0	15.1	5.1	51,681

Zone 3	16.6	15.1	4.7	51,681
Zone 4	16.2	15.1	4.6	51,681
Zone 5	16.0	15.1	4.5	51,681
Zone 6	28.1	30.3	8.0	103,362
Zone 7	14.1	15.1	4.0	51,681
Zone 8	14.1	15.1	4.0	51,681
Zone 9	14.1	15.1	4.0	51,681
Zone 10	17.2	17.7	4.9	60,452

The next location that was analyzed was Tampa, Florida. Similar to Tulsa, this was a cooling dominated region where the cooling load drove the selection of the heat pump. A summary of the details is in Table 2.1.3. The cooling design load varied from 13.3 – 35.7 kW (3.8 – 10.2 Ton) which again drove the selection of the ClimateMaster Tranquility High Efficiency Series model 060 for the smaller zones and two 060 models for the larger zones.

Table 2.1.3 Tampa Equipment Sizing Details

Tampa, Florida Sizing Criteria	**Calculated Design Cooling Load [kW]**	**Calculated Design Heating Load [kW]**	**Calculated Design Cooling Load [Ton]**	**Calculated Design Heating Load [Btu/hr]**
Zone 1	35.7	21.3	10.2	72,537
Zone 2	17.9	10.6	5.1	36,269
Zone 3	15.8	10.6	4.5	36,269
Zone 4	15.4	10.6	4.4	36,269
Zone 5	15.2	10.6	4.3	36,269

Zone 6	26.9	21.3	7.7	72,537
Zone 7	13.4	10.6	3.8	36,269
Zone 8	13.3	10.6	3.8	36,269
Zone 9	14.1	10.6	4.0	36,269
Zone 10	16.3	10.6	4.6	36,269

Sterling, Virginia was also dependent on the cooling load for equipment selection. The cooling design load varied from 13.5 – 33.4 kW (3.8 – 9.5 Ton) which again drove the selection of the ClimateMaster Tranquility High Efficiency Series model 060 for the smaller zones and two 060 models for the larger zones.

Table 2.1.4 Sterling Equipment Sizing Details

Sterling, Virginia Sizing Criteria	Calculated Design Cooling Load [kW]	Calculated Design Heating Load [kW]	Calculated Design Cooling Load [Ton]	Calculated Design Heating Load [Btu/hr]
Zone 1	33.4	33.5	9.5	114,227
Zone 2	16.5	15.5	4.7	53,048
Zone 3	15.0	15.5	4.3	53,048
Zone 4	14.8	15.5	4.2	53,048
Zone 5	14.7	15.5	4.2	53,048
Zone 6	26.0	31.1	7.4	106,097
Zone 7	13.5	15.5	3.8	53,048
Zone 8	13.5	15.5	3.8	53,048
Zone 9	13.5	15.5	3.9	53,048
Zone 10	15.6	18.4	4.4	62,701

Various design strategies were implemented for sizing of the pipes and pumps for this system. Limits on the amount of allowable head loss per 100 foot of pipe were varied as shown in the results section to evaluate energy savings from different approaches. Reverse return piping was used for all scenarios. For this study, a vault with a manifold was used to distribute flow to column headers for each building configuration. The manifold was treated as a fitting with a pressure loss coefficient that is equal across each column resulting in equal flow to each column. The borefield was sized using GLHEPro V.4 where the soil was assumed to be average rock with a conductivity of 1.4 Btu/hr-ft-°F, density of 175lb/ft^3, specific heat of 0.2 Btu/lb-°F, and Volumetric heat of 34.94 Btu/ft^3-°F. Also Table 2.1.5 shows the undisturbed ground temperatures that were used for each location.

Table 2.1.5 Undisturbed Ground Temperatures

	Undisturbed Ground Temperature °C (°F)
Chicago, Illinois	10 (50)
Tulsa, Oklahoma	16.7 (62)
Tampa, Florida	23.9 (75)
Sterling, Virginia	13.3 (56)

2.2 GSHP SYSTEM - HEAT TRANSFER

As discussed above, the temperature of the working fluid has a large impact on the performance of a GSHP system. The heat pump performance is heavily dependent on the entering fluid temperature (EFT). This also affects the pressure loss of the fluid as it circulates through the piping system which affects the pumping energy consumption. There are four sources of heat for the fluid to change the fluid temperature; the pump(s), the heat pump(s), the distribution piping, and the borehole. Each of these will be discussed in depth.

2.2.1 PUMPS

As a pump operates there are some internal energy losses where the energy input is turned to heat. Some of that energy is directly transferred to the fluid inside of the pump and some of the energy is dissipated to the surrounding environment. For this investigation, it was assumed that all of these energy losses were dissipated from the pump, motor, and VFD, where applicable, to the surrounding environment and those losses are already included in the EnergyPlus building loads.

2.2.2 HEAT PUMPS

The building loads are dissipated to and absorbed from the working fluid, in this case assumed to be water at standard conditions. The actual amount of heat transfer is not one for one with the building loads. This is instead a function of the coefficient of performance (COP) of the heat pump. Manufacturer's data normally gives the COP of a heat pump in tabulated form with multiple EFTs and flow rates. The COP can also be broken down into two categories, the COP for heating ($COP_{heating}$) and the COP for cooling ($COP_{cooling}$). This value is a dimensionless value obtained as the ratio of Watts of heating or cooling realized from the heat pump to the Watts of power input to operate the heat pump. The actual amount of heat transferred to or from the water is shown in Equations 2.2.2.1 and 2.2.2.2.

$$\dot{q}_{extracted} = 1 - \frac{1}{COP_{heating}} \tag{2.2.2.1}$$

$$\dot{q}_{rejected} = 1 + \frac{1}{COP_{cooling}} \tag{2.2.2.2}$$

Where $\dot{q}_{extracted}$= Heat Extraction (HE) Rate; W (Btu/hr)

$\dot{q}_{rejected}$= Heat Rejection (HR) Rate; W (Btu/hr)

Note that these equations differ for heating and cooling. The energy input to operate the heat pump's compressor is added to the cooling load for extra heat needing to be rejected from the water loop and the energy is subtracted from the heating load.

The change in temperature of the working fluid is a function of the amount of HR or HE, the flow rate of the fluid, and the EFT to the heat pump. For this, Equation 2.2.2.3 can be used to calculate the change in temperature.

$$\dot{q} = \dot{m}C_p\Delta T \qquad (2.2.2.3)$$

Where $\dot{q}$ = Heat transfer rate, W (Btu/hr)
$\dot{m}$ = Mass flow rate of fluid, kg/s (lbm/s)
Cp = Specific Heat of fluid, kJ/kg-°C (Btu/(lb- °F))
ΔT = Change in Temperature, °C (°F)

2.2.3 DISTRIBUTION PIPING

It is assumed for this thesis that if the temperature inside the building is maintained at room temperature, the heat loss or gain from the pipe to the surrounding area is negligible or the pipe would be insulated.

From the equipment room to the borefield the heat transfer from the pipe to the ground can be calculated by Equation 2.2.3.1.

$$\dot{q} = \frac{(T_g - T_s)}{R'_o} \qquad (2.2.3.1)$$

Where T_g = Temperature of the ground, °C (°F)
T_s = Temperature of the pipe, °C (°F)
R'_o= thermal resistance of pipe, fluid, and ground, C/W (hr-F/Btu)

The resistance of the ground can be defined by Equation 2.2.3.2.

$$R'_g = \frac{(ln\left(\frac{2L}{D}\right)[1 - \frac{ln\left(\frac{L}{2z}\right)}{ln\left(\frac{2L}{D}\right)}]}{2\pi kL} \qquad (2.2.3.2)$$

Where L = Length, m (ft)
D = Pipe outside diameter, m (ft)
z = depth of pipe, m (ft)

k = soil conductivity, W/m-°C (Btu/hr- °F-ft)

Thermal conductivity of the fluid is a function of the fluid temperature. Values for this are tabulated for specific fluids. Thermal conductivities of the pipe are also specific to the type of pipe and pipe thickness, which is assumed to be the accepted standard for ground loops, HDPE pipe. The thermal resistance of HDPE pipe can be calculated by Equation 2.2.3.3.

$$\boldsymbol{R'} = \frac{\boldsymbol{ln}\left(\frac{r_o}{r_i}\right)}{\boldsymbol{2\pi kL}} \tag{2.2.3.3}$$

Where $r_{o,i}$ = radius of pipe, outside and inside, m (ft)
k = thermal conductivity of pipe, W/m-°C(Btu/ft-F)
L = length of pipe, m (ft)

Though some of the heat transfer does happen for this pipe length, it is relatively insignificant compared to the heat transfer in a borehole. For a 200 foot pipe with two inch diameter in ground 67°F carrying 90°F water at 30 gallons per minute would lose approximately 3,000 Btu/hr. At 30 gallons per minute assuming a 10°F temperature change between supply and return would be roughly 150,000 Btu/hr. This would make the heat loss in this section of pipe about 3% of the total heat transfer in a worst case scenario. Though no always the case, for the purposes of this thesis, this heat loss was assumed to be negligible as this is only a three percent loss for a peak condition the building would see approximately one percent of the time.

2.2.4 BOREHOLE

There are many models that can be used to calculate the heat transfer in a borehole. The model used for this simulation is an adaptation of the original model developed by Eskilson (1987). This model was implemented in a Type 620 model in HVACSIM+ which was developed by Xu (Xu and Spitler 2006). A program created by Lu Xing, a Ph. D. candidate at Oklahoma State University incorporated Xu's model to take input from a spreadsheet given the buildings heat pump performance characteristics, the buildings loads, and the mass flow rate of the circulating fluid. The building loads were given as an hourly value. The mass flow rate that was used for these simulations was fixed. This program called HVACSIM+ to run the borefield model and calculate the entering water temperatures, the exiting water temperatures and the total heat transfer rate of the borefield.

2.3 GSHP SYSTEM - PRESSURE LOSS

The frictional pressure losses in a closed system are dependent on both the flow rate and the temperature of the circulating fluid. To determine the pressure loss of a system requires careful analysis of where the fluid is going. GSHP systems have multiple sections that are connected in parallel and depending on what information is known about the system, the flow rate can be solved for using different approaches. For this study, the effects of temperature were ignored.

2.3.1 PIPE FRICTIONAL LOSS

The friction head loss from the fluid to the pipe can be calculated using the friction factor calculated using Churchill's expression (Churchill 1977) shown in Equation 2.3.1.1, 2.3.1.2, and 2.3.1.3. Use of the Churchill expression covers all regimes of fluid flow so only one expression is needed for turbulent flow, laminar flow, and the transition region.

$$\boldsymbol{f} = \mathbf{8}\left[\left(\frac{\mathbf{8}}{\boldsymbol{Re_D}}\right)^{\mathbf{12}} + \frac{\mathbf{1}}{(\boldsymbol{A}+\boldsymbol{B})^{\mathbf{1.5}}}\right]^{\mathbf{1}/\mathbf{12}} \qquad (2.3.1.1)$$

$$\boldsymbol{A} = \left\{ \boldsymbol{2.457ln} \left[\frac{\boldsymbol{1}}{\left({}^{7}/_{\boldsymbol{Re_D}} \right)^{\mathbf{0.9}} + (\mathbf{0.27}\boldsymbol{\varepsilon}/\boldsymbol{D})} \right] \right\}^{\mathbf{16}} \tag{2.3.1.2}$$

$$\boldsymbol{B} = \left(\frac{\mathbf{37{,}530}}{\boldsymbol{Re_D}} \right)^{\mathbf{16}} \tag{2.3.1.3}$$

Where *Re* = Reynolds Number (-)
ϵ = roughness (-)
D = Diameter of pipe, m (ft)

Equation 2.3.1.4 uses the friction factor calculated above to determine the pressure drop in a straight pipe.

$$\boldsymbol{h} = \boldsymbol{f}\ \frac{\boldsymbol{L}}{\boldsymbol{D}} \frac{\boldsymbol{\rho}\overline{\boldsymbol{V}^{\mathbf{2}}}}{\mathbf{2}\boldsymbol{g}} \tag{2.3.1.4}$$

Where *h* = Head loss, m (ft)
L = Length of pipe, m (ft)
D = Diameter of pipe, m (ft)
$\boldsymbol{\rho}$ = fluid density, kg/m3 (lb/ft3)
V = Velocity of fluid, m/s (ft/s)
g = acceleration due to gravity, m/s^2 (ft/s^2)

2.3.2 FITTING AND VALVE PRESSURE LOSS

Each fitting and valve that is installed in a hydronic network has a pressure loss associated with it. Tabulated values can be found for approximate pressure drops in relation to a pressure loss coefficient, K. When calculating the system curve, for each pipe diameter, the K-value for each fitting and valve can be summed together for an equivalent K-value, K_{eq}. Equation 2.3.2.1 can be used to calculate the pressure drop added due to fittings and valves.

$$h = \frac{\boldsymbol{K_{eq}}\boldsymbol{V}^{\mathbf{2}}}{\mathbf{2}\boldsymbol{g}} \tag{2.3.2.1}$$

Where h = head loss, m (ft)
K_{eq} = loss coefficient (-)
V = Fluid velocity, m/s (ft/s)
g = acceleration due to gravity, m/s^2 (ft/s^2)

2.3.3 HEAT PUMP PRESSURE LOSS

Much like fittings and valves, the pressure loss of a heat pump can be approximated with a pressure loss coefficient. Many heat pump manufacturers' gives data that provides the head loss across a heat pump for given flow rates. With the manufacturer's data, the loss coefficient can be calculated using Equation 2.3.2.1. A model can be approximated for Equation 2.3.2.1 using Equation 2.3.3.1.

$$h_{\text{heat pump}} = C*Q^2 \qquad (2.3.3.1)$$

Where $h_{\text{heat pump}}$ = Head loss of heat pump, m(ft)
C = Coefficient
Q = Flow rate of fluid, L/s (GPM)

A model estimating the head loss across a heat pump for a ClimateMaster Tranquility H/V model 060 using Equation 2.3.3.1 is shown in Figure 2.3.3.1.

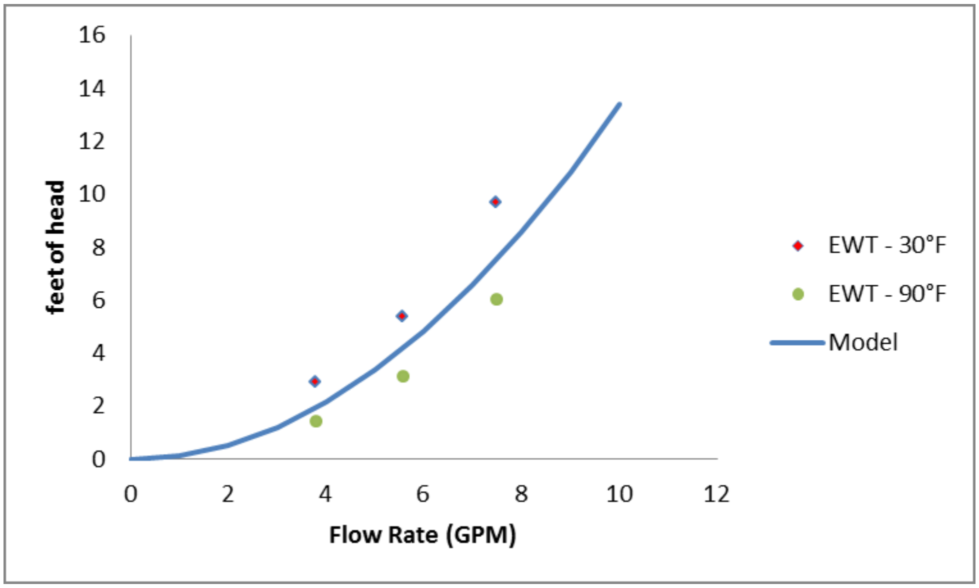

Figure 2.3.3.1 Sample Heat Pump Head Loss

2.3.4 SYSTEM CURVE

For a pumping simulation the total head loss of the system is important. Once the head loss of the whole system has been calculated, the resulting head loss at alternate flow rates can be approximated with Equation 2.3.4.1, if no changes were made to the system.

$$h = C*Q^2 \qquad (2.3.4.1)$$

Where h = Head loss, m (ft)
C = Constant, -
Q = Volumetric Flow Rate, L/s (GPM)

This type of approximation can be used for closed hydronic systems if all heat pumps were left open and only the flow rate changed. GSHP systems have multiple ways to diverge from the normal system curve. With the two-pipe system with a central pumping scheme, the VFD is controlled by a DP sensor. For this type of system, Equation 2.3.4.1 can only be used to approximate the pressure loss outside of the DP sensor which mainly consists of the borefield loop. The pressure loss across the heat pumps would be held constant by the sensor resulting in the overall system acting like an open hydronic system. This is shown in Figure 2.3.4.1. The figure has four curves on it. The borefield acts like a regular closed hydronic system. The heat pump pressure loss is held constant by the DP sensor, and the building distribution piping varies based on which heat pump closes and the layout of the pipes. All of these together make up the system curve. Much like an open system with an elevation change, theoretically at zero flow, there would still be a head loss of 7.3 m (23.5 ft) based on the heat pump.

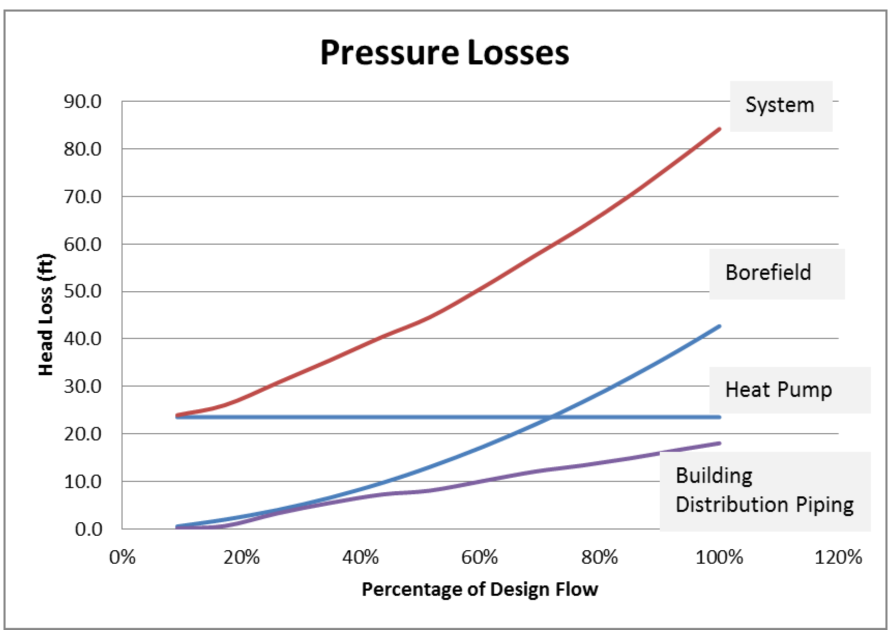

Figure 2.3.4.1 Sample System Curve

For simple parallel loops, such as the borefield, each section can be simplified as a design pressure drop with an equivalent pressure loss coefficient such as the one used in Equation 2.3.4.1. This would also be the case for the one-pipe system if using a VSD.

If the pressure drop across the parallel sections is known, then the flow rate through each section can be solved for directly using Equation 2.3.1.2. At each node of the parallel loops, where the flow distributes and consolidates, each section can only have one pressure. No matter what the intended flow rate is through a section, if left uncorrected a system will naturally balance itself using this principle.

For simple hydraulic systems these can be solved in multiple ways including analytical solutions or using a Newton Raphson (NR) solver. An NR solver uses the NR method to find a solution to a set of equations within specified convergence criteria. In the case of parallel loops, the number of equations would correlate to the number of parallel paths minus one. The NR solver can be used to solve for the flow rates of each path by using pressure balance equations (parallel paths will have an equal pressure drop, not necessarily equal flow rates). For example, take the system in Figure 2.3.4.2; this has three heat pumps (paths) run in parallel. The NR solver can be used to solve for the flow rate through each heat pump. Using a sign convention that going in the direction of the fluid flow is a positive pressure drop and going against the fluid is a negative pressure drop, the pressure drop around loop one and loop two should be zero. This gives us two equations to solve for three unknowns. The third equation can either be a third loop that goes all the way through the system from point one back to point one, or by using the conservation of mass, the flow rate entering point 1 is equal to the flow rate leaving point 1 which is equal to the flow rate entering/leaving point 6. Set the sum of the three heat pump flow rates equal to flow rate 4.

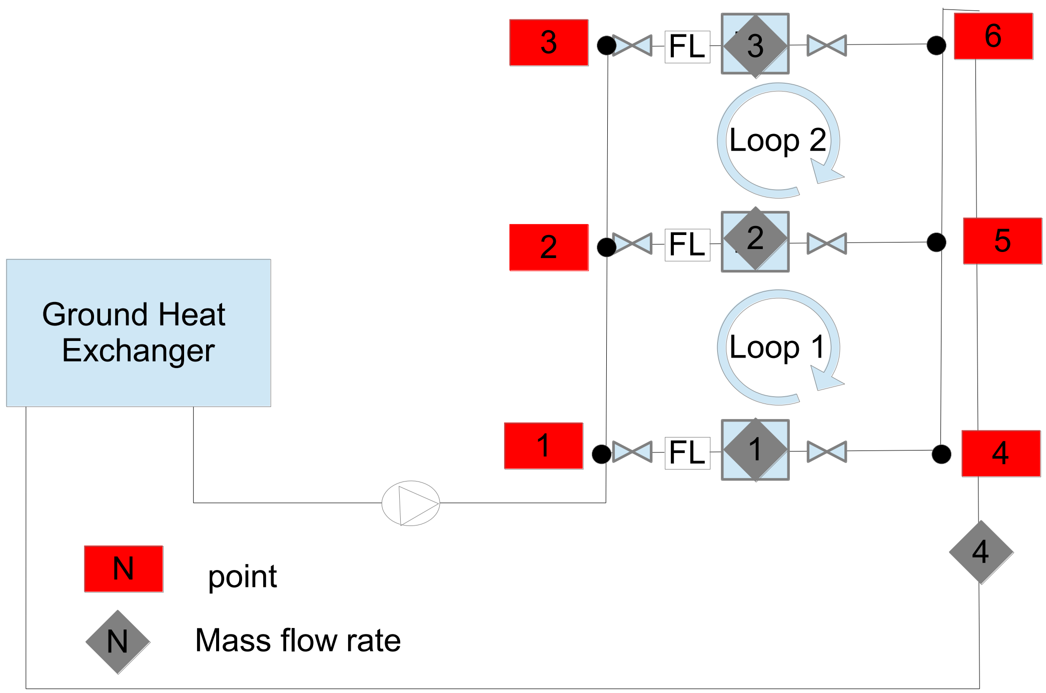

Figure 2.3.4.2 Parallel Loop Example

For the two-pipe system, the total flow rate of the system will always be known. If using a DP sensor to control the VFD, the pump will adjust its speed until the programmed DP is met. Using a NR solver to solve for the speed of the pump will also solve for the flow rate of the pump in that situation. This allows the conservation of mass technique to be used to solve for the independent flow rates in each parallel loop.

2.4 PUMP PRESSURE RISE

2.4.1 CONSTANT SPEED PUMPS

Three of the pumping schemes analyzed use a constant speed pump. The one-pipe system, distributed pumping, and distributed pumping with a hydraulically separated ground loop. For the one-pipe scenario, the pump conditions do not change with the number of heat pumps on. Each heat pump is hydraulically independent of the rest. For this reason, each pump was assumed to always operate at the design condition. For both of the other two pumping schemes, the head loss seen at each pump varies with the number of heat pumps in operation. The extent of how the flow rate and pump pressure rise vary for each condition can only be determined by approximating the performance of real pumps operating data and analyzing the system. Manufacturers typically provide product data that gives the performance of the pump under various conditions. A simplified model for constant speed pumps can use parameter estimation to curve fit the data using Equation 2.4.1.1.

$$\Delta P = C_1 * Q^2 + C_2 * Q + C_3 \tag{2.4.1.1}$$

Where ΔP = Pump Pressure Rise, Pa (ft)
$C_{1\text{-}3}$ = Constants, -
Q = Volumetric Flow Rate, L/s (GPM)

2.4.2 VARIABLE SPEED PUMPS

Variable speed drives have the capability of adjusting the speed of the motor to vary the flow rate. At each speed the pump will ride its curve to equilibrium matching the head loss of the system with the pump pressure rise. It was assumed that for this simulation that the pump was sized appropriately such that the pump pressure rise matched the system losses for each flow rate. Calculation of energy consumption under these conditions is described in the next section.

2.5 ENERGY CONSUMPTION

The primary consumers of electricity in a GSHP system are the heat pump and the circulation pump. For this simulation the control sensors and automatic valves are assumed to be low voltage systems that are powered by the heat pump or the pump itself and have negligible energy consumption.

2.5.1 HEAT PUMP ENERGY CONSUMPTION

Using manufacturer's data the coefficient of performance (COP) of the heat pump can be calculated. Heat pumps do not operate with the same COP for both heating and cooling so they are broken up into $COP_{heating}$ and $COP_{cooling}$. The COP is defined as the ratio of energy output in heating or cooling to the energy input to the heat pump to provide that function. Parameter estimation was used calculate the COP of the heat pump using manufacturers data and Equation 2.5.1.1. Two equation fits were made to account for heating and cooling, but the form of the equation remains the same.

$$\boldsymbol{COP = C1 + C2 * T_{EFT} + C3 * T_{EFT}^2 + C4 * \dot{m} + C5 * \dot{m}^2 + C6 * T_{EFT} * \dot{m}} \quad (2.5.1.1)$$

Where T_{EFT} = Entering Fluid Temperature, °C (°F)
$\dot{\boldsymbol{m}}$ = Mass flow Rate, kg/s (lbm/s)
$C_{1\text{-}6}$ = Constants

With the COP of the heat pump and the heat load on the facility, the energy consumption of the heat pump can be calculated by Equation 2.5.1.2.

$$\text{Energy Consumption} = \frac{\text{Building Load}}{\text{COP}} \quad (2.5.1.2)$$

The building load was discussed previously as being an output from EnergyPlus.

2.5.2 CONSTANT SPEED PUMP ENERGY CONSUMPTION

The energy consumption of a constant speed pump can be calculated using Equation 2.5.2.1.

$$E = \frac{Q * h * SG}{C * \eta} \tag{2.5.2.1}$$

Where E = Energy Input, kW (kW)
Q = Volumetric Flow Rate, L/s (GPM)
h = Pump Head, m (ft)
SG = Specific Gravity, -
C = Unit Conversion, SI = 102 (USCS = 5320)
η = Combined Pump and Motor Efficiency, -

The efficiency will vary with the flow rate and the head produced by the pump. Parameter estimation can be used to estimate the efficiency from the manufacturer's data on the pump curve. For exploration of the effects of efficiency on the performance of some of these systems, the efficiency of the constant speed circulators is calculated in three ways. The first way is to use a constant efficiency. According to Markusson's (2011) research, circulator efficiencies can vary from 10% to 50%, with the latter being achieved experimentally using the ECM "Smart Pumps". It is important to note that this is much higher of an efficiency than what was found by Kavanaugh et. al. (2003) a few years prior for standard circulator pumps. For the one-pipe system, this can be assumed as the circulator is not affected by the status (on/off) of other circulators. We can assume that the pump will operate at the design condition at all times. The pressure drop that the pump sees will vary slightly throughout the year as the temperature of the fluid varies, but this is assumed to be minimal and have a negligible effect on efficiency. For the two-pipe systems that use individual circulator pumps the system curve varies with the number of pumps in operation at the time. For this, two methods could be used to calculate the efficiency of the circulator pump. A pump could be selected for the individual system and the performance of the pump could be approximated using Equation 2.5.2.2.

$$\eta_{circ} = C_1 * Q^2 + C_2 * Q + C_3 \tag{2.5.2.2}$$

Where η_{circ} = Circulator Efficiency, -
$C_{1\text{-}3}$ = Constants, -
Q = Volumetric Flow Rate, L/s (GPM)

The other option would be to use the equations from Sfier et. al. (2005) . This would assume that all circulators have a theoretical pumping power of less than 0.375 kW (0.5 HP). This is shown in Equation 2.5.2.3.

$$\eta_{circ} = 0.06066 * P_{theor}^{0.3543} \tag{2.5.2.3}$$

Where P_{theor} is in Watts

Once the input power required has been calculated, it must be multiplied by the runtime fraction for that hour to calculate the total number of kW-hr that were consumed for that hour. Pumps were found for each design operating point from the Wilo-Select pump selection tool to ensure that efficiencies were realistic with ECM pumps.

2.5.3 VARIABLE SPEED PUMP ENERGY CONSUMPTION

For variable speed pumps many authors have made energy savings claims using the pump and fan affinity laws. These laws state that the energy consumption of a pump is proportional to the flow rate cubed shown in Equation 2.5.3.1

$$P_2 = P_1 \left(\frac{Flow\ Rate_2}{Flow\ Rate_1}\right)^3 \tag{2.5.3.1}$$

In some cases this may be true, such as if the WTW efficiency was constant. As discussed previously, this is not the case. Sfeir et al. (2005) consolidated multiple equations introduced by various authors that can be used in simulations that calculate pump, motor, and VFD efficiencies based on the theoretical power required and the percentage of the full load that the pump is operating at. Equations 2.5.3.2, 2.5.3.3, and 2.5.3.4 are the relevant equations used for this thesis. These equations should be considered valid for nominal power of 0.75 to 37.5 kW (1 to 50 HP)

$$\eta_{\text{pump}} = 0.322 + 1.066\ \text{PFL} - 0.553\ \text{PFL}^2 \tag{2.5.3.2}$$

$$\eta_{motor} = \frac{(79.35 + (14.3 * \text{P\_theor})}{(3.18 + \text{P\_theor}))} * (\frac{(\text{a} * \text{PFL})}{(\text{c} + \text{PFL})} - \text{b} * \text{PFL}) \tag{2.5.3.3}$$

$$\eta_{\text{VFD}} = 87.84 + 0.225 * PFL - 0.001228 * PFL^2 \tag{2.5.3.4}$$

In each equation, PFL is the percentage of the design flow rate that the pump is operating at. For Equation 2.5.3.3, constants *a, b,* and *c* are dependent on the size of the motor. Values for these constants are shown in Table 2.5.3.1.

Table 2.5.3.1 Motor Efficiency Constants after Sfeir, et al. (2005)

Motor Power (HP)	a	b	C
1	144.45	0.16	26.27
1.5 - 5	220.86	0.64	35.56
7.5	171.61	0.41	22.43
10	145.02	0.28	14.58
15 - 25	124.74	0.17	7.75
30 - 60	111.99	0.0798	4.00

Figure 2.5.3.1 shows the efficiency curves for the above equations implemented with a theoretical power requirement of 1 HP.

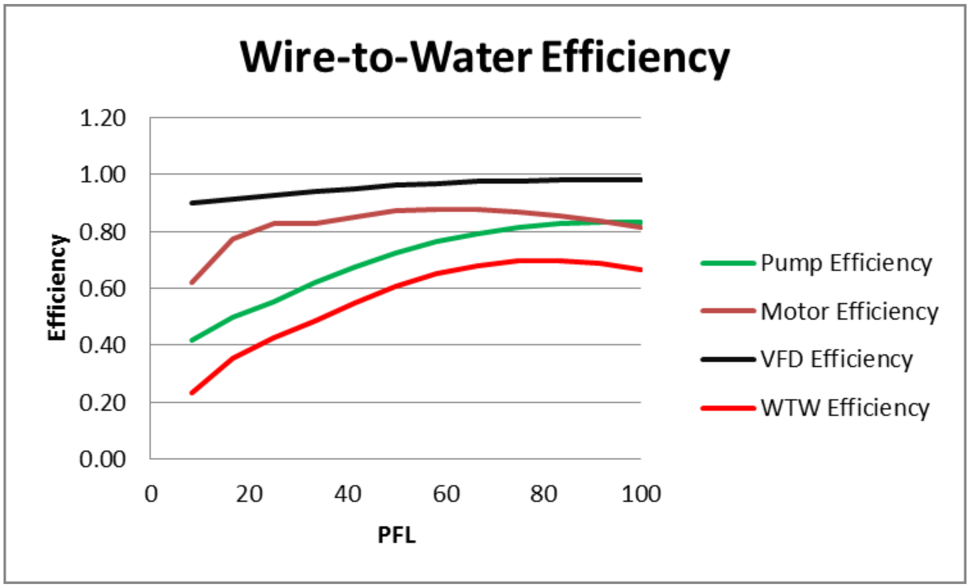

Figure 2.5.3.1 Wire-to-Water Efficiency Curve

Since the VFD on a two-pipe system with central pumping is controlled by a DP sensor, it maintains a constant pressure drop across the heat pumps to ensure they receive a relatively constant flow. For this reason, the system pressure loss curve does not follow a normal quadratic relationship. This also further deviates the actual power requirement from the pump affinity laws prediction as shown in Figure 2.5.3.2.

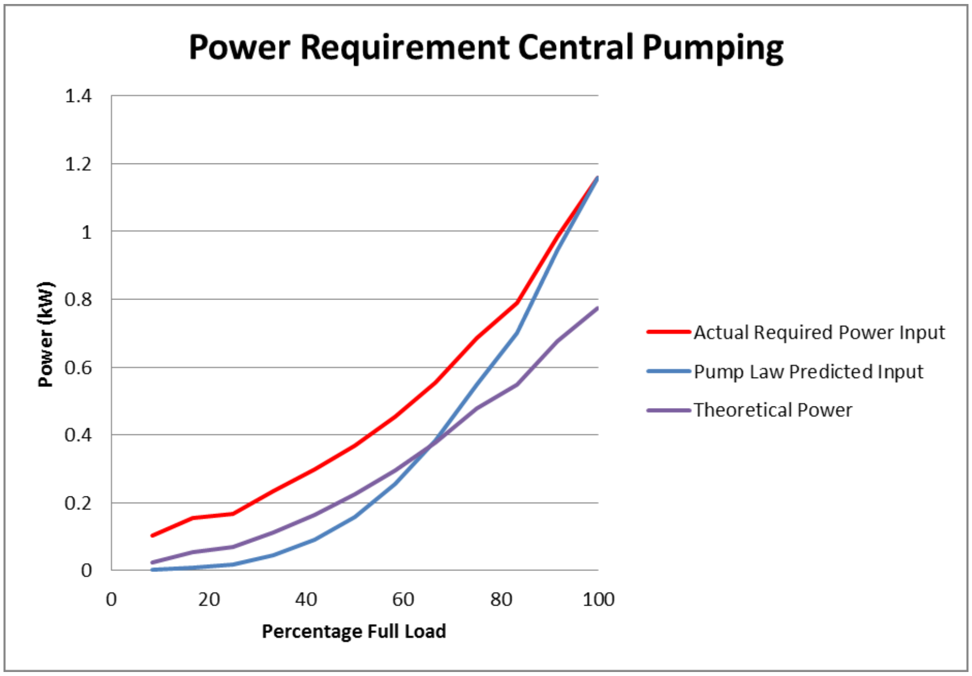

Figure 2.5.3.2 Power Input Requirement

Due to this constant pressure drop, the system acts similarly to an open system in that there is a constant pressure drop and theoretically at zero flow there would still be a constant pressure loss at the heat pumps. If the pump affinity law was used to calculate the energy consumption at low flow rates, the prediction at flows below approximately 70 % would be lower than the theoretical power required by the system. Obviously this is physically impossible without the pump having an efficiency greater than 100%, which to this author's knowledge, does not exist.

By calculating the actual power that would be required based on the efficiency equations above, a power curve can be approximated by a curve fit of the power required to the percentage of full load. Since this system was designed using the same heat pumps and using reverse return piping, the pressure drop across each heat pump is relatively the same and flow rates to each heat pump, when on, are assumed to be equal. This allows the power curve to be a function of only the number of heat pumps on at one time and the average temperature of the circulating fluid and not dependent on which heat pump is on.

For example, in a building with 12 identical heat pumps, configured with two-pipe distribution with central pumping, 0.95 L/s (15 GPM) flow to each heat pump, 6 m (20 ft) spacing between heat pumps, a NR solver was used to solve for the flow rate through each heat pump given a constant pressure differential setting. Figure 2.5.3.3 shows flow rates to each heat pump for 7 - 12 heat pumps in operation. This was simulated by systematically shutting off a heat pump starting from the first one in the line.

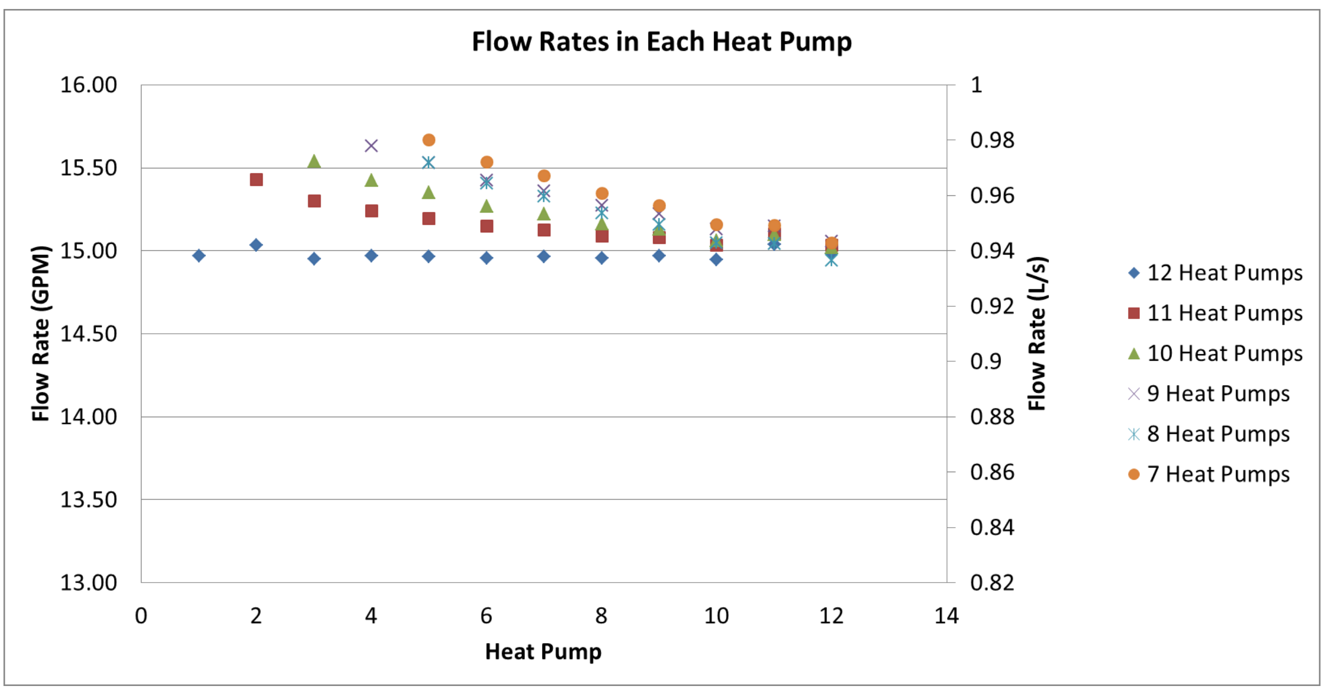

Figure 2.5.3.3 Flow Rate Through Each Heat Pump

This shows that when all heat pumps are in operation, the flow rate varies only .01 L/s (.9 GPM) or roughly 0.6%. This variance increased as pumps were shut off, but peaked at 0.04 L/s (0.59 GPM) or roughly 3.9% with eight heat pumps in operation. In section 1.2.1 it was stated that a 33% change in flow rate has a 4% change in the EER of a heat pump. Extrapolating that to a 3.9% difference in flow rate would equate to about a 0.45% change in EER which is deemed insignificant.

CHAPTER III

APPROXIMATIONS FOR COMPUTATIONAL EFFICIENCY

3.1 SYSTEM FLOW RATE

For each system, the hourly flow rate had to be broken down into sub hourly simulations to facilitate analysis of the many possible part load scenarios. For a system with *N* heat pumps, with each heat pump having its own on/off valve, there are *N!* number of possible flow rates for the central pump. Heat pumps do not typically run for the entire hour. A runtime fraction can be calculated to determine what percentage of the hour each heat pump is running. In a computer simulation it is impossible to know at what time within the hour the heat pump will turn on and run, and if the total runtime fraction will be executed consecutively or if it will be broken up into multiple shorter runtimes. This in turn creates an infinite number of possible flow changes within the hour. Sfeir, et al. (2005) propose taking an average flow rate for the hour as a simplification. Using the four pumping scenarios introduced by Bernier and Bourret (1999) shown in Figure 3.1.1, this can be shown to lead to underestimation of the actual energy consumption. For example, taking an ideal closed loop variable speed pumping scenario where the system curve remains constant and taking the methodology originally introduced by Bernier and Bourret (1999), the energy consumption for each of these flow rates and an average flow rate can easily be calculated. Bernier and Bourret created the pumping profiles in Figure 3.1.1 for annual simulations where the pumps were assumed to be in operation for 8,000 hours per year. For this thesis, this was adapted to sub hourly flow distributions where the profile was applied to one hour of operation instead of 8,000 hours of operation. Each scenario has a distribution of flow rates ranging from the design flow rate to 30% of the design flow rate decreasing in increments of 10%. Since this example is based on using an ideal closed loop pumping system, we can apply the affinity laws showing that the speed of the pump is directly related to the flow rate of the system. This is used only for this example as in a GSHP system, the system curve changes each time a heat pump changes status (on/off). The first flow rate scenario consisted of an even distribution of each of the flow rates, shown as the “Even Distribution” bars in Figure 3.1.1. The second scenario used a bell curve distribution pattern where the majority of the time was spent around 50% flow and less time was spent at full flow rates or minimum flow rates. This is shown as “Bell Shaped Distribution”.

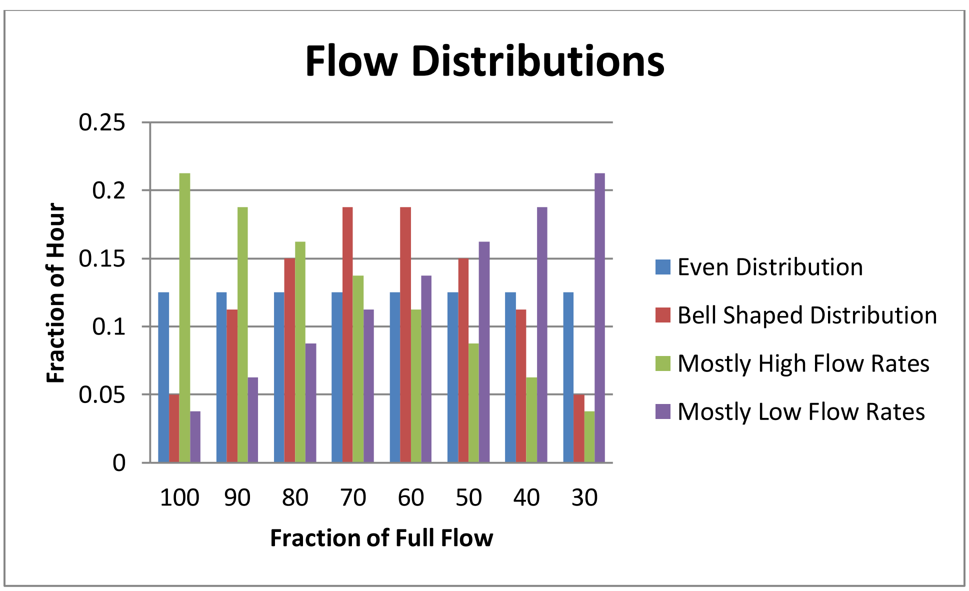

Figure 3.1.1 Flow Distributions

The third and fourth scenarios were mostly higher flow rates and mostly low flow rates respectively. The pump used in this scenario is a Gould 3642 pump with a VFD. The performance of the pump, 5 HP ODP motor, and VFD was curve fit to manufacturers data in a VBA function written by Dr. Jeff Spitler.

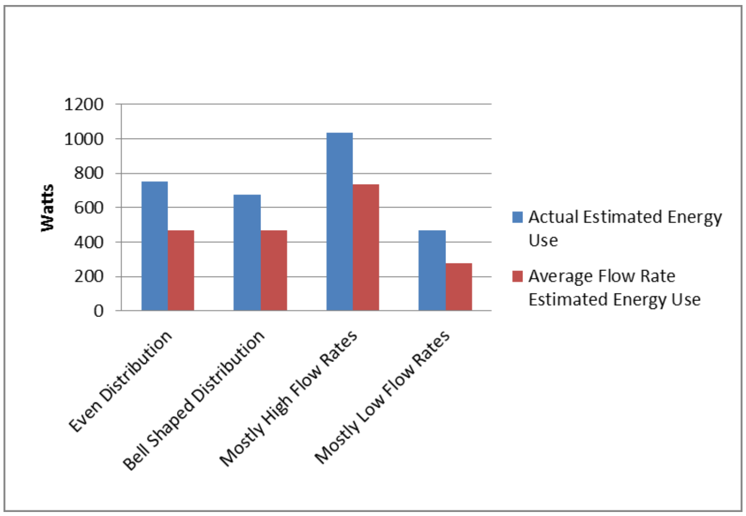

Figure 3.1.2 Comparison of Energy Usage for Different Flow Scenarios

The 100% design flow rate was taken to be 4.5 L/s (71.3 GPM). To calculate the energy consumption for each fraction of an hour, the hourly energy consumption was calculated for the percentage of flow rate for that fraction and then multiplied by the fraction of the hour. The total of these was then summed for the total hourly energy consumption. For the even and bell shaped distributions, the average flow rate was the same. The hourly energy consumption was about 750 watts and 0.678kW-hrs respectively. If using the average flow rate approach, the calculated energy consumption would be nearly 0.470 kW-hrs for both scenarios. This resulted in over 30% underestimation of the energy consumption in each case. The mostly high flow rate scenario underestimated the consumption by 29% and the mostly low flow rates underestimated the consumption by over 40%. This example shows that for a closed hydronic system with a constant system curve, we cannot simply average the flow rate over the hour as this can lead to drastically underestimated energy consumption rates. A GSHP system curve with multiple heat pumps in parallel, changes every time a heat pump opens or closes. A similar result would be seen doing the analysis for this type of system also.

To prevent this, an algorithm was created to simulate random start times for each heat pump within each hourly time step. From the EnergyPlus simulation hourly loads were given from each zone. Based on the capacity of the heat pump that was selected for this study, a run time fraction was calculated to estimate the amount of time through the hour that the heat pump was in operation. Since heat pumps typically have a minimum runtime to prevent short cycling, this was approximated by breaking up each hour into 12 sub-hourly time steps or into 5 minute intervals. For simplification, it was assumed that for each heat pump the total run time would execute in consecutive intervals with no break and would only run on the time steps. This means that the heat pump can only turn on at the beginning of a time step and once the heat pump turns on, it will remain on until the entire load for that hour was met and then would turn off.

Once the RTF was calculated, it was converted into an integer run time based on the values provided in Table 3.1.1.

Table 3.1.1 Heat Pump Run Times

RTF	Run Time	RTF	Run Time
0 < RTF < 1/12	1	6/12 < RTF < 7/12	7
1/12 < RTF < 2/12	2	7/12 < RTF < 8/12	8

2/12 < RTF < 3/12	3	8/12 < RTF < 9/12	9
3/12 < RTF < 4/12	4	9/12 < RTF < 10/12	10
4/12 < RTF < 5/12	5	10/12 < RTF < 11/12	11
5/12 < RTF < 6/12	6	RTF > 11/12	12

To calculate the start time for each heat pump, the eligible start times were created by taking the total number of intervals in the hour and subtracting the run time from it, for example if the heat pump were to run for 2 intervals, (12-2 = 10), the start time could be in any of the first 10 intervals of that hour. To simulate a real system the actual start time was calculated by using a random number generator calculating a random integer within the eligible start times. Since it was assumed that the heat pump would run consecutively until the entire hourly load was met, the time for the heat pump to turn off could be calculated by taking the start time plus the run time. So if for the above example if the random start time were 4, the heat pump would run through interval 4 and 5 and stop at interval 6. This was done for each heat pump and then tracked in a matrix with ones and zeros. Ones were given for when the heat pumps were on and zeros were given for when the heat pumps were off. For each interval, the number of heat pumps in operation could then be counted. For two-pipe systems with a central pump controlled by a DP sensor, to determine the flow rate for each interval, it was approximated that the flow rate through each heat pump was identical. Therefore the total number of heat pumps on was divided by the total number of heat pumps installed in the system to get a percentage of the design flow rate, ie. four heat pumps on in a 12 heat pump system would give 4/12 = 33% of the design flow rate.

Since minimum speeds of VSDs vary from manufacturer and economic effects vary with efficiencies, a minimum speed of 30% of design speed was used. It is interestingly enough for this design, 30% of the design flow rate for this system is where the boreholes transition to laminar flow at about 0.038 L/s (0.6 GPM) flow in a ¾" pipe.

3.2 PUMP ENERGY APPROXIMATION

Since this system was set up with identical heat pumps serving each zone and reverse return piping configuration was used, in central pumping scheme, each heat pump was approximated to see the same flow rate. The pump was then approximated to see only 13 operating conditions ranging from no heat pumps in operation to all 12 heat pumps in operation. The energy consumption of the pumps was calculated for each of the 13 conditions. Since the head loss in the system will vary with the change in temperature of the circulating fluid, only estimating the energy consumption at standard water temperature could lead to significant error in energy consumption. Equation 3.2.1 was used as a power model.

$$\boldsymbol{P = C_1 + C_2 * T_{EFT} + C_3 * T_{EFT}^{\ 2} + C_4 * PFL + C_5 * PFL^2 + C_6 * T_{EFT} * PFL} \qquad (3.2.1)$$

Where P = Power, kW (HP)
$C_{1\text{-}6}$ = Coefficients, -
PFL = Percentage of Full Flow, %
T = Temperature of Fluid, °C (°F)

The power model includes all 13 operating conditions at water temperatures ranging from 0°C (32°F) to 35°C (95°F) and being simulated every 2.8°C (5°F). Using the generalized least squares method of approximation, a curve was fit to the energy consumption of the pump based on the percentage of the design operating condition and the temperature of the fluid. Figure 3.2.1 shows an example of this curve fit for 0°C (32°F), 15 °C (59°F), and 30°C (86°F). In this figure the operating points that were calculated for 12 of the 13 operating conditions are shown in the Actual Power Required data points and the curve fit model is shown as the Power Model.

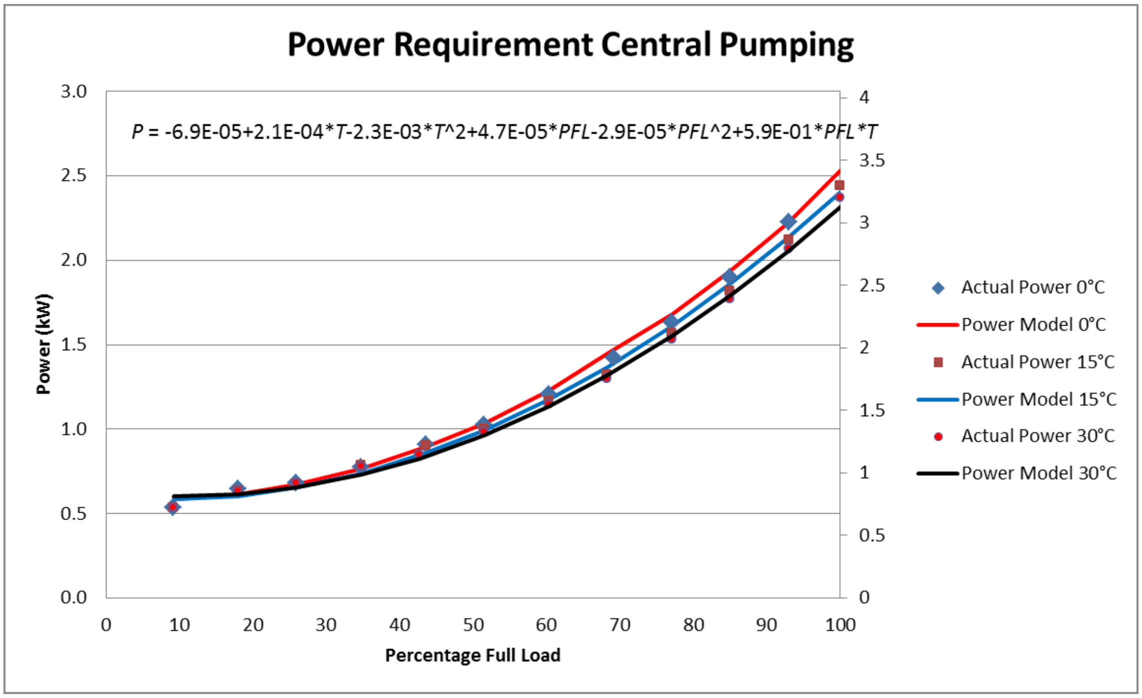

Figure 3.2.1 Pumping Model Curve

Comparing 0°C (32°F) and 30°C (86°F) gives fairly consistent values for caclculated power requirements below 35% of full load flow rates, that being less than a 5% difference. This is the region in which a pump would not operate as it is below the minimum speed of a VSD. In the operable range of a VSD, or flows greater than 35% of full flow, the difference ranges from 7% to 9.3%.

For each interval as described above, the energy consumption was calculated and multiplied for the 1/12th fraction of the hour that it was operating at that point. These values were summed and the hourly energy consumption was calculated. This was done repeatedly for the 8760 hours in the year. Since the primary driver of the start and stop times of this simulation count on a randomly generated number, there is a possibility of there being a wide range of solutions calculated. To verify the consistency of the solutions a test trial was performed for a given scenario for 100 iterations. Each annual energy calculation was stored and compared to the average solution. For this trial the average energy was 3,872 kW-hr. The maximum calculated consumption was 3877 kW-hr and the minimum was 3,867 kW-hr. This had a range of +/- 5 kW-hr or roughly +/- 0.13% difference. Since the variance of this was much less than one percent in a run of 100 iterations, only running one simulation for each scenario would give acceptable results that could be used for comparison.

3.3 HEAT PUMP ENERGY APPROXIMATION

As stated above, the energy consumption of the heat pump is a function of the flow rate of the circulating fluid, the EWT of that fluid, and the building load. To determine this, the COP is calculated for each heat pump, for each time step, as described in section 2.5.1 of this thesis. The building loads were calculated with EnergyPlus. For the two-pipe system with a central pump and for the one-pipe system, the flow rate was picked at a constant volume flow rate when the heat pump was running. This simplifies the problem to one unknown. In a real world system, the actual temperature of the fluid entering a heat pump can vary quite a bit through each 5 minute cycle as described above. If there was no flow for a period of time, there would be a lag time depending on the volume of liquid inside of the system. For this scenario, the system is calculated to contain about 310 gallons of water. With all 12 heat pumps running, the design flow rate is 82 gallons per minute. This means that if assuming plug flow through the pipes, at full load, the fluid will take 3.78 minutes to fully travel through the entire system. The heat pump could then operate for 75% of each cycle with no influence on the temperature of the fluid from the current load for the first 3.78 minutes. This also means the temperature of the fluid is still dependent on the previous loads. In the other case where only 1 heat pump is on, at a flow rate of

6.8 gallons per minute, it would take the water 45.5 minutes to travel through the system. As a result of this, transient effects within the heat pump during start up and shut off, and of the ambiguity of when a heat pump is actually operating throughout the hour, temperatures were calculated on an hourly basis to give an average fluid temperature entering and exiting the heat pumps. For all of the two-pipe scenarios, the temperature entering the heat pump is assumed equal to the temperature of the fluid exiting the borehole. To effectively analyze the one-pipe system and the effects of the temperature change on the last heat pumps in the building the temperature rise across the system was averaged across the hour in the same manner. It was assumed that the load applied by the first heat pump was realized by the second heat pump across the entire hour and so on down the line.

For distributed pumping and distributed with hydraulic separation pumping the flow rate through the heat pump was also assumed to be constant. This would not be the case in a real system and may be situational dependent, but in terms of heat pump energy consumption, this was reasonable. For example, in a system with 12 heat pumps and a design flow rate of 0.948 L/s (15GPM) per heat pump with a design head loss of 15 m (49.3 ft), a pair of Wilo Circo Star 30 BU circulator pumps sized to meet approximately half of the pressure rise at the design flow can be placed in series to meet the design conditions. With all heat pumps operating with the same pumping configuration, each heat pump would see 0.803L/s (12.73 GPM). If only one pump were operating, the pump operating condition would decrease to a total of 7.9 m (25.4 ft) of head and 0.962 L/s (15.25 GPM). Using the parameter estimation model of the COP for a ClimateMaster TR H V 060 heat pump in cooling mode, with an EWT of 20°C (68°F) this would be 5.83 and 5.81 for each respective scenario. This is approximately a 0.3% difference. Again, this is for this situation and with the selected pumps. The shape of the pump curve may drive a different result in another situation, so this assumption may not be valid in all cases.

CHAPTER IV

PUMPING AND PIPING SYSTEM DESIGN TOOL

4.1 PUMP AND PIPE SIZING TOOL

The above equations and assumptions were incorporated into a spreadsheet and Microsoft Visual Basic (VBA) code. The intent of this spreadsheet was to be able to compare the energy costs of different pumping and piping configurations for a commercial GSHP system after the initial loads and borefield size have been calculated by standard sizing practice. This is not means to be used for sizing a borefield, but it can be used to help determine the best configuration from a pumping energy efficiency standpoint. The design tool is set up to calculate various operating data for this system for a 12 heat pump system or less. It will adjust the output based on the number of heat pumps in the system. For more than 12 heat pumps the user would have to modify the code and the spreadsheet. The program utilizes a model developed by Xu and Spitler (2006), which is described later in this thesis, to model the borehole response to heat inputs to calculate the returning water temperature from the borefield. This was then used to determine the EWT for the heat pumps. There are multiple tabs in this spreadsheet that are used for different aspects of the analysis. The tab labeled “main” is for the user to input the specifics of the system. A screen shot for this program is shown in Figure 4.1.1. From this main screen the data is used to initially size the pipes for each section of the system and to calculate the size of the pump required for the system. Each of the user input items to be entered will now be defined.

Required System Data (User Input)

Borefield Data

Number of Columns	7
Number of Rows	6
Borehole Spacing (ft)	20
Borehole Depth (ft)	250.0
Distance from Bldg to borefield (ft)	100
Location of manifold	Vault
Ground Loop Configuration: (Reverse Return / Direct Return)	Reverse

Click here to see a diagram of the borefield set up

Building Data

Type of Building Distribution Network	Two Pipe
Pumping Scheme	Central
Distance From Central Pump to 1st HP (ft)	20
Building Loop Configuration: (Reverse Return / Direct Return)	Reverse

Click here to see a diagram of the GSHP Configuration

Pipe Sizing Criteria. Note: leave blank to use default criteria

Default: pipes 2" and smaller: velocity < 4 fps pipes 2" and greater: head loss < 4 ft/100 ft of pipe	
Head Loss (ft) /100ft	2

Figure 4.1.1 Main Data Entry Tab

The program allows single boreholes, line configurations, and rectangular borefield configurations. The user can specify whether they would like to have direct return piping or reverse return piping for the borefield or the building loop. Figure 5.1.2 shows a pictorial definition of the borefield dimensions. Each column is a parallel line of boreholes that are *N* rows deep. The borehole spacing is assumed to be the same in both the row and column directions. If a vault header is used to distribute flow to each column, the distance to the building is defined as the distance from the central pump of the building, if used, to the edge of the vault. From the vault to each column it was assumed that the distance was spaced according to the borehole spacing provided. For purposes of clarification in this program, a header is defined as pipe that distributes flow to multiple pipes that is generally longer in length, in the case of the borefield, it spans the depth of each column. A manifold is defined as a compact pipe that distributes flow to multiple pipes and generally can be contained in a vault that is buried in the ground or within a single mechanical room.

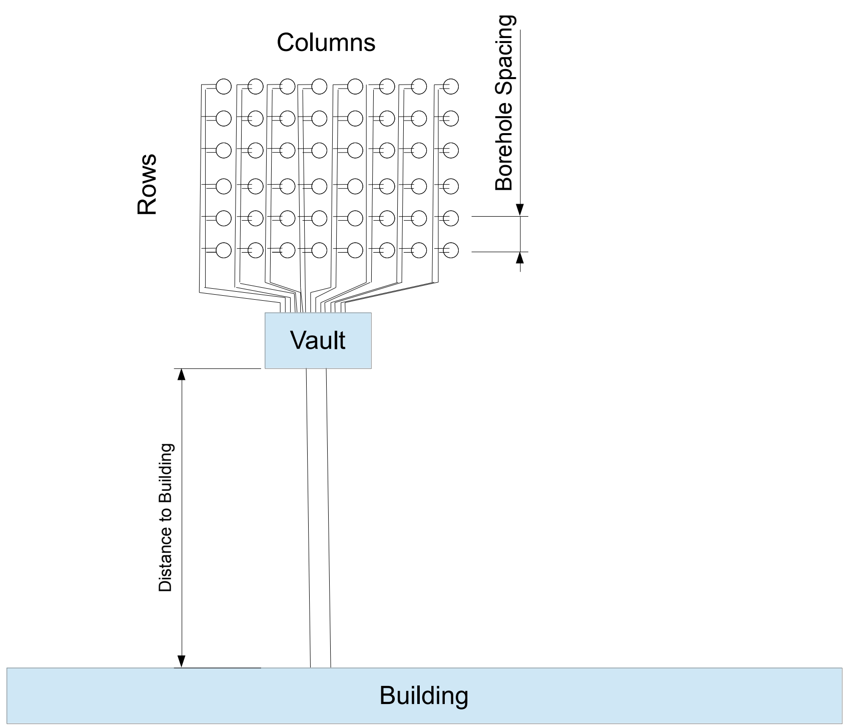

Figure 4.1.2 Borefield Dimensions

Figure 4.1.2 shows a borefield with eight columns and six rows. It has a vault manifold and a direct return header.

Inside the building, the distribution header can be configured in two basic ways. A one-pipe system is shown in Figure 4.1.3.

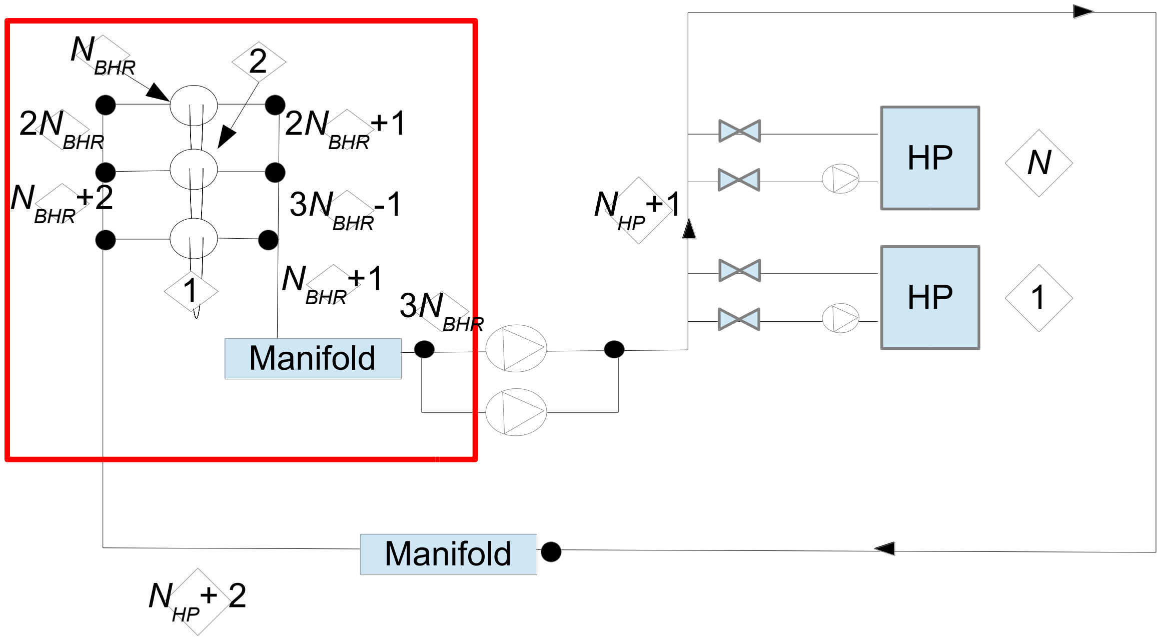

Figure 4.1.3 One-pipe Distribution Header

In this type of system the supply and return is the same pipe. Water is drawn from the main line for each heat pump and expelled back into the same line.

The next type of header is the standard two-pipe header which is shown in Figure 4.1.4. In both Figure 4.1.3 and Figure 4.1.4, the manifold represents where the flow is distributed to the borefield. This could be in an equipment room or close to the borefield in a vault. One manifold is for the flow out to the borefield and the other is to consolidate the flows back into the building. The red box drawn in the figures shows the boundary of the borefield loop. For example, in Figure 4.1.3, the building loop includes the pipe after the distribution manifold/vault, whereas in Figure 4.1.4, this pipe is considered part of the borefield loop.

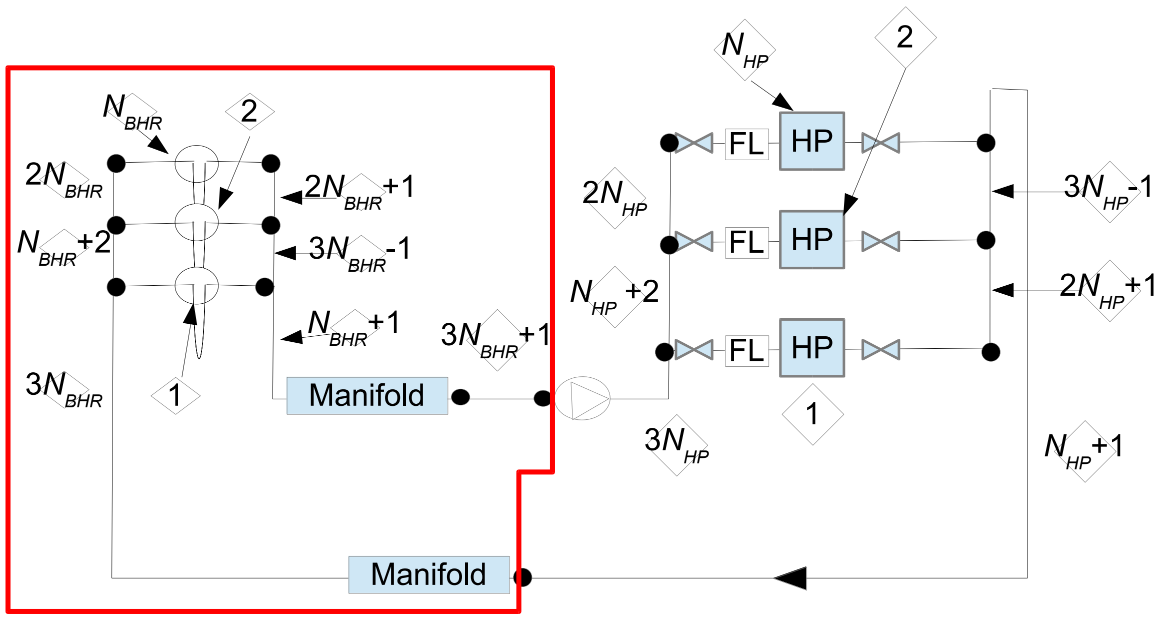

Figure 4.1.4 Two-pipe Distribution Header

This figure shows N_{HP} heat pumps configured in reverse return and a borefield of N_{BHR} rows deep. In this case, the N_{BHR} would be the number of rows in each column and not the total number of boreholes in the system. The flow limiting valves that are shown in this figure are only as a representation of what could be in the system and are not assumed to be installed in the program. The two-pipe header has more individual sections to it. The borefield is set up so that the first borehole section number is $3N_{HP}+1$. The numbering scheme does not change for direct or reverse return headers, rather only for one or two-pipe systems.

There are three pumping schemes that can be used by this program for the two-pipe system: Central pumping, Distributed, or Distributed with Hydraulic Separation. Each one is described below. In the program, cell C3 is where the user can select between a one-pipe and a two-pipe system. Cell C4 will always show the different pumping options, but is not used when One-pipe is selected. There is one default option of central pumping with distributed pumping in the one-pipe system.

Central pumping is where one or more pumps supply flow to the entire system in a central location, typically a mechanical room. Figure 4.1.4 shows two pumps in parallel in a central pumping scheme. This program is only set up to analyze this pumping scheme with a single pump that operates with a variable speed motor. The speed of the motor is controlled by a DP sensor that senses the pressure drop from the supply line, through the heat pump, and back to the return line. This includes the pipe frictional losses, fittings, and the heat pump. The pressure set point should be such that this is at design flow rates. The program uses this set point as ending criteria for a loop. It starts with the design system flow rate and decreases the flow rate until the HP pressure drop is below the set point. For initial system sizing, this function is not used. The program assumes that each heat pump will receive design flow rates. For sizing, it also assumes that each borehole receives equal flow rates.

Distributed pumping has one pump for each heat pump and is sized accordingly to be able to handle the individual heat pump flow rate and the pressure rise of the system when all heat pumps are running. Each pump is demand driven such that it only operates when the heat pump is running. This should only be used for systems with a small number of heat pumps to avoid unnecessarily high pumping power consumption.

Distributed pumping with hydraulic separation refers to each heat pump having its own pump that runs when the heat pump runs. The borefield has its own pump that is hydraulically separated from the building loop by a bypass pipe. This is to allow the circulator pumps to operate independently from the borefield pump. The circulators for each heat pump are sized to be able to provide the design flow rates when all heat pumps are operating.

The distance from the central pump to the first heat pump is defined from the outlet of the pump to where the pipe connects to the supply line feeding the first heat pump. In the heat pump data section of the program, the user should enter the design flow rate for each heat pump, the pressure drop associated with each heat pump, to include any strainers or valves to be installed in the system, the distance from the supply line, assumed to be equal to the distance to the return line, and the distance from the last heat pump. All can be seen in Figure 4.1.5.

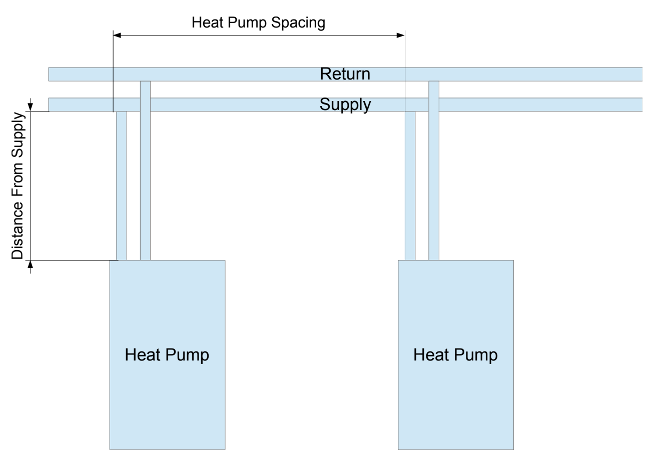

Figure 4.1.5 Heat Pump Dimensions

The user can also set the criteria that are to be used to size the individual sections of pipe. The default criterion is set to four feet per second for pipes two inches and smaller and four feet of head loss per 100 feet of pipe for larger pipes. This is not a recommendation and is only there as it is commonly found in text books for pipe sizing criteria. Using this will result in high system head. If this field is left blank, the program will use this default criteria.

There are links to various diagrams on this page to help the user define what lengths are to be entered. The DP sensor set point is not used for initial sizing of the system and will be described later. Once all relevant data has been entered the user can click on the button labeled "Calculate System Details". This will populate both the Details_SI and Details_US tabs which are for SI units and US customary units.

In these tabs the user can see the pressure loss across the building, the borefield and the total system pressure loss. These cells are set to highlight in red to show when the respective pressure loss is above that recommended by Kirk Mescher in the ASHRAE webinar (2011) of 7.8 m (25 ft) for both the building loop and the borefield loop individually and for a total of 15.6 m (50 ft) of head loss for the total system. This will also have the system flow rate and the theoretical size of the pump required to meet these characteristics. For the heat pump and borehole parallel loops, the user can see the unbalanced flow rates and the required K-value or additional pressure drop that would be necessary to equal out the flows. This would essentially act as simple approximation of a flow limiting valve. The bottom of this tab has the Piping system details where the user can see the lengths, nominal diameter, sectional K-value, and the associated total pressure loss of that section. The previously discussed numbering schemes can be used to determine which section number belongs where.

4.2 SECTION NUMBERING SCHEMES

The numbering of sections of pipe inside the building differ based on the piping configuration used to distribute the fluid. All one-pipe distribution systems use the same scheme which is described in section 4.2.1. For a two-pipe system the numbering scheme will differ only based on the type of return, direct or reverse return as described in section 4.2.2. The borefield will almost always if not always be configured with a two-pipe distribution configuration, which will allow the numbering scheme to be consistent with that of the building loop numbering scheme depending if it is configured with a direct or reverse return. For the borefield, the numbering starts directly after the highest number of sections in the building loop, such that if there were 24 sections in the building loop, the first borehole would be the $(24 + 1) = 25^{th}$ section and so on. For section numbering it is assumed for numbering purposes that the distribution manifold is the distinction from the borefield loop of the piping network and the building loop. It is assumed that each column of boreholes is identical. With this assumption, the pipe size, flow rates, and pressure losses are only calculated for one column. The numbering of each borehole would be the same regardless of which column is used.

The building distribution pipe is the pipe that runs from the outlet of the pump, through the building, and to the manifold that feeds the borefield. The numbers in the diamonds in each figure represent the numbering scheme that was used to track each section of pipe. Sections of pipe are not broken down into commonly purchased lengths; rather each section is broken up by each node represented by the circles in each figure. The nodes are used as points where flow rates can change within the system and the pipe needs to be sized appropriately. With respect to the pumps in the system, the nodes are such that they are on the intake and discharge of the pump. The number one in this figure represents the section of pipe that carries water from the main pipe to the heat pump and back to the main line. The heat pump is treated as a fitting in this section with a user defined flow rate and pressure drop. These numbering schemes are shown in Figures 4.1.3 and 4.1.4 for the one-pipe configuration and two-pipe configuration with reverse return. In both figures the borefield is shown in direct return. This is shown in both figures not as a recommendation but only to represent the numbering scheme for that type of return piping.

4.2.1 ONE-PIPE NUMBERING SCHEME

In the one-pipe system, the first *N* sections represent the sections of pipe connecting the heat pump from the distribution pipe and back to the distribution pipe where there are *N* heat pumps in the system. The distribution pipe is the *N+1th* pipe in the system. The distribution pipe will be sized identically throughout the building since the distribution loop will see identical flow rates. The distribution pipe goes from the outlet of the pump to the distribution manifold that feeds the borefield. The pipe after the distribution manifold to the borefield is defined as $N_{HP} + 2$ and goes till the pipe branches off for the first borehole. The numbering for the borefield starts at $N_{HP} + 3$.

4.2.2 TWO-PIPE REVERSE RETURN NUMBERING SCHEME

Similar to the one-pipe configuration, the numbering of the two-pipe configuration starts with the first heat pump and goes to N_{HP}. The $N_{HP}+1th$ section is the pipe that goes from where the last heat pump in the building connects to the return pipe and goes to the manifold that distributes the fluid to the borefield. The sections in the supply pipe starting after the first heat pump start with $N_{HP} + 2$ and go through $2N_{HP}$. The return pipe numbering starts after the first heat pump starting at $2N_{HP} + 1$ and goes through $3N_{HP}-1$. The pipe that goes from the outlet of the pump to the where the first heat pump branches off is $3N_{HP}$. The pipe section after the distribution manifold for the borefield for both two-pipe distribution configurations will be numbered $3N_{HP} + 3N_{BH}$.

4.2.3 TWO-PIPE DIRECT RETURN NUMBERING SCHEME

A two-pipe configuration with direct return has a similar numbering scheme as reverse return. The first numbering for the heat pumps goes from 1 to N_{HP}. The supply pipe from the first HP to the last HP go from $N_{HP} + 2$ to $2N_{HP}$. Like the reverse return, $N_{HP} + 1$ represents the pipe from the last heat pump to the borefield manifold and $3N$ goes from the pump outlet to the branch of the first HP.

4.3 PUMP ENERGY MODEL

The central pump energy tab is used for its respective design constraints. This will only work if the system was designed for this pumping scheme. The user must specify what DP set point the system should be set at to calculate the total pumping energy consumption. This set point accounts for the pressure drop across the heat pump itself and the lateral lines connecting the heat pump to the main supply and return lines. The efficiencies of the main pump are calculated based on the models described by Sfeir, et al. (2005).

Once the DP set point has been set the user clicks on the button Analyze System. It will then read the details that were calculated by sizing the system on the Main tab and printed to the Details tabs. The number of heat pumps and the nominal pipe diameters are read from the Details_US tab while the pipe lengths and K-values are read from the Details_SI tab. The number of heat pumps could be changed to be read from the Details_SI tab if desired, but is not necessary. The nominal pipe_diameter must be read from the Details_US tab. The pipe properties from various size pipes and types of pipes is read from a program written by Sam Hobson (2006). His program takes inputs on nominal pipe sizes in IP system and returns many features of the pipe such as; inside diameters and roughness ratios, just to name a few. If the user wishes to change a single value of any of these values they must ensure that they change the appropriate tab or it will not register.

The program will then calculate the pressure drop of the system with all heat pumps in operation starting with 2 L/s (31.7 GPM) more than the design flow rate. If the pressure loss in the heat pump section is greater than the set point, it will reduce the flow rate until the set point is satisfied. Once this is satisfied, the WTW efficiency will be calculated and the pump energy consumption at that point will be calculated. It will then repeat the process 11 more times, each time shutting flow off to one heat pump and recalculating the system flow rate until the DP set point is satisfied.

The program simulates a two way shut off valve by setting the fitting loss coefficient (K-value) for a heat pump to 10E8. Since the viscosity of the fluid varies with temperature, the head loss through the system also varies with temperature. For small changes in temperature these effects are typically ignored. The maximum and minimum temperatures for the fluid are 35°C and -5°C (95°F and 23°F) respectively. Because of this, the energy consumption is also calculated for temperatures ranging from 0°C to 35°C(95°F and 0°F), in increments of 5°C (9°F). Once the pump energy consumption has been calculated from all heat pumps on to one heat pump on, the spreadsheet calculates the six coefficients used in Equation 3.2.1, where the *FFL* is assumed to be equivalent to the fraction of the total number of heat pumps in operation at that moment, and temperature. If the DP sensor is set too high, this will be greater than 100%, representing flow rates in excess of the design flow rates.

The distributed pump energy tab is similar to the central pump energy tab with the exception of not requiring a DP set point. Each circulator pump is assumed to consume the same amount of energy. The consumption of one pump is calculated, then multiplied by the number of heat pumps in operation. The pump model used takes the input of the theoretical power required and returns the efficiency of the pump. The efficiency values for this are from actual manufacturers data, this pump would need to be selected for each scenario. This is required to be an actual pump for this scenario to appropriately account for the increase or decrease in flow rate through each heat pump based on the number of heat pumps operating at any given time since the system pressure loss is highly dependent on the number of heat pumps in operation.

The distributed with hydraulic separation tab currently calculates the circulator power in the same way as the distributed pump energy model. This configuration does not include the pressure drop of the boreholes in the calculations, rather only the building pressure drop. Since this part of the program does not include the actual loads of the system, the pump operation of the ground loop pump must be added to the simulation separately from the circulators. There would be too many combinations of flow rates for the primary pump with each number of heat pumps to have in one curve. Instead, two sets of coefficients need to be calculated for use in Equation 3.2.1. One that calculates the pumping power for the individual heat pump circulators based on the number of heat pumps in operation and one that calculates the pumping power for the primary pump.

The one-pipe model does the same thing as the distributed with hydraulic separation tab. The pressure drop used for each pump remains constant regardless of how many pumps are in operation. The energy consumption for one pump operating is calculated then multiplied by how many pumps are in operation. Like the distributed with hydraulic separation tab, the primary pump circulating fluid through the building and the ground loop is calculated separately. The set point for this option is based on the temperature differential across the building loop.

4.4 ANNUAL PUMPING ENERGY CONSUMPTION

In the previous step curves were calculated such that energy consumption of the pumps could be calculated knowing only the percentage of the total design flow rate. Once these are calculated a sub routine called "run_time" is called. This routine reads the run time fractions of each heat pump for each hour and the heating and cooling loads for each heat pump. This routine is broken up into two parts, calculating the number of heat pumps running at a given time and implementing a temperature differential sensor.

The first part of the routine is designed to take a runtime fraction for each hour and determine at what time the heat pump turns on and what time it turns off. To do this each hour was broken up into 12 cycles. This was based on assuming that the heat pumps would have a minimum run time to prevent short cycling and in this case that was assumed to be 5 minutes. The runtime was then converted into the total number of cycles that the heat pump ran during the hour based on Table 4.3.1. If the runtime fraction was zero, then it would obviously have run zero cycles. In this table, the RTF is a the maximum percentage for each number of cycles. This means that if the heat pump had a RTF of slightly higher than 1/12, then it would run for two cycles.

Table 4.3.1 Run Time Fractions

RTF	Cycles	RTF	Cycles
1/12	1	7/12	7
2/12	2	8/12	8
3/12	3	9/12	9
4/12	4	10/12	10
5/12	5	11/12	11
6/12	6	> 1	12

To calculate the start time for each heat pump the number of eligible start times are calculated by taking the number of cycles run that hour subtracted from the total number of cycles. A heat pump that ran for four cycles could then start running at any time before 12-4 = 8th cycle. This is so that no matter what time it starts, it will complete the total run time within that hour. The actual start time is determined by a random number generator. The heat pump is assumed to run for the entire number of cycles in one consecutive run, rather than splitting the run times into multiple cycles within the hour, then shut off.

This is done for each of the 12 heat pumps in this design. The total number of heat pumps running is calculated for each cycle, and then set as a fraction to calculate the Fraction of Full Load (FFL). This would mean if five heat pumps were running at any given time, then the pump would need to provide 5/12th of the total flow rate. Using the curve fit equation developed previously the pump energy consumption for that fraction of the hour could be calculated. This was summed up for each cycle to provide hourly energy consumption and again for the total 8760 hours in the year for the annual pumping energy consumption. This is also used in conjunction with the average hourly temperature of the circulating fluid.

4.5 ANNUAL HEAT PUMP ENERGY CONSUMPTION

The heat pump energy consumption is heavily dependent on the temperature of the entering water. A model was developed by Xu and Spitler (2006) that incorporated the long time-step borehole response model developed by Eskilson (1987) and extended it to short time-steps using a one dimensional numerical model. This model accounts for the thermal mass of the circulating fluid, convective resistances of the fluid, and fluid temperatures. Xu and Spitler created a ground loop heat exchanger component model in HVACSIM+. Lu Xing developed a user interface to be able to utilize this heat exchanger component model with a spreadsheet. The model is called as a sub routine in VBA named “MainDrive”. The explicit details on how this model works are attached in an appendix. From this model the temperature of the fluid entering the heat pump was found as an average temperature for the hour. The energy consumption of the heat pump was then calculated using Equation 2.2.2.2, taking the zone hourly heating or cooling load and dividing it by $COP_{heating}$, or $COP_{cooling}$. In the case where a heat pump performed both heating and cooling, the consumption was computed individually for heating and cooling then added together.

CHAPTER V

RESULTS

The building that was simulated for this research was the strip mall reference building in EnergyPlus V 7.2. The building loads were calculated based on weather data sets from Chicago, Illinois; Tampa, Florida; Tulsa, Oklahoma; and Sterling, Virginia. The heating and cooling sizing criteria used the 99% and 1% dry bulb temperature criteria. Hourly loads for the building in each location are shown in Figure 5.1, where positive values are during heating and negative values are during cooling. Chicago and Sterling are more heavily heating dominant, whereas Tulsa and Tampa are more cooling dominant.

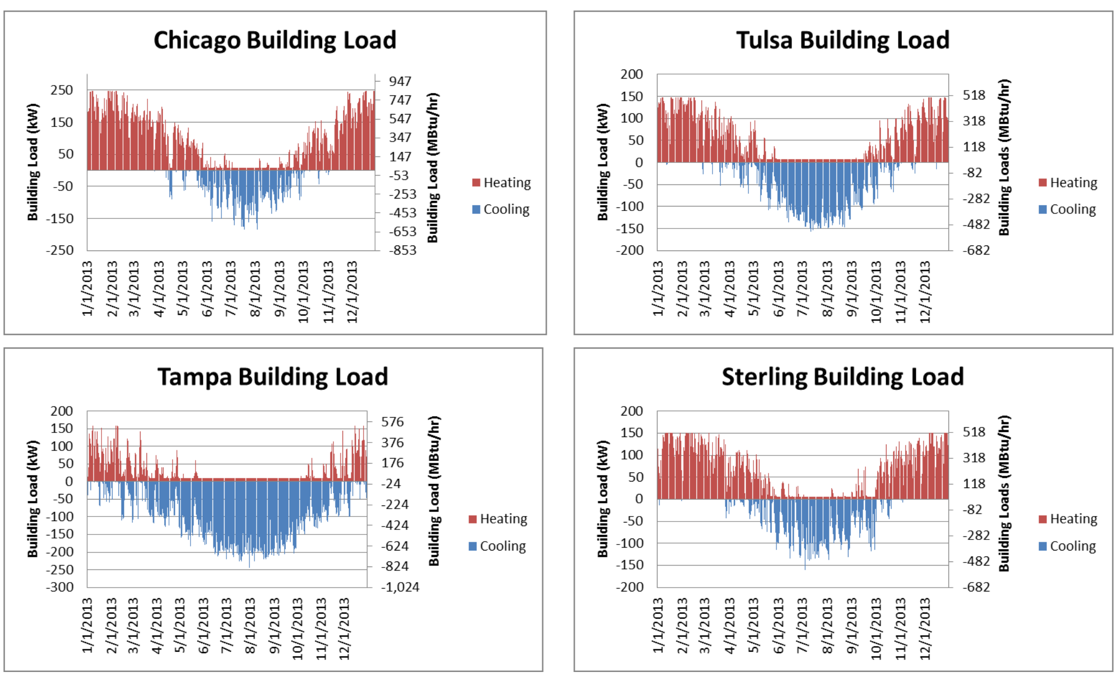

Figure 5.1 Building Loads

The borefield was sized using GLHEPro V4.0. Boreholes had 6.09 m (20 ft) spacing, 110 mm (4.3 in) diameter and nominal borehole pipe diameter was 19 mm (3/4 in) for pipes sized with greater than 0.61 m (2 ft) of head loss per 30.5 m (100 ft) of pipe, and 25.4 mm (1 in) for pipes sized with 0.61 m (2 ft) of head loss per 30.5 m (100 ft) of pipe or less. Since the true ground characteristics could not truly be known without performing a thermal response test at each location, the ground properties were assumed to have the same make up at all locations. This was labeled "Average Rock" in GLHEPro V4.0, where the characteristics are shown in Table 5.1.

Table 5.1 Ground Characteristics of Average Rock

Conductivity W/(m*K) (Btu/(hr*ft^2*R)	**Density** Kg/m^3 (lbm/ft^3)	**Specific Heat** kJ/(Kg*K) (Btu/lbm*F)
2.42 (1.40)	2,803 (175.0)	0.836 (0.0002)

The loads and heat pump selection methodology are described in section 2.1. The heat pump selected for this simulation was the ClimateMaster Tranquility series 060 model. Per the manufacturer's data, the heat pump can operate with flow rates ranging from 0.473 L/s (7.5 GPM) to 0.947 L/s (15 GPM).

5.1 BOREFIELD SIZING RESULTS

Simulations were run for the four locations described in section 5.0. Table 5.1.1 summarizes the borefield sizing results.

Table 5.1.1 Sizing Summary

Location	# of boreholes	Depth m (ft)	Installed Heat Pump Capacity kW (ton)	ft/ton
Chicago	49	138 (453)	246.2 (70)	317
Tampa	49	190 (623)	246.2 (70)	436
Sterling	49	69 (225)	246.2 (70)	158
Tulsa	49	89 (292)	246.2 (70)	204

This table shows that the required feet of borefield per ton of installed capacity ranges considerably based on the location of the building. As noted by Kavanaugh and Kavanaugh (2012b), buildings that have approximately 17.3 m/kW (200 ft/ton) or more are likely to have an ENERGY STAR rating above 90. But as the authors state, the data compared in this article is for buildings where the cooling load determined the borefield size, hence the reason for using the cooling capacity of the heat pumps per foot of installed borehole length rather than heating capacity. For heating dominated locations, the size of the borefield is not driven by the cooling load and would have a larger ft/ton value. The Chicago location was 159% larger than this recommendation, which can be explained by the overwhelmingly heating dominated load profile. Realistically this would be augmented with a supplementary heating source for peak days to reduce the borefield length and could put this within the 17.3 m/kW (200 ft/ton) recommendation, but hybrid systems are beyond the scope of this research. The cooling dominated locations of Tampa and Tulsa along with the other building in Sterling were within the range of surveyed borefields reported in the Kavanaugh and Kavanauagh (2012b) study.

5.2 PIPE SIZING RESULTS

To compare the pumping power requirements for each pipe sizing criteria, the grading scheme shown in Table 1.3.5.1, was used. The grading scheme compares installed pump capacity to the installed cooling capacity W/Ton (HP/100 Tons). The installed pumping power was estimated using the Sfeir et al. (2005) equations. This gave a central pump WTW efficiency of approximately 70% at full flow for all scenarios. This was used in this grade comparison for the central pump in both the two-pipe configuration and the one-pipe and a circulator pump efficiency of 50% was used. Similar to the borefield comparison criteria used in section 5.1, with most locations being sized based on the heating load rather than the cooling load, the values were skewed compared to the original author's comparisons.

The pipe sizing criteria was varied from the standard published recommended pipe sizing criteria of 1.21 m/s (4 ft/s) for pipes 50 mm (2 in) and smaller and a head loss of 1.2 m (4 ft) per 30.5 m (100 ft) of pipe for larger pipes in addition to varying straight head loss allowable per length of pipe from 0.3 m (1 ft) to 1.22 m (4 ft) of head loss per 30.5 m (100 ft).

In ASHRAE research project 1217 TRP, Kavanaugh and Lambert (2004) created the GHPCost piping cost estimator which is available for free download at www.geokiss.com. This program was used to do a rough estimate of what the cost of piping alone would be for each sizing criteria. Since all of the two-pipe configurations are piped roughly the same, the only comparison is that between the two-pipe and the one-pipe piping cost and this has been done only for Chicago. This is not an all-inclusive comparison. There are labor and fitting costs in addition to other installation costs that will differ between the two piping configurations that are unaccounted for. Table 5.2.1 shows the comparison for the two-pipe configuration and the one-pipe configuration costs to include the cost of piping for the building loop, headers, boreholes, and for grouting the boreholes.

Table 5.2.1 Pipe Cost Comparison

	Piping and Grout	
	One-Pipe	**Two-Pipe**
(4 fps<2"; 4' head loss/100'>2")	$321,162	$322,639
4' head loss/100'	$321,219	$322,636
3' head loss/100'	$321,371	$323,182
2' head loss/100'	$318,513	$320,708
1' head loss/100'	$318,530	$321,291

As one would expect, as the head loss in the system increases, the cost of the pipe decreases, or alternatively, as the pipe size increases, the cost increases. When decreasing the allowable head loss from 0.934 m (3 ft) per 30.5 m (100 ft) to 0.623 m (2 ft) per 30.5 m (100 ft), the cost goes against the general trend and decreases. This is caused by a decrease in the overall cost of the borehole as the borehole transitions from a 26.7 mm (3/4 in) pipe to a 33.4 mm (1 in) pipe at that point. The larger diameter pipe decreases the cost of grouting the borehole and results in a less expensive borehole if the possibility that the borehole size may be changed is neglected. This is potentially not significant as the borehole pipes are typically sized separately as a part of sizing the borefield and may be sized by different criteria. From purely a pipe sizing methodology standpoint, using 0.623 m (2 ft) per 30.5 m (100 ft) to size pipes is the most economical criteria.

5.2.1 PUMP SIZING RESULTS

Table 5.2.1.1 shows the results of the total theoretical pumping power required for the traditional two-pipe system with central pumping and the one-pipe system for the Chicago location. The flow rate that was required for sizing was determined by ensuring the heat pump met the design loads as discussed in section 2.1, in this case was 0.947 L/s (15 gpm) with an associated head loss of 7.5 m (24 ft) across the heat pump. The theoretical pumping power was then converted into actual pumping power to be analyzed for a grade.

Table 5.2.1.1 Pumping Grades

Sizing Criteria	**Two-pipe Theoretical Pumping Power** kW (HP)	**One-pipe Theoretical Pumping Power** kW (HP)	**Two-pipe Installed W/ton (HP /100 ton)**	**One-pipe Installed W/ton (HP /100 ton)**	**Two-pipe Grade**	**One-pipe Grade**
4 ft/s – 2” and smaller and 4 ft/100 ft – greater than 2”	2.73 (3.67)	2.75 (3.70)	56 (7.5)	49 (6.6)	B	B
4 ft/100 ft	2.71 (3.63)	2.73 (3.66)	55 (7.4)	48 (6.5)	B	B
3 ft/100 ft	2.60 (3.48)	2.67 (3.58)	53 (7.1)	48 (6.4)	B	B
2 ft/100 ft	1.62 (2.17)	1.54 (2.07)	33 (4.4)	30 (4.0)	A	A
1 ft/100 ft	1.45 (1.94)	1.48 (1.98)	30 (4.0)	28 (3.8)	A	A

For the traditional pipe sizing criteria the system would get a pumping grade of a ‘B’. This was also the grade when 1.2 m (4 ft) per 30.5 m (100 ft) of pipe and 0.914 m (3 ft) per 30.5 m (100 ft) of pipe. When 0.6 m (2 ft) per 30.5 m (100 ft) of pipe the total pumping power decreases by 38% and the grade transitions from a ‘B’ to an ‘A’. Coincidently this transition happened for both the one-pipe and two-pipe configurations at this point.

For each location, the pipe layout for the building remained the same resulting in no change in building interior pipe lengths. The size of the borefield however varied for each respective site resulting in the pumps needing to be sized for each location. Using the above conditions for pipe sizing, similar results were obtained for each location, that being a grade transition point when 0.6 m (2 ft) per 30.5 m (100 ft) of pipe was used as the pipe sizing criteria. Using more conservative values for pipe sizing such as 0.3 m (1 ft) per 30.5 m (100 ft) of pipe resulted in only marginal reductions in pumping power compared to the reduction realized going from 0.914 m (3 ft) to 0.6 m (2 ft) per 30.5 m (100 ft) of pipe. 0.6 m (2 ft) per 30.5 m (100 ft) of pipe was used to size the pipes for the remainder of this research unless specified.

5.3 CHICAGO ENERGY CONSUMPTION RESULTS

The pumping energy consumption and heat pump energy consumption was calculated for multiple design and operating conditions for the Chicago location. The first configuration that was analyzed was the one-pipe configuration. Since the pipe sizing criteria of 0.6 m (2 ft) per 30.5 m (100 ft) of pipe was where the design first achieved an ‘A’ rating, this criteria was used for the analysis. The design for the one-pipe system called for a primary circulation pump sized for 11.34 L/s (180 GPM) with a design head loss of 6.5 m (20.9 ft). This resulted in a central pump with a theoretical pumping power of 0.71 kW (0.95 HP) and 12 circulators with a theoretical pumping power of 0.066 kW (0.088 HP) each. The calculated central pump efficiency at full flow was 67% and the circulators were assumed to be at the maximum available circulator efficiency of 50%. To optimize the performance of the one-pipe configuration, the set point for the temperature differential (ΔT) sensor was varied to find the set point with the optimal system performance. The ΔT set the flow rate based on the total building load such that across the building the temperature of the fluid would increase by the value of the set point. A summary of the results is displayed in Figure 5.3.1.

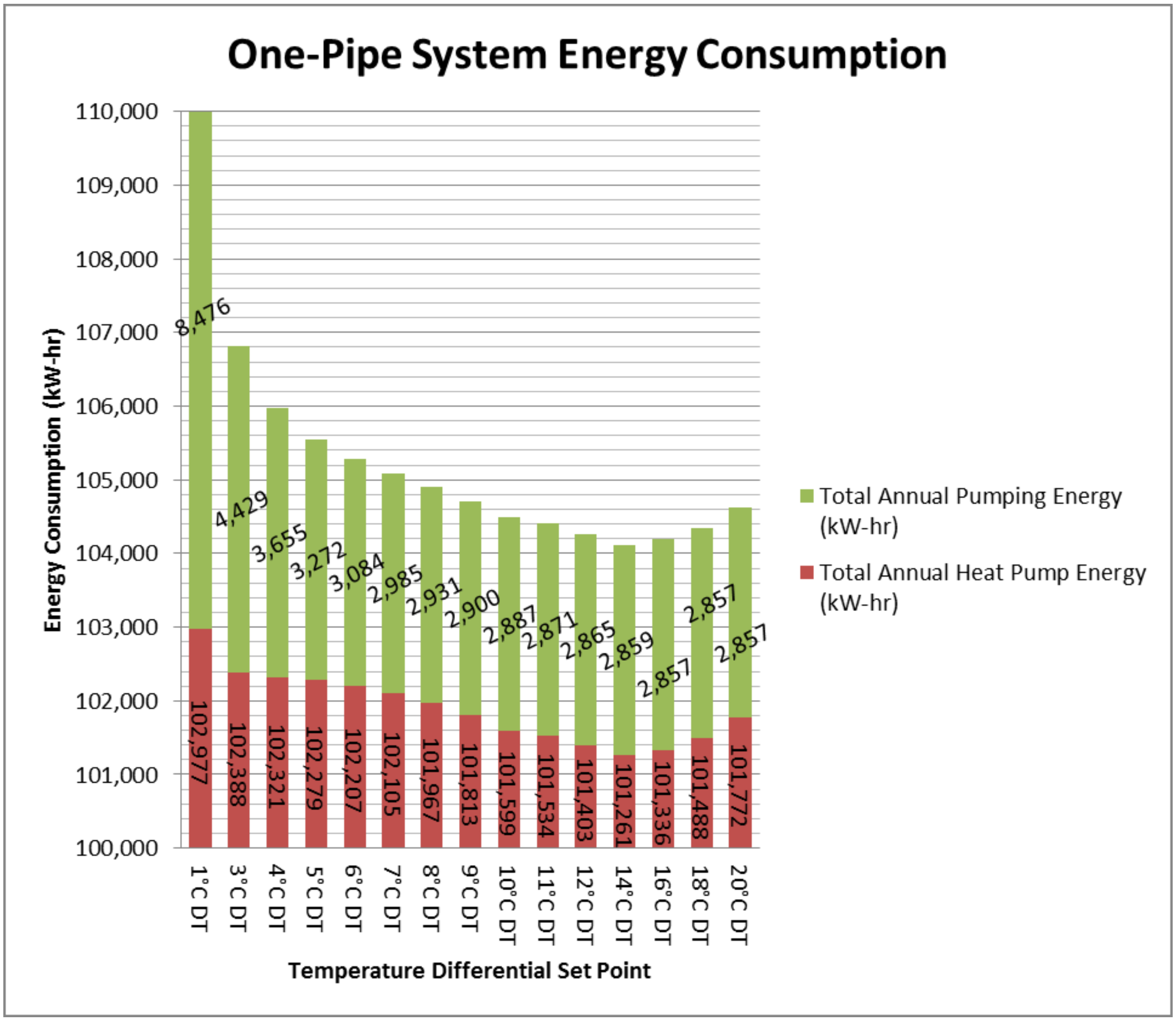

Figure 5.3.1 Chicago One-Pipe System Energy Consumption

From these simulations, the ΔT set point with the best overall system performance was 14°C (25.2°F) with a total annual system energy consumption of 104,120 kW-hr. This set point was much higher than what was expected. It was expected that as the ΔT across the building was increased, the heat pump performance would steadily decrease. This ended up being not true to a certain extent. With a ΔT set point of 1°C (1.8°F), the primary circulation pump was running at full speed almost all of the time it was running which was about 11.34 L/s (180 GPM). Because of this, the pump should not have had a VSD, rather could have been a constant speed pump. As the ΔT set point was increased, the pump was allowed to operate at lower speeds and the returning temperature from the borefield was higher in heating and cooler in cooling. This can be seen in Figure 5.3.2 for the case a peak heating scenario and in Figure 5.3.3 for the peak cooling scenario.

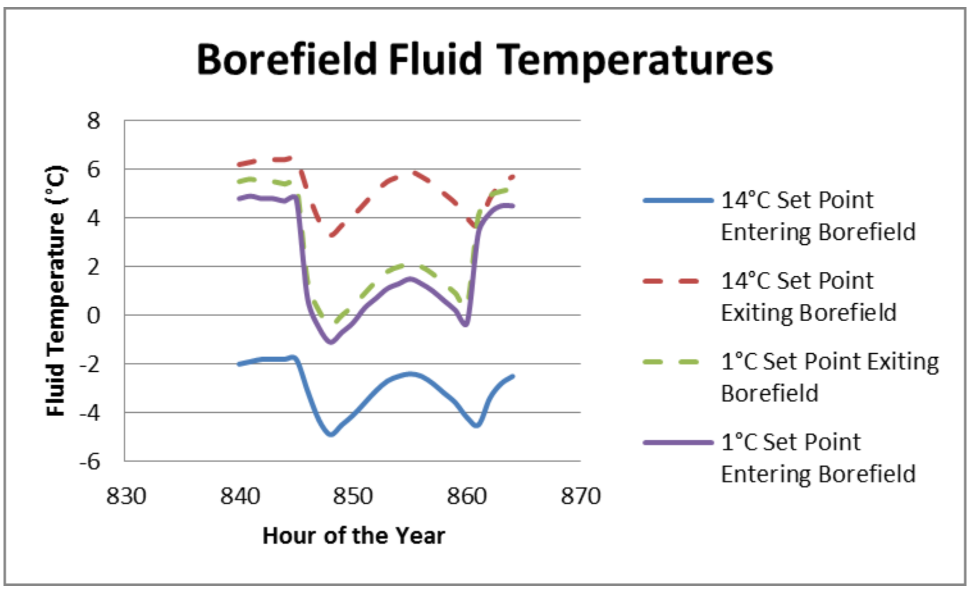

Figure 5.3.2 Chicago One-Pipe Peak Heating GHX Inlet and Outlet Temperatures

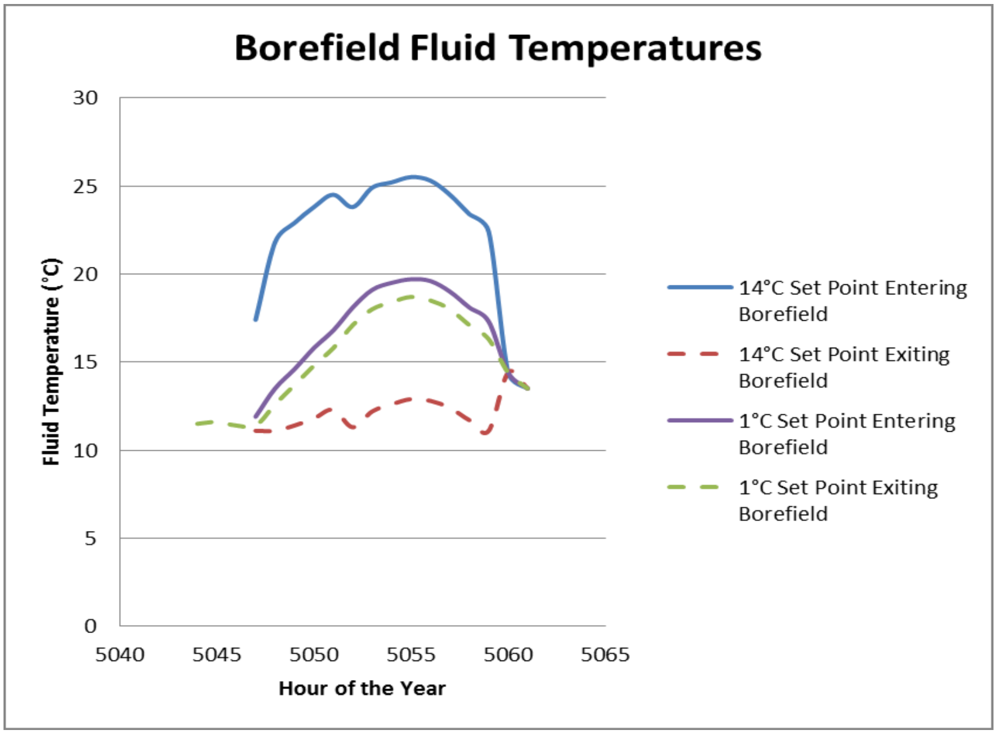

Figure 5.3.3 Chicago One-Pipe Peak Cooling GHX Inlet and Outlet Temperatures

In this figure the two solid lines represent the temperature of the fluid entering the ground heat exchanger. The dashed lines represent the fluid temperatures leaving the ground heat exchanger. As one would expect, the higher ΔT set point has a lower entering temperature. What was not expected was that the heat transfer rate of the ground increased with slower flow rates resulting in higher exiting fluid temperatures from the borefield. A set point of 14°C (25.2°F) did not result in a 14°C (25.2°F) difference in the temperature leaving the borefield to the temperature entering the borefield. This was due to the minimum flow rate of the pump being set at 30% of full flow. The flow rate required for the set point was below this value so the pump could not operate at the appropriate speed. Since a majority of the time, the building heating load is not at peak conditions the actual flow rate required to match the 14°C (25.2°F) was lower than the minimum flow rate of the primary pump. This resulted in a temperature difference lower than the set point. In Figure 5.3.2, the peak heat transfer rate from the fluid to the ground heat exchanger was 175.3kW for the 14°C (25.2°F) set point, and 174.2 kW for the 1°C (1.8°F) set point. As a result of this increased heat transfer to the ground heat exchanger and the fluid returning to the building at a higher temperature in heating, lower temperature in cooling, the COP was higher in the first couple of heat pumps in the building and lower in the last heat pumps in the building. This is demonstrated for the same time period as in Figure 5.3.2, in Figure 5.3.4. The average COP for heating of the first heat pump was 3.5 where the average COP of the last heat pump was 3.36. This compared rather well to the manufacturers rating of 3.1 for heating. The increase in COP was significant enough to decrease the overall heat pump energy consumption despite a lower COP in the last heat pump in the building compared to a more conservative ΔT. This effect was not prevalent in cooling. The temperature of the fluid was over 5°C (9°F) higher coming out of the building for the 14°C (25.2°F) set point.

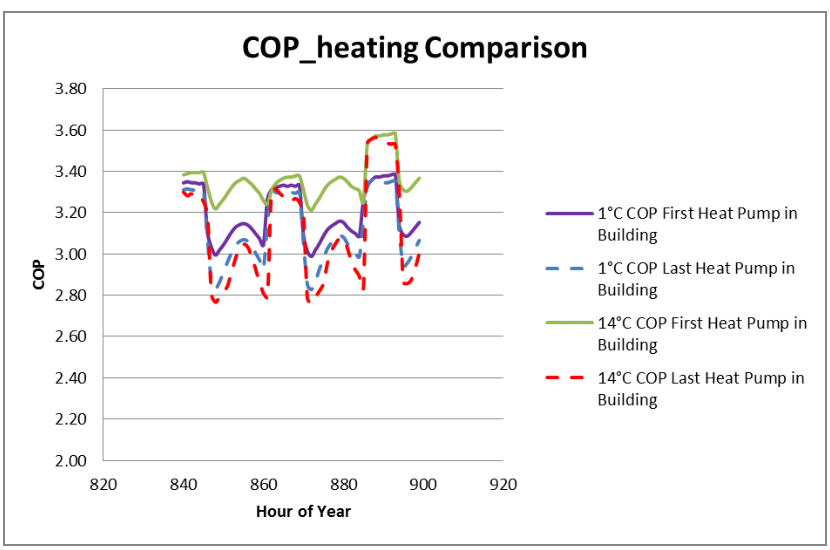

Figure 5.3.4 Chicago One-Pipe Heating COP Comparison for First and Last Heat Pumps

The performance of the two scenarios was similar in cooling as in heating. For the 14°C (25.2°F) simulation, the COP for the first heat pump was much higher than the COP of the last heat pump. The other significant part is the performance of the first heat pump was only slightly impacted at the peak cooling condition. This was heavily influenced by the 10°C (50°F) undisturbed ground temperature and that this is a heating dominated climate. The average COP for cooling of the first and last heat pumps were 5.97 and 5.52 respectively. The manufacturer specified a COP of 4.3 for cooling.

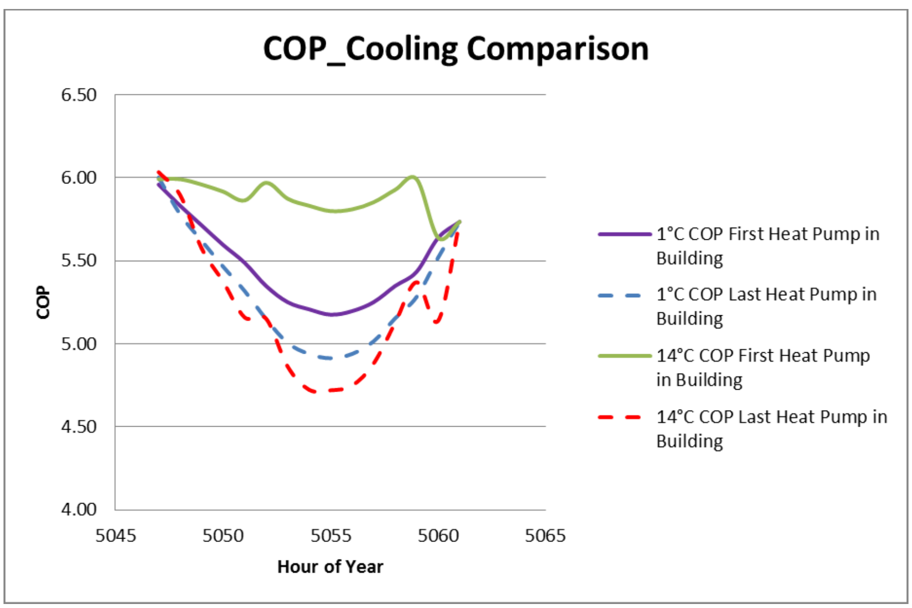

Figure 5.3.5 Chicago One-Pipe COP Cooling Comparison for First and Last Heat Pumps

As expected the pumping energy consumption decreased consistently as the temperature differential was increased. The individual circulator energy consumption remained the same regardless of the set point; however the primary pump energy consumption increased as much as

15 times the 14°C (25.2°F) value when the set point was at 1°C (1.8°F). The relationship can be seen in Figure 5.3.6.

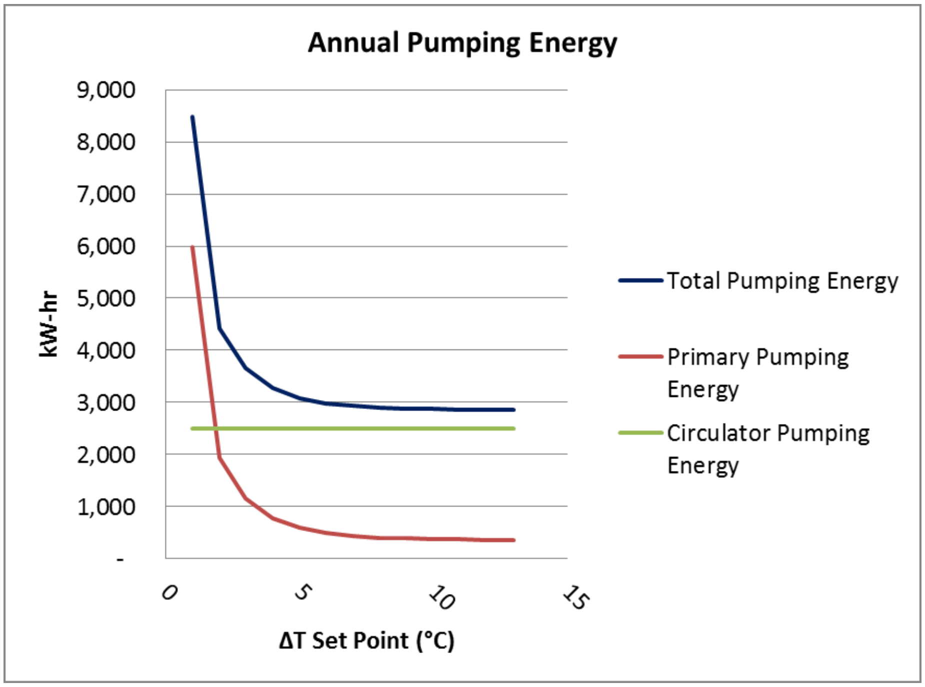

Figure 5.3.6 Chicago One-Pipe Pumping Energy Consumption Analysis

The energy consumption increased rapidly for set points smaller than 5°C (9°F). For set points greater than this, the decrease in energy consumption was minimal. For set points above 14°C (25.2°F), there was no decrease in energy consumption. This was the point at which for all heat pump operating conditions, the pump was operating at the minimum set flow rate of 30% of the design flow rate which was approximately 3.4 L/s (54 gpm). There was a concern if the flow rate dropped too low, that the system would recirculate fluid through the heat pump without going through the borefield. Due to the minimum flow rate being well above the 0.95 L/s (15 gpm) that was required by each individual heat pump, this was assumed to be a non-issue.

The two-pipe configuration was analyzed next. This was analyzed with three separate pumping schemes; central, distributed, and distributed with hydraulic separation. The central pumping scheme was optimized by changing the DP sensor set point from 53.8 kPa (18 ft) to 83.7 kPa (28 ft). This had a theoretical pumping power requirement of 1.62 kW (2.17 HP). The results for this are shown in Figure 5.3.5. Additionally shown are the results for the two-pipe configuration with distributed pumping. This pumping scheme had a primary theoretical pumping power of 0.135 kW (0.181 HP) for each circulator with a total theoretical requirement of 1.62 kW (2.17 HP). Both schemes were designed for a 15 GPM flow rate through each heat pump with 74 kPa (24.5 ft) of head. For the central pumping schem, by changing the DP set point, this varied the COP of the heat pump. As the flow rate through each heat pump increased, the performance of the heat pump increased. This in turn required a greater pumping energy. 65.8 kPa (22 ft) is bolded and circled in Figure 5.3.7 as the optimal set point. The total system energy consumption at this set point was 102,367 kW-hr. This included the heat pump energy consumption of 98,784 kW-hr and a pumping energy consumption of 3,583 kW-hr. The two-pipe configuration with distributed pumping was slightly higher than this with 98,558 kW-hr in heat pump energy consumption, 6,109 kW-hr in pumping, for a total of 104,667 kW-hr. Surprisingly the heat pump energy consumption for the distributed pumping was less than both the one-pipe configuration and the two-pipe configuration with central pumping. The reason for this will be described after the two-pipe configuration with central pumping. The distributed circulator pumping efficiency used the equations from Sfeir et al. (2005) and was approximately 29%. This was changed to 50% in conjunction with the one-pipe configuration to see what the performance would be with the best pumps on the market, and the pumping energy was reduced to 3,475 kW-hr. This brought the total system energy consumption to 102,033 kW-hr which was lower than the central pumping scheme.

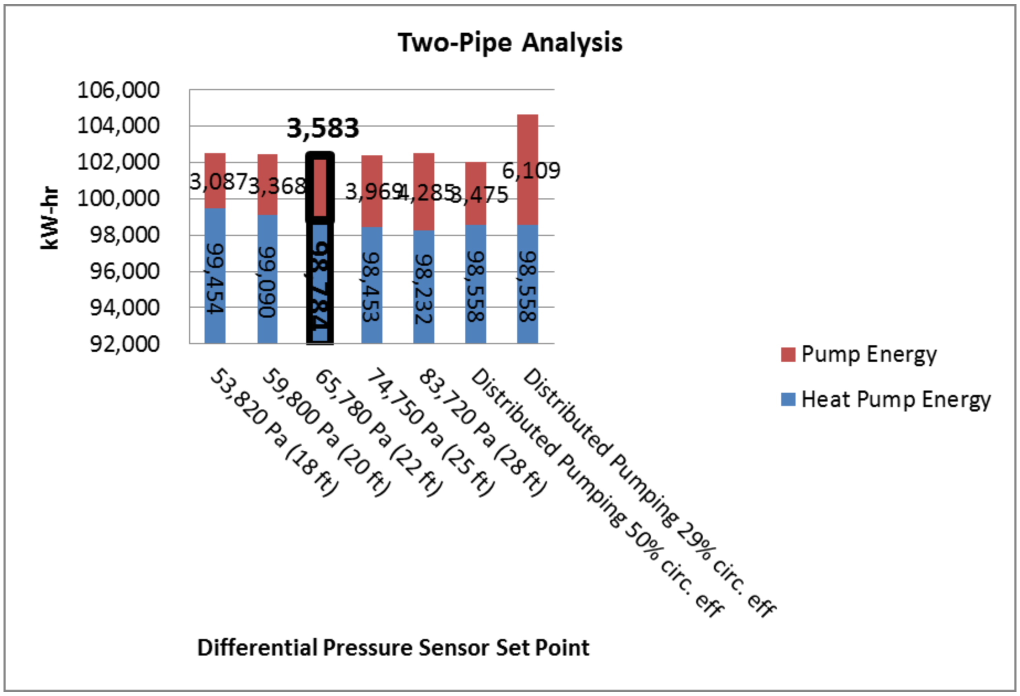

Figure 5.3.7 Chicago Two-Pipe with Central Pumping and Distributed System Energy

The distributed pumping with hydraulic separation scheme was designed with the same operating conditions as above, with 0.947 L/s (15 GPM) flow rate through each heat pump with 74 kPa (24.5 ft) of head. This had a theoretical pumping requirement of 0.5 kW (0.68 HP) for the central primary pump and 0.093 kW (0.125 HP) for each individual circulator for a total theoretical pumping power of 1.62 kW (2.17 HP). This was optimized by changing the speed of the primary pump based on the load in the building. The set point was designed to be the temperature rise of the fluid across the building. There was a DT sensor that could be adjusted to pump fluid to the ground loop based on the temperature rise across the building. There was not a clear optimum found for this scheme, but the decrease in energy consumption stopped at 15°C (27°F). For DT set points above 7°C (12.6°F) there was minimal savings as the primary pump was almost always operating at the minimum flow rate. Any increase in the performance from the heat pump was solely due to the improved performance of the GHX. A summary of the results is shown in Figure 5.3.8.

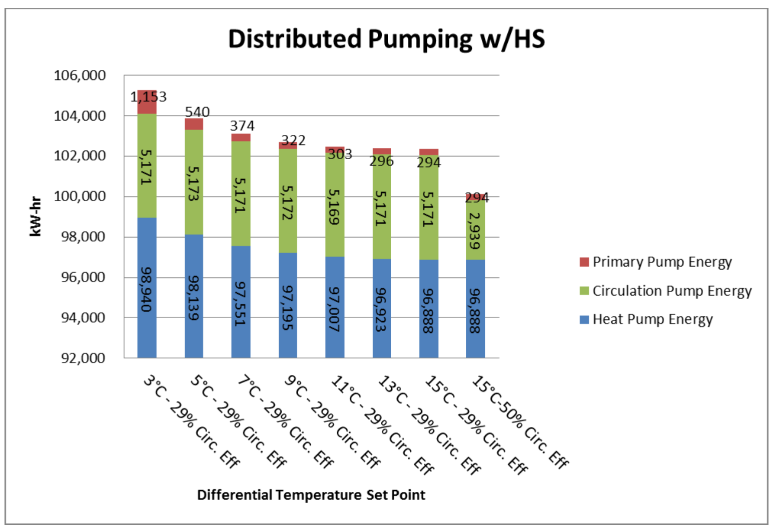

Figure 5.3.8 Chicago Two-Pipe Distributed w/HS System Energy

The circulator pumping efficiency used the equations from Sfeir et al. (2005) and was approximately 29%, same as with distributed pumping. This was also changed to 50% to see what the performance would be with the best pumps on the market verses the pump that was selected, and the circulator pumping energy was reduced from 5,171 kW-hr to 2,939 kW-hr with a total system energy consumption of 100,121 kW-hr.

Since the heat pumps were configured in parallel for all two-pipe configurations, they all saw the same temperature of fluid and were assumed to have equal flow rates. For central pumping, the optimal solution had a flow rate of 0.88 L/s (14 GPM) for each heat pump at design conditions and a total system flow rate of 10.6 L/s (168 GPM). Unlike the one-pipe system which varied the primary pump flow rate by 9.71 L/s (154 GPM) from 11.4 L/s (180 GPM) down to 3.4 L/s (54 GPM), the total system flow rate through the primary pump only varied by 0.76 L/s (12 GPM). As a result there was negligible change in the fluid temperature exiting the borefield or the heat transfer to the borefield. Temperatures entering the ground loop were slightly lower than that of the 1°C (1.8°F) set point performance for the one-pipe configuration yet slightly higher than the 14°C (25.2°F) set point. For distributed pumping the circulator flow rate was 0.836 L/s (13.25 GPM). This resulted in similar temperatures as the central pumping scheme. Both central pumping and distributed pumping temperatures entering and exiting the borefield are shown if Figure 5.3.9 for the same peak heating time period used in the one-pipe analysis.

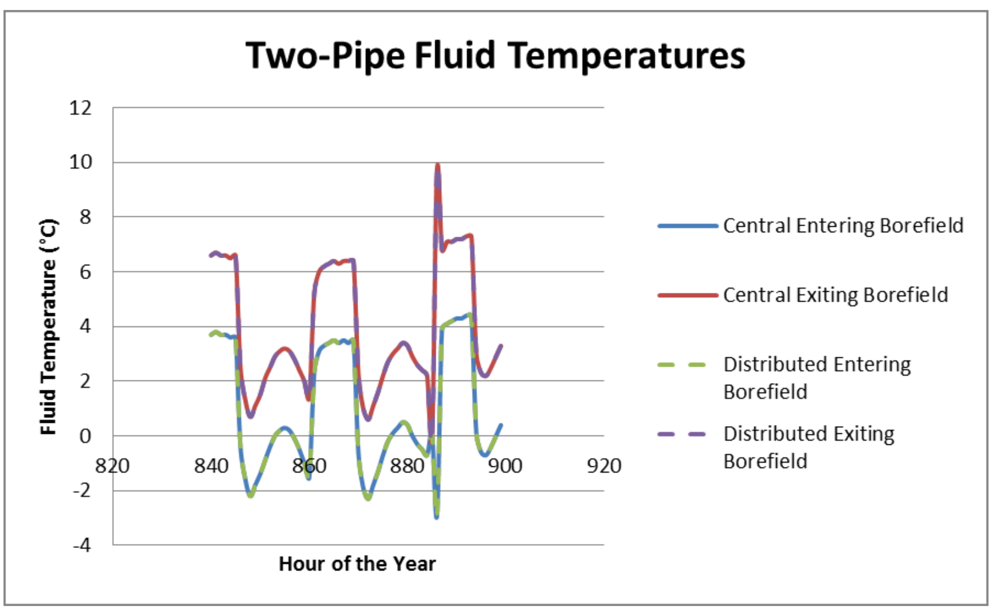

Figure 5.3.9 Chicago Two-Pipe Central/Distributed Peak Heating GHX Temps

With all 12 heat pumps in operation, the flow rate for the distributed pumping scheme was 0.836 L/s (13.25 GPM) which was only 0.047 L/s (0.75 GPM) less than the optimal design of the central pumping scheme. For both the central pumping and distributed, the temperature rise of the fluid across the building during peak heating was approximately 3°C (5.4°F). The sharp spikes in all of the figures showing fluid temperature appear after a short period of no heat pumps in operation as the fluid has had time to dissipate or absorb heat while not operating. In the distributed pumping with hydraulic separation, the entering and exiting temperatures as shown in Figure 5.3.10 were lower than the two other two-pipe pumping schemes, but similarly to the one-pipe configuration, the exiting temperature of the fluid from the borefield was higher (in heating) resulting in higher COP's. In part load operation, the temperature difference of the fluid entering and exiting the borefield was often less than the set point due to the minimum speed of the primary pump not being able to slow down to the required flow rate.

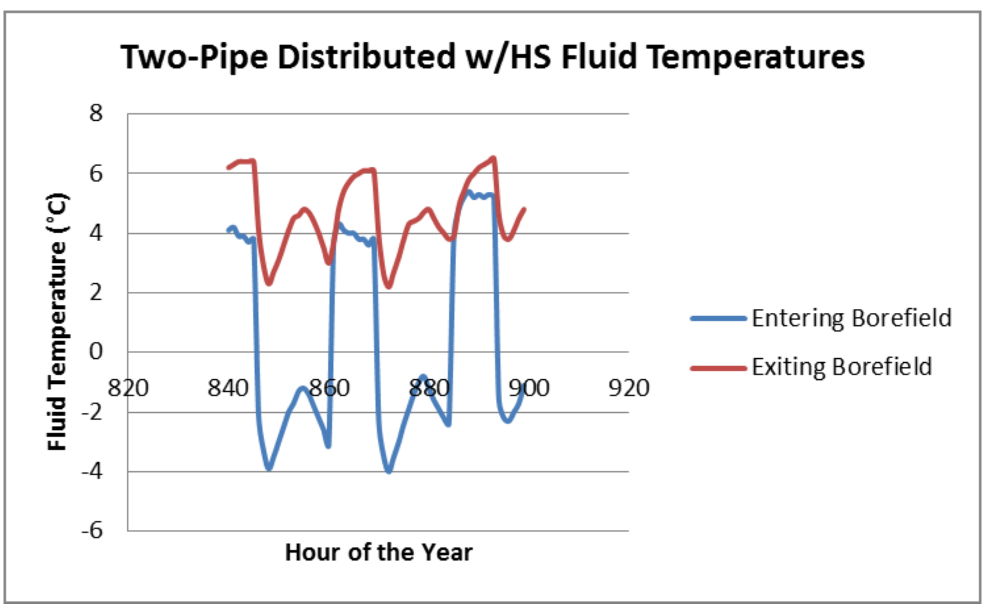

Figure 5.3.10 Chicago Two-Pipe Distributed w/HS Peak Heating GHX Temps

Figures 5.3.11 shows the peak cooling temperatures entering and exiting the borefield for both the central and distributed pumping schemes. The results agreed with the results from heating. The temperatures exiting the borefield were approximately 2 degrees cooler in cooling than the one-pipe configuration with 1°C (1.8°F) DT set point.

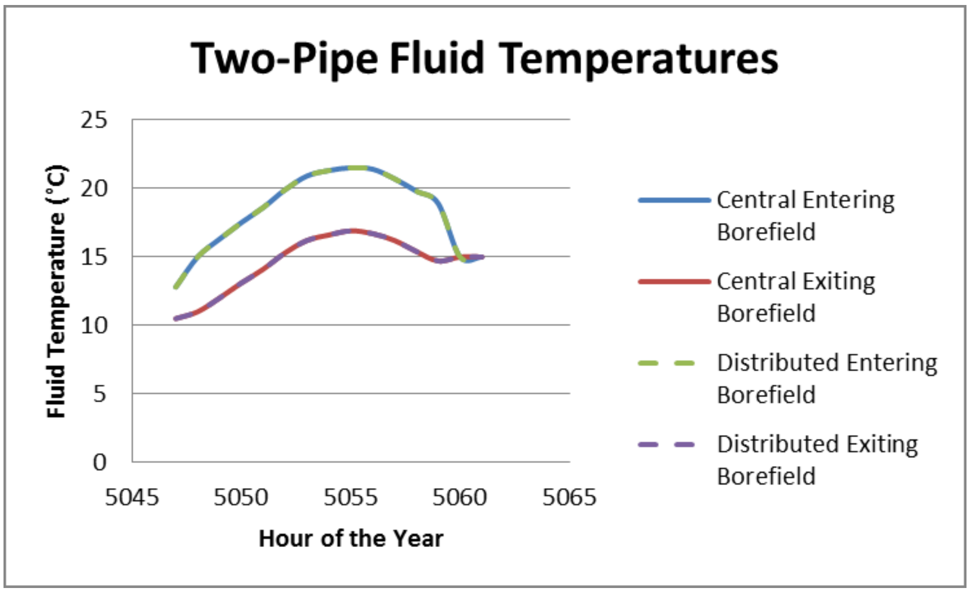

Figure 5.3.11 Chicago Two-Pipe Central Pump Peak Cooling GHX Inlet and Outlet Temperatures

Figure 5.3.12 shows the borefield temperatures for the two-pipe configuration with central pumping with hydraulic separation using the 15°C (27°F) set point. The exiting temperature does not exceed 15°C (59°F). This results in higher COP's than the other two pumping schemes with the two-pipe configuration.

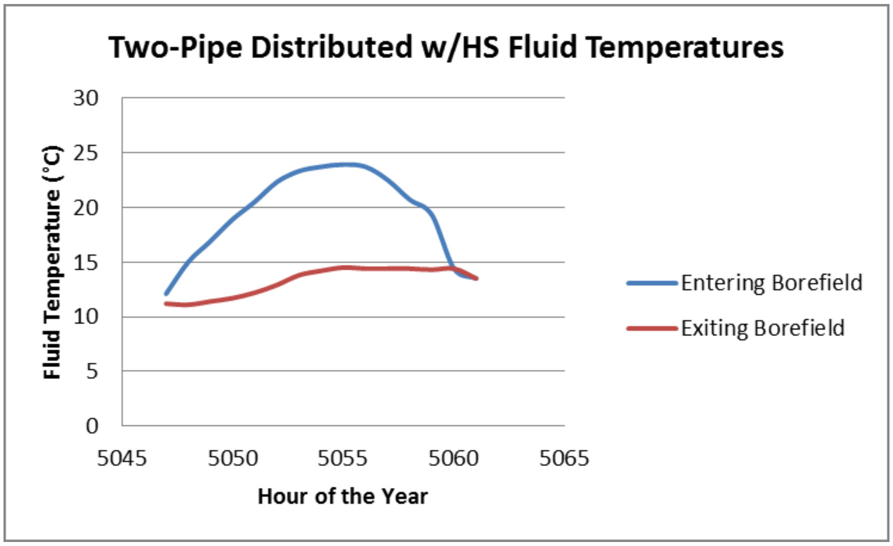

Figure 5.3.12 Chicago Two-Pipe Distributed w/HS Peak Cooling GHX Inlet and Outlet Temps

Comparing the heating COP's seen at peak conditions shown in Figure 5.3.13, the central pumping and distributed pumping schemes were almost identical. The distributed with hydraulic separation pumping scheme had a consistently higher COP than both central and distributed pumping, which were all almost identical. Also shown in this Figure 5.3.13 is the COP for a 83,720 Pa (28 ft) set point. The increased performance from the extra flow to each heat pump was negligible. The average COP for heating with central pumping was 3.44, for distributed pumping it was 3.45, and for distributed pumping with hydraulic separation it was 3.48.

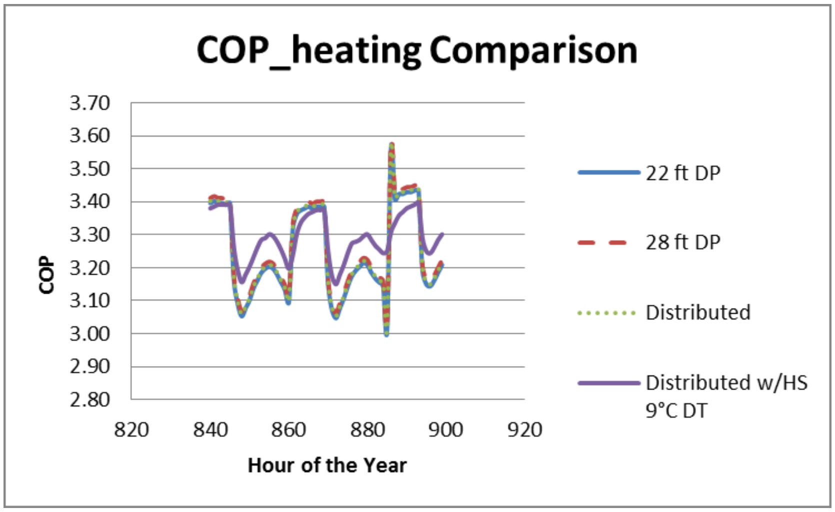

Figure 5.3.13 Chicago Two-Pipe Peak Heating COP Comparison

For cooling the results agreed with the heating results. The distributed with hydraulic separation pumping scheme saw consistently higher COP's than the other two pumping schemes. This is shown in Figure 5.3.14. There were only minor differences between the COP for 83,720 Pa (28 ft) and the 65,780 Pa (22 ft) set points. The average COP for cooling for the central pumping scheme was 5.84, 5.86 for distributed pumping, and 6.38 for distributed pumping with hydraulic separation. The temperature advantage of the distributed pumping with hydraulic separation scheme was significant.

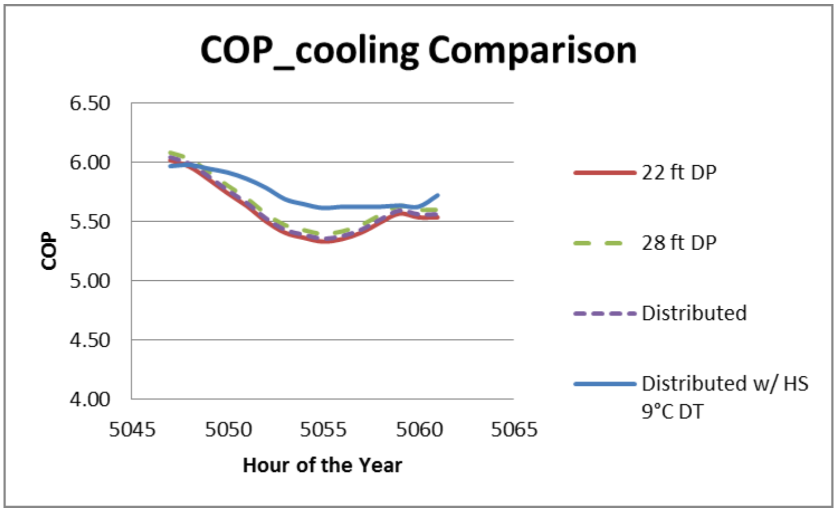

Figure 5.3.14 Chicago Two-Pipe Peak Cooling COP Comparison

A summary of the piping and pumping options is shown in Table 5.3.1. This shows that when using high efficiency circulators, the two-pipe configuration, distributed pumping with hydraulic separation has the lowest annual energy consumption. This was with 50% circulator efficiency for both the distributed pumping and the distributed pumping with hydraulic separation. The decreased flow rate to the GHX resulting in a higher heat transfer rate to the ground allowed the distributed pumping with hydraulic separation scheme to have the most efficient heat pump performance. The one-pipe configuration had the lowest pump energy consumption. The one-pipe configuration total system energy consumption was within 4% of the two-pipe with distributed pumping with hydraulic separation. Therefore, it is likely that the decision as to which system should be used may come down to life cycle cost. If the one-pipe system has sufficient first cost savings, it may be preferable.

Table 5.3.1 Chicago Energy Rollup

Pumping Scheme	Circulator Pump Energy (kW-hr)	Primary Pump Energy (kW-hr)	Heat Pump Energy (kW-hr)	Total System Energy (kW-hr)
One-Pipe	2,494	365	101,261	104,120
Two-Pipe - Central	-	3,583	98,784	102,367
Two-Pipe - Distributed	3,475	-	98,558	102,033
Two-Pipe - Distributed w/ HS	2,939	294	96,888	100,121

5.4 TAMPA ENERGY CONSUMPTION RESULTS

The next location that was analyzed was in Tampa, Florida. The same pipe sizing methodology was used for this location, 0.6 m (2 ft) per 30.5 m (100 ft) of pipe. This again resulted in an 'A' rating for the pumping power in the design. Table 5.4.1 shows the theoretical pumping power for each pumping scheme.

Table 5.4.1 Tampa Theoretical Pumping Power

Pumping Scheme	Theoretical Primary Pump kW (HP)	Theoretical Individual Circulator Pump kW (HP)	Total Theoretical Pumping Power kW (HP)
One-Pipe	0.82 (1.09)	0.07 (0.093)	1.65 (2.22)
Two-Pipe w/ Central Pumping	1.73 (2.32)	-	1.73 (2.32)
Two-Pipe w/ Distributed Pumping	-	0.144 (0.193)	1.73 (2.32)
Two-Pipe w/ Central Pumping with HS	0.61 (0.82)	0.093 (0.125)	1.73 (2.32)

Tampa is a cooling dominated region that has a much higher undisturbed ground temperature than Chicago. The optimal ΔT set point for the one-pipe configuration was at 7°C (12.6°F). This resulted in a heat pump energy consumption of 98,343 kW-hr, with a total pumping energy consumption of 3,279 kW-hr. The results for 1°C (1.8°F) through 13°C (23.4°F) are shown in Figure 5.4.1.

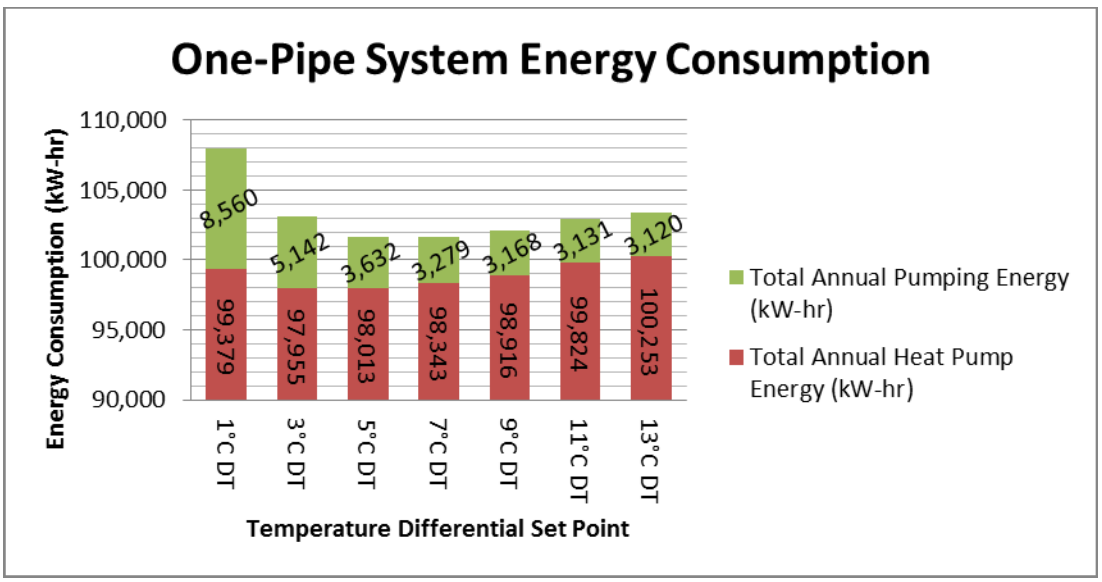

Figure 5.4.1 Tampa One-Pipe System Energy Consumption

As expected, as the ΔT set point increased, the pumping energy decreased for all cases. This also decreased the performance of the heat pump with one exception. For the 1°C (1.8°F) set point the heat pump energy consumption was higher than the 3°C (5.4°F) set point. The reason for this is the higher temperature of the fluid returning from the borefield. For example, Table 5.4.2 shows the average temperatures of the fluid entering the first heat pump and average temperatures leaving the last heat pump for both heating and cooling for both of these set points and the optimal set point of 7°C (12.6°F). In heating, the average fluid temperature of the 1°C (1.8°F) set point does not change much, less than 1°C (1.8°F) across the building. For the 3°C (5.4°F) set point, the water enters 0.25°C (0.45°F) warmer and leaves about 1°C (1.8°F) cooler. This does not have an appreciable impact on the performance. For cooling however, the average temperature entering the first heat pump was about 0.85°C (0.53°F) cooler and leaves the last heat pump 1°C (1.8°F) warmer. Transitioning from the 3°C (5.4°F) set point to the 7°C (12.6°F) set point, the average temperature was 0.05 °C (0.09°F) entering the first heat pump and was higher leaving the last heat pump decreased 2.5°C (4.5°F). In cooling, the average temperature entering the first heat pump decreases as the set point increases while the average temperature leaving the last heat pump increased.

Table 5.4.2 Tampa One-Pipe Average Fluid Temperatures

	°C (°F)	°C (°F)	°C (°F)
Set Point	1 (1.8)	3 (5.4)	7 (12.6)
Heating			
Entering First Heat Pump	24.65 (76.46)	24.91 (76.84)	24.96 (76.93)
Leaving Last Heat Pump	24.02 (75.24)	23 (73.40)	20.5 (68.90)
Cooling			
Entering First Heat Pump	26.25 (79.25)	25.4 (77.72)	24 (75.20)
Leaving Last Heat Pump	27.2 (80.96)	28.1 (82.58)	30.4 (86.72)

The average COP in heating was relatively unaffected by the change in set point based on the range of set points examined in this study. This was the case as most of the heating loads required less than the minimum flow rate of the pump to achieve the temperature difference. The average cooling COP was slightly lower for both the first heat pump and the last heat pump going from the 1°C (1.8°F) set point to the 3°C (5.4°F) as shown in Table 5.4.3. Transitioning to the

7°C (12.6°F) set point, the average COP of the first heat pump increased 0.15 and the average COP of the last heat pump decreased 0.14. As a result of the increased COP from 1°C (1.8°F) set point to the 3°C (5.4°F) the heat pump performed better at 3°C (5.4°F). The decreased COP from 3°C (5.4°F) set point to the 7°C (12.6°F) set point caused the performance of the heat pump to be degraded.

Table 5.4.3 Tampa One-Pipe Seasonal HP COP

Set Point	1°C (1.8°F)	3°C (5.4°F)	7°C (12.6°F)
Heating			
COP First Heat Pump	4.23	4.24	4.24
COP Last Heat Pump	4.21	4.21	4.21
Cooling			
COP First Heat Pump	4.35	4.45	4.60
COP Last Heat Pump	4.16	4.18	4.04

The energy consumption of the pumps is shown in Figure 5.4.2; the circulators were independent of the set point, but the primary pump consumed less energy as the set point increased. Above set points of 4°C (7.2°F) there were minimal pumping energy savings.

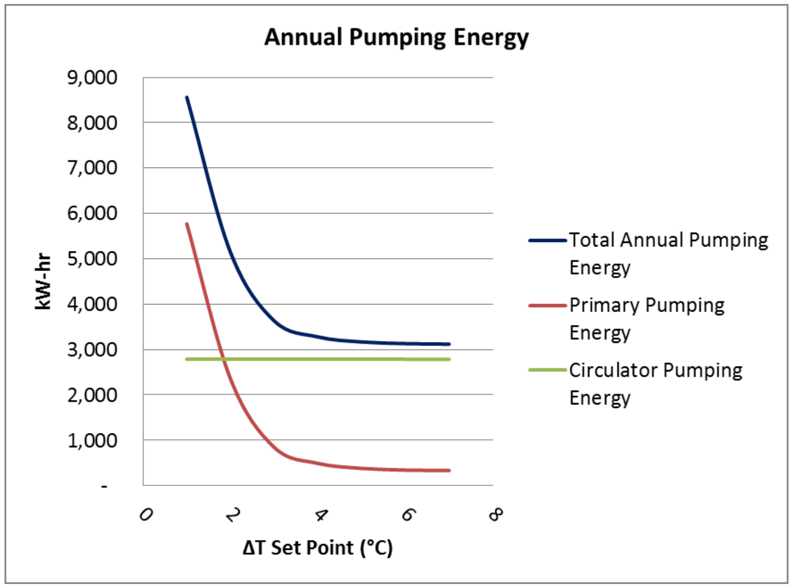

Figure 5.4.2 Tampa One-Pipe Pumping Energy Consumption Analysis

The amount of savings due to the increased ΔT decreased as the flow rate required dropped below the minimum flow rate of the pump. For set points above 7°C (12.6°F) the pump almost always operated at the minimum speed.

During peak heating conditions the temperature change of the fluid across the building stayed below the set point. Figure 5.4.3 shows the temperature of the fluid entering the borefield and exiting the borefield for a peak heating condition. From this plot, the temperatures entering the borefield have a wide range, however the temperature of the fluid exiting the borefield fluctuates less than 2°C (3.6°F). This shows that the borefield is oversized for the heating load and has a much greater capacity to handle a larger heating load.

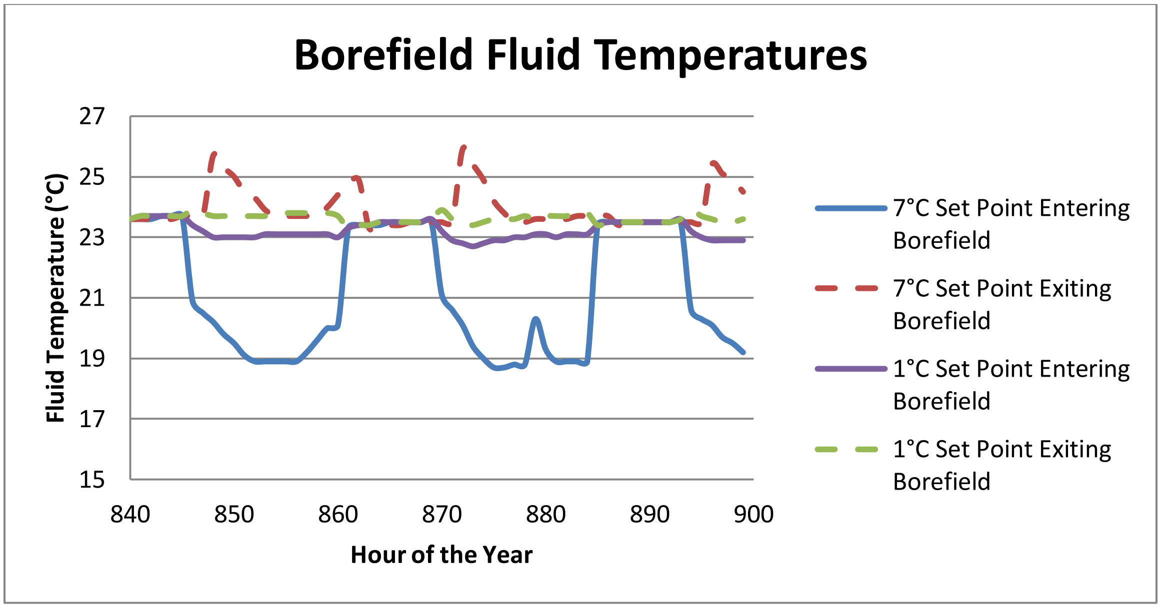

Figure 5.4.3 Tampa One-Pipe Peak Heating GHX Inlet and Outlet Temperatures

In cooling, the temperature change follows the set point rather well. This can be seen in Figure 5.4.4, which shows two days of peak cooling. This means that the pump is operating as designed and can provide sufficient flow to match the temperature rise. The part in Figure 5.4.4 where all of the temperatures are equal is when the pumps are off during the night time set back of the facility. The temperature of the fluid returning from the borefield increases through the day as the borefield heats up and gradually decreases overnight.

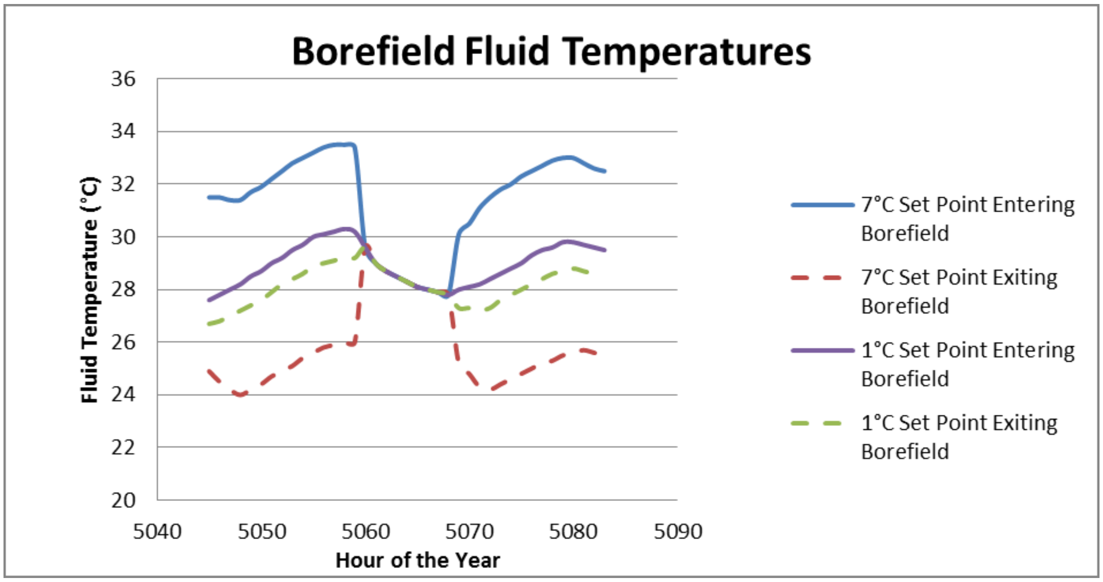

Figure 5.4.4 Tampa One-Pipe Peak Cooling GHX Inlet and Outlet Temperatures

As a result from the mild fluctuations in temperatures in the ground loop during heating, the COPs varied less than (0.12). The COPs appear to spike high and low during operation which can be explained by initial temperatures leaving the borefield in the morning when the heat pumps turn on recovering from the night time set back. This initial load will be larger than most of the heating load through the day.

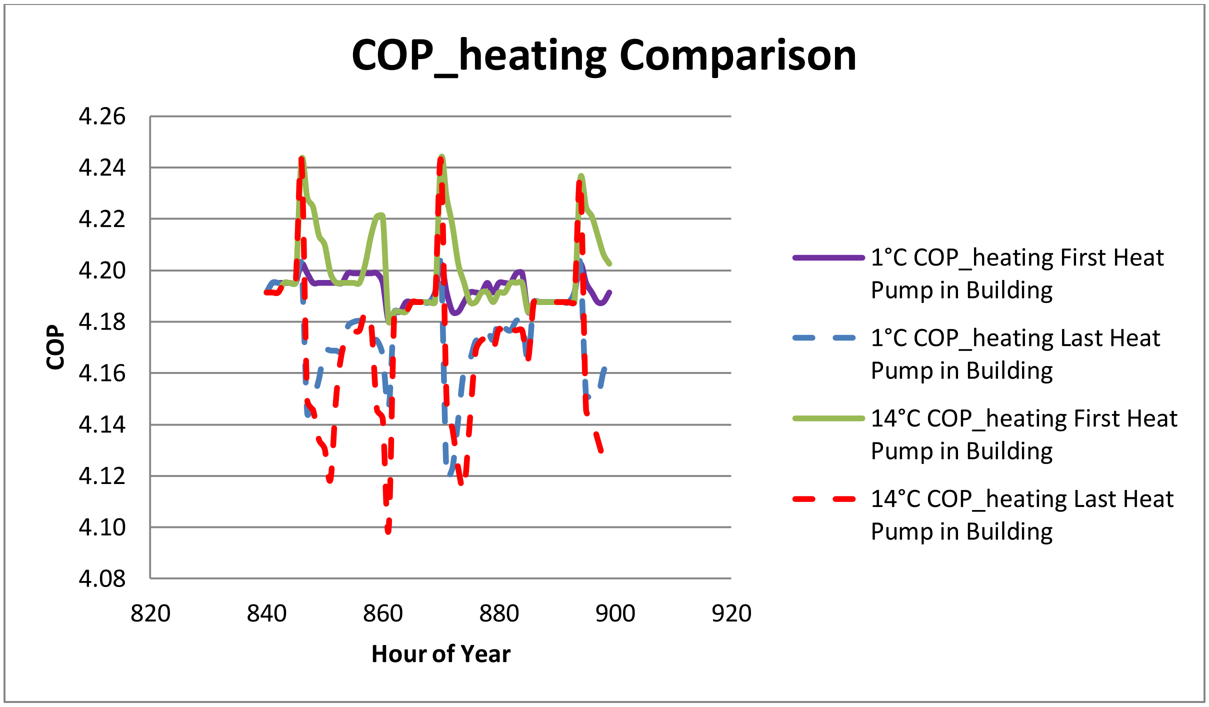

Figure 5.4.5 Tampa One-Pipe Heating COP Comparison for First and Last Heat Pumps

While in cooling the COPs changed accordingly to the temperature of the fluid. The COP decreased as the temperature of the fluid increased throughout the day, as shown in Figure 5.4.6. The performance of the heat pump for the 7°C (12.6°F) set point varied less and performed better throughout the day than the 1°C (1.8°F).

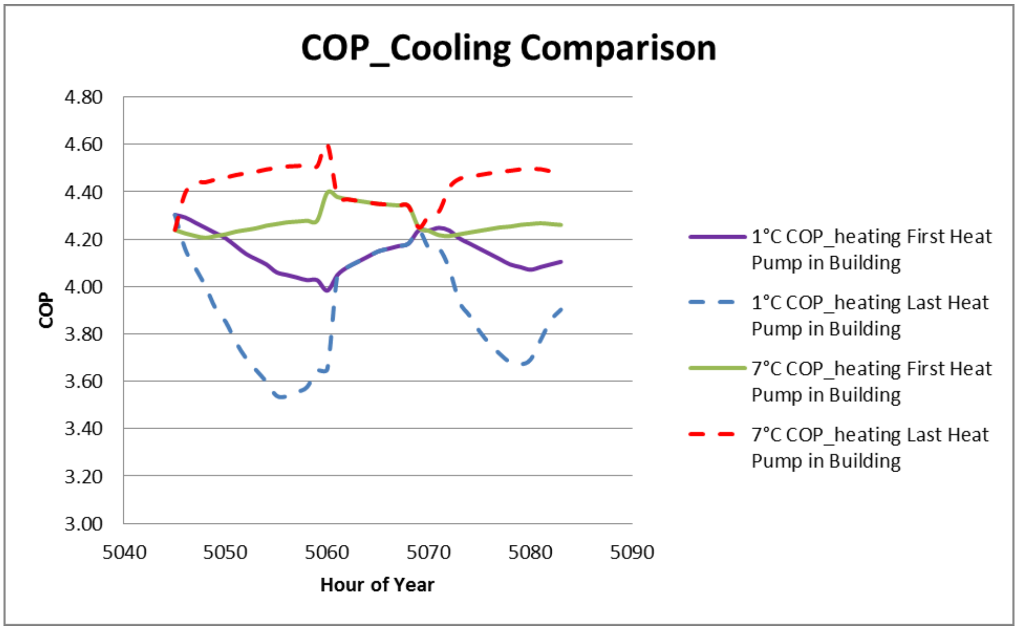

Figure 5.4.6 Tampa One-Pipe Cooling COP Comparison for First and Last Heat Pumps

The two-pipe system with central pumping required a higher flow rate than in Chicago for the optimal performance. Figure 5.4.7 shows the results as the ΔT set point was varied from 53,820 Pa (18 ft) to 95,680 Pa (32 ft) with the lowest energy consumption being at the 83,720 Pa (28 ft) set point. Also shown in Figure 5.4.7 is the results from the two-pipe system with distributed pumping. The flow rate through each heat pump at full load for the 83,720 Pa (28 ft) set point was 1 L/s (15.9 GPM). The distributed configuration had a slightly worse performance and had a large pumping load when using the efficiencies described by Sfier et. al (2005). When the efficiency was changed to assume 50% efficient pump, the total system energy consumption was 0.3% more than the optimal solution for central pumping.

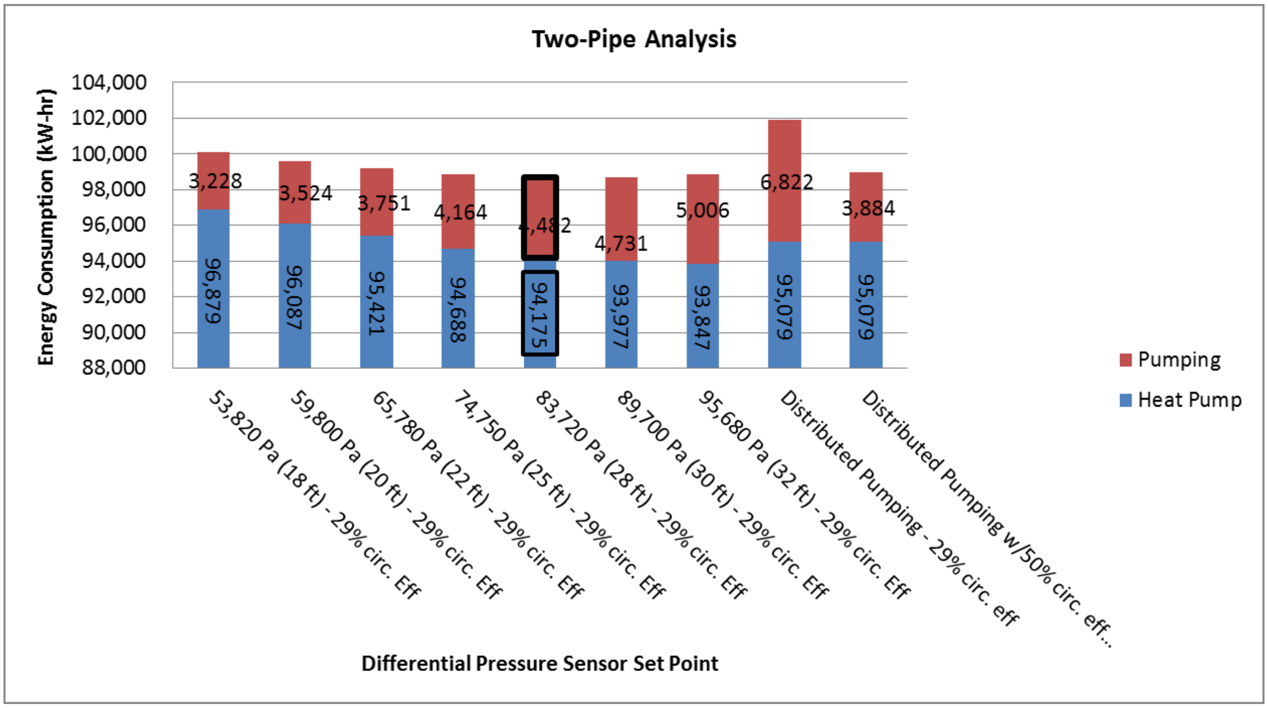

Figure 5.4.7 Tampa Two-Pipe with Central Pumping and Distributed System Energy

The two-pipe system with distributed pumping with hydraulic separation decreased energy consumption as the ΔT set point increased. Simulations were run from 3°C (5.4°F) to 17°C (30.6°F). As the temperature increased the total system energy consumption decreased. These results are shown in Figure 5.4.8. These results are consistent with the results from the Chicago simulations. As the set point increased the heat pump performance increased. Also, for ΔT set points above 9°C (16.2°F) the primary pump operated at the minimum speed for most conditions and did not see any energy savings; this set point will be used to compare against the other configurations.

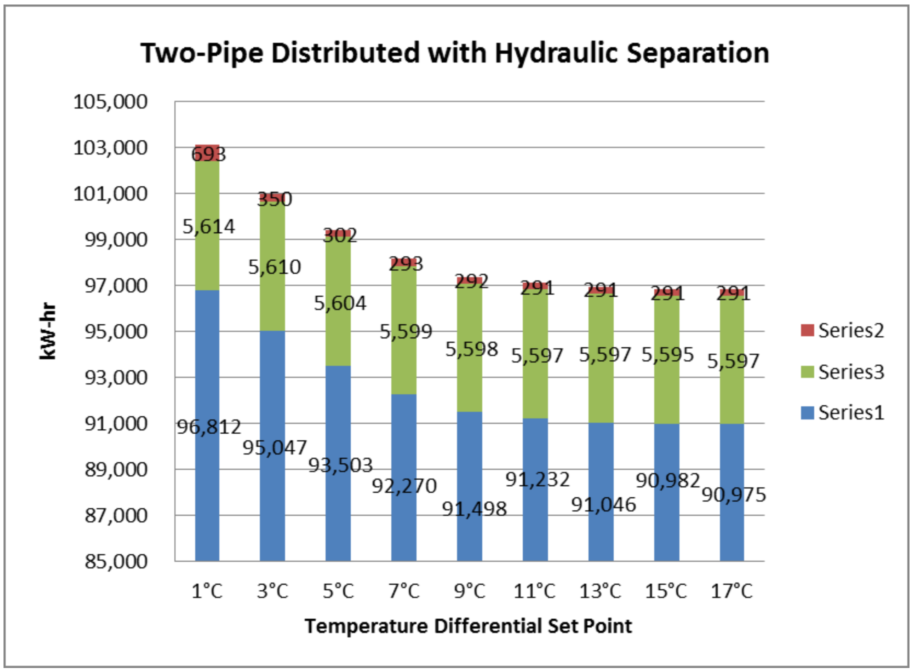

Figure 5.4.8 Tampa Two-Pipe Distributed w/HS System Energy

COP's for the peak heating condition for the three two-pipe pumping schemes varied at most during the peak heating condition by 0.15. The average COP for heating was 4.24 for all pumping schemes. For cooling, the distributed pumping, central pumping, and distributed-with-hydraulic-separation pumping schemes were 4.48, 4.52, and 4.60 respectively. The cooling COP was the driving factor in why the distributed-with-hydraulic-separation pumping scheme had the lowest heat pump energy consumption. COP's for a sample peak cooling day is shown in Figure 5.4.9. The COP for distributed-with-hydraulic-separation is typically 0.2 higher than the other two pumping schemes through the day.

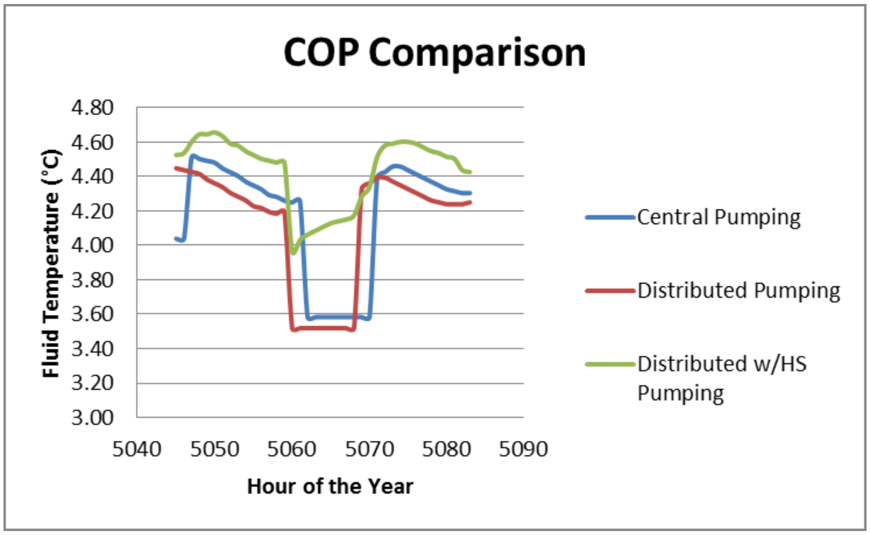

Figure 5.4.9 Tampa Two-Pipe Peak Cooling COP Comparison

A summary of the energy consumption for each piping configuration and pumping scheme is shown in Table 5.4.4. The two-pipe-distributed and the two-pipe distributed-with-hydraulic-separation circulator pump energy is shown with the results of using a 50% efficient circulator.

Table 5.4.4 Tampa Energy Rollup

Pumping Scheme	Circulator Pump Energy (kW-hr)	Primary Pump Energy (kW-hr)	Heat Pump Energy (kW-hr)	Total System Energy (kW-hr)
One-Pipe	2,787	492	98,343	101,622
Two-Pipe - Central	-	4,482	94,175	98,658
Two-Pipe - Distributed - 50% Circ Eff	3,884	-	95,079	98,962
Two-Pipe - Distributed - 29% Circ Eff	6,822	-	95,079	101,900.29
Two-Pipe - Distributed w/ HS -	3,184	292	91,498	94,974
Two-Pipe - Distributed w/ HS - 50% Circ. Eff	5,598	292	91,498	94,974

The one-pipe system consumed 7% more than the two-pipe system with distributed pumping with hydraulic separation. This was solely due to the increased energy consumption of the heat pumps. The one-pipe configuration had the lowest pump energy consumption of all 4 pumping schemes.

5.5 TULSA ENERGY CONSUMPTION RESULTS

The next location that was analyzed was in Tulsa, Oklahoma. The same pipe sizing methodology was used for this location, 0.6 m (2 ft) per 30.5 m (100 ft) of pipe. This again resulted in an 'A' rating for the pumping power in the design. Table 5.5.1 shows the theoretical pumping power for each pumping scheme.

Table 5.5.1 Tulsa Theoretical Pumping Power

Pumping Scheme	Theoretical Primary Pump kW (HP)	Theoretical Individual Circulator Pump kW (HP)	Total Theoretical Pumping Power kW (HP)
One-Pipe	0.61 (0.82)	0.070 (0.093)	1.44 (1.94)
Two-Pipe w/ Central Pumping	1.52 (2.04)	-	1.52 (2.04)
Two-Pipe w/ Distributed Pumping	-	0.127 (0.170)	1.52 (2.04)
Two-Pipe w/ Central Pumping with HS	0.40 (0.54)	0.093 (0.125)	1.52 (2.04)

Tulsa is a cooling dominated region that also has a much higher undisturbed ground temperature than Chicago. The results for 1°C (1.8°F) through 16°C (28.8°F) are shown in Figure 5.5.1. For this location, there was no optimal solution reached. The energy consumption decreased as the ΔT set point increased. The set point of 15°C (28.8°F) was chosen to be a stopping point for analysis since the pumping was equal to the lower set point of 13°C (23.4°F).

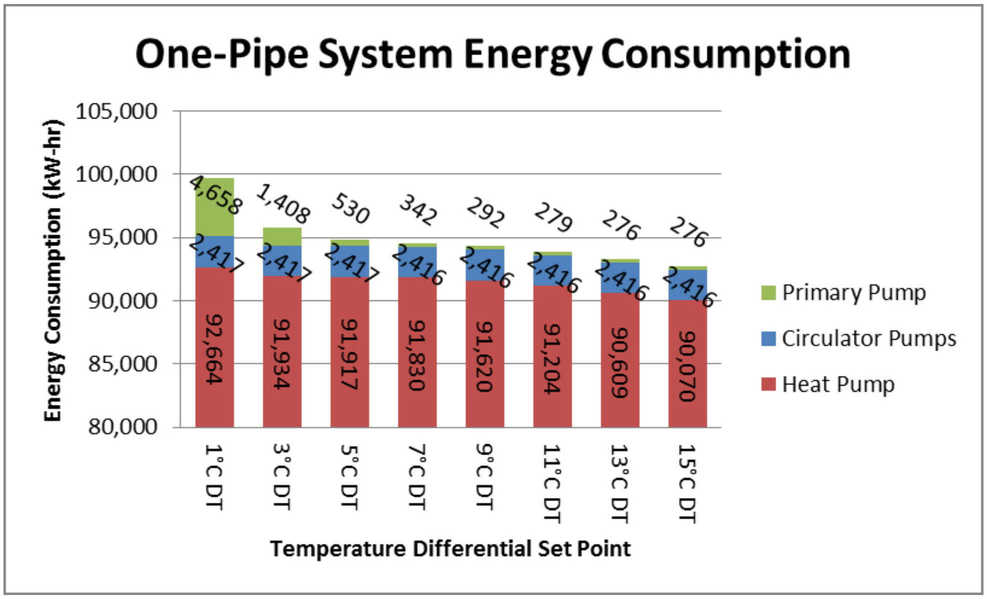

Figure 5.5.1 Tulsa One-Pipe System Energy Consumption

Looking at the temperatures of the fluid entering the borefield and the temperature leaving the borefield for the 1°C (1.8°F) and 15°C (27°F) set point, shown in Figure 5.5.2, the pump is able to provide sufficient flow to meet the 1°C (1.8°F) set point, but the minimum speed of the pump prevents it from operating at the 15°C (27°F) set point. The actual temperature difference for this peak heating condition peaks at about 10°C (18°F).

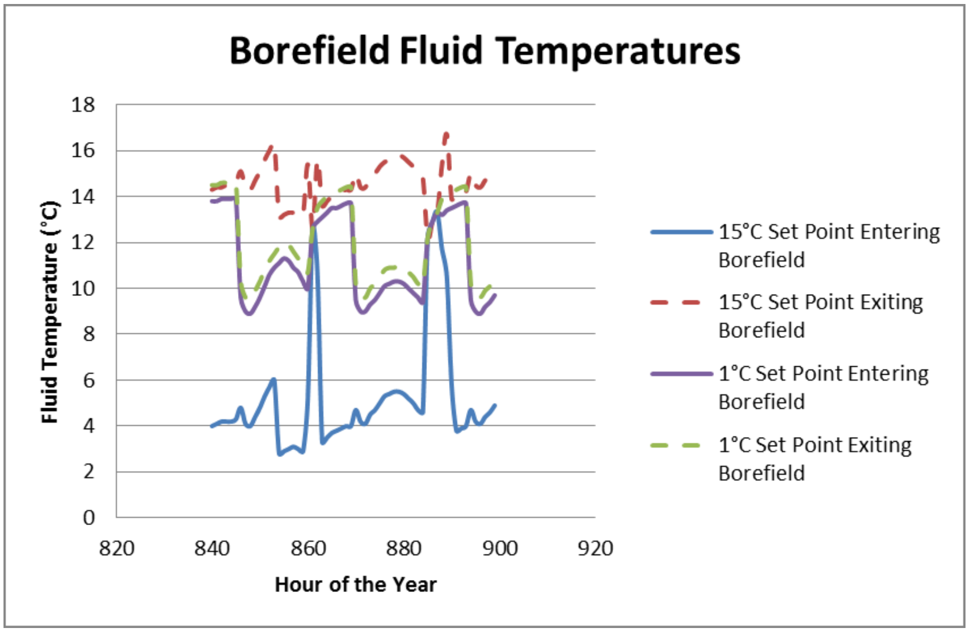

Figure 5.5.2 Tulsa One-Pipe Peak Heating GHX Inlet and Outlet Temperatures

Looking at the peak cooling condition, pump is sufficiently able to keep the temperature difference for both set points. At the peak condition, the 15°C (27°F) set point enters the borefield about 5°C (9°F) warmer than the 1°C (1.8°F) set point. The fluid leaves the borefield about 6°C (10.8°F) cooler. Despite having much warmer temperatures entering the borefield, the fluid returns from the borefield cooler with a higher ΔT set point.

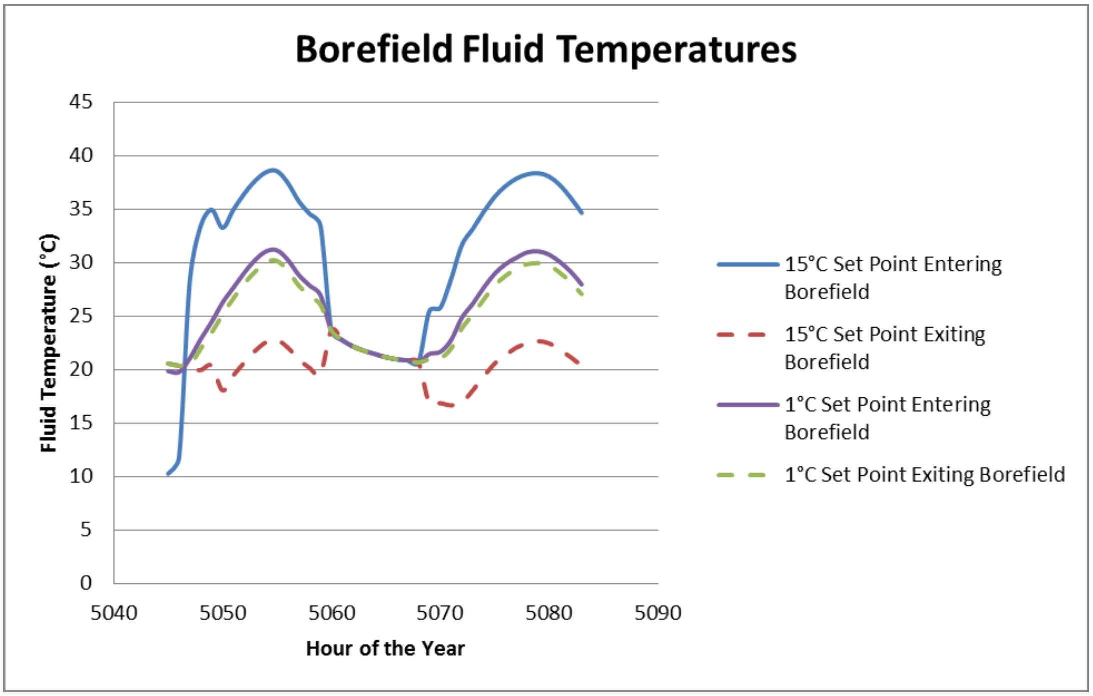

Figure 5.5.3 Tulsa One-Pipe Peak Cooling GHX Inlet and Outlet Temperatures

The average temperatures of the fluid entering the first heat pump and leaving the last heat pump are shown in Table 5.5.2 for both heating and cooling. Comparing the 1°C (1.8°F) and 15°C (27°F) set points, for heating, the temperature entering the first heat pump was 1.3°C (2.34°F) higher, while the temperature leaving the last heat pump was 7.5°C (13.5°F) cooler. During cooling, the temperature entering the first heat pump was 3.9°C (7.02°F) cooler, while the temperature leaving the last heat pump was 7.7°C (7.7°F) warmer.

Table 5.5.2 Tulsa One-Pipe Average Fluid Temperatures

	°C (°F)	°C (°F)
Set Point	1 (1.8)	15 (27)
Heating		
Entering First Heat Pump	14.4 (57.92)	15.7 (60.26)
Leaving Last Heat Pump	13.7 (56.66)	6.2 (43.16)
Cooling		
Entering First Heat Pump	23.2 (73.76)	19.3 (66.74)
Leaving Last Heat Pump	24.1 (75.38)	31.8 (89.24)

The average COP in heating was relatively unaffected by the change in set point based on the range of set points examined in this study. This was the case as most of the heating loads required less than the minimum flow rate of the pump to achieve the temperature difference. In cooling, the first heat pump for the 15°C (27°F) was 0.44 higher than the 1°C (1.8°F) set point while the last heat pump was 0.01 lower. Both heating and cooling COPs are shown in Table 5.5.3.

Table 5.5.3 Tulsa One-Pipe Average COP

Set Point	1°C (1.8°F)	15°C (27°F)
Heating		
COP First Heat Pump	3.8	3.86
COP Last Heat Pump	3.77	3.76
Cooling		
COP First Heat Pump	4.68	5.12
COP Last Heat Pump	4.54	4.53

The energy consumption of the pumps is shown in Figure 5.5.4, the circulators were independent of the set point, but the primary pump consumed less energy as the set point increased. The difference in pumping energy from the 9°C (16.2°F) to the 11°C (19.8°F) set point was about 1% savings. Above the 11°C (19.8°F) set point there were no additional savings.

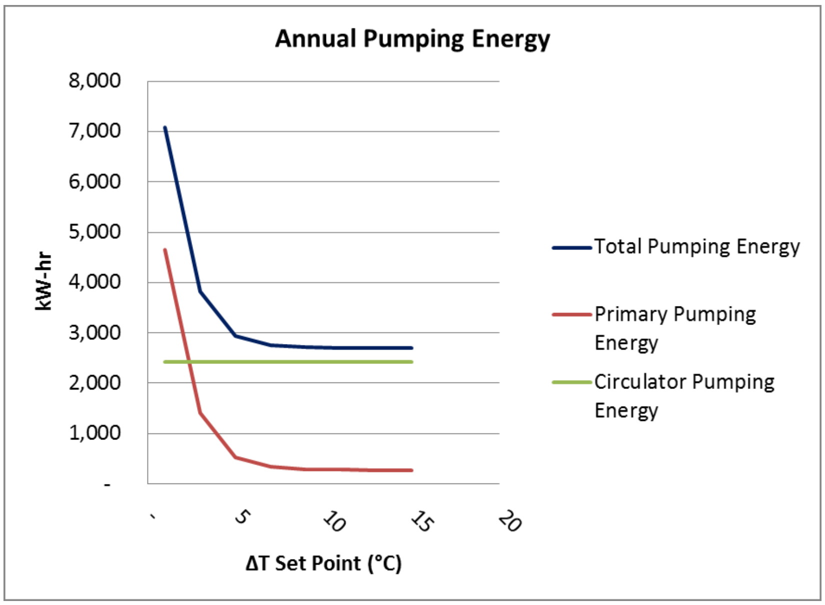

Figure 5.5.4 Tulsa One-Pipe Pumping Energy Consumption Analysis

The two-pipe configuration with central pumping was analyzed from the 53,820 Pa (18 ft) set point to the 95,680 Pa (32 ft) set point. The results are summarized in Figure 5.5.5. The optimal set point was 83,720 Pa (28 ft). This had a pumping energy of 4,022 kW-hr, a heat pump energy of 87,944 kW-hr, and a total system energy consumption of 91,966 kW-hr. Comparably, the distributed pumping had only a 0.5% higher consumption of heat pump energy. The pumping energy consumption was 47% higher when using the 29% efficient pump. If using the 50%

efficient circulator pump, this would drop to a 16% decrease in pumping energy over central pumping.

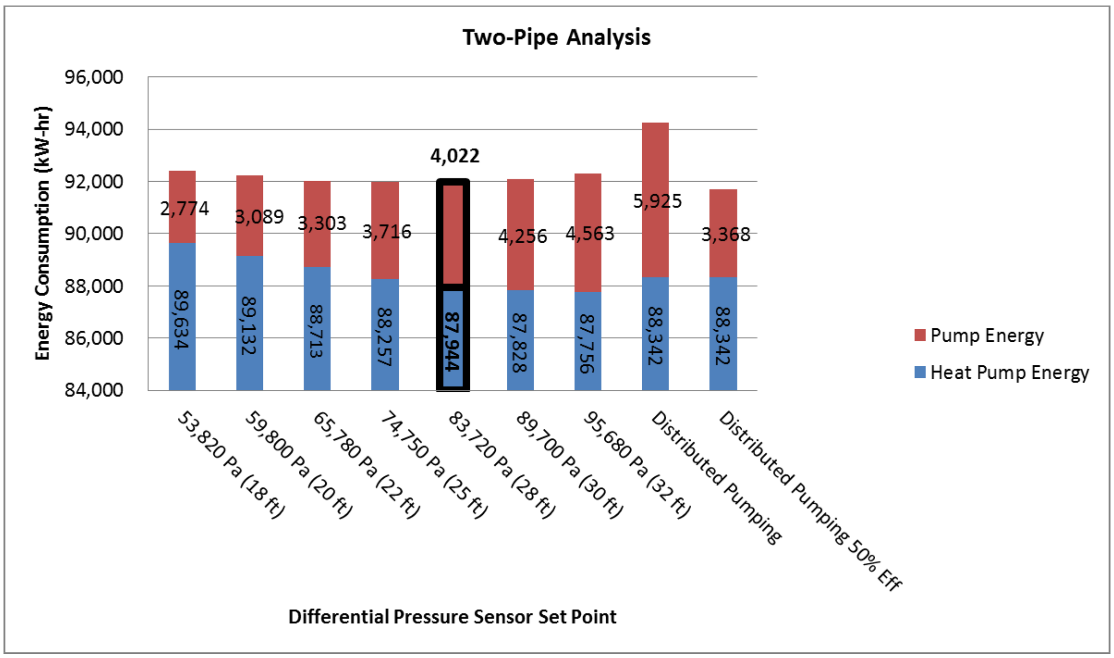

Figure 5.5.5 Tulsa Two-Pipe with Central Pumping and Distributed System Energy

The distributed-with-hydraulic-separation pumping scheme was analyzed from the 5°C (9°F) ΔT set point to the 15°C (27°F) set point. The total energy consumption decreased as the DT set point increased for all cases, though for each 2°C (3.6°F) increase in set point the decrease in energy savings was less than 1%. Both the pumping energy and heat pump energy consumptions decreased with increasing ΔT set points.

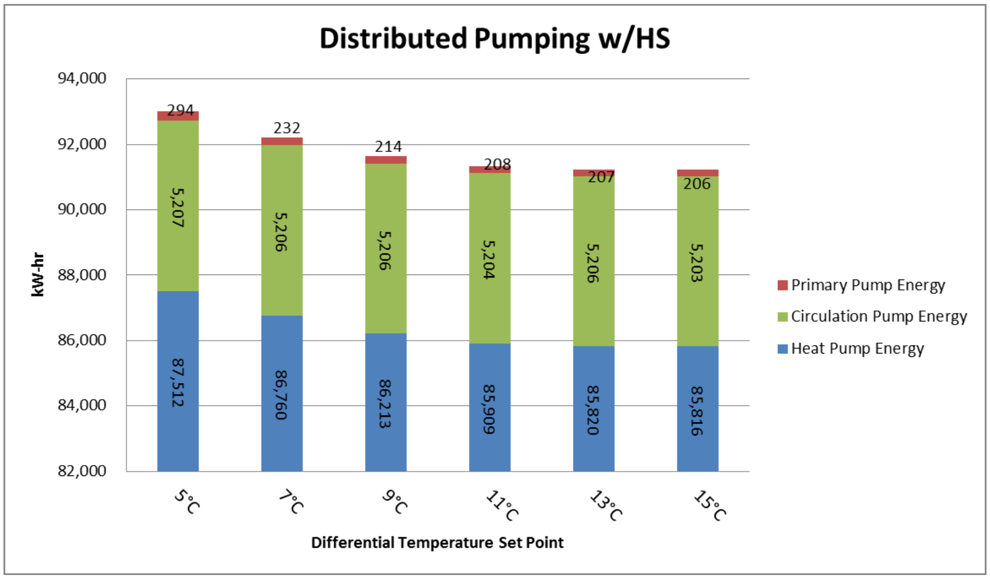

Figure 5.5.6 Tulsa Two-Pipe Distributed w/HS System Energy

The temperatures entering and exiting the borefield at a peak cooling condition are shown in Figure 5.5.7 for all three two-pipe pumping schemes. The distributed-with-hydraulic-separation scheme consistently had lower temperatures returning from the borefield compared to the other two pumping schemes.

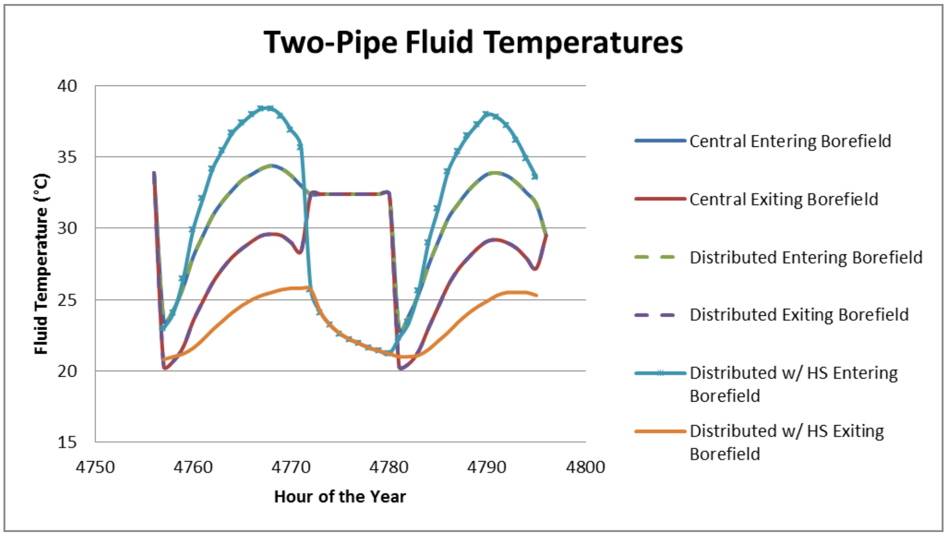

Figure 5.5.7 Tulsa Two-Pipe Peak Cooling GHX Temps

The result from this was a higher COP for cooling under most conditions for the distributed with hydraulic separation scheme. The peak condition's COP is shown in Figure 5.5.8.

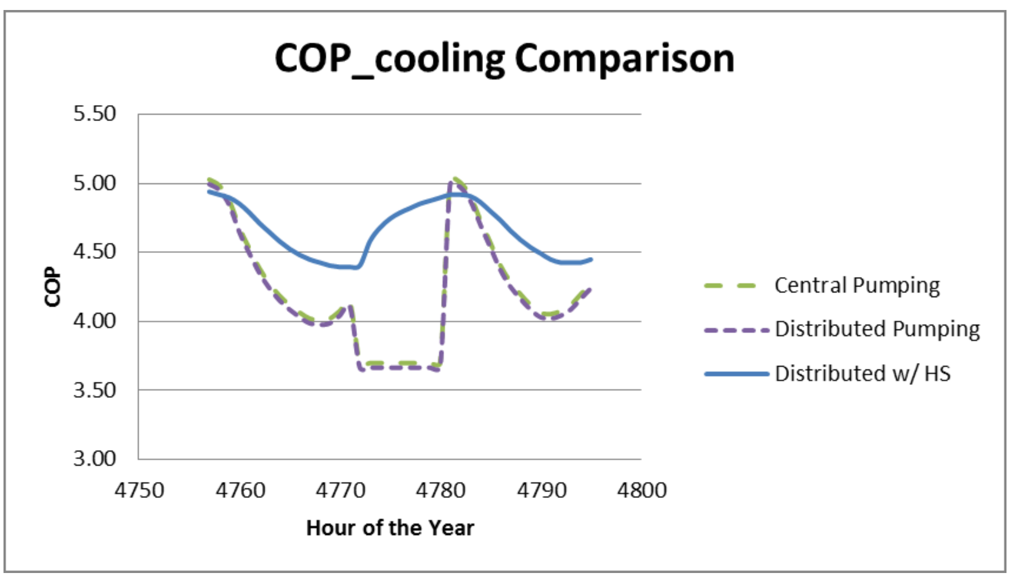

Figure 5.5.8 Tulsa Two-Pipe Peak Cooling COP Comparison

In heating the average COP was almost the same for all three pumping schemes as shown in Table 5.5.4. In cooling however, as discussed in Figure 5.5.8, the distributed with hydraulic separation scheme had much cooler temperatures returning from the borefield which created a much higher COP than the other two pumping schemes. This resulted in a 2% higher COP over central pumping or about 0.1.

Table 5.5.4 Tulsa Two-Pipe Average COP

Set Point	COP
Heating	
Central	3.84
Distributed	3.83
Distributed w/ HS	3.85
Cooling	
Central	4.86
Distributed	4.83
Distributed w/ HS	4.96

Comparing the performance for all three two-pipe pumping schemes with the one-pipe configuration, the two-pipe configuration with distributed pumping with hydraulic separation is the best performing system. A summary of the optimal solution for each pumping scheme is presented in Table 5.5.5. For all schemes, the circulator efficiency was assumed to be a constant 50%. With 50% circulator efficiency, the one-pipe system was still only 4% higher in energy consumption for this location. With the Sfeir et al. (2005) calculated efficiency of 29%, this dropped to 1%, however the distributed with hydraulic separation pumping scheme was still the best performing pumping scheme.

Table 5.5.5 Tulsa Energy Rollup

Pumping Scheme	Circulator Pump Energy (kW-hr)	Primary Pump Energy (kW-hr)	Heat Pump Energy (kW-hr)	Total System Energy (kW-hr)
One-Pipe	2,416	276	90,070	92,762
Two-Pipe - Central	-	4,022	87,944	91,966
Two-Pipe - Distributed	3,368	-	88,342	91,710
Two-Pipe - Distributed w/ HS	3,164	206	85,816	89,186

5.6. STERLING ENERGY CONSUMPTION RESULTS

Sterling, Virginia is a heating dominated region, though less so than Chicago, Illinois. A breakdown of the theoretical pumping sizes is shown in Table 5.6.1. Analysis was done for each two-pipe configuration using circulator pump efficiencies calculated by the equations from Sfeir et al. (2005). Once the best solution is found, the circulator pumping efficiency is changed to 50% to evaluate the performance with the best available efficiency on the market. The primary pump wire to water efficiency was 65%, 66%, and 67% for the two-pipe distributed pumping with hydraulic separation, one-pipe, and two-pipe with central pumping configurations respectively. The one-pipe circulator efficiency was assumed to be 50% for all ΔT set points. The one-pipe configuration requires the smallest theoretical pumping power of all configurations.

Table 5.6.1 Sterling Theoretical Pumping Power

Pumping Scheme	Theoretical Primary Pump kW (HP)	Theoretical Individual Circulator Pump kW (HP)	Total Theoretical Pumping Power kW (HP)
One-Pipe	0.57 (0.76)	0.070 (0.093)	1.4 (1.88)
Two-Pipe w/ Central Pumping	1.48 (1.98)	-	1.48 (1.98)
Two-Pipe w/ Distributed Pumping	-	0.123 (0.165)	1.48 (1.98)
Two-Pipe w/ Central Pumping with HS	0.36 (0.49)	0.093 (0.125)	1.48 (1.98)

The one-pipe configuration was analyzed by changing the ΔT set point from 1°C (1.8°F) to 11°C (19.8°F). The optimal solution for this location was at 5°C (9°F) where the heat pump energy consumption was 82,963 kW-hr and the total pumping energy consumption was 2,700 kW-hr. The heat pump energy consumption decreased the 1°C (1.8°F) set point to the 4°C (7.2°F) set point, then increased for all additional increases in the ΔT set point. Pumping energy decrease rapidly from set points 1°C (1.8°F) to 4°C (7.2°F) then gradually decreased for ΔT above this. To meet the 1°C (1.8°F) the primary pump was operating at full speed under most conditions, however was able to slow down for higher set points.

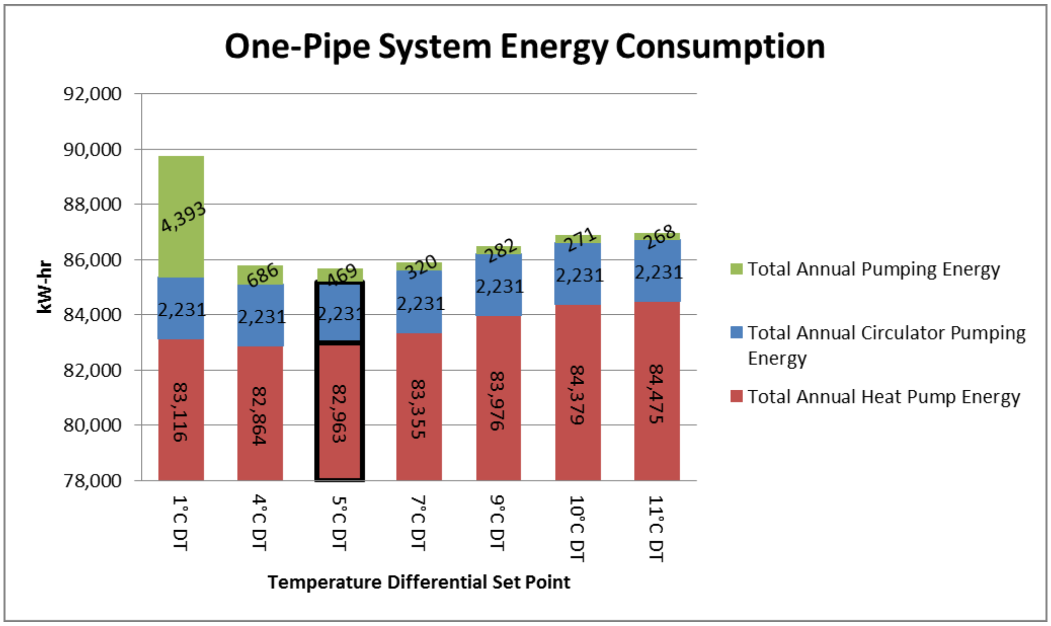

Figure 5.6.1 Sterling One-Pipe System Energy Consumption

Consistent with the other locations, generally the higher the ΔT set point in heating returned warmer fluid temperatures from the borefield with cooler temperatures going to the borefield. This can be seen in Figure 5.6.2. The alternative can be seen in Figure 5.6.3 for cooling, where the higher the set point, the cooler the return temperature.

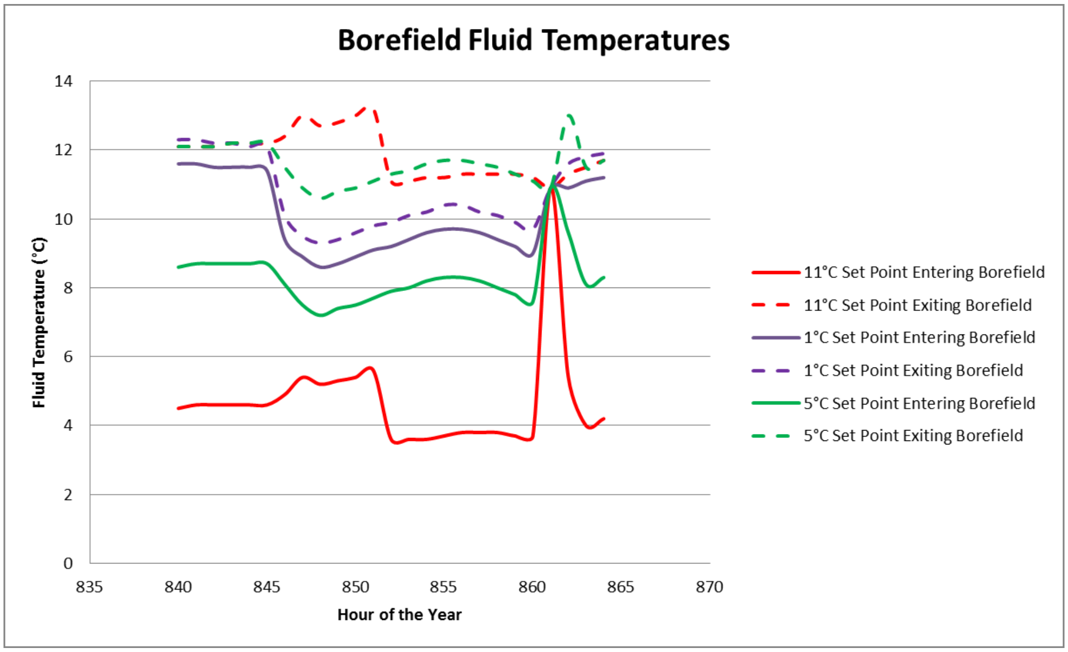

Figure 5.6.2 Sterling One-Pipe Peak Heating GHX Inlet and Outlet Temperatures

Unlike the peak heating condition, in the peak cooling condition, there is relatively little difference in the temperature of the fluid leaving the borefield for the 5°C (9°F) and the 11°C (19.8°F) set points. This likely means that the local ground temperature of the GHX is closer to 15°C (59°F) than the undisturbed ground temperature of 13.3°C (56°F) and no further heat exchange is possible.

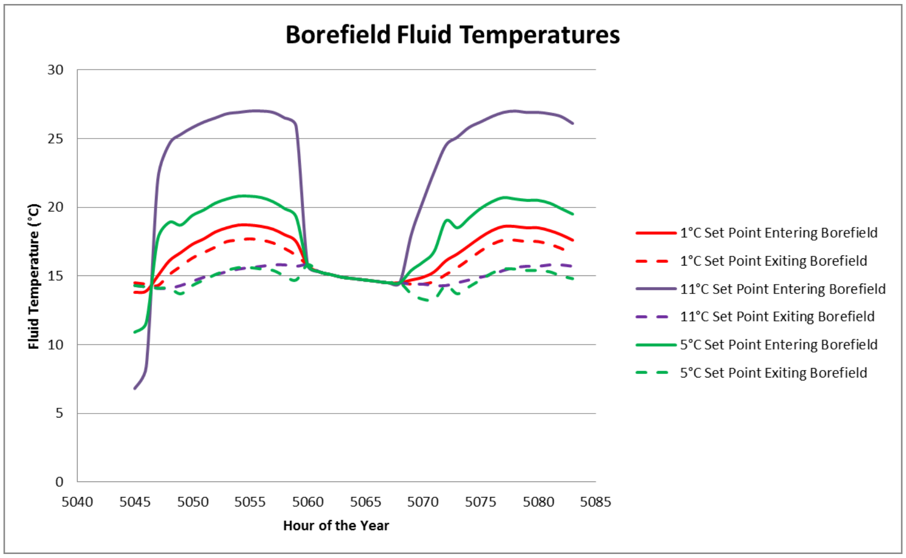

Figure 5.6.3 Sterling One-Pipe Peak Cooling GHX Inlet and Outlet Temperatures

Comparing the average fluid temperatures of the fluid in heating, the temperature entering the GHX (leaving the last heat pump) decreased with the increase in ΔT set point as shown in Table 5.6.2. The returning temperature of the 5°C (9°F) set point was however the highest of all set points. In cooling, the temperatures entering the GHX were consistently increasing with the increase in ΔT, but again, the 5°C (9°F) set point had the lowest returning temperature.

Table 5.6.2 Sterling One-Pipe Average Fluid Temperatures

	°C (°F)	°C (°F)	°C (°F)
Set Point	1 (1.8)	5 (9)	11 (19.8)
Heating			
Entering First Heat Pump	12.58 (54.64)	12.69 (54.84)	12.53 (54.55)
Leaving Last Heat Pump	11.5 (52.70)	9.4 (48.92)	5.43 (41.77)
Cooling			
Entering First Heat Pump	15.18 (59.32)	14.08 (57.34)	14.28 (57.70)
Leaving Last Heat Pump	15.99 (60.78)	18.16 (64.69)	23.22 (73.80)

This resulted in better COPs overall for the 5°C (9°F) set point. Table 5.6.3 shows the average COPs for the first and last heat pumps in heating and in cooling. On average the first heat pump operated with better COPs than the other two set points. The last heat pump was better than the 11°C (19.8°F) set point, but worse than the 1°C (1.8°F) set point.

Table 5.6.3 Sterling One-Pipe Average COP

Set Point	1°C (1.8°F)	5°C (9°F)	11°C (19.8°F)
Heating			
COP First Heat Pump	3.7	3.72	3.71
COP Last Heat Pump	3.66	3.64	3.6
Cooling			
COP First Heat Pump	5.56	5.67	5.65
COP Last Heat Pump	5.44	5.35	5.19

For the pumping energy, the circulators saw a constant energy consumption across all set points. The primary pump energy decrease as the ΔT set point increased. For the 1°C (1.8°F) set point the pump ran at full speed almost all of the time while for the 11°C (19.8°F) set point the pump operated at the minimum speed most of the time.

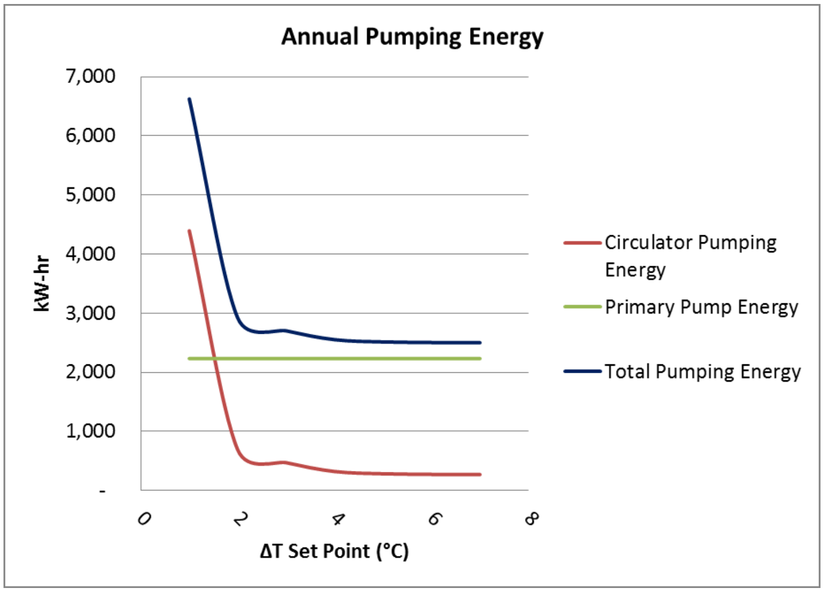

Figure 5.6.4 Sterling One-Pipe Pumping Energy Consumption Analysis

The two-pipe configuration with central pumping was analyzed for DP set points of 53,820 Pa (18 ft) to 83,720 Pa (28 ft). The optimal set point was 65,780 Pa (22 ft) where the heat pump energy consumption was 81,062 kW-hr, the pumping energy consumption was 3,072 kW-hr, for a total of 84,134 kW-hr. The results are shown in Figure 5.6.5. Also shown are the results for the distributed pumping with the calculated pump efficiency and the assumed 50% pump efficiency. Distributed pumping with the calculated efficiency had a 2.5% higher energy consumption than the optimal central pumping set point, whereas the 50% efficient distributed pumping had a 0.3% less energy consumption.

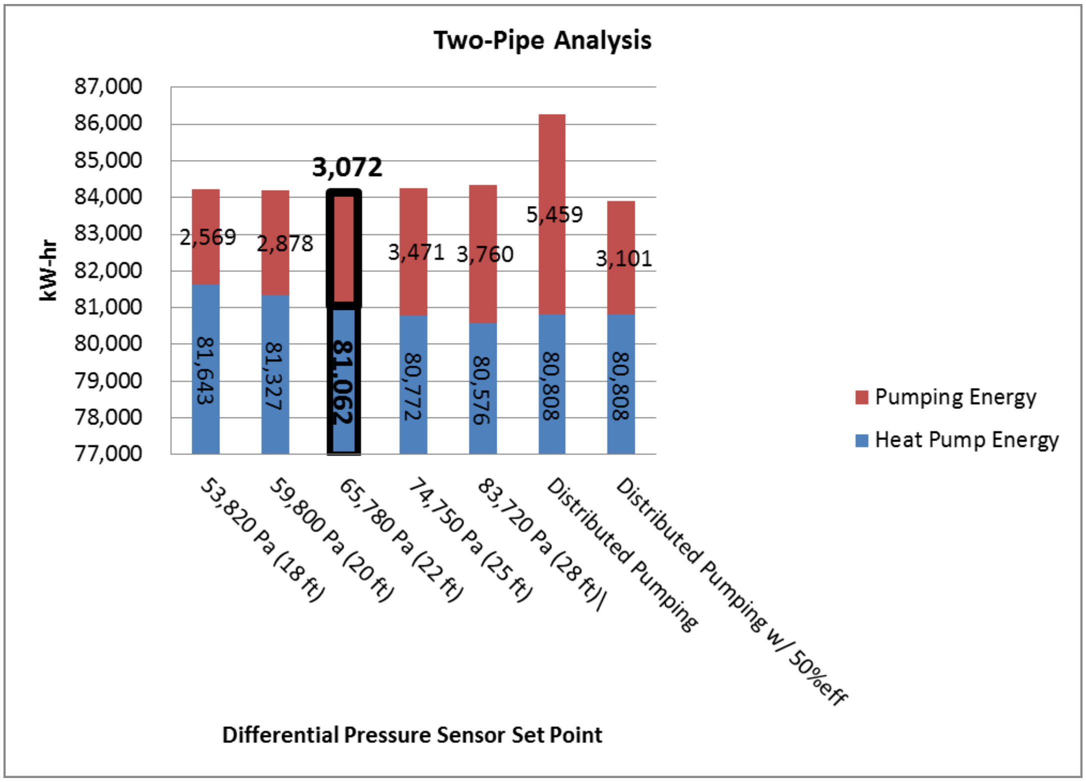

Figure 5.6.5 Sterling Two-Pipe with Central Pumping and Distributed System Energy

The two-pipe configuration with distributed pumping with hydraulic separation was analyzed with ΔT set points ranging from 3°C (5.4°F) to 13°C (23.4°F) with the optimal set point being at 7°C (12.6°F) where the heat pump energy consumption was 80,282 kW-hr, the pumping energy consumption was 5,098 kW-hr, with a total system energy consumption of 85,380 kW-hr.

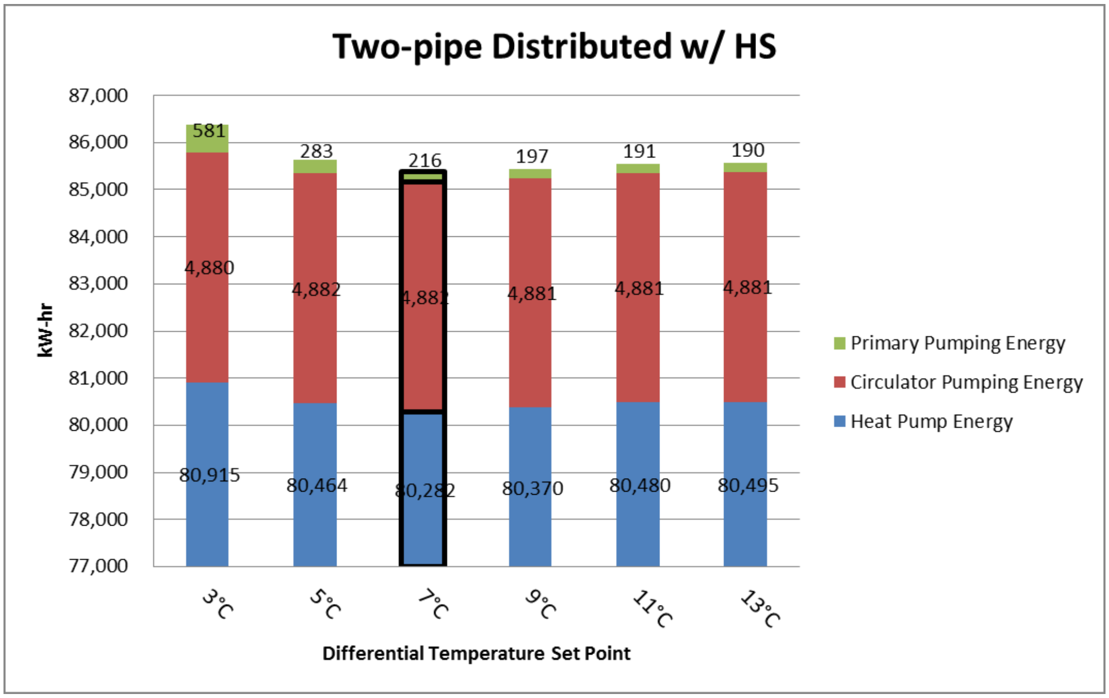

Figure 5.6.6 Sterling Two-Pipe Distributed w/HS System Energy

Figure 5.6.7 shows a comparison of the borefield entering and exiting temperatures for both central pumping and distributed-pumping-with-hydraulic-separation. Not shown is distributed pumping as the temperatures were close to identical to that of the central pumping. The central pumping temperatures had a slightly smaller ΔT across the borefield to include a slightly lower entering temperature, but was warmer upon exiting the GHX.

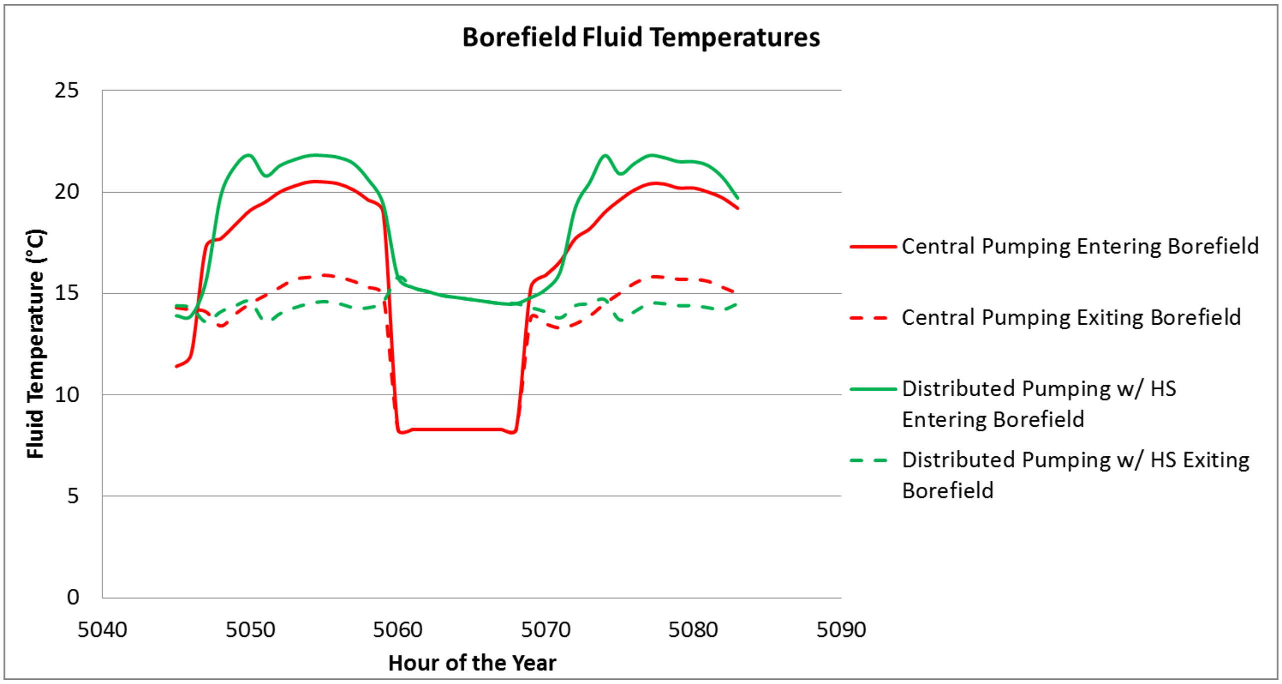

Figure 5.6.7 Sterling Two-Pipe Peak Cooling GHX Temps

As a result of the warmer exiting temperature the COP for the central pumping scheme was slightly lower than the distributed pumping with hydraulic separation scheme. Additionally despite similar returning temperatures, the distributed pumping scheme had slightly better COPs than the central pumping scheme as shown in Figure 5.6.8.

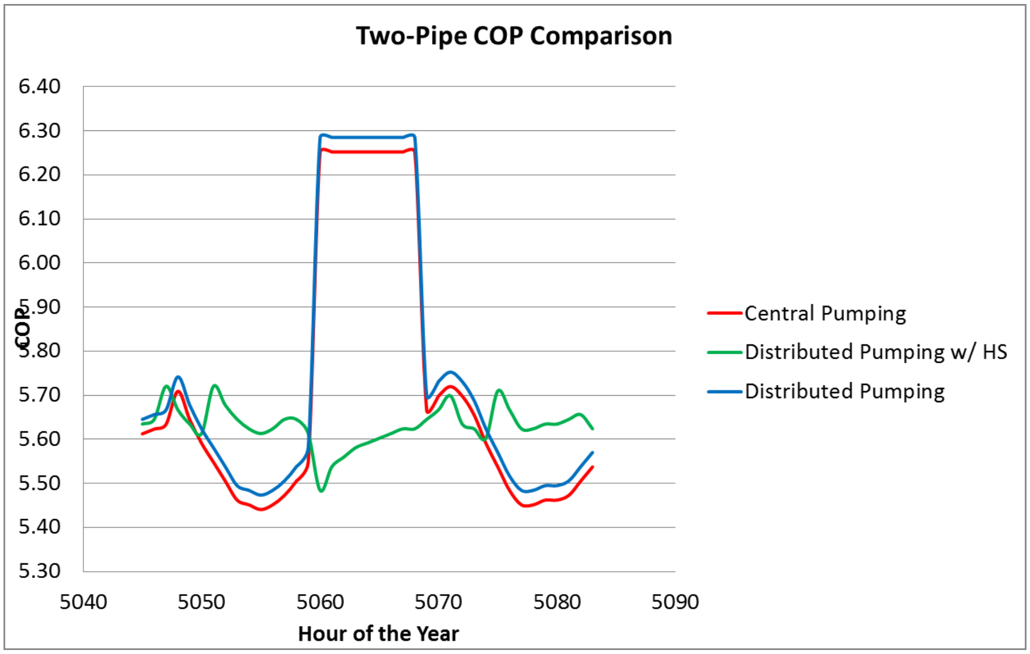

Figure 5.6.8 Sterling Two-Pipe Peak Cooling COP Comparison

The average COP for each of the three pumping schemes were very similar, varying by only 0.02 or 0.5% in heating and 0.05 or 0.08% in cooling. Average COPs for both heating and cooling are shown in Table 5.6.4.

Table 5.6.4 Sterling Two-Pipe Average COP

Set Point	COP
Heating	
Central	3.71
Distributed	3.72
Distributed w/ HS	3.73
Cooling	
Central	5.63
Distributed	5.66
Distributed w/ HS	5.68

A summary of the individual pumping energy, heat pump energy, and total system energy consumptions is shown in Table 5.6.5. For this location the best performing system was the two-pipe configuration with distributed pumping with hydraulic separation with a 50% efficient pump. The one-pipe configuration was the poorest performing configuration but was still within 3.5% of the best performing configuration.

Table 5.6.5 Sterling Energy Rollup

Pumping Scheme	Circulator Pump Energy (kW-hr)	Primary Pump Energy (kW-hr)	Heat Pump Energy (kW-hr)	Total System Energy (kW-hr)
One-Pipe	**2,231**	**320**	**83,355**	**85,906**
Two-Pipe - Central	**-**	**3,072**	**81,062**	**84,134**
Two-Pipe - Distributed	**3,101**	**-**	**80,808**	**83,908**
Two-Pipe - Distributed w/ HS	**2,987**	**216**	**80,282**	**83,485**

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

6.1 CONCLUSIONS

A program was designed to help in the analysis of four separate pumping and piping schemes. This program is able to size the piping network of a GSHP with both direct and reverse return piping. It can adjust the spacing of each heat pump as well as simulate having a header in an equipment room or installing a vault next to the borefield. The pipe sizing criteria can be specified by the user to evaluate the required pumping power for each criterion. The spacing of boreholes, the number of boreholes, and the depth of boreholes can all be specified. Once sized the user can simulate the performance of the heat pump and pumps for a typical meteorological year. This incorporates using HVACSIM+ to simulate the performance of the borefield to give the returning fluid temperatures.

A sample strip mall's energy performance for 10 zones with a total of 12 heat pumps has been simulated in four locations. Various design conditions have been used to size the pipes used to distribute the fluid ranging from 1.21 m/s (4 ft/s) for pipes 50 mm (2 in) and smaller and a head loss of 1.2 m (4 ft) per 30.5 m (100 ft) of pipe for larger pipes through a straight head loss of 1.2 m (4 ft) per 30.5 m (100 ft) of pipe. It was determined that 0.6 m (2 ft) per 30.5 m (100 ft) of pipe resulted in a grade of 'A' by the established pumping scale (Kavanaugh et al. 2003) while a more conservative sizing criteria created a much larger initial purchase price for the pipe. This was then used as the design criteria for all four locations.

For each location, a detailed analysis of four separate pumping and piping configurations was analyzed and optimized. One pumping scheme for the one-pipe configuration, and three pumping schemes for the two-pipe configuration, central pumping, distributed pumping, and distributed pumping with hydraulic separation. The performance of the heat pump was considered along with the performance of the primary pump and individual circulator pumps where applicable. For all four locations, the one-pipe configuration had the lowest pumping energy consumption (assuming a 50% efficient circulator) while also having the largest heat pump energy consumption. This ended up consuming the most energy at each location, but at most was 7% worse than the best performing system, three of the four locations were within 5%. The two-pipe configuration with distributed pumping with hydraulic separation was the best performing pumping scheme at all four locations, but only when the best efficiency on the market for circulator pumps (50%) is used. If the equations from Sfeir et al. (2004) where the peak circulator pumping efficiency was 29% for all locations, the two-pipe configuration with central pumping was the best performing system. The two pipe configuration with distributed pumping was the second best performing system, again with the 50% efficient circulators but was the 3 worst system with the 29% efficient circulators.

Looking into the performance of the borefield, as the amount of fluid pushed through the borefield decreased, the temperature rise across the building increased. Unexpectedly, the temperature of the fluid returning from the borefield was cooler during cooling and warmer during heating resulting in better overall heat pump performance in most conditions. This was also dependent on the undisturbed ground temperature and whether or not the building was in a balanced environment or if it was dominated by either heating or cooling.

For the one-pipe configuration, generally as the ΔT set point increased the system had a better overall performance. In the heating dominated locations, there was an optimal ΔT set point found, where in the cooling dominated locations there was only a decrease in energy as the ΔT set point increased until the pump was operating at minimum flow at all times in which no more savings were realized.

The two-pipe configuration with central pumping had an optimal set point of 65,780 Pa (22 ft) for both heating dominated locations and an optimal set point of 83,720 Pa (28 ft) for both of the cooling dominated locations. For both types of locations, the design set point was 74,750 Pa (25 ft).

The two-pipe configuration with distributed pumping was a well performing system overall. The pumping energy consumption was highly dependent on the efficiency of the circulator pumps that were chosen. Assuming 50% efficient circulation pumps are used, this configuration was the second best performing out of the four pumping schemes. If normal efficiency pumps were used, as defined by Sfeir et al. (2005), this increased the pumping energy by 76%.

The two-pipe configuration with distributed pumping with hydraulic separation was the best performing pumping scheme for all four locations. This was assuming 50% efficient circulation pumps. If Sfeir et al. (2005) equations were used to calculate the efficiency, which was about 29%, the pumping energy would increase about 63%. With the exception of the Sterling location, this pumping scheme was more efficient with higher ΔT set points.

6.2 RECOMMENDATIONS

A lot of the benefits of decreased energy consumption for pumping in a GSHP come at a cost. To decrease the installed pumping power, the pipe sizes must be increased which increases installed pipe costs. In general, the pumping energy consumption for most of these simulations were in the order of 2%-6% of the total system energy consumption. This is consistent with the recommendations of Kavanaugh et al. (2003) that a good design should have pumping energy consumption of no more than 5% of the total energy consumption. Recommendations for further research could include the use of hybrid systems. Using a hybrid system such as a cooling tower in a cooling dominated system or a boiler in a heating dominated region could decrease the annual energy consumption and decrease initial installation costs by reducing the total required borehole depth. Furthermore, decoupling the primary pump size from the flow rate required by the heat pumps could be explored to see if installing a larger pump or a smaller pump would be more energy efficient. Also, a model could be developed for the modeling of ECM pumps to be incorporated into the program that was developed for this research. Finally, research could be done to analyze a more detailed cost estimate between all configurations.

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APPENDICES

Question: How to connect a heat pump model written in Excel/VBA program to a ground heat exchanger model (Type 620) written in HVACSIM+?

Suppose there is a ground source heat pump (GSHP) system as shown in Figure 1. The ground heat exchanger extracts or rejects certain amount of heat from the ground, the heat is transferred through the heat pump to the house, in order to meet the building heating loads $q_{heating}$ and building cooling loads $q_{cooling}$. The building heating and cooling loads are required as inputs for the model.

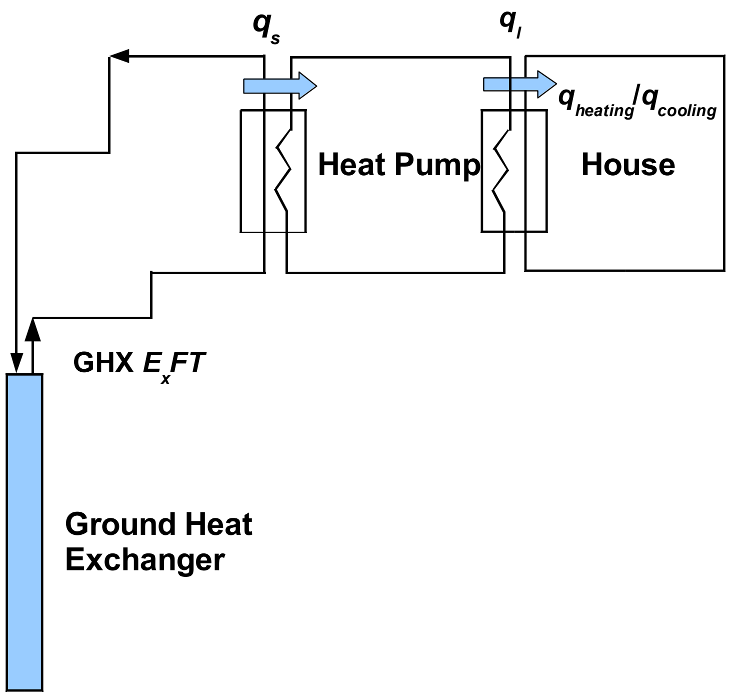

Figure 1: Schematic of the Problem

In this problem, the heat transfer rate between heat pump and ground heat exchanger q_s is a function of Ground heat exchanger exiting fluid temperature (GHX E_xFT) and building heating loads $q_{heating}$ and building cooling loads $q_{cooling}$, the GHX E_xFT is also a function of q_s as well. So there are two unknowns, GHX E_xFT and q_s, and two models - a heat pump model and ground heat exchanger model can be used to solve this problem.

How to solve this then? Figure 2 shows the flow chart of the ground source heat pump system model. For the first iteration, assume a initial value for the GHX E_xFT, run the heat pump model, which reads the building heating and cooling loads, GHX E_xFT as inputs and calculates the heat transfer rate between ground heat exchanger and heat pump q_s. Then the ground heat exchanger model reads the q_s as inputs and calculates the GHX E_xFT. Update the value of GHX E_xFT and run the heat pump model again, continue iterating until the absolute different of GHX E_xFT and updated GHX E_xFT is less than the convergence criteria. Then stop the iteration. The main subroutine is written in VBA program and pasted in APPENDIX A.

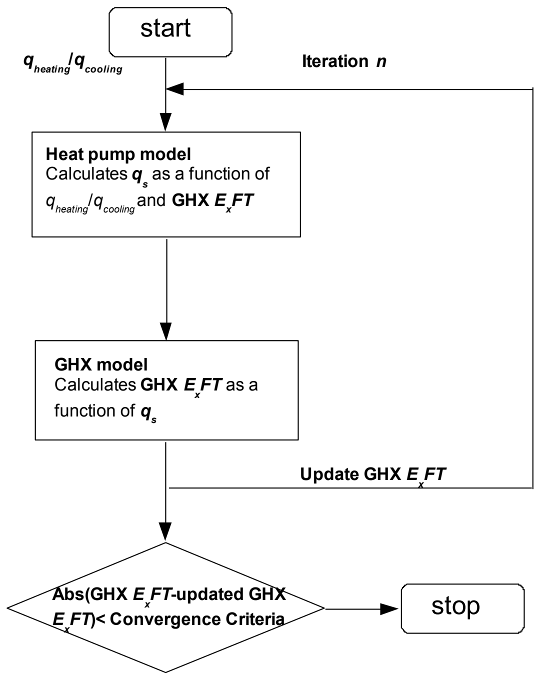

Figure 2: The flow chart of the ground source heat pump system model

Heat Pump Model

The heat pump model is written in the VBA program and the subroutine has been pasted in Appendix B. It is assumed that the ground heat exchanger is connected to a heat pump model in series. Hourly heating and cooling loads applied on the heat pump are estimated from a building simulation program, such as EnergyPlus. Users are required to input the information of the modeled building, including geometry, construction information, internal heat gains and weather data. Then the program will estimate the hourly heating and cooling loads of the building.

The model uses two polynomial equations to relate heating loads to heat extraction and cooling loads to heat rejection, then the total heat extraction rate from the ground heat exchanger q_s can be calculated as:

$$q_s = q_{heating}\left(C_{abs,1}ExFT_{GHX}^{0} + C_{abs,2}ExFT_{GHX}^{1} + C_{abs,3}ExFT_{GHX}^{2}\right) - q_{cooling}\left(C_{rej,1}ExFT_{GHX}^{0} + C_{rej,2}ExFT_{GHX}^{1} + C_{rej,3}ExFT_{GHX}^{2}\right) \quad (1)$$

Where:

q_s is the heat extracted by the ground heat exchanger and transfer to the heat pump, in KW or Btu/hr;

$q_{heating}$ is the building heating loads, in KW or Btu/hr;

$q_{cooling}$ is the building cooling loads, in KW or Btu/hr;

$ExFT_{GHX}$ is the ground heat exchanger exiting fluid temperature, in °C or °F;

$C_{abs,i}$ and $C_{rej,i}$ are values used for determining the ratio of heating or cooling, i could be 1, 2 or 3;

For a specific heat pump, the ratio of heating/cooling at different heat pump *EFT* (GHX E_xFT) can be calculated from the data provided by the manufacture. The value of $C_{abs,i}$ and $C_{rej,i}$ can then be found by curve fitting the manufacturer's data.

Ground Heat Exchanger Model

The ground heat exchanger model is written in HVACSIM+, to run it would require four files, modsim.exe, inputfile.dat, simtemp.dfn and simtemp.bnd. The modsim.exe is a executive tool for running the model. The inputfile.dat records the total simulation time and time steps for the simulation, etc. The simtemp.dfn files records the ground heat exchanger parameters. The simtemp.bnd records the heat transfer rate between the ground heat exchanger and the heat pump q_s and the heat exchanger mass flow rate for each time step. The model will output the result in simtemp.out.

The ground heat exchanger model written in the VBA program basically write the input file such as simtemp.dfn and simtemp.bnd and run the modesim.exe, then reads the output from the file simtemp.out.. The simtemp.dfn is written from base.dfn (provides the format for the dfn file) and Type620.par (provide the ground heat exchanger parameters) by running the subroutine CompileDfnFile. Users can generate the Type620.par from GLHEPRO program. The simtemp.bnd is written from the output of the heat pump model, the heat transfer rate between the ground heat exchanger and heat pump q_s and the mass flow rate of the ground heat exchanger by running Subroutine WriteInputToGHXModel..The it reads the output such as GHX E_xFT and *EFT* from the file simtemp.out. The subroutine of the GHX model is provided in APPENDIX C.

Appendix A: The Main Subroutine

```
Sub MainDrive()

' ===============

'  PURPOSE OF THIS SUBROUTINE

'  To calculate the entering and exiting fluid temperature and ground heat exchanger for the

'  Ground Source Heat Pump System

'*************************************************************************

'  METHDOLOGY EMPLOYED

'  None

'  REFERENCES

'  None

'  CREATED BY

'  Lu Xing

'  Mechanical and Aerospace Engineering

'  Oklahoma State University

'  DATE WRITTEN

'  Dec. 14, 2012

'  REVISED BY

'  None

'  DATE REVISED

'  None

'  REVIEWER

'  None

'*************************************************************************==========================================
===================

'Input

Dim C_ABS(3) As Double, C_REJ(3) As Double

Dim BuildingHeatingLoads(HourInAYear) As Double, BuildingCoolingLoads(HourInAYear) As Double,
MassFlowRateFluidGHX(HourInAYear) As Double

'Internal variables

Dim ExFTGHX(NumOfIteration, HourInAYear) As Double, EFTGHX(NumOfIteration, HourInAYear) As Double,
EFTHeatPump(NumOfIteration, HourInAYear) As Double

Dim HeatExtractionRateOfGHX(NumOfIteration, HourInAYear) As Double
```

```
Dim QTotalHeating(NumOfIteration, HourInAYear) As Double

Dim MaxDifference(NumOfIteration) As Double


'Read the input of the model

Call GetInputParameters(C_ABS, C_REJ, BuildingHeatingLoads, BuildingCoolingLoads, MassFlowRateFluidGHX)


'Write the simtemp.dfn file from type 620.par

Call CompileDfnFile


IterationGHX = 0

Do

   IterationGHX = IterationGHX + 1

   For Hour = 1 To HourInAYear

   'update the EFTHeatPump

   If IterationGHX = 1 Then

   EFTHeatPump(IterationGHX, Hour) = EFTInitial

   Else

   EFTHeatPump(IterationGHX, Hour) = ExFTGHX(IterationGHX - 1, Hour) 'The entering heat pump temperature is the same as the exiting GHX temperature

   End If


   'Calculate the total heating rate needs to be provided by the ground heat exchanger for each hour

   Call HeatPumpModel(C_ABS, C_REJ, BuildingHeatingLoads(Hour), BuildingCoolingLoads(Hour), EFTHeatPump(IterationGHX, Hour), QTotalHeating(IterationGHX, Hour))

   Next Hour


   'Run the GHX model, reading heat transfer rate on the source side as inputs, output the EWT, entering fluid temperature to the heat pump

   Call RunGHXModel(IterationGHX, MassFlowRateFluidGHX, QTotalHeating, EFTGHX, ExFTGHX, HeatExtractionRateOfGHX)


   If IterationGHX <> 1 Then
```

```
    'Calculate the maximum difference between TempOutletGHX of last iteration and this iteration between 8760 hours

    Call IterationDifferenceCalc(IterationGHX, ExFTGHX, MaxDifference(IterationGHX))

  End If


Loop While (MaxDifference(IterationGHX) > CovergeCriteria Or IterationGHX = 1)

Call WriteOutput(IterationGHX, EFTGHX, ExFTGHX, HeatExtractionRateOfGHX)


End Sub
```

Appendix B: Heat Pump Model

```
Sub HeatPumpModel(ByRef C_ABS() As Double, ByRef C_REJ() As Double, QH As Double, QC As Double, EWT As Double, QTotalHeating As Double)

' ===============

'  PURPOSE OF THIS SUBROUTINE

'  To calculate the COP and heat transfer rate on the load and source side

'  based on known EFT on load and source side and volume flow rate on load

'  and source side


'****************************************************************************

'  METHDOLOGY EMPLOYED

'  None


'  REFERENCES

'  None


'  CREATED BY

'  Lu Xing

'  Mechanical and Aerospace Engineering

'  Oklahoma State University
```

```
'  DATE WRITTEN

'  Dec 14, 2012


'  REVISED BY

'  None


'  DATE REVISED

'  None


'  REVIEWER

'  None

'************************************************************************==============================================
====================


'Inputs

'C_ABS:              The curve fit coefficients that give the ratio of heat extraction to heating load as a function of EWT  (-)

'C_REJ:              The curve fit coefficients that give the ratio of heat rejection to cooling load as a function of EWT    (-)

'QH:                 The heating required by the system   (KW)

'QC:                  The cooling required by the system   (KW)

'EWT:                 Entering Fluid Temperature enter the heat pump (C)


'Internal variables

'QHeating:            Heating load of the ground heat exchanger(KW)

'QCooling:            Cooling load of the ground heat exchanger(KW)


'Outputs

'QTotalHeating :               The total heating load needs to be provided by the ground heat exchanger(KW)


Dim QHeatingRate As Double, QCoolingRate As Double


QHeatingRate = QHeating(C_ABS, QH, EWT)
```

```
QCoolingRate = QCooling(C_REJ, QC, EWT)

QTotalHeating = QHeatingRate - QCoolingRate

End Sub

Function QHeating(ByRef C_ABS() As Double, QH As Double, EWT As Double)
'To determine the Heat extraction according to certain heat input(QH) entering the heat pump(w/m)
'Lu Xing
'Last updated 10/20/2008

'Inputs
'C_ABS:          The curve fit coefficients that give the ratio of heat extraction
'                to heating load as a function of EWT  (-)
'QH:             The heating required by the system   (w)
'EWT:            Entering Fluid Temperature enter the heat pump (C)
'EWTF:           Entering Fluid Temperature enter the heat pump (F)

'Internal variables
'Heat:           The ratio of heat extracted to heat(QH) provided by the heat pump (-)

'Outputs
'QHeating:       Heating load exit the heat pump towards the ground(W/m)

Dim i As Integer
Dim Heat As Double, EWTF As Double
Dim F() As Double

ReDim F(3)
```

```
'curve C_ABS is served in temperature with unit F
EWTF = 32 + (9 / 5) * EWT

Heat = 0

'when EWTF is smaller than zero, put it in right order
If EWTF < 0 Then
   EWTF = 40
End If

F(1) = 1
F(2) = EWTF
F(3) = EWTF ^ 2

'calculate the heat load exit the heat pump and towards the ground
For i = 1 To 3
   Heat = Heat + C_ABS(i) * F(i)
Next i

QHeating = Heat * QH

End Function
Function QCooling(ByRef C_REJ() As Double, QC As Double, EWT As Double)
'To determine the Heat Rejection according to the Cool input(QC) entering the heat pump(w/m)
'Lu Xing
'Last updated 10/20/2008

'Inputs
'C_REJ:          The curve fit coefficients that give the ratio of heat rejection to
'             cooling load as a function of EWT    (-)
```

```
'QC:            The cooling required by the system   (w)

'EWT:            Entering Fluid Temperature enter the heat pump (C)

'EWTF:            Entering Fluid Temperature enter the heat pump (F)



'Internal variables

'Cool:            The ratio of Heat rejected to cool(QC) provided by the heat pump (-)


'Outputs

'QCooling:            Cooling load exit the heat pump towards the ground(W/m)


Dim i As Integer

Dim Cool As Double, EWTF As Double

Dim F() As Double


ReDim F(3)


'change the unit from C to F

EWTF = 32 + (9 / 5) * EWT


Cool = 0


'when EWTF is smaller than zero, put it in right order

If EWTF < 0 Then

   EWTF = 40

End If


F(1) = 1

F(2) = EWTF

F(3) = EWTF ^ 2
```

```
'calculate the heat and cool load exit the heat pump and towards the ground

For i = 1 To 3

   Cool = Cool + C_REJ(i) * F(i)

Next i

QCooling = Cool * QC

End Function
```

Appendix C: Run GHX Model

```
Sub RunGHXModel(IterationGHX As Integer, _
            ByRef MassFlowRateFluidGHX() As Double, ByRef QTotalHeating() As Double, _
            ByRef EFTGHX() As Double, ByRef ExFTGHX() As Double, ByRef HeatExtractionRateOfGHX() As Double)
' ===============
'  PURPOSE OF THIS SUBROUTINE
'  To calculate the outlet fluid temperature from the ground heat exchanger, if we know the building heating and cooling loads,
'  This is done by conecting a ground heat exchanger (Type 620 in HVACSIM+)to a idea heater(Type 643 in HVACSIM+)

'***************************************************************************
'  METHDOLOGY EMPLOYED
'  None
'  REFERENCES
'  None
'  CREATED BY
'  Lu Xing
'  Mechanical and Aerospace Engineering
'  Oklahoma State University
'  DATE WRITTEN
'  Dec 14, 2012
'  REVISED BY
'  None
'  DATE REVISED
'  None
'  REVIEWER
'  None
'***************************************************************************================================================================
'***************************Call the exe*********************************************************
Dim Status As Variant
Dim FilePath  As String

'Write the simtemp.bnd file
Call WriteInputToGHXModel(IterationGHX, MassFlowRateFluidGHX, QTotalHeating)

 'change the default directory to current dictory
  FilePath = ThisWorkbook.Path
  ChDrive FilePath
  ChDir FilePath

  'run the exe file

  Status = ExecCmd(ThisWorkbook.Path & "\Modsim.exe")
```

```
 'read the output fromt he exe file
 If Status = 1 Then

   Call ReadOutputOfGHXModel(IterationGHX, EFTGHX, ExFTGHX, HeatExtractionRateOfGHX)

 Else

   MsgBox "It failes to run the exe"

 End If

End Sub
```

VITA

Benjamin William Rhoda

Candidate for the Degree of

Master of Science

Thesis: INVESTIGATION OF PUMPING AND PIPING SYSTEM DESIGN FOR COMMERCIAL GROUND SOURCE HEAT PUMPS

Major Field: Mechanical Engineering

Biographical:

Education:

Completed the requirements for the Master of Science in Mechanical Engineering at Oklahoma State University, Stillwater, Oklahoma in December, 2013.

Completed the requirements for the Bachelor of Science in Mechanical Engineering at the University of Minnesota, Minneapolis, Minnesota/USA 2006.

Experience: United States Air Force: May 2006 to Present. In which time, May 2006 to May 2009, served as a Mechanical Engineer at Grand Forks Air Force Base in Grand Forks, North Dakota. May 2009 to May 2011, Served as a Project Engineer at Andersen Air Force Base, Yigo, Guam. May 2011 to present. Studied Mechanical Engineering at Oklahoma State University.

May 2006 to May 2009 – Mechanical Engineer

Professional Memberships: ASHRAE Student Member,

Name: Benjamin William Rhoda Date of Degree: December, 2013

Institution: Oklahoma State University Location: Stillwater, Oklahoma

Title of Study: INVESTIGATION OF PUMPING AND PIPING SYSTEM DESIGN FOR COMMERCIAL GROUND SOURCE HEAT PUMPS

Pages in Study: 158 Candidate for the Degree of Master of Science

Major Field: Mechanical Engineering

Scope and Method of Study: This study focused on the simulation of previously researched design methodologies and pumping strategies and compared them in detailed simulations to see the impacts of certain designs on the performance of the pumping energy and the performance of the heat pumps. This included both direct and reverse return systems, pipe sizing criteria, central and distributed pumping, and a combined central with distributed pumping, the effects of differential temperature set point, and differential pressure set point. This study also compared the traditional two-pipe distribution system to the one-pipe distribution system.

Findings and Conclusions: Through the analysis, the most important factor in pumping performance was the control mechanisms used and pump efficiency. The variance in energy consumption between the four configurations that were studied was at most 7%. Using 0.61 m (2 ft) of head loss per 30.5 m (100 ft) of pipe as a pipe sizing technique was the transition point for the pumping power to go from a 'B' to an 'A' grade. Ensuring the pump is equipped to be shut off when no heat pumps are operating can save significant amounts of energy. The one-pipe system had the lowest pumping energy of all four configurations but also had the highest heat pump energy consumption. As a result, the one-pipe system consumed the most energy for all four locations that were analyzed. The best performing system in all four locations was the two-pipe configuration with distributed-pumping-with-hydraulic-separation assuming a circulator pumping efficiency of 50% as is attainable with ECM pumps. The temperature differential set point to control the ground loop fluid flow rate varied depending on the location. If a standard induction motor pump is used, the two-pipe system with central pumping was the best performing configuration.

ADVISER'S APPROVAL: Dr J. D. Spitler