# ANALYSIS OF ENERGY AND SAVINGS OF USING GROUND LOOP OR STEAM TO CHANGE TEMPERATURE OF THE BULDING HEAT PUMP LOOP IN NORM ASBJORNSON HALL

by

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# GLOSSARY

| | |
|---|---|
| MSU | Montana State University |
| LEED | Leadership in Energy and Environmental Design |
| MMBTU | Million British Thermal Unit |
| PGS | Pump for Geothermal System |
| DWH | Domestic Water Heater |
| LWT | Leaving Water Temperatures |
| HX | Heat Exchanger |
| BT | Buffer Tank |
| HPL-RT | Heat Pump Loop- Return Temperature |
| EWT | Entering Water Temperature |
| GPM | Gallons Per Minute |
| HP | Heat Pump |
| NAH | Norm Asbjornson Hall |
| GSHP | Ground Source Heat Pump |
| DOAS | Dedicated Outdoor Air System |
| HPL | Heat Pump Loop |

# ABSTRACT

The need for efficient and sustainable environmental conditioning systems in buildings has become increasingly important in the face of rising energy costs and environmental concerns. This thesis aims to assess the optimization of the control logic to maximize energy savings and costs associated with utilizing ground loop or steam to modify the temperature of a heat pump loop in ground source heat pumps (GSHP) in Norm Asbjornson Hall (NAH) building at Montana State University (MSU).

The study begins by providing a comprehensive review of existing literature on GSHP systems, their working principles, and the various methods employed to alter the temperature of the heat pump loop. The research methodology involves determining the conditions under which it is economically viable to operate ground loops and/or a steam heat exchanger to maintain the heat pump loop temperature within a set operating range. This is done by deriving an equation that utilizes the coefficient of performance (COP) and entering water temperature (EWT) of the heat pump loop. Energy and cost analysis is then conducted to assess the energy efficiencies for different cases.

The findings reveal that both steam and ground loops can effectively alter the temperature of heat pump loops, providing enhanced temperature control and increased energy efficiency. The analysis shows that each strategy does have important financial and environmental implications, nevertheless. Due to the equipment, infrastructure, and operational expenditures, steam injection is primarily utilized to raise the loop's temperature for heating mode only, and at extreme situations when the ground loop cannot provide enough energy to maintain the heat pump loop temperatures. However, compared to steam injection, ground loops, which can be used for both heating or cooling, offer significant energy savings and lower long-term maintenance costs, albeit needing a sizable initial investment.

In summary, the thesis explores how to optimize control logic to save energy and costs using ground loop or steam to adjust building heat pump loop temperature. The study evaluates energy, cost, and environmental impact of the proposed control logic optimization approach. The findings aim to provide insights into informed decision-making regarding the adoption of this alternative method.

# CHAPTER ONE

# INTRODUCTION

## General Background

Geothermal energy plays a significant role in the sustainable and energy-efficient operations of Montana State University's (MSU) Norm Asbjornson Hall (NAH). A ground source heat pump (GSHP) system is used in the building to regulate the temperature by tapping into the earth's natural energy. GSHPs are among the most energy-efficient technologies for HVAC and water heating, using less energy than traditional methods. They transfer heat to or from the ground, taking advantage of the earth's constant temperatures. GSHPs are recognized as efficient and sustainable heating and cooling systems, offering long-term cost savings, and reduced environmental impact. They use significantly less energy than can be produced by burning fuel in a boiler or furnace or by using resistive electric heaters.

The GSHP system in Norm Asbjornson Hall consists of a network of underground pipes, known as ground loops or boreholes, which are drilled deep into the ground. These loops circulate a heat transfer fluid, typically a mixture of water and glycol, which extracts heat from the ground during the heating season and dissipates excess heat during the cooling season.

This process significantly reduces the reliance on traditional heating systems, such as combustion or electric resistance heaters, resulting in substantial energy savings. During the cooling season, the GSHP system reverses its operation. It extracts excess heat from the building and transfers it to the cooler ground, effectively using the earth as a heat sink. This process eliminates the need for traditional air conditioning systems, such as chillers or cooling towers, thereby reducing energy consumption and associated costs.

Moreover, using ground loop in NAH exemplifies MSU's dedication to sustainable building design and energy management. It serves as a model for integrating renewable energy systems into educational facilities, demonstrating the potential for widespread adoption of ground loop technologies in the heating-dominated region.

As the first LEED Platinum building on MSU's campus, NAH elevates sustainable design to a new level. Utilizing multiple unglazed transpired solar collectors and GSHP systems, the facility is predominantly heated and cooled by natural sources. (U.S. Green Building Council USGBC, n.d.)

A built-in data collection system tracks building performance for optimum long-term efficiency while serving as an instructional tool for students. Because the building typically generates more energy than it uses, excess heat can be transferred to support surrounding facilities. (U.S. Green Building Council USGBC, n.d.)

NAH Heat Pump Loop

Most energy systems in NAH utilize heat pumps connected by a common heat pump (HP) water circulating loop. This HP loop system consists of GSHPs, ground loops, storage tank, steam heat exchanger, and circulating pumps. The current system allows the HP loop temperature to float between minimum and maximum temperatures (currently 45°F-85°F) before activating the ground loops or steam heat exchanger.

Figure 1. Norm Asbjornson Hall

Problem Statement

The purpose of this thesis is to determine a control logic optimization approach for when to turn on the ground loop or steam heat exchanger in the building HP loop system to optimize the HP loop temperature for maximum efficiency and cost savings. The temperature of the HP loop must be maintained within a range of 45°F-85°F.

## Background Study

Joshua Talbert (2014) conducted a study where he performed an energy assessment of buildings using photographic-based model development and modeling calibration. The purpose of the study was to evaluate the effectiveness of modern energy modeling tools, such as EnergyPlus, in reducing the time required for accurate model development. The modeling results supported the idea that these tools can significantly reduce campus building-related greenhouse gas emissions. (J. Talbert, 2014)

Joshua Hays (2019) developed a method to assess a mini-district energy loop using historical utility data. The study involved creating a simple building model based on energy modeling from building construction specifications and inputs from the University Facilities

Engineers at MSU. The model was calibrated for heat demand and validated using metered data from a high-energy-use laboratory building on campus. Additionally, the model was converted to be heat pump compatible, enabling internal thermal energy sharing within the building. The research demonstrated that high-energy-use laboratories have significant potential for internal energy savings. As a recommendation, it was suggested that these buildings should be converted to heat pump-compatible buildings to maximize energy sharing capabilities. (J. Hays, 2019)

Oladayo (2021) conducted a study to identify various heat sources and heat sinks available in a building and assess the energy savings from connecting these systems in a heat pump loop. The research used Barnard Hall on campus as a case study, and if the results were promising, similar models could be adopted for other buildings and connected to an overall heat pump loop on campus. The project also provided ways of assessing or measuring the energy value of various systems in any building and determining the energy savings of connecting those systems in a heat pump loop. The study showed that this provided some energy savings opportunities. Increasing the COP of heat pumps can generate more savings. The COP for the heat pumps used in this simulation were operating within the range of 3.5 – 5.5; hence more energy savings can be generated if increased. (Oladayo Oladeji, 2021)

Overall, all three studies highlight the importance of energy modeling in assessing and optimizing building energy performance. They showcase the potential of modern energy modeling tools and techniques in reducing greenhouse gas emissions and maximizing building energy efficiency.

Typically, the working fluid circulating through a ground loop has a higher temperature than the outdoor air during winter and a lower temperature during summer. As a result, utilizing this fluid as a heat source, a ground loop proves to be more efficient than using outdoor air. When GSHPs share a common ground loop, with some units providing cooling and others providing heating, moving heat between the heat pumps becomes technically feasible. Moreover, the demand for multiple heat pumps within a shared ground loop is well-balanced and falls within a predictable range. In that case, the length of the ground heat exchanger can be significantly reduced. This reduction in size makes GSHPs a more economically viable option for urban properties that lack sufficient ground area to accommodate ground heat exchangers. (Reid Hart)

The preceding investigation had also observed that an elevation in the entering water temperature of the GSHP for hot water ($GSHP_{hw}$) resulted in a decline in the power demanded by the compressor for the identical condensation temperature, culminating in an enhanced COP for the $GSHP_{hw}$. Additionally, if the setpoint temperature for hot water is curtailed throughout the cooling season (which leads to a reduction in the condensation temperature of the $GSHP_{hw}$), the COP of the $GSHP_{hw}$ would experience a further increase. Compared to making mechanical modifications to the compressor frequency range, reducing the setpoint temperature for hot water is a more straightforward and practical approach. In the deployed system, the service hot water temperature had been invariably fixed at 122 °F during the summer and 131 °F in winter. Implementing an outdoor air reset control (OARC) for service hot water has the potential to augment the COP of the $GSHP_{hw}$, which can eventually lead to the reduction of yearly electricity consumption.

Utilizing GSHP systems can result in significant energy savings of up to 40% over a one-year operational period compared to conventional air-to-water heat pumps (Urchueguía, et al., 2008). However, to fully optimize the system's energy efficiency, it is crucial to consider the entire system, including auxiliary equipment consumption, rather than solely focusing on the heat pump itself. Achieving an integrated optimal operation requires adapting the heat pump capacity and additional equipment operation to match the thermal energy demand of the building.

Similarly, a paper with an example of a building in Cyprus is used to evaluate how building design and thermal sources directly affect primary energy saving. This study investigates the feasibility of implementing GSHP systems in Cyprus, focusing on single-family and multi-family buildings. The authors use EnergyPlus software to analyze these buildings' energy requirements for heating and cooling. The results indicate that multi-family buildings with lower surface area to volume (A/V) ratios offer greater energy, environmental, and economic benefits compared to single-family buildings with higher A/V ratios. GSHP systems can achieve significant primary energy savings, ranging from 1.0% to 7.3% in single-family buildings and 18.4% to 23.5% in multi-family buildings, with even higher savings in colder areas. Assessing CO2 emissions, substituting conventional systems with GSHP reduces emissions by 19.0% to 24.1% in single-family buildings. At the same time, the results vary for multi-family buildings, showing a decrease of 4.7% to an increase of 10.5%. Notably, the largest CO2 reductions occur in the mild climate conditions of Nicosia, while the cold area of Saittas experiences an increase in emissions. In conclusion, this study underscores the potential of GSHP systems for energy savings and CO2 emission reduction in Cyprus, highlighting the importance of considering building type and climate conditions when evaluating the feasibility and advantages of GSHP installations. (Michopoulos et al., 2016)

GSHPs, also known as geothermal heat pumps (GHPs), are highly regarded for their exceptional performance in heating, cooling, and water heating. These systems offer enhanced comfort, substantial electrical energy consumption and demand reductions, low maintenance requirements, and environmental benefits. When compared to air source heat pumps, GSHPs exhibit several advantages: they consume less energy, utilize the more stable energy source of the earth or groundwater instead of air, eliminate the need for supplemental heating in extremely cold temperatures, require less refrigerant, have simpler designs with lower maintenance needs, and are not exposed to weathering due to their location. However, the primary drawback is their higher initial capital cost, which can be around 30-50% more expensive than air-source units. This is mainly due to the additional expenses involved in burying heat exchangers or providing a well for accessing the energy sources (Hepbasli et al., 2003).Despite the higher upfront investment, the overall cost of GSHPs over their lifespan is lower, resulting in long-term savings.

GHPs use heat from the ground to reduce energy costs and carbon emissions associated with heating and cooling compared to alternative methods. The heat output from a GHP is typically 3 or 4 times greater than the electrical input, resulting in large energy cost savings and carbon savings. However, the extent of savings and whether savings will be achieved at all will depend on the design and operation of the system. The biggest design factor in determining the efficiency of the GHP system is the distribution system and the temperatures used in it. Controls also have a major bearing on the eventual system efficiency. Assuming the control system is well designed and installed and that the design temperatures have been chosen to maximize the efficiency of the system, then very high system efficiencies can result. GHPs are characterized by high capital costs and low operational costs compared to other HVAC systems. Their overall economic benefit depends primarily on the relative costs of electricity and fuels, which are highly variable over time and across the world. In applications where heating and cooling are required simultaneously for part of the year, further efficiency benefit can be derived by using heat pump products designed to do this. Carbon savings will depend on the carbon factor of electricity as well as the fuel it displaces in addition to the COP or seasonal coefficient of performance (SCOP) of the system.

## Heat Pump System

Heat pump systems play a pivotal role in providing efficient heating and cooling solutions in modern buildings, including university facilities. The heat pump loop, a crucial component of these systems, enables the exchange of thermal energy with the building's surroundings. Traditionally, the temperature control of the heat pump loop has been achieved through conventional means, such as boilers or cooling towers. However, technological advancements have opened new avenues for optimizing the efficiency and cost-effectiveness of heat pump systems.

This research is focused on two distinct approaches to temperature control: using ground loop or steam. The ground loop taps into the earth's subsurface to exchange heat with the heat pump loop. This geothermal method harnesses the stable temperature of the ground to regulate the heat pump loop temperature, offering an environmentally friendly and potentially cost-effective alternative. Furthermore, steam-based temperature control involves the introduction of steam into the heat pump loop, elevating its temperature. Steam, known for its high latent heat and widespread availability in university buildings, presents an intriguing solution for enhancing the overall efficiency of heat pump systems.

The primary objective of this study is to conduct a comprehensive analysis of the costs and savings associated with using steam or ground loop for temperature control in a university building's heat pump loop. This analysis encompasses various factors, including initial investment requirements, ongoing operational expenses, maintenance and repair costs, energy consumption, and the long-term financial benefits resulting from energy savings. By meticulously comparing these two approaches, we strive to determine the most economically viable and sustainable solution for temperature control in the heat pump loop of a university building.

The outcomes of this research hold significant implications for sustainable building design and energy management within the academic sector. Identifying the most cost-effective and energy-efficient method for temperature control in heat pump systems can pave the way for enhanced energy performance, reduced operational costs, and a diminished carbon footprint in university buildings.

Vapor Refrigeration Cycle in Heat Pumps

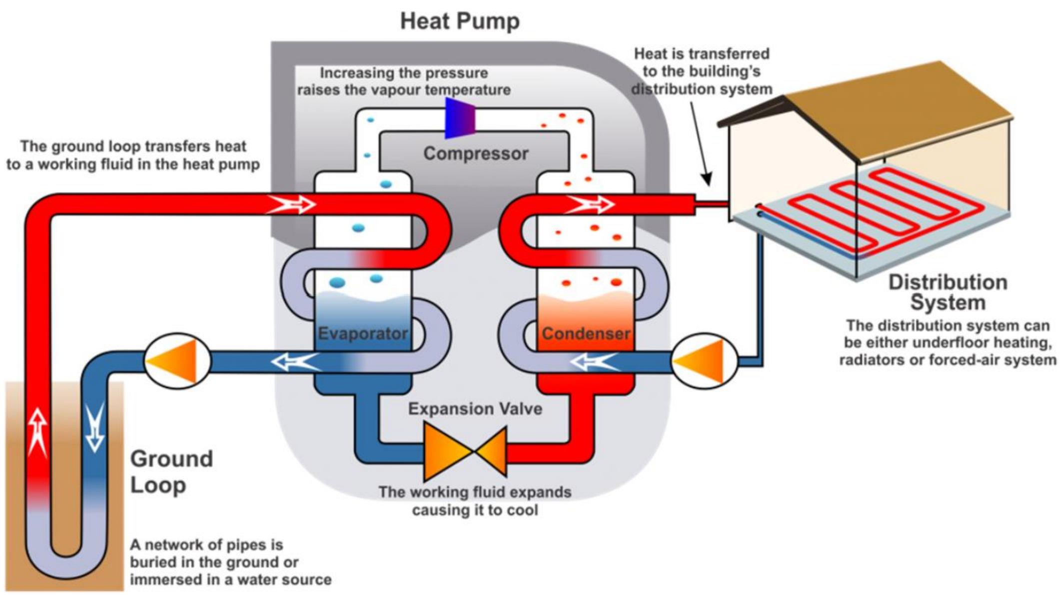


Figure 2. Schematic of vapor-compression refrigeration cycle in typical GSHP (Slagle, n.d.)

The vapor-compression refrigeration cycle is a process that uses the physics of phase change heat transfer and the unique properties of a refrigerant to transfer heat from a relatively cold source to a hot medium. Simply stated, refrigeration systems efficiently move heat from a cold source to a hot heat sink (normally air). The components in the basic refrigeration system are the compressor, condenser, expansion valve, and evaporator. (Slagle, n.d.). Figure 2 shows a schematic of vapor-compression refrigeration cycle in typical GSHP.

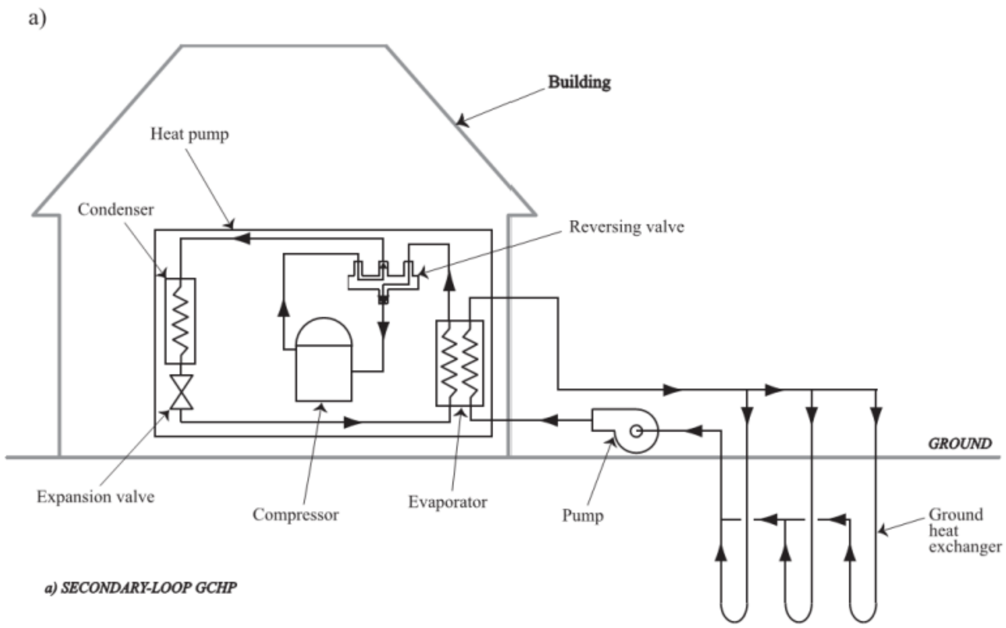


Figure 3. Schematic of Closed Loop Ground Source Heat Pump

GSHP primarily use water-source heat pumps (WSHP), which makes them unique, efficient, and usually more expensive to install for each ground loop. A conventionally designed water-source heat pump system would incorporate a boiler as a heat source during the winter heating operation and a cooling tower to reject heat during the summer cooling operation. This system type is also sometimes called a boiler/tower water-loop heat pump system. The water loop circulates to the entire water source heat pumps connected to the system. The boiler and the cooling tower provide a constant water-loop temperature, allowing the water-source heat pumps to operate efficiently. Figure 3 shows a schematic of closed loop GSHP.

GSHPs take advantage of the earth's thermodynamic properties and use the ground (or in some cases groundwater) as the heat source during the winter heating operation and as the heat sink during the summer cooling operation. The average ground temperature remains consistent throughout the year, resulting in significantly better efficiencies for a GSHP as opposed to an air-source heat pump. The GSHPs may be subject to higher temperature fluctuations than conventional water-source heat pumps but not as high as air-source heat pumps. Consequently, most manufacturers have developed extended-range systems that operate more efficiently while subjected to the extended-temperature range of the water loop. Like WSHPs, GSHPs use a water loop between the heat pumps and the heat source/heat sink (the earth). The primary exception is the direct-expansion ground-source heat pump. Temperatures below the ground surface do not fluctuate significantly through the day or the year, as do ambient air temperatures. For this reason, the GSHPs remain highly efficient throughout the year in virtually any climate. The optimization of GSHP systems requires a comprehensive understanding of the factors that influence their performance and energy efficiency. The studies discussed above provide valuable insights into the thermal properties of the ground heat exchanger, the compressor energy savings, and the cost of steam, which are crucial in determining the optimal conditions for using the steam heat exchanger in the heat pump loop of a GSHP system.

Several types of heat pumps are available, each with its advantages and disadvantages. Here is a summary of the information found in the search results:

1. Air-source heat pumps: The most common type of heat pump, which transfers heat between the house and the outside air. They can reduce electricity use for heating by approximately 50% compared to electric resistance heating such as furnaces and baseboard heaters. They can also

dehumidify better than standard central air conditioners, resulting in less energy usage and more cooling comfort in summer months.

2. WSHPs are divided into water-to-water and water-to-air. Water-to-water heat pumps are used to heat and cool water.

3. GSHPs use the constant temperature of the earth as the exchange medium instead of the outside air temperature. They are more expensive to install but can be more efficient than air-source heat pumps.

4. Ductless mini-split heat pumps are similar to air-source heat pumps but do not require ductwork. They are ideal for buildings without existing ductwork or for room additions.

5. Absorption heat pumps: These use heat or thermal energy as their energy source and are also called gas-fired heat pumps.

5. Hybrid heat pumps combine a heat pump with a gas furnace, providing efficient heating in cold climates.

In summary, several types of heat pumps are available, each with its advantages and disadvantages. The most common types are air-source, water-source, and geothermal heat pumps, but there are also ductless mini-split, absorption, and hybrid.

## Ground Source Heat Pump

GSHP is a system that utilizes the ground, groundwater, or surface water as a heat source and sink. Different types of GSHP systems are based on the type of exterior heat exchange system used. These include ground-coupled heat pumps (GCHPS), ground-water heat pumps (GWHPS), and surface-water heat pumps (SWHPS). GCHPS involves closed-loop piping systems buried in the ground, GWHPS uses open-loop piping systems with water wells, and

SWHPS utilizes closed-loop piping coils or open-loop systems connected to lakes, streams, or other reservoirs.

In a GSHP system, heat pumps are in buildings and cool by removing indoor heat and rejecting it to the ground loop. In heating mode, the process is reversed as heat is removed from the ground loop by the heat pumps and delivered to the building. GSHP systems have been traditionally used in residential buildings but are now increasingly being utilized in commercial and institutional sectors as well.

The economics of GSHP systems can be attractive in larger buildings due to their ability to provide comfort and high efficiency without the need for elaborate equipment and controls. Simple design approaches can offset the added cost of ground heat exchangers and reduce maintenance requirements. However, simply attaching a ground heat exchanger, groundwater loop, or surface-water coils to conventional water-cooled without taking into account specific features of the HVAC systems can result in higher installation costs, poor efficiency, and added maintenance requirements. Therefore, typical installation recommendations and conventional approaches need to be amended to fully take advantage of GSHP systems. (Kavanaugh)

## Ground Coupled Heat Pump

GCHPs are a subset of GSHPs and are often called closed loop ground source heat pumps. A GCHP refers to a system consisting of a network of heat pumps linked to a closed-ground heat exchanger buried in the soil.

# CHAPTER TWO

# METHODOLOGY

The methodology for the thesis includes a review of NAH building HP loop configuration. Furthermore, this thesis assesses the determination of cost and energy savings of using a ground loop and steam heat exchanger, to increase or decrease the building heat pump loop temperature within the current operating range in the heat pump loop.

The first section mentions that the HP loop temperatures are controlled between 45°F and 85°F in heating and cooling modes. The COP of the heat pumps is evaluated, and it is found that COP for HPs in heating mode increases as loop temperature increases, whereas in cooling mode as loop temperature increases, the COP decreases. Energy sources and costs are also evaluated, where steam from the MSU heating plant costs $10 / MMBTU (million BTU) and the electricity from the utility provider costs $0.10 / KWH. The cost analysis for the use of the steam heat exchanger to increase heat pump loop temperature within the current operating range is determined.

Likewise in the next section, the condition at which it makes sense to use the ground loop in the heat pump loop is evaluated. For the determination of ground temperature, the ground loop information from direct digital controls (DDC) for about six hours of ground loop operation on September 7, 2022, is provided, which is taken for reference for the calculation. The ground temperature (Tg) is determined from the initial temperature (To) after the circulating pump start. The heat transfer rate is determined and used to determine the UA (Heat Transfer Coefficient *Area) value. For ground heat exchanger calculation, a condensing or counterflow heat exchanger is used. The pump power and pump law determine the pump energy for the use in ground loop. Finally, with the analysis of sensitivity to change of value (COV) trending, it is concluded that the evaluation of UA value by this process can be a sensible approximation for the ground loops for the calculation.

Heat pumps are widespread in buildings as a critical component of energy systems. These heat pumps are connected through a shared circulating loop for heat pump water. This system includes various elements like heat pumps, ground loops, a storage tank, a steam heat exchanger, and circulating pumps. The purpose of this interconnected system is to efficiently transfer and distribute heat throughout the building.

To optimize the use of ground loops or steam heat exchanger in the heat pump loop in a building, a control logic optimization decision matrix is developed to determine different conditions that could make sense to alter the heat pump loop temperature.

## Energy and Cost Analysis for Use of Ground Loop and Steam Heat Exchangers in the HP Loop

A critical feature of the current system is its ability to allow the temperature of the heat pump loop to vary within a predefined range. Currently, this range is between 45°F and 85°F. The system ensures optimal performance and energy efficiency by allowing the temperature to fluctuate within these limits. It takes advantage of the ground loops, thermal storage capabilities and the heat pumps' ability to operate within a range of temperatures.

However, it's important to note that the system's specific temperature range and activation thresholds may vary depending on factors such as the building's design, particular requirements, and environmental conditions.

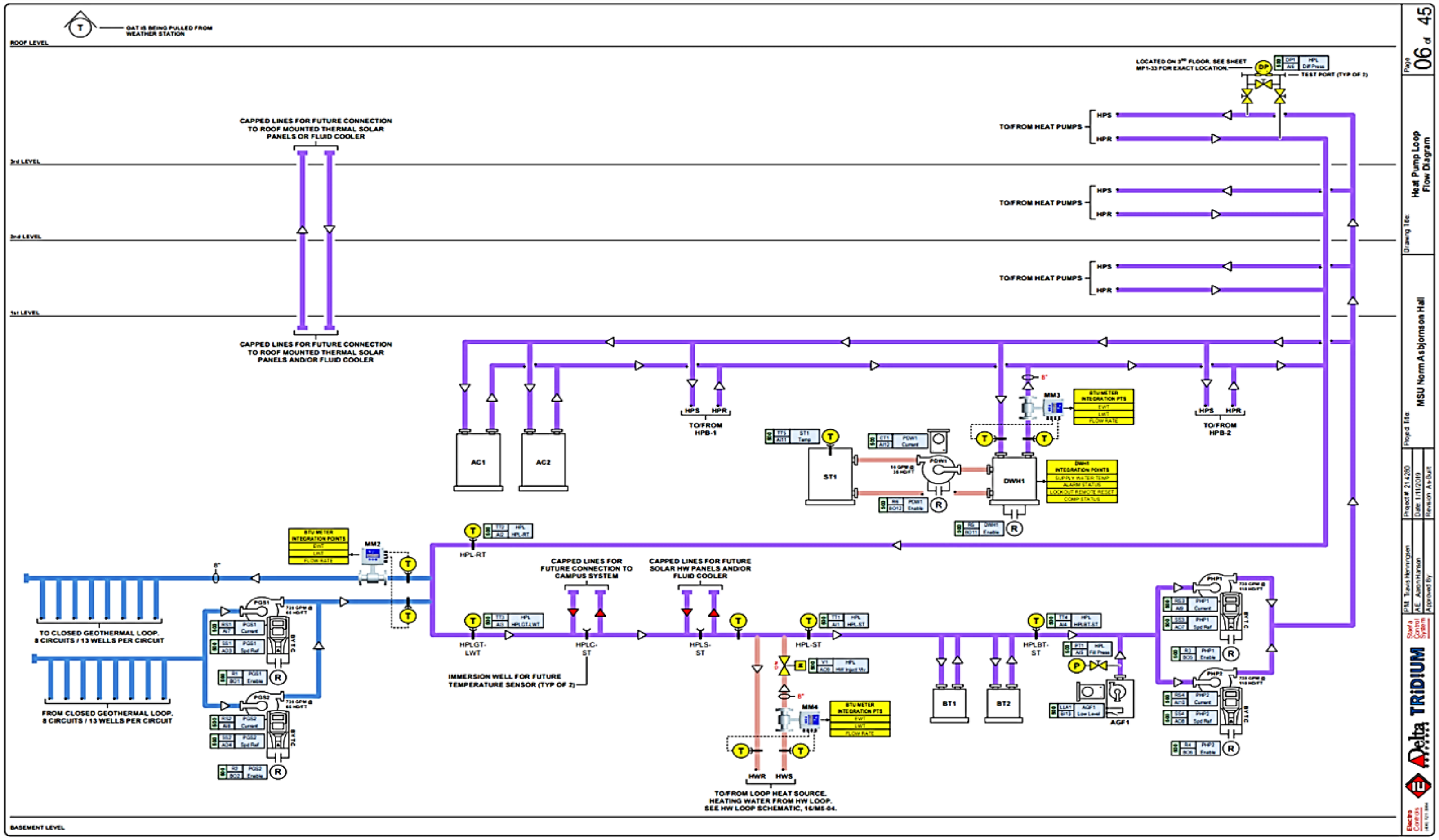

Figure 4. Heat Pump Loop Model in the NAH

Figure 4 above is simplified into Figure 5 showing the schematic Heat Pump Loop for NAH.

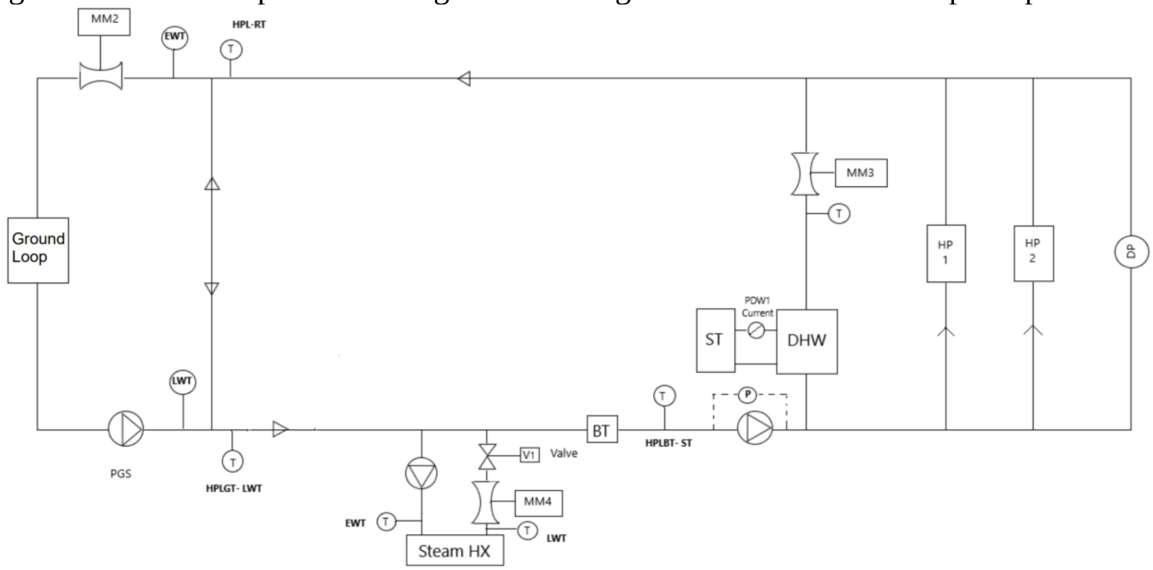


Figure 5. Simplified Schematic Heat Pump Loop in NAH

In the figure above,

| | |
|---|---|
| MM2/MM3/MM4 | Flowmeters |
| V1 | Valve |
| EWT | Entering Water Temperature |
| HPL | Heat Pump Loop |
| RT | Return Temperature |
| LWT | Leaving Water Temperatures |
| PGS | Pump for Geothermal System |
| HPLGT | LWT from Ground loop |
| Steam HX | Steam Heat Exchanger |
| BT | Buffer Tank |
| HPLBT-ST | Temperature leaving the Buffer Tank |
| P | Pump |
| DWH | Domestic Water Heater |
| HP1 | Heat Pump Loop 1 |
| HP2 | Heat Pump Loop 2 |

Figure 6. Ground Loop Water Supply and Ground Loop Water Return from Heat Pump Loop System in NAH

## Coefficient of Performance

The COP is a measure of the efficiency of a heat pump system. It is defined as the ratio of the desired heat output (in heating mode) or cooling output (in cooling mode) to the required input energy. When it comes to water source heat pumps, the entering water temperature can significantly impact the COP. The COP and its relationship with the entering water temperature for GSHP in both heating and cooling modes are discussed for both heating and cooling mode below.

In heating mode, the COP of a GSHP increases as the entering water temperature increases. This is because the heat pump has to work less hard to extract heat from the ground when the ground is warmer. A typical COP for a GSHP in heating mode is between 3 and 4, but COPs of up to 5 are possible with well-designed systems.

In cooling mode, the COP of a GSHP decreases as the entering water temperature increases. This is because the heat pump has to work harder to remove heat from the building when the ground is warmer. A typical COP for a GSHP in cooling mode is between 2 and 3, but COPs of up to 4 are possible with well-designed systems.

To the information about the COP characteristics of a similar model of GSHP used for NAH, in both heating and cooling modes with respect to entering water temperature, McQuay WSHP is referred for the reference to plot the COP graph with respect to the temperature. Figure 7 shows the COP for EWT in the HP loop for heating and cooling mode. This heat pump model provides close and detailed information based on the specific system in question.

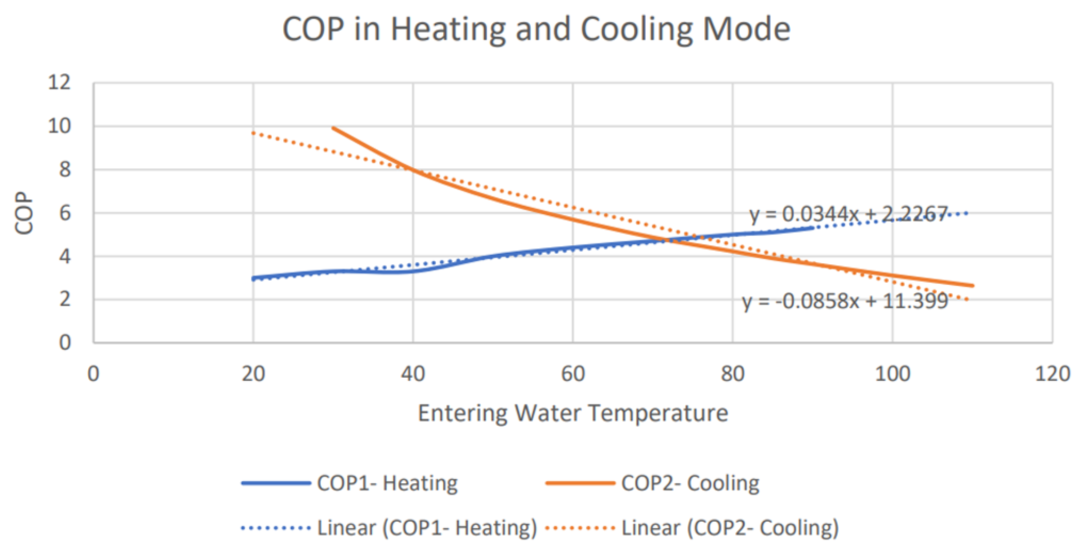


Figure 7. Coefficient of Performance (COP) w.r.t the Increasing Entering Water Temperature in Heating and Cooling Modes.

<u>Coefficient of Performance in Heating Mode.</u> COP represents the relationship between the cooling or heating effect produced by an air conditioner or heat pump and the net work input required for its operation. It is calculated by comparing the cooling or heating effect to the net compression work input, both measured in the same units. Firstly, the energy sources and costs are also evaluated, where steam from the MSU heating plant costs $10 / MMBTU (million BTU) and the electricity from the utility provider costs $0.10 / KWH. The cost analysis for the use of the steam heat exchanger to increase heat pump loop temperature within the current operating range is determined.

| EXPANSION TANK | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| PLAN CODE | SYSTEM | MFGR | MODEL NUMBER | ESTIMATED SYSTEM VOLUME | TOTAL VOLUME | ACCEPT. VOLUME | PRECHARGE PRESSURE (PSI) | SIZE |
| ET-1, 2, 3, 4 | HEAT PUMP LOOP | TACO | CA-700 | 17,300 GAL | 185 | 185 | 39 | 81"H X 30" DIA |

Figure 8. From Mechanical Schedules" M6-01", Expansion Tank.

System Volume for Steam Heat Exchanger (V) = Total Volume- Volume of Ground Heat Exchanger

Where, the total volume calculated for the entire Ground Heat Exchanger for Norm Asbjornson Hall from Major Geothermal System Designer is given to be 13098.3 gal.

Thus, V =17300 gal-13098.3 gal = 4200 gallons

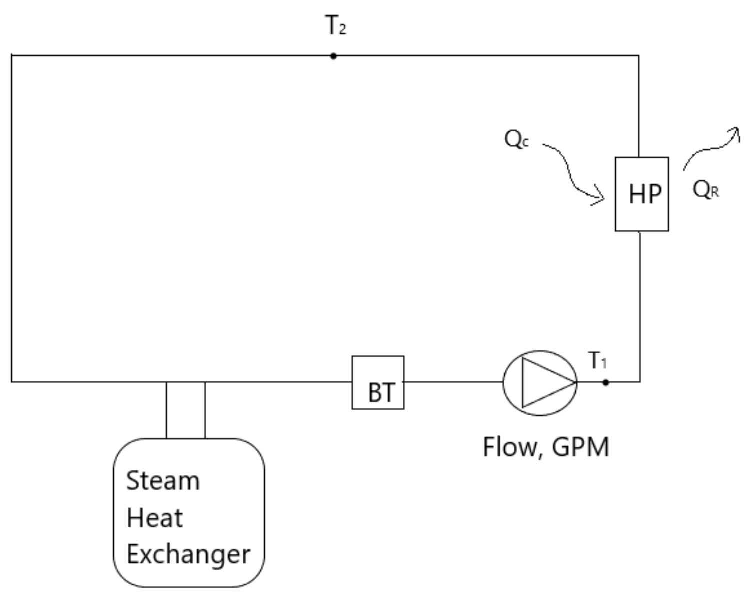


Figure 9. Heat Pump Loop with Steam Heat Exchanger

To estimate savings for a mechanical heat pump, we need to know how much energy we will save and its value. We can determine the relationships between work input and heat added to water in heating mode using a parameter known as the heat pump Coefficient of Performance ($COP_{Heating}$). Figure 9 shows a typical heat pump loop with steam heat exchanger. Equation (1) indicates $COP_{Heating}$is governed by the heat added to water in heating mode that is a function of the overall flowrate GPM and temperature difference in the HPL, and the net work into the system.

$\mathrm{COP}_{\mathrm{Heating}} = \frac{Q_{WH}+W}{W}$ , where $Q_{WH} = Q_R - Q_C = 500 * GPM * (T_s - T_R)$

(1)

Where,

| | |
|---|---|
| $Q_{wh}$ | Heat added to water in heating mode, |
| $Q_c$ | Heat Added by Compressor |
| $Q_R$ | Heat added in the room |
| W | Net Work put into the system |
| $T_s$ | Supply Temperature to the HP |
| $T_R$ | Return Temperature to the HP |

For 4200 gallons of system volume and $(\Delta T)=(T_1-T_2)=10\,^{\circ}F$, the cost of increasing heat pump loop water temperature using steam is

$$\text{Cost of using steam} = V * \frac{8.3\text{lbs}}{\text{gallons}} * \frac{1\text{Btu}}{\text{lb}^{\circ}\text{F}} * (\Delta T) * SC = \$3.487$$

The cost analysis for the use of steam heat exchanger to increase heat pump loop temperature within the current operating range is determined using the COP of the water source heat pump for both heating and cooling modes. The system volume is evaluated using the total volume minus the volume of the ground heat exchanger. The cost of increasing heat pump water temperature using steam is determined by using system volume and steam cost for delta T. Likewise, compressor energy saving of delta T in heat pump loop water temperature is determined by Heat added or removed (based on whether it is in heating mode or cooling mode or both) and COP in heating or cooling mode. Hence, the condition for the steam injection that makes economic sense is evaluated.

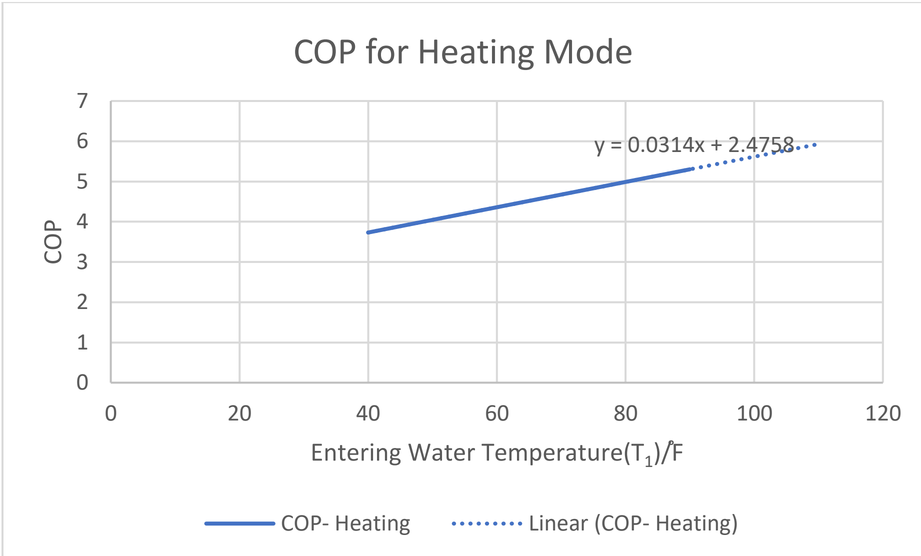


Figure 10. COP w.r.t to Increasing Entering Water Temperature in Heating Mode from 40°F.

Equation (2) is derived from the line of best fit (linear trend line) the COP as function of EWT of the HPL.

$$COP= 0.314(T_1) +2.4758 \tag{2}$$

$$COP_{Heating} = \frac{Q_{Wh} + W}{W}$$

$$W \quad = Q_{WH}\left(\frac{1}{COP_{heating} - 1}\right)$$

Electrical energy savings for arbitrary temperature difference($\Delta T$) in Heat Pump loop water temp = $Q_{WH}\left(\frac{1}{COP(T_1)-1} - \frac{1}{COP(T_2)-1}\right)$

To derive formula stating the relationship between steam cost (SC), electrical cost (EC), Entering Water Temperature ($T_1$), and system volume (V) that defines when it makes sense to use steam to increase loop temperature.

Total Saving in heating from T1 for $\Delta T$= 10 ̊F in the Heat Pump = Steam Cost

Equation (3) is for the condition where compression energy saving is greater than the cost of using steam heat exchanger for arbitrary ΔT.

$$Q_{WH}\left(\frac{0.0314\Delta T}{(0.0314(T_1)+1.4758)(0.0314(T_1+\Delta T)+1.4758}\right)*EC$$
$$> V*\frac{8.3lbs}{gallons}*\frac{1Btu}{lb}{}^{\circ}F*(\Delta T\_steam)*SC)$$

(3)

The heat added by water in heating mode $Q_{WH}$ is determined by the balance point equation above.

When ΔT= ΔT_steam = 10 ̊F and T1= 50 ̊F, $\mathrm{Q_{WH}} = 3{,}342{,}795.712$ Btu = 3.34 MMBtu, which is the balance point of the equation.

This means makes it sense to use Steam Heat Exchanger when the heat added by water in heating mode is more than 3.3 MMBTU at EWT of 50 ̊F and temperature difference between EWT is 10 ̊F. It is because in this condition the compressor energy saving is higher than the cost of steam. The first part of the equation gives the compressor energy saving for the temperature difference of ΔT in while the cost of steam is calculated using the system volume and ΔT.

Thus, when considering whether to use a steam heat exchanger in a water heat pump loop, the cost of steam and the compressor energy savings of the heat pump must be compared. If the compressor energy savings of the heat pump is more than the cost of using steam in the water heat pump loop, then it makes sense to use a steam heat exchanger.

<u>Coefficient of Performance in Cooling Mode.</u> The equation for COP in cooling is derived as Equation (4), for the condition in Figure 8. Figure 11 shows that COP in cooling mode decreases with the EWT in the heat pump loop.

$$COP_{Cooling} = \frac{Q_{WC} - W}{W}$$

(4)

$$COP_{Cooling} *W+W= Q_{WC}$$

$$W = Q_{WC}\left(\frac{1}{1 + COPc}\right)$$

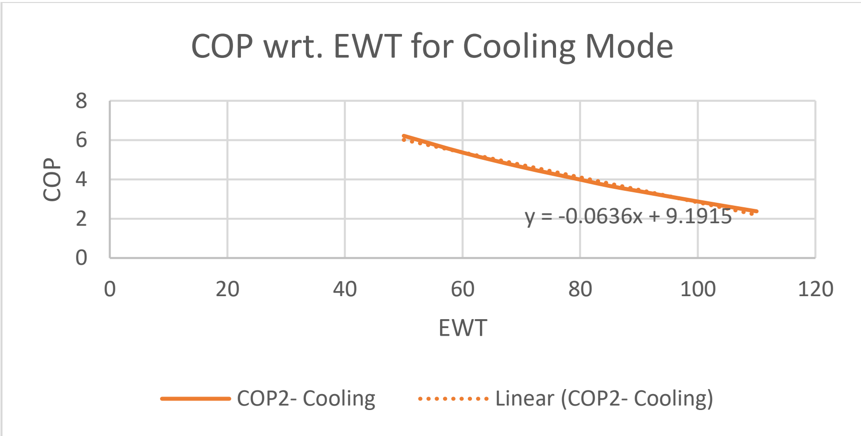


Figure 11. COP for Increase in Entering Temperature of Water and Trendline of the Change.

From the trendline of the change of COP in cooling mode in the graph above is used to derive the equation of COP as a function of the temperature. Equation (5) gives the linear trend equation for the COP for cooling mode which is a function of EWT in HPL.

$$COP_c= -0.0636T_1+9.1915$$

(5)

$$W = Q_{WC}\left(\frac{1}{(10.1915 - 0.0636T_1)}\right)$$

Electrical Energy Increase for a ΔT temperature difference in Heat Pump loop water temp

$$= Q_{WC}\left(\frac{1}{COP(T_1)+1} - \frac{1}{COP(T_2)+1}\right)$$

$$= Q_{WC}\left(\frac{1}{COP(T_1)+1} - \frac{1}{COP(T_1+\Delta T)+1}\right)$$

$$= Q_{WC}\left(\frac{0.0636\Delta T}{(-0.0636(T_1)+10.1915)(-0.0636(T_1+\Delta T)+10.1915)}\right) * EC$$

Equation (6) is for the condition where compressor energy saving HP loop energy increase in HP in cooling mode is greater than the cost of steam injection for arbitrary temperature difference ΔT. This is used for determining the equation for actual capacity of heat pumps in cooling mode as a function of overall flow in the heat pump loop and temperature rise.

$$Q_{WC}\left(\frac{-0.0636\Delta T}{(-0.0636(T_1)+10.1915)(-0.0636(T_1+\Delta T)+10.1915)}\right) * EC >$$

$$> V * \frac{8.3lbs}{gallons} * \frac{1Btu}{lb}{}^{\circ}F * (\Delta T\_steam) * SC) \tag{6}$$

For the calculation of the actual capacity of heat pumps in heating and cooling modes as a function of overall flow in the heat pump loop and temperature rise, the equation is given as

Q(Btu/hr) = 500 x Flow rate (GPM) x Temperature difference(°F) (7)

Where,

| | |
|---|---|
| Q(Btu/hr) | Heat transfer in the heat pump |
| GPM | Flow rate in Gallons per minute, |

$\Delta T_{HP} = T_{S(supply)}$-$T_{R(return)}$ Temperature difference between supply and return in Heat Pump (°F)

Conversion Factor =500 = 60 min/hr * 8.33 lbs/gallon * 1 Btu/lb°F.

For standard conditions in which density is 62.4 lb/ft$^3$ and specific heat is 1 Btu/lb°F.

Determining the proportions of heat pumps in heating and cooling modes.

Depending on the demand for heating and cooling in the building, the proportions of heat pump in heating and cooling modes varies. The proportion of heat pumps in heating mode versus cooling mode is an important metric for evaluating the performance of a heat pump system. This is because the efficiency of a heat pump can vary significantly depending on the mode that it is operating in. The proportion of heat pumps in heating and cooling modes is the function of the overall flowrate of the building heat pump loop and, the temperature difference between the building heat loop supply and return; and is evaluated for the temperature difference in the heat pump loop supply and return where $\Delta T_{HP}= T_{Supply} - T_{Return.}$

If all valves are 2-way type, HP is in heating mode then; it would have about 8°F temperature drop in HP Loop, and HP is in cooling mode then, it would have about 10°F temperature rise in HP Loop. The proportion of HPs in cooling and heating mode is determined based on temperature difference between heat loop supply and return. Likewise, the percentage of HPs in heating and cooling mode can be determined if HP Flowrate capacity is given.

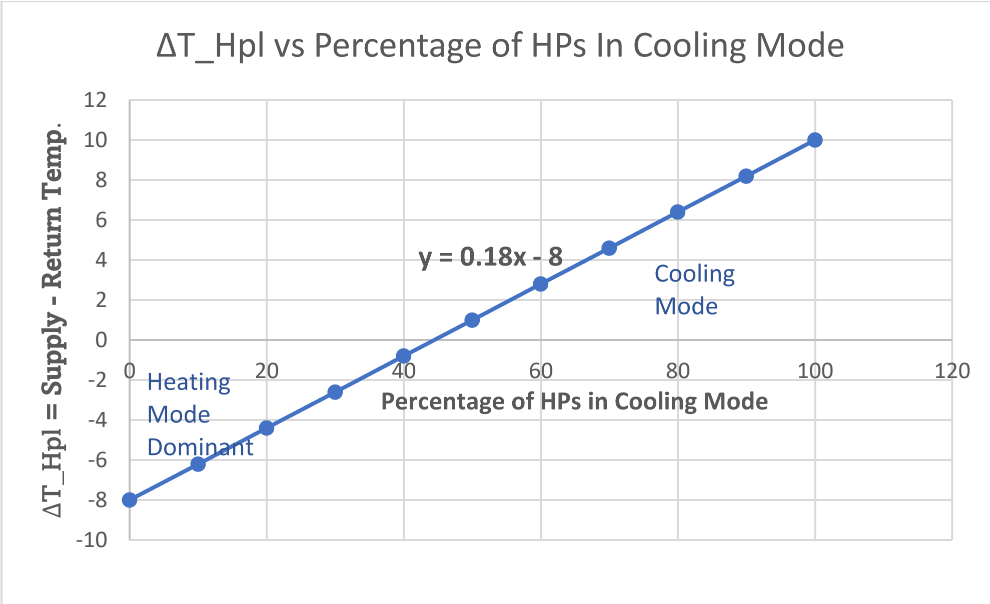


Figure 12. Graph for the temperature difference in proportion of HP in cooling mode

Figure 12 shows the graph the temperature difference as a function of the proportion of HPs in cooling mode. Equation (8) denotes the temperature difference between the supply and return of the heat pumps $\Delta T_{HP}$ to the proportion of HPs in cooling mode C. And the proportion of HPs in heating is 100-C.

$$\Delta T_{HP}=\frac{10*C-8*(100-C)}{100}$$

$$\Delta T_{HP}=0.18C-8 \quad (8)$$

For total nominal capacity from the given GPM, the HP flowrate capacity is given in Equation (9).

$$\text{HP Flowrate Capacity}= 3\ \text{GPM/Ton} \quad (9)$$

Table 1. Calculation of the proportion of HPs in cooling and heating mode using the graph

| GPM_HPL (Given: 3 GPM/Ton) | Total Nominal Capacity (Ton) | ΔT_HP | Proportion of HPs in Cooling Mode | Proportion of HPs in Heating Mode | Capacity in Cooling Mode (Ton) | Capacity in Heating Mode (Ton) |
|---|---|---|---|---|---|---|
| 140 | 46.67 | 10 | 100.00 | - | 46.67 | - |
| | | 8 | 88.89 | 11.11 | 41.48 | 5.19 |
| | | 6 | 77.78 | 22.22 | 36.30 | 10.37 |
| | | 4 | 66.67 | 33.33 | 31.11 | 15.56 |
| | | 2 | 55.56 | 44.44 | 25.93 | 20.74 |
| | | 1 | 50.00 | 50.00 | 23.33 | 23.33 |
| | | 0 | 44.44 | 55.56 | 20.74 | 25.93 |
| | | -1 | 38.89 | 61.11 | 18.15 | 28.52 |
| | | -2 | 33.33 | 66.67 | 15.56 | 31.11 |
| | | -2 | 33.33 | 66.67 | 15.56 | 31.11 |
| | | -6 | 11.11 | 88.89 | 5.19 | 41.48 |
| | | -8 | - | 100.00 | - | 46.67 |

Table 1 shows the proportion of HPs in heating mode and cooling mode is 1:1 at $\Delta T_{HP}$ 1 F and the flowrate of 140 GPM as highlighted. Equation (8) and (9) above are solved for the case where we have 720 overall hp loop water flow and 4 F HP Loop temperature rise. The percentage of HPs in cooling mode is solved using Equation (8).

$$C=\frac{\Delta T_{HP}+8}{0.18}$$

Hence, the condition for compressor energy saving that would make economic sense in the system with some heat pumps in heating mode and some in cooling mode is derived from GPM and $\Delta T_{HP}$ as equation (10.1) as a function of COP in heating mode and COP in the cooling mode.

$$\text{Energy Saving in Heating Mode} = Q_{WH}\left(\frac{1}{COP_h(T_1)-1}-\frac{1}{COP_h(T_1+\Delta T)-1}\right)$$

$$\text{Energy Loss in Cooling Mode} = Q_{wc}\left(\frac{1}{COP_c(T_1)+1}-\frac{1}{COP_c(T_1+\Delta T)+1}\right)$$

Equation (10.1) gives when compressor energy cost is greater than the cost of steam injection for the condition some heat pumps are in heating mode, and some are in cooling mode.

$$\{Q_{WH}\left(\frac{1}{COP_h(T_1)-1}-\frac{1}{COP_h(T_1+\Delta T)-1}\right)-Q_{wc}\left(\frac{1}{COP_c(T_1)+1}-\frac{1}{COP_c(T_1+\Delta T)+1}\right)\}*\mathrm{EC}>$$

$$(\mathrm{V}*\frac{8.3\mathrm{lbs}}{\mathrm{gallons}}*\frac{1\mathrm{Btu}}{\mathrm{lb}}{}^{\circ}\mathrm{F}*(\Delta\mathrm{T\_steam})*\mathrm{SC}) \tag{10.1}$$

Where,

| | |
|---|---|
| $Q_{wh}$ | Heat added to water in heating mode. |
| $Q_{wc}$ | Heat removed from water in cooling mode. |
| GPM_HPL | Flowrate of the heat pump loop |
| $\Delta T_{HP} = T_{Supply} - T_{Return.}$ | Temperature difference in the heat pump loop supply and return. |

$Q_{WH} = \frac{1}{3}\mathrm{GPM} - \frac{\Delta T_{HP}+8}{0.18}$ and $Q_{WC} = \frac{\Delta T_{HP}+8}{0.18}$, derived from eq (10.1) for simplifying equation in terms of GPM and $\Delta T_{HP.}$

$$\{Q_{WH}\left(\frac{1}{COP_h(T_1)-1}-\frac{1}{COP_h(T_1+\Delta T)-1}\right)-Q_{WC}\left(\frac{1}{COP_c(T_1)+1}-\frac{1}{COP_c(T_1+\Delta T)+1}\right)\}*\mathrm{EC}>$$

$$\left(\mathrm{V}*\frac{8.3\mathrm{lbs}}{\mathrm{gallons}}*\frac{1\mathrm{Btu}}{\mathrm{lb}}{}^{\circ}\mathrm{F}*(\Delta\mathrm{T\_steam})*\mathrm{SC}\right)$$

OR, $\{(\frac{1}{3}GPM - \frac{\Delta T_{HP}+8}{0.18})\left(\frac{0.0314\Delta T}{(0.0314(T_1)+1.4758)(0.0314(T_1+\Delta T)+1.4758}\right) - (\frac{\Delta T_{HP}+8}{0.18})\left(\frac{0.0636\Delta T}{(-0.0636(T_1)+10.1915)(-0.0636(T_1+\Delta T)+10.1915)}\right)\}*EC > (V*\frac{8.3lbs}{gallons}*\frac{1Btu}{lb}{}^{\circ}F*(\Delta\mathrm{T\_steam})*SC$

$$\text{OR, }\{(\frac{1}{3}GPM\left(\frac{0.0314\Delta T}{(0.0314(T_1)+1.4758)(0.0314(T_1+\Delta T)+1.4758}\right) -$$
$$\frac{\Delta T_{HP}+8}{0.18}\left(\frac{0.0314\Delta T}{(0.0314(T_1)+1.4758)(0.0314(T_1+\Delta T)+1.4758}\right)\} -$$
$$(\frac{\Delta T_{HP}+8}{0.18})\left(\frac{0.0636\Delta T}{(-0.0636(T_1)+10.1915)(-0.0636(T_1+\Delta T)+10.1915)}\right)\} * \mathrm{EC} > (\mathrm{V} * \frac{8.3\mathrm{lbs}}{\mathrm{gallons}} * \frac{1\mathrm{Btu}}{\mathrm{lb}}{}^{\circ}\mathrm{F} *$$
$$(\Delta\mathrm{T\_steam}) * \mathrm{SC})$$

$$\frac{1}{3}GPM\left[(\left(\frac{0.0314\Delta T}{(0.0314(T_1)+1.4758)(0.0314(T_1+\Delta T)+1.4758}\right) -\right.$$
$$\frac{\Delta T_{HP}+8}{0.18}\left\{\left(\frac{0.0314\Delta T}{(0.0314(T_1)+1.4758)(0.0314(T_1+\Delta T)+1.4758}\right) -\right.$$
$$\left.\left(\frac{-0.0636\Delta T}{(-0.0636(T_1)+10.1915)(-0.0636(T_1+\Delta T)+10.1915)}\right)\right\}\Bigg] * \mathrm{EC} > (\mathrm{V} * \frac{8.3\mathrm{lbs}}{\mathrm{gallons}} * \frac{1\mathrm{Btu}}{\mathrm{lb}}{}^{\circ}\mathrm{F} * (\Delta\mathrm{T\_steam}) * \mathrm{SC})$$

(10.2)

Hence, the required condition that would make economic sense to turn on steam heat exchanger in the system with some heat pumps in heating mode and some in cooling mode is derived from GPM and $\Delta T_{HP}$ is given by Equation (10.3) simplified by Equation (10.2). We will evaluate the conditions in table 8 to see if there is any condition which would make sense to turn on steam heat exchanger.

$$\mathrm{GPM} > 3\left(\frac{0.0314\Delta T}{(0.0314(T_1)+1.4758)(0.0314(T_1+\Delta T)+1.4758}\right)\{\left(\mathrm{V} * \frac{8.3\mathrm{lbs}}{\mathrm{gallons}} * \frac{1\mathrm{Btu}}{\mathrm{lb}}{}^{\circ}\mathrm{F} * (\Delta \mathrm{T_{steam}}) * \mathrm{SC}\right)$$
$$+ \left(\frac{5.56(\Delta T+8)(0.0006\,T_1\,\Delta T^2 - 0.02330\,\Delta T^2 + 0.0006\,{T_1}^2\,\Delta T - 0.04660\,T_1\,\Delta T + 3.12289\,\Delta T)}{(0.0314(T_1)+1.4758)(0.0314(T_1+\Delta T)+1.4758)(0.0636(T_1)-10.1915)(0.0636(T_1+\Delta T)-10.1915)}\right)$$

(10.3)

<u>Ground Loop Properties</u>

For the determination of the current ground loop temperature in GSHP system, an iterative method is used by using the initial ground loop temperature when the ground loop was turned on and water temperature entering the ground loop at the current time.

The DDC information is used to assess the initial ground loop temperature (Ti) and final temperature (To) through ground loop; and the ground temperature ($T_g$). Figure 13 and Figure 14 indicated the time period the ground loop circulating pump is turned on from DDC information, for September 2022. This is taken as a reference to calculate the ground properties and heat transfer in the ground loop.

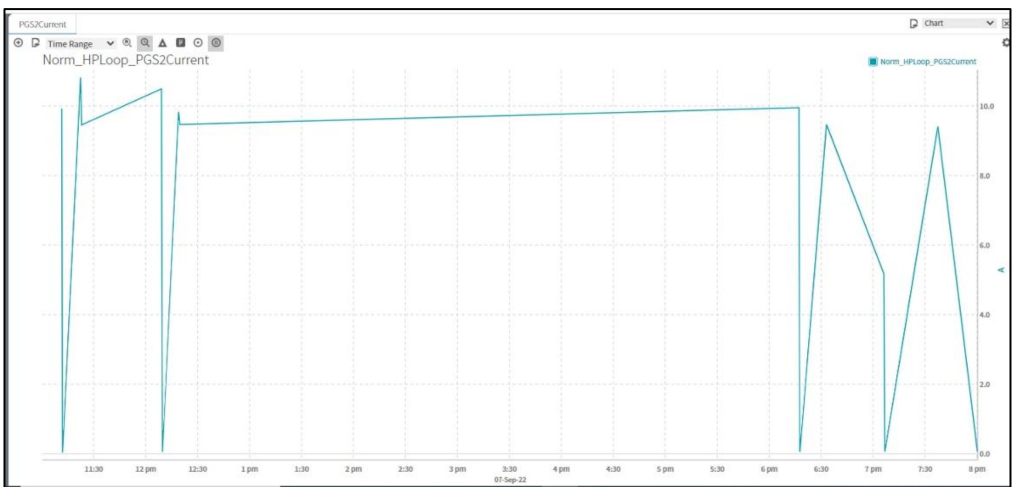


Figure 13 DDC trend for circulating pump PGS2 for ground loop current for time for the condition when the pump is on

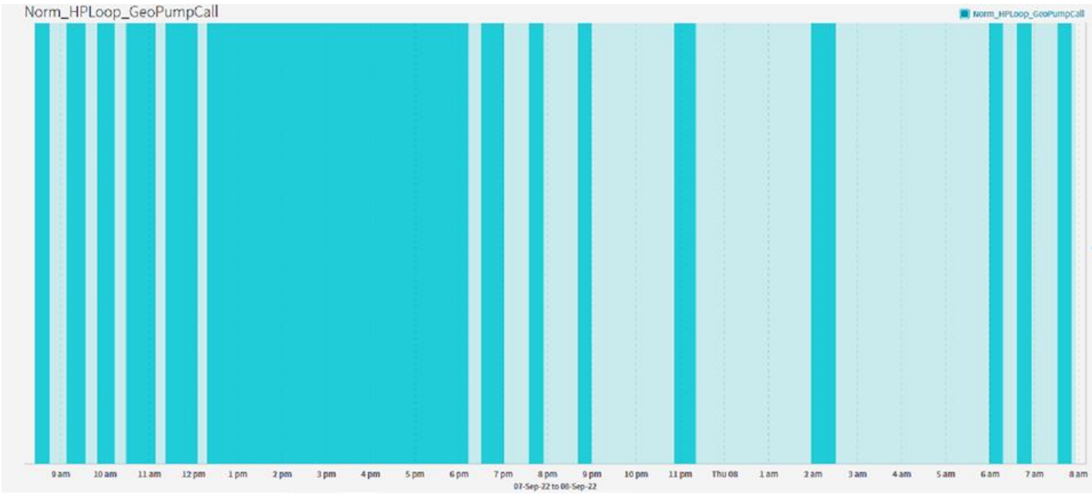

Figure 14. HP Loop Ground Loop Circulating Pump Call from DDC for the duration when the pump is on

The ground loop pump operating at about 10 amps during Sept 7, 2022, period based on the DDC trend and using the pump amperes graph, which is 20 HP pump about 27 FLA. Based on the average flowrate from database, the flowrate is assumed to be 300 GPM. The ground temperature $T_g$ can be determined from the initial temperature after starting the circulating pump. Figure 15 indicates that ground loop supply temperature ($T_i$) is 78°F. Figure 16 indicates return temperature ($T_o$) from HPL is 68°F and ground temperature $T_g$ is 66°F.

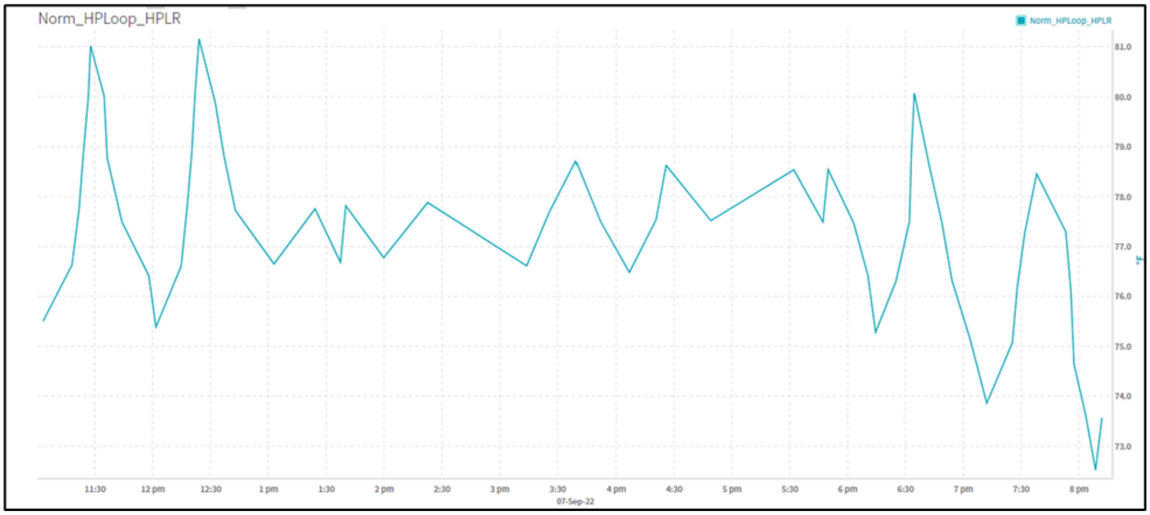


Figure 15. Supply Temperature of the ground loop over the period of 8 hours for 7th September 2022

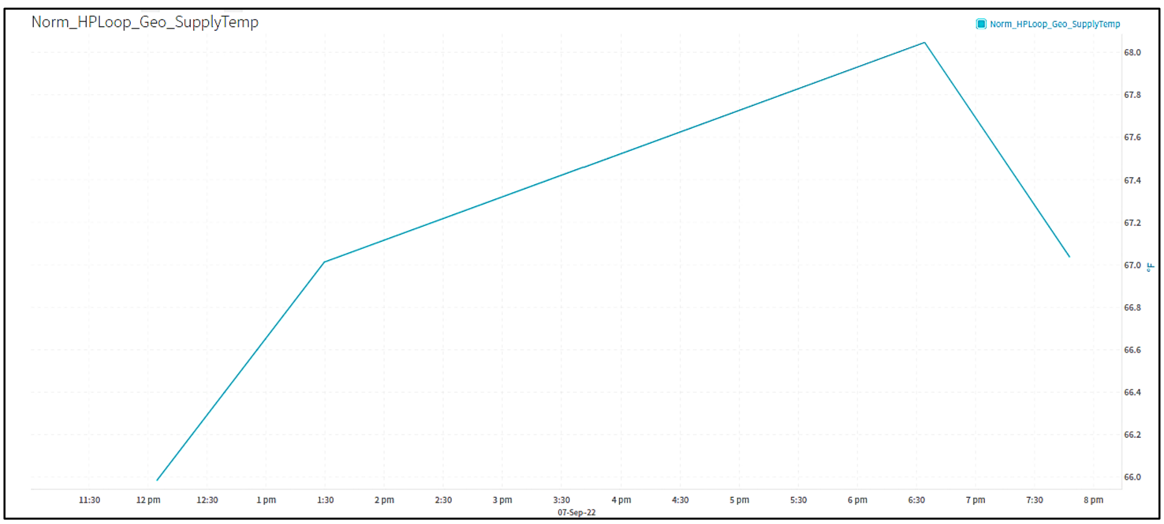


Figure 16. Return Temperature from the HP loop from 11:30 am to 8 pm on September 7th, 2022.

Heat Transfer in the Ground Loop

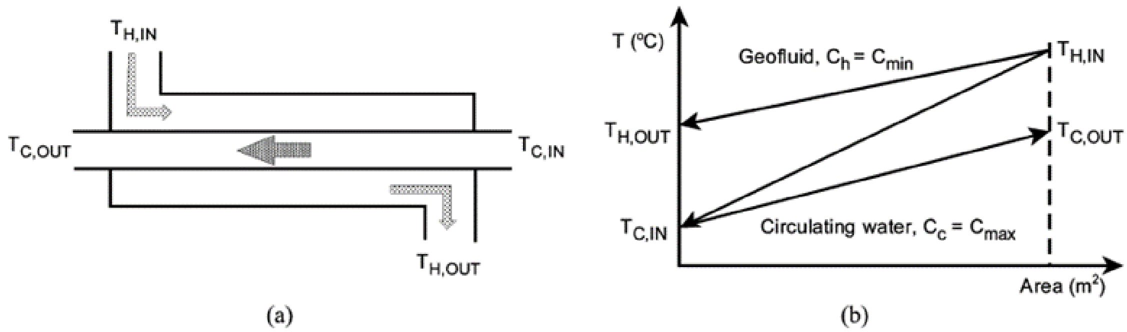


Figure 17. (a) Schematic and (b) Temperature Profile of Counter-Flow heat Exchanger

Figure 17 shows the schematic and temperature profile of a counterflow heat exchanger. The formulas used in the heat exchanger calculator are essential in determining the heat transfer rate and optimizing the heat exchanger design for maximum efficiency. The Equation (11) indicate the heat rate transfer Q is directly correlated to the overall coefficient of heat transfer from fluid to fluid U, and the heat transfer area A of the heat exchanger associated with U, and the log mean temperature difference (LMTD) Δtm,

$$Q = UA\,\Delta tm, \tag{11}$$

For a counterflow or condensing heat exchanger with a constant U, LMTD is calculated as Δtm as shown in equation (12), which is function of inlet and outlet temperatures of hot and cold fluids.

$$\Delta tm = Cf\,(T_{H,\,IN} - T_{C,OUT}) - (T_{H,OUT} - T_{C,IN}) / \ln(T_{H,IN} - T_{C,OUT})/(T_{H,OUT} - T_{C,IN}), \tag{12}$$

Where,

$T_{C,\,IN}$ Inlet Temperature of Cold Fluid

| | |
|---|---|
| $T_{C, OUT}$ | Outlet temperature of Cold Fluid |
| $T_{H, OUT}$ | Outlet temperature of Hot Fluid |
| $T_{H, IN}$ | Inlet Temperature of Hot Fluid |
| $C_f$ | Correction Factor for the Heat Exchanger |

The product of heat transfer coefficient and area of the ground loop (UA) is a constant for the ground loop that should not change.

The log mean temperature difference (LMTD) is a mathematical formula used in heat exchanger calculations to calculate the temperature difference between the hot and cold fluids. The LMTD is used to determine the heat transfer rate and is calculated using the inlet and outlet temperatures of both fluids.

The heat transfer rate is mostly governed by the product UA, which is the product of surface area and heat transfer coefficient. UA can always be improved by changing the heat exchanger design or just increasing the surface area. It is important to consider the correction factor, the overall coefficient of heat transfer, and the heat transfer area when calculating the rate of heat transfer. By optimizing the design of the heat exchanger, maximum efficiency can be achieved.

The formulas used in the heat exchanger calculator are as follows:

Q = FLOW x HEAT CAPACITY x FLUID TD, where Q is the rate of heat transferred, FLOW is the fluid flow rate, HEAT CAPACITY is the heat capacity of the fluid, and FLUID TD is the temperature difference between the inlet and outlet of the fluid.

The following steps are used to calculate the new outlet temperature:

1. Calculate the heat capacity of both fluids at the measured conditions.
2. Calculate the LMTD for the measured or design conditions.

3. Calculate the heat exchanger U x A (overall heat transfer coefficient multiplied by the heat transfer area).
4. Calculate the new heat exchanger effectiveness based on the NTU and type of heat exchanger.
5. Calculate the new outlet temperatures based on the new heat transfer rate and fluid flows. For example, the calculation of a 300 GPM (gallons per minute) flow rate gives a heat transfer rate of Q = 500 x GPM x DT = 500 x 300 x 10 F = 1.5 MMBH, which can serve as a reference for spreadsheet calculations.

Time t1 is the time shortly after starting the pump. Water temperature at outlet To is assumed to be ground temperature Tg1.Time t2 is about one hour after t1 and represents the time that stagnant water has cleared from wells and the system has stabilized. We assume that Tg has not changed since t1.

Calculate the UA of the ground loop system based on parameters at t2. The product of the heat transfer coefficient and area of the ground (UA) is a constant for the ground loop that should not change. Time t2a (not shown on in the graph above) Time between t2 and t3 that represents end of ground loop use period. Calculate ground temperature Tg2a using parameters at time t2a.Then, the time t3 is future time when want to know if should activate ground loop system. In the next step we assume temperature of the ground at this time Tg3 is same as calculated at time t2a. Lastly, it is determined if Tg3 can be used for heating or cooling the heat pump loop. And calculate expected Q3 based on Tg3=Tg2a, Ti, UA, and GPM.

Sensitivity to change of value (COV) trend. For the method used to determine the UA and heat transfer rate from spreadsheet shown in Table 2, the supply and return temperature of the ground loop with the analysis of sensitivity to change of value (COV) trending, it is concluded that the evaluation of UA value by this process can be a sensible approximation for the ground loop for now. However, it is important to consider the limitations of the analysis as it should also consider other factors such as the depth of the ground loop, the type of layers of ground, and the water flow rate through the ground loop.

Table 2. Heat Transfer Rate and UA Value with variation in Entering Water Temperature in Heat Pump Loop

| HX DESIGN OR MEASURED PARAMETERS | DESIGN PARAMETERS OR MEASURED VALUES | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| COLD FLUID INLET TEMP (F) | 76.4 | 77.0 | 77.4 | 78.0 | 78.6 | 77.0 | 77.0 | 77.0 |
| COLD FLUID OUTLET TEMP (F) | 67.0 | 67.0 | 67.0 | 67.0 | 67.0 | 66.1 | 66.3 | 66.6 |
| HOT FLUID INLET TEMP (F) | 66.0 | 66.0 | 66.0 | 66.0 | 66.0 | 66.0 | 66.0 | 66.0 |
| HOT FLUID OUTLET TEMP (F) | 66.2 | 66.2 | 66.2 | 66.2 | 66.2 | 66.2 | 66.2 | 66.2 |
| | | | | | | | | |
| TEMP DIFFERENCE AT INLET OF HX | -10.2 | -10.8 | -11.2 | -11.8 | -12.4 | -10.8 | -10.8 | -10.8 |
| TEMP DIFFERENCE AT OUTLET OF HX | -1.0 | -1.0 | -1.0 | -1.0 | -1.0 | -0.1 | -0.3 | -0.6 |
| CALCULATED LMTD OR MTD | -4.0 | -4.1 | -4.2 | -4.4 | -4.5 | -2.3 | -2.9 | -3.5 |
| | | | | | | | | |
| COLD FLUID FLOWRATE (CFM OR GPM) | 300.0 | 300.0 | 300.0 | 300.0 | 300.0 | 300.0 | 300.0 | 300.0 |
| COLD FLUID HEAT CAPACITY (0.92 FOR AIR, 500 FOR WATER) | 500.00 | 500.00 | 500.00 | 500.00 | 500.00 | 500.00 | 500.00 | 500.00 |
| COLD FLUID CONDUCTANCE, CCOLD = FLOW x CP | 150,000 | 150,000 | 150,000 | 150,000 | 150,000 | 150,000 | 150,000 | 150,000 |
| | | | | | | | | |
| HOT FLUID FLOWRATE (CFM, GPM, LB/HR) | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 |
| HOT FLUID HEAT CAPACITY (0.92 AIR, 500 WATER, 950 STEAM) | 1,000.00 | 1,000.00 | 1,000.00 | 1,000.00 | 1,000.00 | 1,000.00 | 1,000.00 | 1,000.00 |
| HOT FLUID CONDUCTANCE, CHOT = FLOW x CP | 1,000 | 1,000 | 1,000 | 1,000 | 1,000 | 1,000 | 1,000 | 1,000 |
| | | | | | | | | |
| **MEASURED HEAT TRANSFER (SET UP FOR COLD SIDE HX)** | **-1,410,000** | **-1,500,000** | **-1,560,000** | **-1,650,000** | **-1,740,000** | **-1,635,000** | **-1,605,000** | **-1,560,000** |
| **CALCULATED UA = Q/LMTD (OR MTD) OF HEAT EXCHANGER** | **355,931** | **364,216** | **369,493** | **377,071** | **384,280** | **715,447** | **547,766** | **442,057** |

The graph in Figure 18, is extracted from Table 2 to examine how the heat transfer rate and UA value vary in supply temperature. The variation of calculated UA value from the equation vs the measured heat rate value from the spreadsheet is compared. The temperature is varied by 0.1 ̊F.

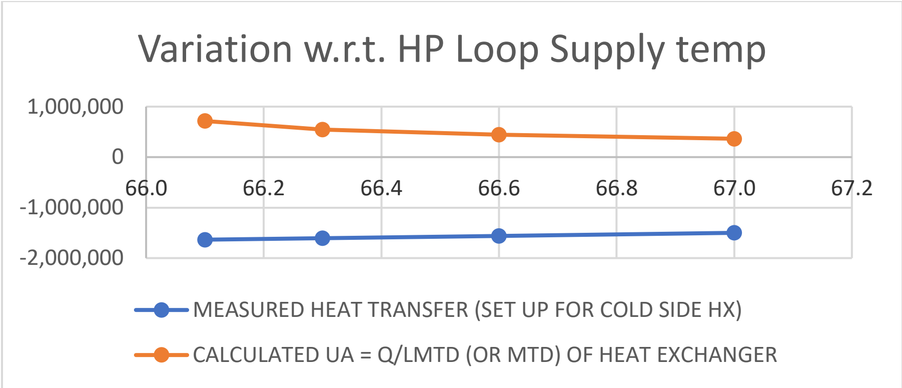


Figure 18. Heat Transfer Rate and UA Value with variation in inlet and outlet temperature from Table 2

Determination of accuracy from the review of the sensitivity to the COV trend. From Table 3 for EWT and LWT is a change of value trend, to review the sensitivity of the COV trend the cold inlet temperature is increased by 1°F in the second case compared to the first case, it shows the change in UA value by 23,000. Likewise, for the third case, the cold outlet temperature is decreased by 1° F, which shows a dramatic increase in UA value by increase of more than 100,000. Therefore, we can say that the change in HP loop return temperature (cold fluid outlet temperature) shows that the result is sensitive to the change in its value.

Table 3. Heat Transfer Rate and UA Value with HP loop Return temperature variation.

| HX DESIGN OR MEASURED PARAMETERS | DESIGN PARAMETERS OR MEASURED VALUES | | | |
|---|---|---|---|---|
| COLD FLUID INLET TEMP (F) | 77.0 | 77.0 | 77.0 | 77.0 |
| COLD FLUID OUTLET TEMP (F) | 66.1 | 66.3 | 66.6 | 67.0 |
| HOT FLUID INLET TEMP (F) | 66.0 | 66.0 | 66.0 | 66.0 |
| HOT FLUID OUTLET TEMP (F) | 66.2 | 66.2 | 66.2 | 66.2 |
| | | | | |
| TEMP DIFFERENCE AT INLET OF HX | -10.8 | -10.8 | -10.8 | -10.8 |
| TEMP DIFFERENCE AT OUTLET OF HX | -0.1 | -0.3 | -0.6 | -1.0 |
| CALCULATED LMTD OR MTD | -2.3 | -2.9 | -3.5 | -4.1 |
| | | | | |
| COLD FLUID FLOWRATE (CFM OR GPM) | 300.0 | 300.0 | 300.0 | 300.0 |
| COLD FLUID HEAT CAPACITY (0.92 FOR AIR, 500 FOR W/ | 500.00 | 500.00 | 500.00 | 500.00 |
| COLD FLUID CONDUCTANCE, CCOLD = FLOW x CP | 150,000 | 150,000 | 150,000 | 150,000 |
| | | | | |
| HOT FLUID FLOWRATE (CFM, GPM, LB/HR) | 1.0 | 1.0 | 1.0 | 1.0 |
| HOT FLUID HEAT CAPACITY (0.92 AIR, 500 WATER, 950 S | 1,000.00 | 1,000.00 | 1,000.00 | 1,000.00 |
| HOT FLUID CONDUCTANCE, CHOT = FLOW x CP | 1,000 | 1,000 | 1,000 | 1,000 |
| | | | | |
| **MEASURED HEAT TRANSFER (SET UP FOR COLD SIDE** | **-1,635,000** | **-1,605,000** | **-1,560,000** | **-1,500,000** |
| **CALCULATED UA = Q/LMTD (OR MTD) OF HEAT EXCH/** | **715,447** | **547,766** | **442,057** | **364,216** |

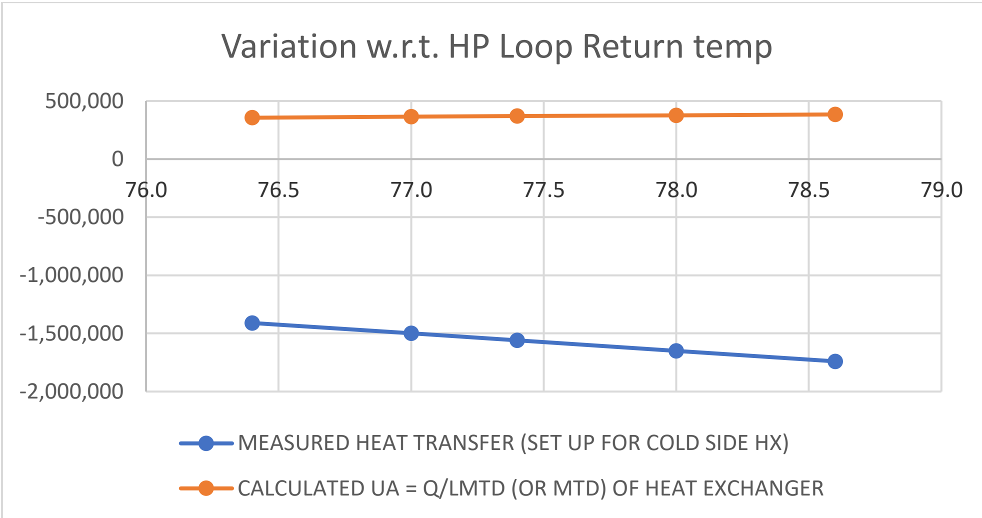


Figure 19. Heat Transfer Rate and UA Value with variation in ground temperature from Table 3

Figure 19 shows the variation in the heat transfer with respect to the increasing return temperature in HP loop is increasing linearly with the increase of temperature. However, UA value is nearly constant in this case.

However, increasing the HP loop supply temperature has greater fluctuation in the UA value. Additionally, the change in heat transfer value is at lower rate than the one in the first case.

Pump Energy for Ground Loop Circulating Pump.

Equation (13) indicated the pump energy calculation, where pump power is function of the flowrate in the circulating pump. The pump energy evaluation will allow us to determine the cost of using the ground loop.

$$\text{Pump Power} = \frac{\text{GPM\_p x } \Delta\text{h}}{3960 \text{ x } \eta_p} \tag{13}$$

Where,

| | |
|---|---|
| GPM_p | Water Circulating Rate (gallons per minute) |
| $\eta_p$ | Pump Efficiency |
| $\Delta h$ | Pump head (feet of water) |

For Ground Loop Circulation pump. The

GPM_p = water circulating pump (GPM) = 300 GPM

$\eta_p$ = Pump efficiency = 0.6 (value assumed for typical pump efficiencies)

$\Delta h$ = Pump head, feet of water = 65 feet of head (from the Mechanical Schedule for PGS)

Hence, Pump Power = $\frac{300*65}{3960*0.6}$ = 8.2 hp

Q(Btu/hr) = Heat transfer in the pump

= Time Period for the given system volume * Pump Power

= ($17,300$ gallons/720 gal/min) *(1h/60min) * 8.2 hp

= 3.28 hp.h = 2.45 Kw.h

The conditions that make economic sense to use ground loop for heating mode, cooling mode, and, both heating and cooling mode are evaluated. Thus, to determine the cost benefit of using a ground loop to increase or decrease heat pump loop temperature, firstly, the temperature difference differs from that used in the steam heat exchanger. Additionally, for the ground loop calculation, the pump energy used in the ground loop is determined using pump power and pump law. The equations for three conditions are listed below.

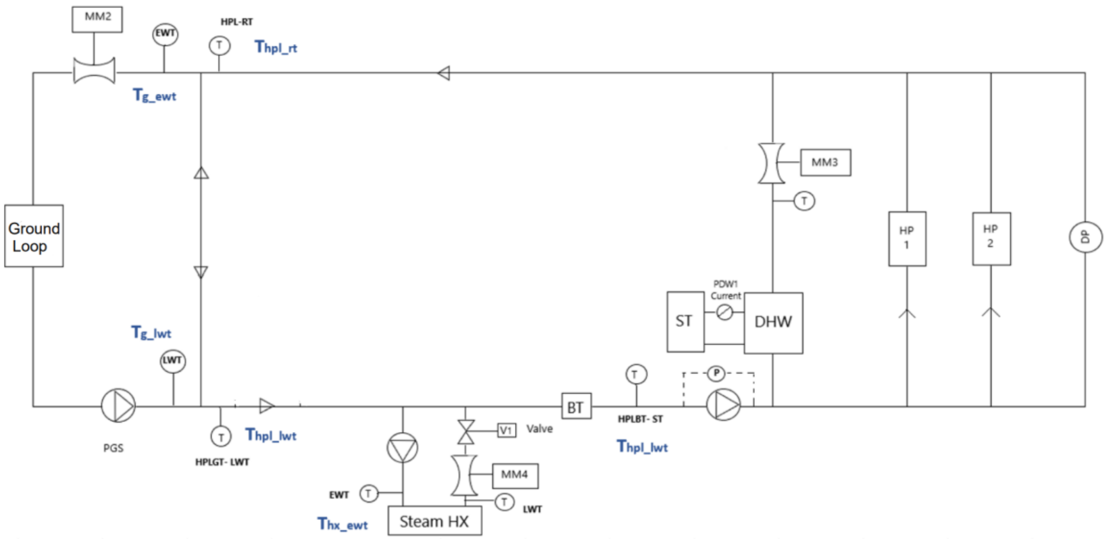


Figure 20. Schematic of GSHP with Temperatures Labeled for Ground Loop.

The temperatures in the heat pump loop are listed below,

| | |
|---|---|
| $T_{hpl\_rt}$ | Return water temperature from the heat pump loop |
| $T_{g\_ewt}$ | EWT in the ground loop |
| $T_{g\_lwt}$ | LWT from the ground loop |
| $T_{hplgt\_lwt}$ | LWT from the ground loop |
| $T_{hx\_ewt}$ | EWT in the steam heat exchanger for the heat pump loop |
| $T_{hpl\_bt}$ | Water temperature in the buffer tank of the HP Loop |

From the temperatures at various states in the heat pump loop, we can calculate the temperature difference for the following:

$\Delta T_{HPL}$ = Temperature difference between heat loop supply and return of heat pumps.

$= T_{hpl\_bt} - T_{hpl\_rt}$

$\Delta\ T_{d\_gw}$ = Temperature change of water entering the ground loop and water leaving the ground loop

$= T_{hpl\_rt} - T_{hplgt\_lwt}$

Hence, $\Delta T_{d\_gw} = T_{hpl\_rt} - T_{hplgt\_lwt}$, which is the difference of supply temperature to the ground loop and return temperature from the ground loop

Where $T_{hpl\_lwt}$ is the achieved by calculating the balance points of flowrates in the heat exchanger for ground loop and heat pump loop.

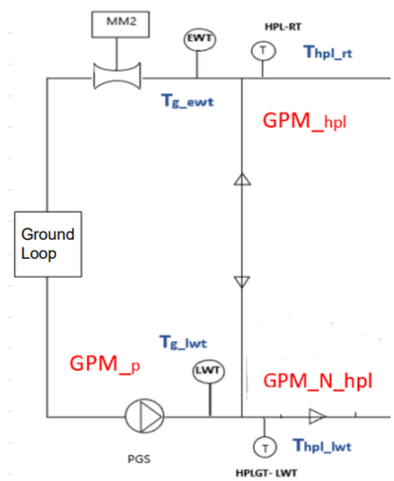


Figure 21. Schematic of a section of heat pump loop to demonstrate the flowrate distribution to calculate new flowrate outside group loop.

Where,

$GPM\_{hpl}$ is the total flowrate of the Heat pump loop.

$\Delta T_{HP}$ is the temperature difference between heat loop supply and return of heat pumps.

$GPM\_{p}$ = Flowrate in the circulating GSHP

$GPM\_{N\_hpl}$ = New Flowrate entering the heat pump loop

Hence using the flowrates of the overall building HPL system and circulating pump,

$$T_{hplgt\_lwt} = \frac{(GPM_p)*(T_{g_{lwt}})+[GPM_{N\_hpl}*\{T_{hpl\_rt}\}]}{GPN_{hpl}} \tag{14}$$

Table 4. Calculation of LWT from Ground Loop from LMTD Method and flowrate balance

| For Heat Pump Loop | T_g_ewt | Thpl_rt | T_g_lwt | | GPM_hpl | GPM_p | Tpl_lwt |
|---|---|---|---|---|---|---|---|
| For LMTD Calculation | T_ci | T_hi | T_ho | T_co | | | |
| 1 | 60 | 65 | 62.4 | 60.2 | 720 | 300 | 63.92 |
| 2 | 60 | 70 | 64.8 | 60.4 | 720 | 300 | 67.83 |
| 3 | 60 | 75 | 67.2 | 60.5 | 720 | 300 | 71.75 |
| 4 | 60 | 80 | 69.7 | 60.7 | 720 | 300 | 75.71 |
| 5 | 60 | 85 | 72.1 | 60.9 | 720 | 300 | 79.63 |

Table 4 shows the calculation of LWT from ground loop for EWT of 60 ̊F for ground loop at varying return temperature from the HPL, for overall flowrate of the HPL at 720 GPM and flowrate of the ground loop circulating pump at 300 GPM by using Equation (14).

The conditions for when to turn on the ground loop in the heat pump loop are evaluated by analyzing if the compression energy saving is greater than the cost of using ground loop circulating pump energy for Equation (15.1), (15.2) and (15.3).

Condition I: For heating Mode only

$$Q_{WH}\left(\frac{0.0314\varDelta \mathrm{T_{d\_gw}}}{(0.0314(\mathrm{T_{hpl\_rt}})+1.4758)(0.0314(\mathrm{T_{hpl\_rt}}+\varDelta \mathrm{T_{d\_gw}})+1.4758)}\right)*EC > \frac{\mathrm{GPM\_p}*\ \Delta \mathrm{h}}{3960*\ \eta_p}*EC \tag{15.1}$$

Where, $Q_{wh}$ =Heat added to water in heating mode

GPM_p = Flowrate in the circulating GSHP

And ($\frac{\text{GPM\_p} * \Delta\text{h}}{3960 * \eta_p} * EC$) is the cost of using the circulating pump for the GSHP and this value is constant throughout the heating mode, cooling mode or, both heating and cooling mode.

Condition II: For Cooling Mode only

$$Q_{WC}\left(\frac{-0.0636\Delta\text{T}_{\text{d\_gw}}}{(-0.0636(\text{T}_{\text{hpl\_rt}}) + 10.1915)(-0.0636(\text{T}_{\text{hpl\_rt}} + \Delta\text{T}_{\text{d\_gw}}) + 10.1915)}\right) * EC > \frac{\text{GPM\_p} * \Delta\text{h}}{3960 * \eta_p} * EC \tag{15.2}$$

Where, $Q_{wc}$- Heat removed from water in cooling mode.

Condition III: For some HPs in cooling mode and some in heating Mode

$$\frac{1}{3}GPM\_hpl\left[\left(\frac{0.0314\Delta\text{T}_{\text{d\_gw}}}{(0.0314(\text{T}_{\text{hpl\_rt}}) + 1.4758)(0.0314(\text{T}_{\text{hpl\_rt}} + \Delta\text{T}_{\text{d\_gw}}) + 1.4758}\right) - \frac{\Delta T_{HP} + 8}{0.18}\left\{\left(\frac{0.0314\Delta\text{T}_{\text{d\_gw}}}{(0.0314(\text{T}_{\text{hpl\_rt}}) + 1.4758)(0.0314(\text{T}_{\text{hpl\_rt}} + \Delta\text{T}_{\text{d\_gw}}) + 1.4758}\right) - \left(\frac{-0.0636\Delta\text{T}_{\text{d\_gw}}}{(-0.0636(\text{T}_{\text{hpl\_rt}}) + 10.1915)(-0.0636(\text{T}_{\text{hpl\_rt}} + \Delta\text{T}_{\text{d\_gw}}) + 10.1915)}\right)\right\}\right] * \text{EC} > \frac{\text{GPM\_p} * \Delta\text{h}}{3960 * \eta_p} * EC \tag{15.3}$$

# CHAPTER THREE

## RESULTS AND DISCUSSION

### Evaluation of When It Makes Sense to Turn on Ground Loop or Steam Injection

A control optimization logic decision matrix is developed which is a tool used to determine when to activate the ground loop in a building's heat pump loop system. The goal is to find the most efficient way to heat or cool the building while also keeping the heat pump loop temperature within the desired range. Based on these factors, the matrix determines whether to activate the ground loop, or steam heat exchanger. By using this tool, building owners can save energy and money while also ensuring that their buildings are comfortable and well-maintained.

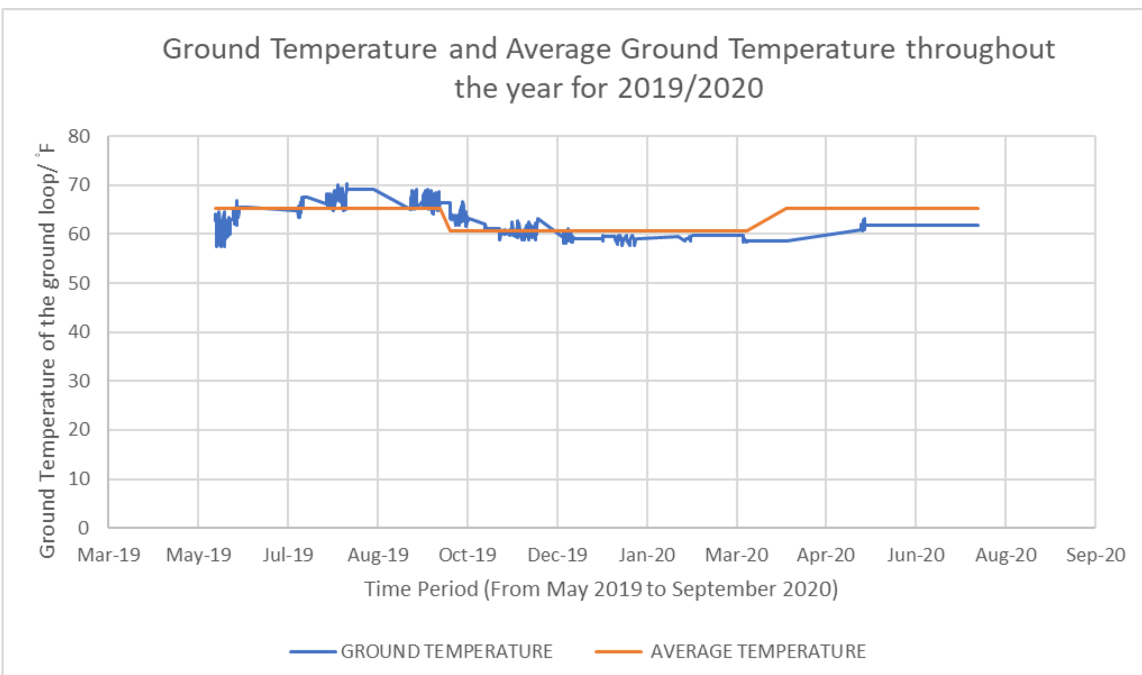


Figure 22. Ground Loop Temperature throughout the year from July 2019 to September 2020 for NAH Building achieved from NAH Database

Figure 22 shows the ground loop temperature throughout the year from May 2019 to September 2020 for NAH from the building management system (BMS) database. This graph gives us the general trend of ground loop temperatures for a one-year period from 2019 to 2020. Using this data, we determined the average ground temperature of 65.3°F for the summer season from April to September and 60.5 °F for the winter season from October to March.

## Energy and Cost Saving Evaluations

The energy savings between improved COP with increasing EWT in heating mode to the ground loop pump energy is evaluated for EWT of 40 ̊F to 60 ̊F. The compressor energy of the HP is the function of COP which depends on EWT. Table 5 shows the COP for HP from McQuay Entity for increasing EWT, and the COP derived from the linear trend line equation from equation (2).

Table 5. COP for increasing EWT from McQuay Entity and Linear Trend Line

| Entering Water Temperature (EWT) | COP Heating from Data | COP1-Heating from Linear Equation |
|---|---|---|
| | from McQuay Entity | COP=0.0314*(EWT)+2.4758 |
| 40 | 3.60 | 3.73 |
| 45 | 3.70 | 3.89 |
| 50 | 4.00 | 4.05 |
| 55 | 4.20 | 4.20 |
| 60 | 4.40 | 4.36 |

The pump energy of the circulating pump for ground loop is determined for the time period required to flow water across the total system volume.

$GPM_p$ = water circulating rate (GPM) = 300 GPM

$\eta_p$ = Pump efficiency = 0.6 (value assumed for typical pump efficiencies)

$\Delta h$ = Pump head, feet of water = 65 feet of head (from the Mechanical Schedule for PGS)

Hence, Pump Power = $\frac{300*65}{3960*0.6}$

= 8.2 hp

Q(Btu/hr) = Heat transfer in the pump = Total Pump Energy

= Time Period for the given system volume * Pump Power

= (17, 300 gal/720 gal/min) *(1h/60 min) * 8.2 hp

= 3.28 hp.h = 2.45 Kw.h

Table 6. Compressor energy for COP @EWT and Ground Loop Pump Energy

| | Heat added to the water in heating mode | Compressor Work/ kWh = Qwh/( COP-1) *293 | | | | | Ground Loop (GL) Pump Energy = [Flowrate(GPM) X Systema Heat(m)]/(3960 * Efficiency)] x Time Period of GL Use | |
|---|---|---|---|---|---|---|---|---|
| Percent of Heat Added | Qwh/ MMBTU | W@ 40 EWT / kWh | W@ 45 EWT / kWh | W@ 50 EWT / kWh | W@ 55 EWT / kWh | W@ 60 EWT / kWh | Pump Energy @ Δ 5°F | Pump Energy @ Δ 10°F |
| 0 | - | - | - | - | - | - | 2.45 | 2.45 |
| 10 | 0.29 | 32.46 | 31.25 | 28.13 | 26.37 | 24.82 | 2.45 | 2.45 |
| 20 | 0.58 | 64.91 | 62.51 | 56.26 | 52.74 | 49.64 | 2.45 | 2.45 |
| 30 | 0.86 | 97.37 | 93.76 | 84.38 | 79.11 | 74.46 | 2.45 | 2.45 |
| 40 | 1.15 | 129.82 | 125.01 | 112.51 | 105.48 | 99.28 | 2.45 | 2.45 |
| 50 | 1.44 | 162.28 | 156.27 | 140.64 | 131.85 | 124.09 | 2.45 | 2.45 |
| 60 | 1.73 | 194.73 | 187.52 | 168.77 | 158.22 | 148.91 | 2.45 | 2.45 |
| 70 | 2.02 | 227.19 | 218.77 | 196.90 | 184.59 | 173.73 | 2.45 | 2.45 |
| 80 | 2.30 | 259.64 | 250.03 | 225.02 | 210.96 | 198.55 | 2.45 | 2.45 |
| 90 | 2.59 | 292.10 | 281.28 | 253.15 | 237.33 | 223.37 | 2.45 | 2.45 |
| 100 | 2.88 | 324.55 | 312.53 | 281.28 | 263.70 | 248.19 | 2.45 | 2.45 |

Table 7. Compressor energy saving between improved COP @EWT vs Ground Loop Pump

| ENERGY SAVING BETWEEN INPROVED COP@EWT VS GROUND LOOP PUMP ENERGY | | | | |
|---|---|---|---|---|
| | Compressor work saving : Wsavings in KWh= Wold-(WNew+ GL Circulating Pump Work) @ Delta T=5 | | | |
| Qwh/ MMBTU | From 40 to 45 EWT | From 45 to 50 EWT | From 50 to 55 EWT | From 55 to 60 EWT |
| 0.29 | (1.25) | 0.68 | (0.69) | (0.90) |
| 0.58 | (0.05) | 3.80 | 1.07 | 0.65 |
| 0.86 | 1.16 | 6.93 | 2.82 | 2.20 |
| 1.15 | 2.36 | 10.05 | 4.58 | 3.75 |
| 1.44 | 3.56 | 13.18 | 6.34 | 5.31 |
| 1.73 | 4.76 | 16.30 | 8.10 | 6.86 |
| 2.02 | 5.96 | 19.43 | 9.86 | 8.41 |
| 2.30 | 7.17 | 22.55 | 11.61 | 9.96 |
| 2.59 | 8.37 | 25.68 | 13.37 | 11.51 |
| 2.88 | 9.57 | 28.80 | 15.13 | 13.06 |

Table 6 shows the compressor energy calculated for the various heat added to the water in heating mode, and the ground loop circulating pump energy. Table 7 shows the compressor energy saving between improved COP at EWT to ground loop pump energy. The grey highlighted portion of Table 7 shows the compressor energy saving in increasing the EWT by 5°F. Likewise, Table 8 shows the cost of using HP at different EWT, and the cost of using the ground loop circulating pump.

Table 8. Costs for improved COP @EWT vs Ground Loop Pump Cost

| | Heat added to the water in heating mode | Cost of Compressor Work in Dollars = Qwh/( COP-1) *293* $34/MMBTU | | | | | Cost of Ground Loop Pump Work | |
|---|---|---|---|---|---|---|---|---|
| Percent of Heat Added | Qwh/ MMBTU | Cost for W@ 40 EWT | Cost for W@ 45 EWT | Cost for W@ 50 EWT | Cost for W@ 55 EWT | Cost for W@ 60 EWT | Electric Cost of Circulating Pump @ Δ 5°F | Electric Cost of Circulating Pump @ Δ 10°F |
| 0 | - | $0.00 | $0.00 | $0.00 | $0.00 | $0.00 | 83.30 | 83.30 |
| 10 | 0.29 | $1,103.48 | $1,062.61 | $956.35 | $896.58 | $843.84 | 83.30 | 83.30 |
| 20 | 0.58 | $2,206.97 | $2,125.23 | $1,912.70 | $1,793.16 | $1,687.68 | 83.30 | 83.30 |
| 30 | 0.86 | $3,310.45 | $3,187.84 | $2,869.06 | $2,689.74 | $2,531.52 | 83.30 | 83.30 |
| 40 | 1.15 | $4,413.93 | $4,250.45 | $3,825.41 | $3,586.32 | $3,375.36 | 83.30 | 83.30 |
| 50 | 1.44 | $5,517.42 | $5,313.07 | $4,781.76 | $4,482.90 | $4,219.20 | 83.30 | 83.30 |
| 60 | 1.73 | $6,620.90 | $6,375.68 | $5,738.11 | $5,379.48 | $5,063.04 | 83.30 | 83.30 |
| 70 | 2.02 | $7,724.38 | $7,438.29 | $6,694.46 | $6,276.06 | $5,906.88 | 83.30 | 83.30 |
| 80 | 2.30 | $8,827.86 | $8,500.91 | $7,650.82 | $7,172.64 | $6,750.72 | 83.30 | 83.30 |
| 90 | 2.59 | $9,931.35 | $9,563.52 | $8,607.17 | $8,069.22 | $7,594.56 | 83.30 | 83.30 |
| 100 | 2.88 | $11,034.83 | $10,626.13 | $9,563.52 | $8,965.80 | $8,438.40 | 83.30 | 83.30 |

Likewise, Table 9 shows the cost saving between improved COP at increasing EWT to the ground loop pump. Like energy saving, there is cost benefit in using ground loop for heat added to the system is more than around 0.58 MMBTU. From Table 9, we can see that for 0.58 MMBTU heat added to the water in heating mode for entering water temperature to the heat pump between 45 ̊F to 55 ̊F is positive, which means that it makes sense to use ground loop in the case where heat added to the system is more than 0.58 MMBTU for improvement of EWT to the heat pumps by 5 ̊F for the heating mode.

Table 9. Costs saving between improved COP @EWT with Ground Loop Pump Cost

| Qwh/ MMBTU | Compressor work Cost saving : Wsavings in dollars = Wold-(WNew+ GL Circulating Pump Cost) @ Delta T=5 | | | |
|---|---|---|---|---|
| | From 40 to 45 EWT | From 45 to 50 EWT | From 50 to 55 EWT | From 55 to 60 EWT |
| 0.29 | ($42.43) | $22.96 | ($23.53) | ($30.56) |
| 0.58 | ($1.56) | $129.22 | $36.24 | $22.18 |
| 0.86 | $39.31 | $235.48 | $96.02 | $74.92 |
| 1.15 | $80.18 | $341.75 | $155.79 | $127.66 |
| 1.44 | $121.05 | $448.01 | $215.56 | $180.40 |
| 1.73 | $161.92 | $554.27 | $275.33 | $233.14 |
| 2.02 | $202.79 | $660.53 | $335.10 | $285.88 |
| 2.30 | $243.66 | $766.79 | $394.88 | $338.62 |
| 2.59 | $284.53 | $873.05 | $454.65 | $391.36 |
| 2.88 | $325.40 | $979.31 | $514.42 | $444.10 |

The result for compressor energy saving with respect to the entering water temperature to the heat pump loop is evaluated. Moreover, the cost saving is also calculated for different entering water temperature to the heat pump loop. Hence, there are energy and cost savings in using ground loop for practical scenarios.

Likewise, the energy and cost analysis of the use of steam injection in the HPL is done using the tabulated analysis in Table 10 is used to solve for Compressor Energy Saving when we increase the entering temperature of the heat pump by ΔT of 5 °F and 10 ̊F is evaluated for EWT of 20 ̊F to 90 ̊F for the heating mode. The calculation data is to determine the balance point temperature in which steam injections makes sense from BTU standpoint for overcoming COP difference.

Table 10. Energy Analysis for Steam Heat Exchanger Use

**ENERGY ANALYSIS**

Steam Work = System Volume * 8.3* ΔT* 0.000293

| Steam Work @ Δ 5°F | Steam Work @ Δ 10°F | Steam Work @ Δ 15°F |
|---|---|---|
| 51.07 | 102.14 | 153.21 |

| Heat added to the water in heating mode | Compressor Work (W)/ kWh = $Q_{wh}$/( COP-1) *293 | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Qwh/ MMBTU | W@ 20 EWT / kWh | W@ 25 EWT / kWh | W@ 30 EWT / kWh | W@ 35 EWT / kWh | W@ 40 EWT / kWh | W@ 45 EWT / kWh | W@ 50 EWT / kWh | W@ 55 EWT / kWh | W@ 60 EWT / kWh | W@ 65 EWT / kWh | W@ 70 EWT / kWh | W@ 75 EWT / kWh | W@ 80 EWT / kWh | W@ 85 EWT / kWh | W@ 90 EWT / kWh | |
| - | - | - | - | - | - | - | - | - | - | - | - | - | - | - | - | |
| 0.29 | 40.11 | 37.32 | 34.90 | 32.77 | 30.89 | 29.21 | 27.71 | 26.35 | 25.12 | 23.99 | 22.97 | 22.03 | 21.16 | 20.36 | 19.62 | |
| 0.58 | 80.22 | 74.65 | 69.80 | 65.55 | 61.78 | 58.42 | 55.41 | 52.69 | 50.23 | 47.99 | 45.94 | 44.06 | 42.32 | 40.72 | 39.23 | |
| 0.86 | 120.33 | 111.97 | 104.70 | 98.32 | 92.67 | 87.63 | 83.12 | 79.04 | 75.35 | 71.98 | 68.91 | 66.08 | 63.48 | 61.08 | 58.85 | |
| 1.15 | 160.44 | 149.30 | 139.60 | 131.09 | 123.56 | 116.84 | 110.82 | 105.39 | 100.46 | 95.98 | 91.88 | 88.11 | 84.64 | 81.44 | 78.46 | |
| 1.44 | 200.55 | 186.62 | 174.51 | 163.87 | 154.45 | 146.05 | 138.53 | 131.73 | 125.58 | 119.97 | 114.85 | 110.14 | 105.80 | 101.80 | 98.08 | |
| 1.73 | 240.66 | 223.95 | 209.41 | 196.64 | 185.34 | 175.26 | 166.23 | 158.08 | 150.69 | 143.97 | 137.81 | 132.17 | 126.96 | 122.15 | 117.70 | |
| 2.02 | 280.77 | 261.27 | 244.31 | 229.41 | 216.23 | 204.48 | 193.94 | 184.43 | 175.81 | 167.96 | 160.78 | 154.19 | 148.12 | 142.51 | 137.31 | |
| 2.30 | 320.88 | 298.60 | 279.21 | 262.18 | 247.12 | 233.69 | 221.64 | 210.78 | 200.93 | 191.96 | 183.75 | 176.22 | 169.28 | 162.87 | 156.93 | |
| 2.59 | 360.99 | 335.92 | 314.11 | 294.96 | 278.01 | 262.90 | 249.35 | 237.12 | 226.04 | 215.95 | 206.72 | 198.25 | 190.44 | 183.23 | 176.54 | |
| 2.88 | 401.10 | 373.25 | 349.01 | 327.73 | 308.90 | 292.11 | 277.05 | 263.47 | 251.16 | 239.95 | 229.69 | 220.28 | 211.61 | 203.59 | 196.16 | |
| 3.00 | 417.82 | 388.80 | 363.55 | 341.39 | 321.77 | 304.28 | 288.59 | 274.45 | 261.62 | 249.94 | 239.26 | 229.46 | 220.42 | 212.07 | 204.33 | Checking for $Q_{wh}$ > |
| 5.00 | 696.36 | 648.00 | 605.92 | 568.98 | 536.28 | 507.13 | 480.99 | 457.41 | 436.04 | 416.57 | 398.77 | 382.43 | 367.37 | 353.45 | 340.56 | 2.88 MMBTU (max |
| 6.00 | 835.63 | 777.60 | 727.11 | 682.77 | 643.53 | 608.56 | 577.19 | 548.89 | 523.25 | 499.89 | 478.52 | 458.91 | 440.84 | 424.15 | 408.67 | heat added in |

| Qwh/ MMBTU | Compressor work saving : $W_{savings}$ in KWh= $W_{old}$-($W_{New}$+ Steam Work) @ Delta T=5 | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | From 20 to 25 EWT | From 25 to 30 EWT | From 30 to 35 EWT | From 35 to 40 EWT | From 40 to 45 EWT | From 45 to 50 EWT | From 50 to 55 EWT | From 55 to 60 EWT | From 60 to 65 EWT | From 65 to 70 EWT | From 70 to 75 EWT | From 75 to 80 EWT | From 80 to 85 EWT | From 85 to 90 EWT | |
| | in KWh | | | | | | | | | | | | | | |
| 0.29 | (48.28) | (48.65) | (48.94) | (49.19) | (49.39) | (49.56) | (49.71) | (49.84) | (49.95) | (50.04) | (50.13) | (50.20) | (50.27) | (50.33) | |
| 0.58 | (45.50) | (46.22) | (46.81) | (47.30) | (47.71) | (48.06) | (48.35) | (48.61) | (48.83) | (49.02) | (49.19) | (49.34) | (49.47) | (49.58) | |
| 0.86 | (42.71) | (43.80) | (44.69) | (45.42) | (46.03) | (46.55) | (47.00) | (47.38) | (47.71) | (47.99) | (48.25) | (48.47) | (48.67) | (48.84) | |
| 1.15 | (39.93) | (41.38) | (42.56) | (43.54) | (44.35) | (45.05) | (45.64) | (46.15) | (46.58) | (46.97) | (47.30) | (47.60) | (47.86) | (48.10) | |
| 1.44 | (37.14) | (38.95) | (40.43) | (41.65) | (42.68) | (43.54) | (44.28) | (44.91) | (45.46) | (45.94) | (46.36) | (46.73) | (47.06) | (47.35) | |
| 1.73 | (34.36) | (36.53) | (38.30) | (39.77) | (41.00) | (42.04) | (42.92) | (43.68) | (44.34) | (44.92) | (45.42) | (45.87) | (46.26) | (46.61) | |
| 2.02 | (31.57) | (34.10) | (36.17) | (37.89) | (39.32) | (40.53) | (41.56) | (42.45) | (43.22) | (43.89) | (44.48) | (45.00) | (45.46) | (45.87) | |
| 2.30 | (28.79) | (31.68) | (34.04) | (36.00) | (37.64) | (39.02) | (40.21) | (41.22) | (42.10) | (42.87) | (43.54) | (44.13) | (44.66) | (45.13) | |
| 2.59 | (26.00) | (29.26) | (31.92) | (34.12) | (35.96) | (37.52) | (38.85) | (39.99) | (40.98) | (41.84) | (42.60) | (43.26) | (43.86) | (44.38) | |
| 2.88 | (23.22) | (26.83) | (29.79) | (32.23) | (34.28) | (36.01) | (37.49) | (38.76) | (39.86) | (40.82) | (41.66) | (42.40) | (43.05) | (43.64) | |
| 3.00 | (22.05) | (25.82) | (28.90) | (31.45) | (33.58) | (35.39) | (36.92) | (38.25) | (39.39) | (40.39) | (41.26) | (42.04) | (42.72) | (43.33) | Checking for $Q_{wh}$ > |
| 5.00 | (2.71) | (8.99) | (14.12) | (18.37) | (21.92) | (24.93) | (27.49) | (29.70) | (31.60) | (33.27) | (34.73) | (36.01) | (37.15) | (38.17) | 2.88 MMBTU (max |
| 6.00 | 6.96 | (0.58) | (6.73) | (11.83) | (16.10) | (19.70) | (22.78) | (25.42) | (27.71) | (29.71) | (31.46) | (33.00) | (34.37) | (35.59) | heat added in heating |

| Qwh/M MBTU | Compressor work saving : $W_{savings}$= $W_{old}$-($W_{New}$+ Steam Work) @ Delta T=10 | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | From 20 to 30 EWT | From 25 to 35 EWT | From 30 to 40 EWT | From 35 to 45 EWT | From 40 to 50 EWT | From 45 to 55 EWT | From 50 to 60 EWT | From 55 to 65 EWT | From 60 to 70 EWT | From 65 to 75 EWT | From 70 to 80 EWT | From 75 to 85 EWT | From 80 to 90 EWT | |
| | in KWh | | | | | | | | | | | | | |
| 0.288 | (96.93) | (97.59) | (98.13) | (98.58) | (98.96) | (99.28) | (99.55) | (99.79) | (99.99) | (100.17) | (100.33) | (100.47) | (100.60) | |
| 0.576 | (91.72) | (93.04) | (94.12) | (95.02) | (95.77) | (96.41) | (96.96) | (97.43) | (97.85) | (98.21) | (98.52) | (98.80) | (99.05) | |
| 0.864 | (86.51) | (88.48) | (90.10) | (91.45) | (92.59) | (93.55) | (94.37) | (95.08) | (95.70) | (96.24) | (96.71) | (97.13) | (97.51) | |
| 1.152 | (81.30) | (83.93) | (86.09) | (87.89) | (89.40) | (90.68) | (91.78) | (92.73) | (93.55) | (94.27) | (94.91) | (95.46) | (95.96) | |
| 1.44 | (76.09) | (79.38) | (82.08) | (84.33) | (86.22) | (87.82) | (89.19) | (90.38) | (91.41) | (92.31) | (93.10) | (93.80) | (94.42) | |
| 1.728 | (70.89) | (74.83) | (78.07) | (80.77) | (83.03) | (84.96) | (86.60) | (88.03) | (89.26) | (90.34) | (91.29) | (92.13) | (92.87) | |
| 2.016 | (65.68) | (70.28) | (74.06) | (77.20) | (79.85) | (82.09) | (84.02) | (85.67) | (87.11) | (88.37) | (89.48) | (90.46) | (91.33) | |
| 2.304 | (60.47) | (65.73) | (70.05) | (73.64) | (76.66) | (79.23) | (81.43) | (83.32) | (84.97) | (86.41) | (87.67) | (88.79) | (89.78) | |
| 2.592 | (55.26) | (61.17) | (66.04) | (70.08) | (73.48) | (76.37) | (78.84) | (80.97) | (82.82) | (84.44) | (85.86) | (87.12) | (88.24) | |
| 2.88 | (50.05) | (56.62) | (62.02) | (66.52) | (70.29) | (73.50) | (76.25) | (78.62) | (80.67) | (82.47) | (84.05) | (85.45) | (86.69) | |
| 3.00 | (47.88) | (54.73) | (60.35) | (65.03) | (68.97) | (72.31) | (75.17) | (77.64) | (79.78) | (81.65) | (83.30) | (84.76) | (86.05) | Checking for $Q_{wh}$ > |
| 5.00 | (11.70) | (23.12) | (32.49) | (40.29) | (46.85) | (52.42) | (57.19) | (61.30) | (64.87) | (67.99) | (70.74) | (73.17) | (75.32) | 2.88 MMBTU (max |
| 6.00 | 6.38 | (7.31) | (18.56) | (27.93) | (35.80) | (42.48) | (48.20) | (53.13) | (57.42) | (61.17) | (64.46) | (67.37) | (69.96) | heat added in |

Table 10 shows that from BTU standpoint steam injection doesn't work for a system of 2.88 MMBTU which is also the maximum capacity in heating mode. Steam injection can be seen to be makes sense around 6 MMBTU with EWT 20 ̊F for a temperature increase by 10 ̊F. Energy savings are realized at this point as shown in Table 10. However, this hypothetical system would never exist in reality since no GSHP system would have that low of entering water (i.e., 20 ̊F).

Table 11. Cost Analysis (cost of compression work and saving of compression work for different temperature difference) for Steam Heat Exchanger Use

COST ANALYSIS

| Cost of Steam Work = System Volume * 8.3 * ΔT * 0.000293 * $10 /MMBTU | | |
|---|---|---|
| Steam Work @ Δ 5°F | Steam Work @ Δ 10°F | Steam Work @ Δ 15°F |
| 510.70 | 1,021.40 | 1,532.10 |

| Heat added to the water in heating mode | Cost of Compressor Work (W) in Dollars = Qwh/( COP-1) *293* $34/MMBTU | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Qwh/ MMBTU | Cost for W@ 20 EWT | Cost for W@ 25 EWT | Cost for W@ 30 EWT | Cost for W@ 35 EWT | Cost for W@ 40 EWT | Cost for W@ 45 EWT | Cost for W@ 50 EWT | Cost for W@ 55 EWT | Cost for W@ 60 EWT | Cost for W@ 65 EWT | Cost for W@ 70 EWT | Cost for W@ 75 EWT | Cost for W@ 80 EWT | Cost for W@ 85 EWT | Cost for W@ 90 EWT |
| - | $0.00 | $0.00 | $0.00 | $0.00 | $0.00 | $0.00 | $0.00 | $0.00 | $0.00 | $0.00 | $0.00 | $0.00 | $0.00 | $0.00 | $0.00 |
| 0.29 | $1,363.75 | $1,269.04 | $1,186.64 | $1,114.28 | $1,050.24 | $993.17 | $941.97 | $895.80 | $853.94 | $815.81 | $780.95 | $748.94 | $719.46 | $692.21 | $666.94 |
| 0.58 | $2,727.50 | $2,538.09 | $2,373.28 | $2,228.57 | $2,100.49 | $1,986.33 | $1,883.94 | $1,791.59 | $1,707.87 | $1,631.63 | $1,561.90 | $1,497.89 | $1,438.92 | $1,384.41 | $1,333.89 |
| 0.86 | $4,091.25 | $3,807.13 | $3,559.92 | $3,342.85 | $3,150.73 | $2,979.50 | $2,825.91 | $2,687.39 | $2,561.81 | $2,447.44 | $2,342.85 | $2,246.83 | $2,158.38 | $2,076.62 | $2,000.83 |
| 1.15 | $5,455.00 | $5,076.18 | $4,746.56 | $4,457.13 | $4,200.98 | $3,972.66 | $3,767.88 | $3,583.18 | $3,415.75 | $3,263.26 | $3,123.80 | $2,995.78 | $2,877.83 | $2,768.82 | $2,667.77 |
| 1.44 | $6,818.75 | $6,345.22 | $5,933.20 | $5,571.42 | $5,251.22 | $4,965.83 | $4,709.86 | $4,478.98 | $4,269.68 | $4,079.07 | $3,904.75 | $3,744.72 | $3,597.29 | $3,461.03 | $3,334.72 |
| 1.73 | $8,182.50 | $7,614.27 | $7,119.83 | $6,685.70 | $6,301.46 | $5,958.99 | $5,651.83 | $5,374.78 | $5,123.62 | $4,894.89 | $4,685.70 | $4,493.67 | $4,316.75 | $4,153.24 | $4,001.66 |
| 2.02 | $9,546.25 | $8,883.31 | $8,306.47 | $7,799.98 | $7,351.71 | $6,952.16 | $6,593.80 | $6,270.57 | $5,977.56 | $5,710.70 | $5,466.65 | $5,242.61 | $5,036.21 | $4,845.44 | $4,668.60 |
| 2.30 | $10,910.00 | $10,152.36 | $9,493.11 | $8,914.26 | $8,401.95 | $7,945.32 | $7,535.77 | $7,166.37 | $6,831.49 | $6,526.52 | $6,247.60 | $5,991.55 | $5,755.67 | $5,537.65 | $5,335.55 |
| 2.59 | $12,273.74 | $11,421.40 | $10,679.75 | $10,028.55 | $9,452.19 | $8,938.49 | $8,477.74 | $8,062.17 | $7,685.43 | $7,342.33 | $7,028.55 | $6,740.50 | $6,475.13 | $6,229.86 | $6,002.49 |
| 2.88 | $13,637.49 | $12,690.45 | $11,866.39 | $11,142.83 | $10,502.44 | $9,931.65 | $9,419.71 | $8,957.96 | $8,539.37 | $8,158.14 | $7,809.51 | $7,489.44 | $7,194.58 | $6,922.06 | $6,669.43 |

| Qwh/ MMBTU | Saving of Compressor work Cost @Δ T=5 °F : $W_{savings}$ in dollars = $W_{old}$-($W_{New}$+ Steam Work Cost) | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | From 20 to 25 EWT | From 25 to 30 EWT | From 30 to 35 EWT | From 35 to 40 EWT | From 40 to 45 EWT | From 45 to 50 EWT | From 50 to 55 EWT | From 55 to 60 EWT | From 60 to 65 EWT | From 65 to 70 EWT | From 70 to 75 EWT | From 75 to 80 EWT | From 80 to 85 EWT | From 85 to 90 EWT |
| | in dollars | | | | | | | | | | | | | |
| 0.29 | ($415.99) | ($428.29) | ($438.34) | ($446.66) | ($453.62) | ($459.50) | ($464.52) | ($468.84) | ($472.58) | ($475.84) | ($478.69) | ($481.21) | ($483.45) | ($485.44) |
| 0.58 | ($321.29) | ($345.89) | ($365.99) | ($382.62) | ($396.54) | ($408.31) | ($418.35) | ($426.98) | ($434.45) | ($440.97) | ($446.69) | ($451.73) | ($456.19) | ($460.17) |
| 0.86 | ($226.58) | ($263.48) | ($293.63) | ($318.58) | ($339.46) | ($357.12) | ($372.17) | ($385.12) | ($396.33) | ($406.11) | ($414.68) | ($422.24) | ($428.94) | ($434.91) |
| 1.15 | ($131.88) | ($181.08) | ($221.27) | ($254.54) | ($282.39) | ($305.92) | ($326.00) | ($343.26) | ($358.21) | ($371.24) | ($382.67) | ($392.76) | ($401.69) | ($409.65) |
| 1.44 | ($37.17) | ($98.67) | ($148.92) | ($190.50) | ($225.31) | ($254.73) | ($279.82) | ($301.40) | ($320.09) | ($336.38) | ($350.67) | ($363.27) | ($374.44) | ($384.38) |
| 1.73 | $57.53 | ($16.27) | ($76.56) | ($126.46) | ($168.23) | ($203.53) | ($233.65) | ($259.54) | ($281.97) | ($301.52) | ($318.66) | ($333.78) | ($347.19) | ($359.12) |
| 2.02 | $152.23 | $66.14 | ($4.21) | ($62.42) | ($111.15) | ($152.34) | ($187.47) | ($217.68) | ($243.84) | ($266.65) | ($286.66) | ($304.30) | ($319.93) | ($333.86) |
| 2.30 | $246.94 | $148.54 | $68.15 | $1.62 | ($54.07) | ($101.15) | ($141.30) | ($175.82) | ($205.72) | ($231.79) | ($254.65) | ($274.81) | ($292.68) | ($308.60) |
| 2.59 | $341.64 | $230.95 | $140.51 | $65.65 | $3.01 | ($49.95) | ($95.12) | ($133.96) | ($167.60) | ($196.92) | ($222.64) | ($245.33) | ($265.43) | ($283.33) |
| 2.88 | $436.35 | $313.36 | $212.86 | $129.69 | $60.09 | $1.24 | ($48.95) | ($92.10) | ($129.48) | ($162.06) | ($190.64) | ($215.84) | ($238.18) | ($258.07) |

| Qwh/ MMBTU | Saving of Compressor work Cost @Δ T= 10 °F: $W_{savings}$ in dollars= $W_{old}$-($W_{New}$+ Steam Work) @ Delta T=10 | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | From 20 to 30 EWT | From 25 to 35 EWT | From 30 to 40 EWT | From 35 to 45 EWT | From 40 to 50 EWT | From 45 to 55 EWT | From 50 to 60 EWT | From 55 to 65 EWT | From 60 to 70 EWT | From 65 to 75 EWT | From 70 to 80 EWT | From 75 to 85 EWT | From 80 to 90 EWT |
| | in dollars | | | | | | | | | | | | |
| 0.29 | ($844.29) | ($866.64) | ($885.00) | ($900.28) | ($913.13) | ($924.03) | ($933.36) | ($941.42) | ($948.41) | ($954.53) | ($959.91) | ($964.66) | ($968.88) |
| 0.58 | ($667.18) | ($711.87) | ($748.61) | ($779.16) | ($804.85) | ($826.66) | ($845.33) | ($861.43) | ($875.43) | ($887.66) | ($898.41) | ($907.92) | ($916.37) |
| 0.86 | ($490.07) | ($557.11) | ($612.21) | ($658.04) | ($696.58) | ($729.29) | ($757.29) | ($781.45) | ($802.44) | ($820.79) | ($836.92) | ($851.18) | ($863.85) |
| 1.15 | ($312.96) | ($402.35) | ($475.82) | ($536.93) | ($588.31) | ($631.92) | ($669.26) | ($701.47) | ($729.45) | ($753.92) | ($775.43) | ($794.45) | ($811.34) |
| 1.44 | ($135.85) | ($247.59) | ($339.42) | ($415.81) | ($480.04) | ($534.55) | ($581.22) | ($621.49) | ($656.47) | ($687.05) | ($713.94) | ($737.71) | ($758.82) |
| 1.73 | $41.26 | ($92.83) | ($203.03) | ($294.69) | ($371.76) | ($437.18) | ($493.19) | ($541.51) | ($583.48) | ($620.18) | ($652.45) | ($680.97) | ($706.31) |
| 2.02 | $218.37 | $61.93 | ($66.63) | ($173.57) | ($263.49) | ($339.81) | ($405.16) | ($461.53) | ($510.50) | ($553.31) | ($590.95) | ($624.23) | ($653.79) |
| 2.30 | $395.48 | $216.69 | $69.76 | ($52.46) | ($155.22) | ($242.44) | ($317.12) | ($381.54) | ($437.51) | ($486.44) | ($529.46) | ($567.49) | ($601.28) |
| 2.59 | $572.59 | $371.46 | $206.16 | $68.66 | ($46.94) | ($145.08) | ($229.09) | ($301.56) | ($364.52) | ($419.57) | ($467.97) | ($510.75) | ($548.76) |
| 2.88 | $749.71 | $526.22 | $342.55 | $189.78 | $61.33 | ($47.71) | ($141.05) | ($221.58) | ($291.54) | ($352.70) | ($406.48) | ($454.02) | ($496.25) |

Meanwhile, for cost, we can see from Table 8 that for ΔT=5 ̊F, there is $1.24 cost saving from increasing EWT from 45 ̊F to 50 ̊F, which is the only reasonable point to use steam heat exchanger from temperature range of 45 ̊F-85 ̊F standpoint. Therefore, we do not observe a significant condition where steam injection would be beneficial.

For instance, when a reasonable temperature where EWT= 60 ̊F and ΔT is 10 ̊F, the Qwh value is 4.03 MMBTU which is far more than the maximum capacity of 2.88 MMBTU heat in heating mode given cost saving. However, this case is only a theoretical scenario and does not happen practically or in reality.

Table 12. Analysis of Saving of Compressor Work Cost

| Qwh/ MMBTU | Saving of Compressor work Cost @Δ T= 10 °F: $W_{savings}$ in dollars= $W_{old}$-($W_{New}$+ Steam Work) @ Delta T=10 | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | From 20 to 30 EWT | From 25 to 35 EWT | From 30 to 40 EWT | From 35 to 45 EWT | From 40 to 50 EWT | From 45 to 55 EWT | From 50 to 60 EWT | From 55 to 65 EWT | From 60 to 70 EWT | From 65 to 75 EWT |
| | in dollars | | | | | | | | | |
| 0.29 | ($844.29) | ($866.64) | ($885.00) | ($900.28) | ($913.13) | ($924.03) | ($933.36) | ($941.42) | ($948.41) | ($954.53) |
| 0.58 | ($667.18) | ($711.87) | ($748.61) | ($779.16) | ($804.85) | ($826.66) | ($845.33) | ($861.43) | ($875.43) | ($887.66) |
| 0.86 | ($490.07) | ($557.11) | ($612.21) | ($658.04) | ($696.58) | ($729.29) | ($757.29) | ($781.45) | ($802.44) | ($820.79) |
| 1.15 | ($312.96) | ($402.35) | ($475.82) | ($536.93) | ($588.31) | ($631.92) | ($669.26) | ($701.47) | ($729.45) | ($753.92) |
| 1.44 | ($135.85) | ($247.59) | ($339.42) | ($415.81) | ($480.04) | ($534.55) | ($581.22) | ($621.49) | ($656.47) | ($687.05) |
| 1.73 | $41.26 | ($92.83) | ($203.03) | ($294.69) | ($371.76) | ($437.18) | ($493.19) | ($541.51) | ($583.48) | ($620.18) |
| 2.02 | $218.37 | $61.93 | ($66.63) | ($173.57) | ($263.49) | ($339.81) | ($405.16) | ($461.53) | ($510.50) | ($553.31) |
| 2.30 | $395.48 | $216.69 | $69.76 | ($52.46) | ($155.22) | ($242.44) | ($317.12) | ($381.54) | ($437.51) | ($486.44) |
| 2.59 | $572.59 | $371.46 | $206.16 | $68.66 | ($46.94) | ($145.08) | ($229.09) | ($301.56) | ($364.52) | ($419.57) |
| 2.88 | $749.71 | $526.22 | $342.55 | $189.78 | $61.33 | ($47.71) | ($141.05) | ($221.58) | ($291.54) | ($352.70) |
| 4.03 | $1,457.16 | $1,144.40 | $887.38 | $673.58 | $493.82 | $341.23 | $210.60 | $97.90 | $0.00093 | ($85.59) |

From the cost analysis using Table 12 in excel and the condition we got for heating mode, we can say that it wouldn’t make sense to use steam injection for the given conditions when some systems are in heating, and some are in cooling modes. It is because the value of $Q_{wh}$ in both heating and cooling mode would be even lower than that of its value when all the systems are only in the heating mode. This is because the $Q_{wh}$ would be decreasing as some systems will be in cooling mode and it would be much more difficult to achieve balance point in the reasonable temperature range in this condition.

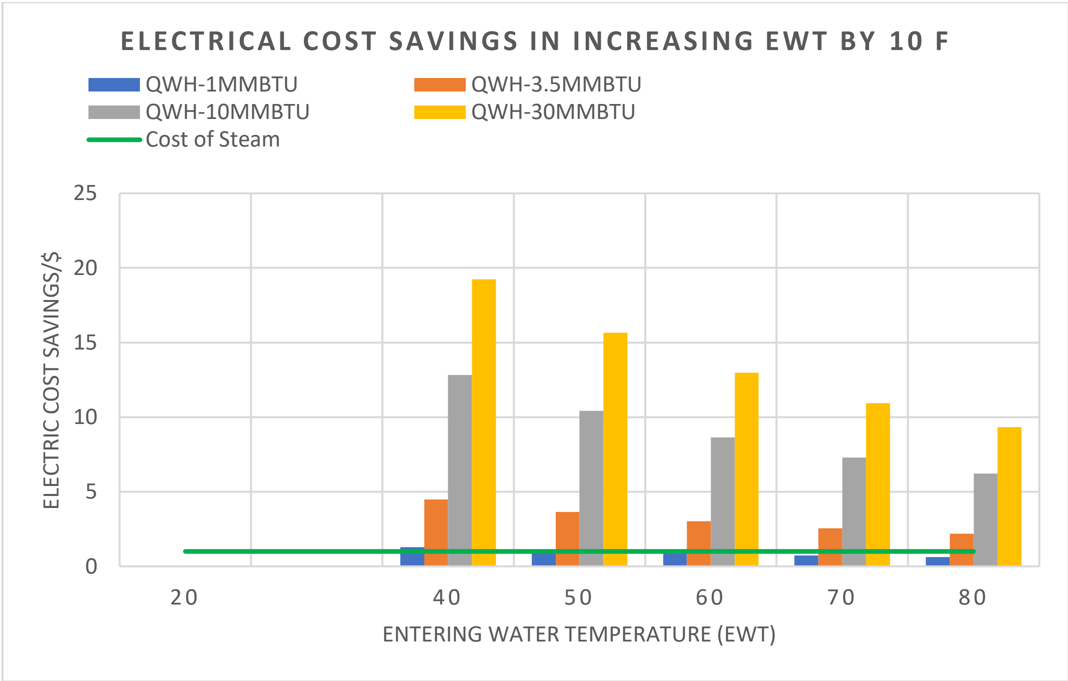


Figure 23. Electric Cost Savings in increasing EWT by 10 F for varying Qwh and Cost of Steam

From Figure 23 we can observe the EWT 50°F if you increase the temperature by 10°F we can get a break-even point where the cost of using compressor energy is balanced. The heat added by water in heating mode (Q_WH) is from 1 MMBTU to 30 MMBTU range. The Q_WH value above 2.88 MMBTU is not realistic and EWT below 45̊F practically does not occur in the heat pump loop in NAH, hence, it does not make sense to turn on steam heat exchanger in NAH in the general operation of the heat pump loop.

Decision Matrix to Activate the Ground Loop

A control optimization logic decision matrix is used to determine when to turn on the ground loop in a building heat pump loop system. The goal of the matrix is to find the most efficient way to heat or cool the building, while also keeping the HP loop temperature within the desired range. The decision matrix can help to save energy and money, by creating control optimization logic for turning on the ground loop only when necessary.

Inputs for Control Logic Optimization. The control logic loop for the decision matrix to determine when to turn on the ground coupled loop in the GSHP system based on the entering water temperature in the heat pump loop and, the temperature difference between the leaving water temperature in the heat pump loop and the entering water temperature in the heat pump loop:

The inputs are:

1. Entering water temperature in the heat pump loop (EWT_hpl)
2. Leaving water temperature in the heat pump loop (LWT_hpl)
3. Temperature difference between the leaving water temperature in the heat pump loop and the entering water temperature in the heat pump loop (ΔT_hpl)

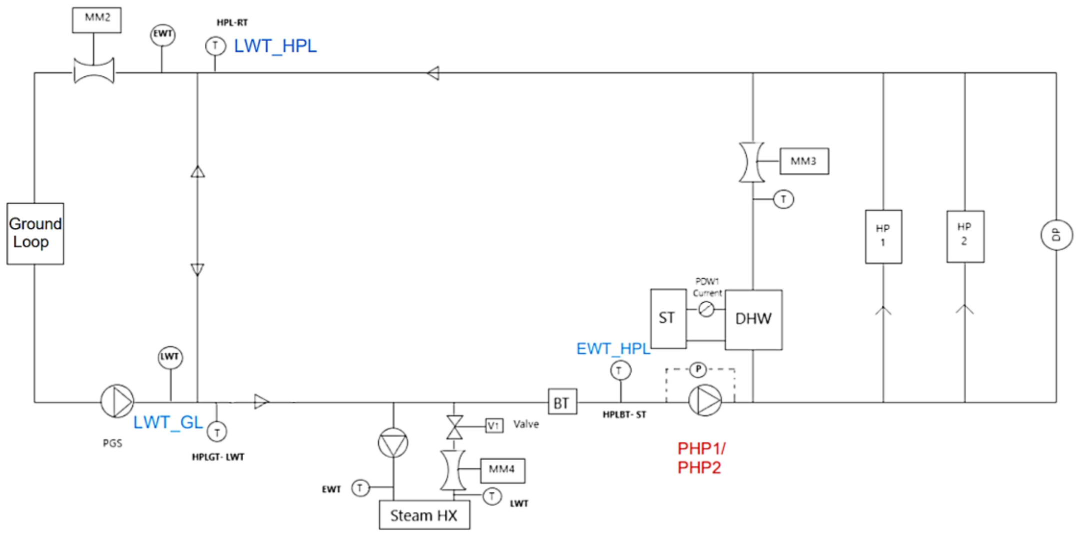


Figure 24. PHP1/PHP2, EWT_HPL and LWT_GL in the NAH Heat Pump Loop for Decision Matrix

<u>Determining Flowrate of Heat Pump Loop (PHP1/PHP2).</u> The flowrate and daily volume requirement for a GSHP depends on its size, specific design characteristics, water flow per extracted BTU/hour of heating, hours per day of operation, and the temperature of the local ground water supply. For system volume of 4200 gallons and assuming the run time for evaluating whether to turn on the ground loop is 30 minutes. We know that,

Flowrate for 30 minutes run time = 4200 gallons/30 minutes = 140 GPM

The pump flowrate in a GSHP loop with a VFD pump can be determined by using the pump curve and pump law. The pump curve is a graph that shows the relationship between the flowrate and head pressure for a given pump.

| PUMP SCHEDULE | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| PLAN CODE | MANUFACTURER | MODEL NUMBER | IMP. SIZE | RPM | GPM | FT OF HD | ELECTRICAL DATA | | | FLUID | VFD | REMARKS |
| | | | | | | | HP | VOLT | PH | | | |
| P-HP1, P-HP2 | TACO | FI-5011 | 11" | 1750 | 720 | 110 | 40 | 460 | 3 | 35% PG | YES | SEE NOTES, HEAT PUMP BUILDING LOOP |
| P-GS1, P-GS2 | TACO | FI-5009C | 8.9" | 1750 | 720 | 65 | 20 | 460 | 3 | 35% PG | YES | SEE NOTES, HEAT PUMP GROUND LOOP |
| P-HW1, P-HW2 | TACO | FI-4011C | 9.6 | 1750 | 475 | 90 | 20 | 460 | 3 | 35% PG | YES | SEE NOTES, HOT WATER HEATING LOOP |
| P-DW1 | TACO | 1915 | 5.8 | 1750 | 14 | 35 | 0.75 | 460 | 3 | WATER | NO | DOMESTIC WATER HEATING LOOP - NO GLYCOL IN THIS LOOP |

PUMP NOTES:
1. PROVIDE WITH BASE MOUNTED PUMPS WITH SUCTION DIFFUSER, GROUTED BASE PER DETAIL 4/M5-011
2. PROVIDE APPROPRIATE SEALS FOR PROPYLENE GLYCOL USE WHERE GLYCOL SPECIFIED.
3. PROVIDE WITH PREMIUM EFFICIENCY INVERTER DUTY MOTORS WHERE VFD USE IS NOTED.
4. FLUID PROPERTIES: SEE NOTES ON THIS SHEET FOR GLYCOL PERCENTAGE. SELECT PUMPS USING THIS FLUID.
5. SEE 4/M5-01 FOR PUMP DETAIL.
6. T.C. CONTRACTOR TO PROVIDE VFD'S WITH FACTORY DISCONNECT WHERE NOTED.
7. P-DW1 IS SHOWN ON PLUMBING PLAN PH1-01. SEE 1/PH1-01 AND 18/PH1-01.

Figure 25 Pump Schedule of PHP1/PHP2

For the evaluation of threshold flowrate for GSHP for turning on the ground loop, the flowrate for PHP1/PHP2 is determined. For the heat pump size Figure 25 shows the pump schedule for PHP1/PHP2 which is Taco Pump Model Number F1-5011.

To evaluate the flowrate of HPL (PHP1/PHP2), pump curve and pump performance curve is used from the information on the mechanical schedule for the pump from the entering water temperature in the heat pump loop, and the frequency of 54.1 Hz for the PHP1 pump is used as shown in the graph in Figure 26 for September 2019.

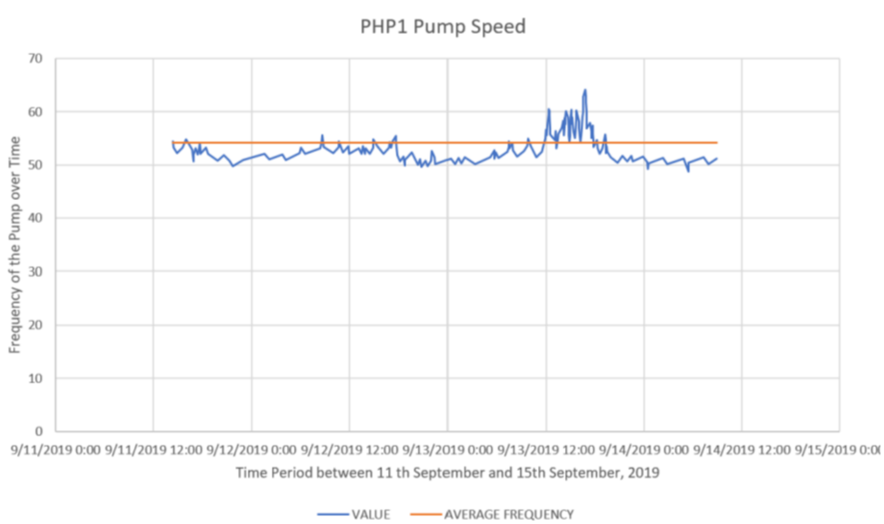

Figure 26. PHP1 Speed trend plot for time 11th September to 19th September 2019 with average speed plotted to be 54.1 Hz.

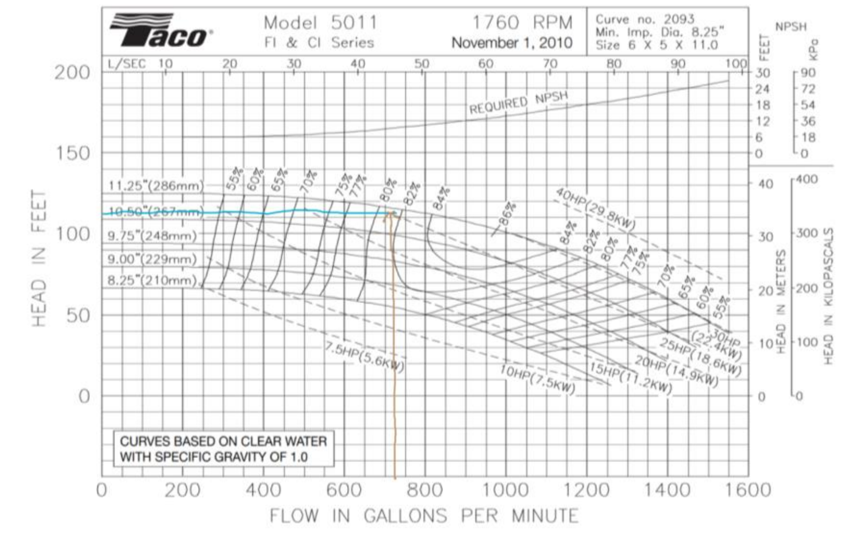


Figure 27. Pump Curve for Taco Pump Model Number F1-5011

Figure 26 shows that for HPL (PHP1), we found has the frequency of the pump is 54.1 Hz for the period of 4 days 11th September to 19th September 2019. The value 54.1 Hz is taken for the determination of the flowrate of PHP1 pump.

Figure 27 shows the pump curve, for 110 feet of head and 720 GPM operating at 60Hz (which is for the given pump performance curve for the given pump model). To find the current flow rate at a frequency of 54.1 Hz (from data base for PHP1/PHP2), we make use of the Affinity law. The Affinity law is stated in Equation (16).

$$\frac{Q1}{Q2} = \frac{n1}{n2} \text{ and } \frac{n1}{n2} = \frac{F1}{F2}\text{, thus } \quad \frac{Q1}{Q2} = \frac{F1}{F2} \tag{16}$$

Where,

Q = Flow rate (GPM),

n = rotational speed, and

F= frequency.

When Q1=720 GPM, F1= 60 HZ, F2 = 54.1 Hz, Q2 =?

Applying equation (16) above to get Q2, we have Q2= 649.2 GPM which is the flowrate of PHP1 which is greater than 140 GPM. Hence, the control logic Flowrate makes sense.

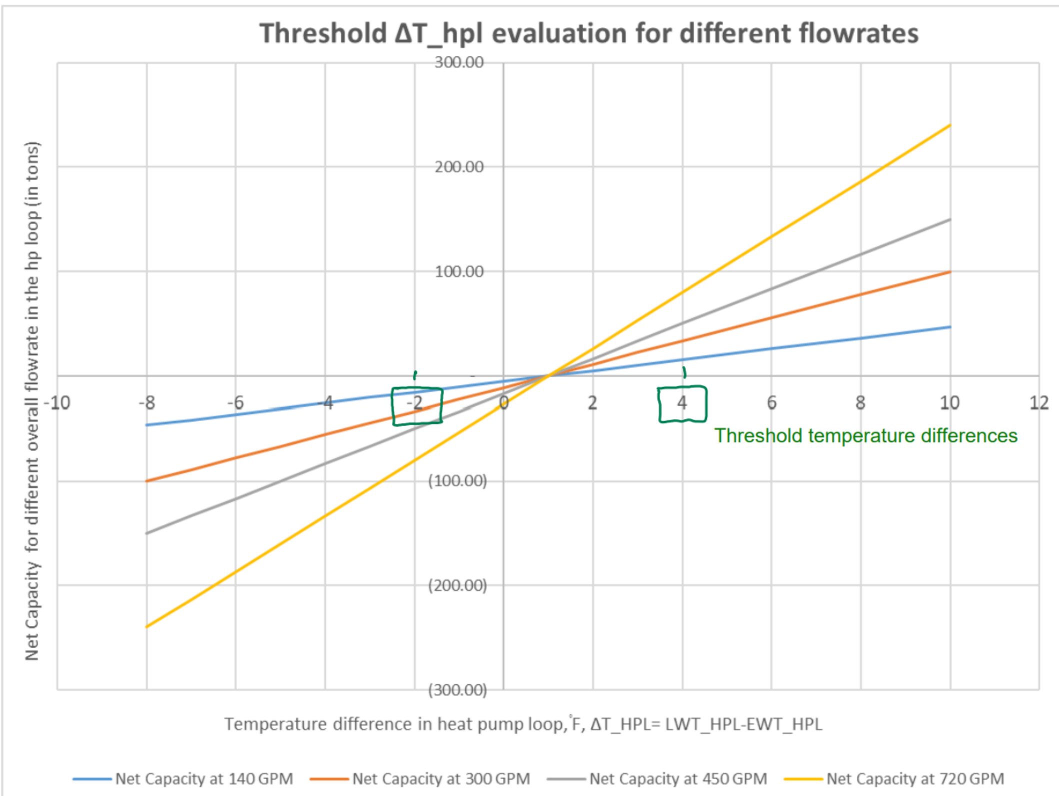


Figure 28. Graphical Analysis of threshold for ΔT_HPL for the activation of ground loop

Figure 28 shows that with the increasing overall flowrate of the heat pump loop system, the net capacity of heat pumps turned on in cooling mode demonstrated in the graph by positive net capacity value and the net capacity of heat pumps in heating mode demonstrated by the negative net capacity value. It shows that for the HPs in the cooling dominant with the net capacity is 15.6 tons or more for the lowest flowrate evaluated which is 140 GPM when $\Delta T\_HPL$ is greater than 4°F, whereas, when HPs are in heating dominant mode the net capacity is 33.3 tons or more when $\Delta T\_HPL$ is less than -2°F. 15.6 tons of net capacity or above is a good determination of the evaluation of the heat pump capacity to activate the ground loop.

Control Logic Decision Matrix on when to turn on the ground loop.

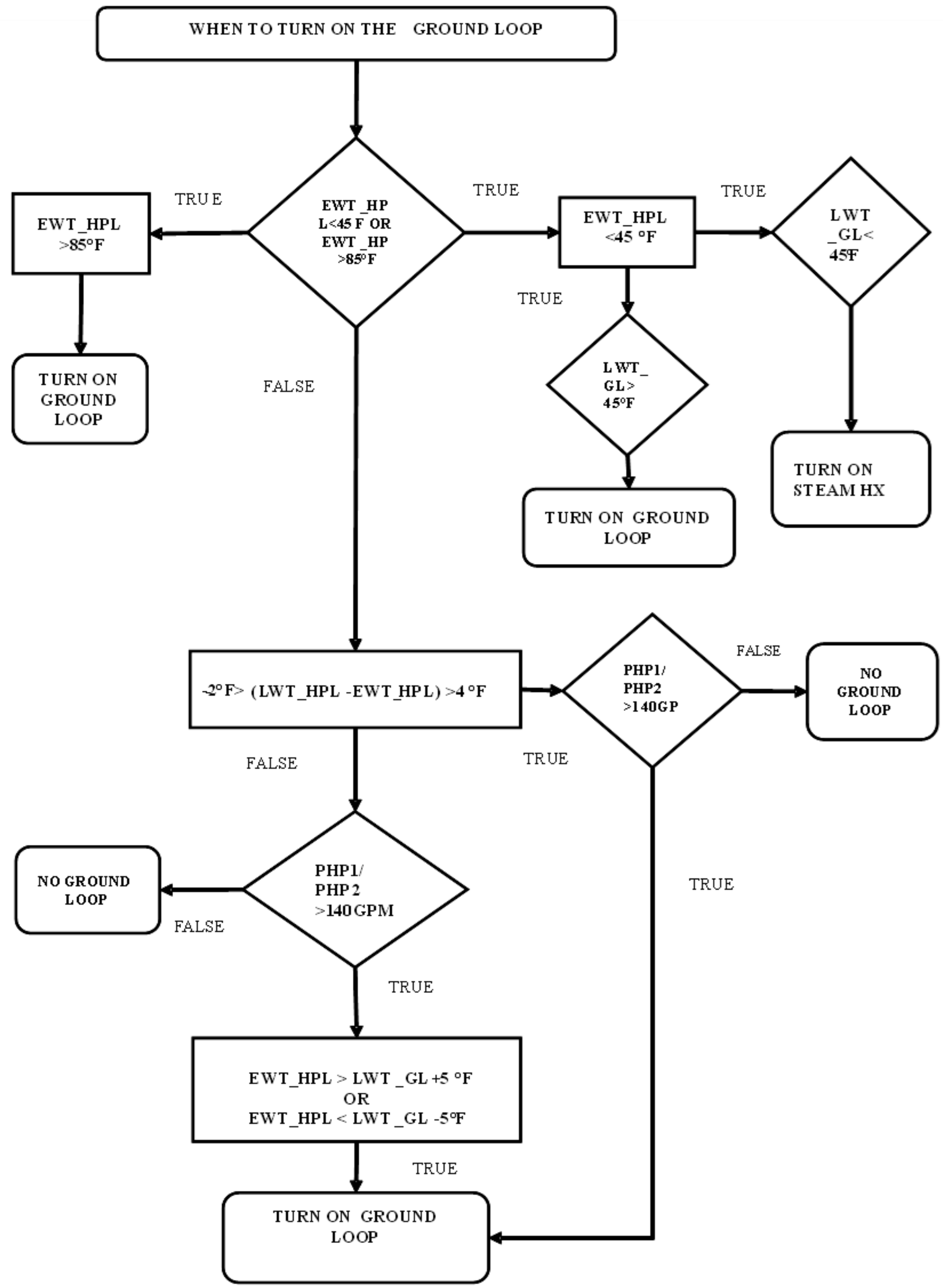


Figure 29. Flowchart (Logic Diagram) on when to turn on the ground loop.

Figure 29 is the control logic loop for the decision matrix on to determine when to turn on the ground loop in a GSHP system based on the entering water temperature in the heat pump loop (EWT_HPL) and the temperature difference between the leaving water temperature in the ground loop and the entering water temperature in the heat pump loop (ΔT_HPL) can vary depending on the specific design and requirements of the GSHP system. The control logic loop typically involves comparing the EWT_HPL and ΔT_HPL values to predefined thresholds or setpoints. For determining the optimal conditions for activating the ground loop, the flowchart for control logic for decision matrix is developed. The approach is based on the following steps:

1. Evaluate the entering water temperature in the heat pump loop (EWT_HPL). If EWT_HPL is less than 45°F or greater than 85°F, the ground loop is turned on. However, if EWT_HPL is less than 45°F and the leaving water temperature from the ground loop (LWT_GL) is also less than 45°F, the steam heat exchanger is activated.
2. Calculate the temperature difference between the entering water temperature in the heat pump loop and the leaving water temperature from the ground loop (ΔT_HPL). This temperature differential serves as a key parameter in determining the activation of the ground loop.
3. If EWT_HPL is more than 45°F or less than 85°F, and ΔT_HPL is greater than 4°F or less than -2°F, the flowrate of the heat pump should exceed 140 GPM to activate the ground loop. If this condition is not met, the ground loop remains inactive.
4. In cases where ΔT_HPL is less than 4°F or greater than -2°F, the flowrate of the heat pump loop should still exceed 140 GPM to activate the ground loop. Additionally, if the flowrate of PHP1 exceeds 140 GPM, EWT_HPL should be either greater than the sum of

LWT_GL and 5°F or less than LWT_GL minus 5°F. These conditions ensure that the temperature difference of 5°F between the ground loop and the buffer tank is within an acceptable range for activating the ground loop. (Chen & Richman, 2020)

5. Monitor the EWT_HPL and ΔT_HPL values continuously.
6. And continue the process once again at step 1.

The control logic optimization approach presented in this thesis has the potential to significantly improve the performance and energy efficiency of GSHP systems. By determining the optimal conditions for activating the ground loop, the approach can help to ensure that the ground loop is only used when it is necessary, which can lead to significant energy savings.

The approach is based on a few key factors, including the entering water temperature in the heat pump loop, the temperature difference between the entering and leaving water temperatures, and the flowrates of the heat pump loop. It's important to note that the specific thresholds or setpoints used in the control logic loop may vary depending on the design and requirements of the GSHP system. By considering all these factors, the approach is able to determine the optimal conditions for activating the ground loop.

# CHAPTER FOUR

# CONCLUSION

In conclusion, analyzing the energy savings and cost benefits associated with using a ground loop and steam to change the temperature of the heat pump loop in a building has provided valuable insights into the economic implications of these alternative methods.

The control logic optimization approach plays a crucial role in determining the activation of the ground loop in the heat pump loop of a GSHP system. A control logic decision matrix was developed to optimize the activation of the ground loop in NAH building at MSU based on specific criteria. Hence, the decision matrix developed for NAH can be applied for buildings with similar system design and operation by determining the building's specific threshold for flowrate and temperatures. The following steps are employed in this approach:

1. Initially, the entering water temperature in the heat pump loop (EWT_HPL) is evaluated. If EWT_HPL is less than 45°F and greater than 85°F, the ground loop is turned on. However, if EWT_HPL is less than 45°F and the leaving water temperature from the ground loop (LWT_GL) is also less than 45°F, the steam heat exchanger is activated. This step ensures efficient utilization of the ground loop based on temperature thresholds.
2. The temperature difference between the entering water temperature in the heat pump loop and the leaving water temperature from the ground loop is calculated as ΔT_HPL. This temperature differential serves as a key parameter in determining the activation of the ground loop.

3. If EWT_HPL is more than 45°F and less than 85°F, the value of ΔT_HPL is examined. If ΔT_HPL is greater than 4°F or less than -2°F, the flowrate of the heat pump loop in the system should exceed 140 GPM (the required flowrate for 30 minutes run time for the system volume) to activate the ground loop. If this condition is not met, the ground loop remains inactive.
4. In cases where ΔT_HPL is less than 4°F or greater than -2°F, the flowrate of the heat pump should still exceed 140 GPM to activate the ground loop. Additionally, suppose the flowrate of the HP loop exceeds 140 GPM. In that case, EWT_HPL should be either greater than the sum of LWT_GL and 5°F or less than LWT_GL minus 5°F, where 5°F is considered as a rule of thumb for ensuring that the temperature difference between the ground loop and the buffer tank is within an acceptable range for activating the ground loop. (Chen & Richman, 2020)

The control logic optimization approach aims to enhance the performance and energy efficiency of GSHP systems by optimizing the conditions for activating the ground loop. In general, to optimize energy savings with a GSHP, the ground loop is activated in specific cases. The specific temperature difference between the water leaving the buffer tank and the ground temperature to trigger the ground loop will depend on the GSHP system's design. A temperature difference of at least 5 °F between the ground temperature and buffer tank in the HP loop is a good indication to turn on the ground loop (rule of thumb) which is referred for the NAH, (Chen & Richman, 2020). If the water temperature leaving the buffer tank is close to the ground temperature, the ground loop may not be necessary as the heat pump might be able to provide sufficient heating or cooling without it.

The results of the study showed that the proposed approach can effectively optimize the activation of the ground loop and improve the performance and energy efficiency of the GSHP system. The approach can be used to determine the optimal conditions for activating the ground loop in a variety of buildings and climates.

In addition, this study's outcome demonstrated that using the steam heat exchanger would only make sense if the leaving water temperature of the ground loop is less than 45°F, which is only evaluated for theoretical condition does not practically happen in the ground loop in NAH. Hence, it does not make sense to use steam injection practically as the leaving water temperature of the ground loop is very rarely less than 45°F. In this scenario, it is more efficient to first use a ground loop, if possible, to optimize the performance and energy efficiency of the GSHP system.

# FUTURE WORKS AND RECOMMENDATIONS

The economic implications of using ground loops or steam injection to change the temperature of the heat pump loop in a building can be further investigated for better understanding of the long-term costs and savings associated with each approach. Additionally, evaluating the long-term performance and sustainability of GSHP systems with different temperature control methods, considering their scalability and applicability, will yield more meaningful results. Furthermore, developing a more sophisticated control logic optimization approach for determining the activation of the ground loop in the heat pump loop of a GSHP system can lead to more accurate results. The control logic optimization approach presented in the thesis is a good foundation, but it can be improved by incorporating more factors into the decision-making process, such as the weather forecast and the building's energy usage patterns.

# REFERENCES CITED


Steve Kavanaugh, K. R. (n.d.). Geothermal Heating and Cooling Design of Ground-Source Heat Pump Systems. W. Stephen Comstock.

Climo, M., Lind, L., Carey, B., & Bendall, S. (2014). The rise and rise of geothermal heat pumps in New Zealand. IPENZ Transactions, 41(March), 1–7. Retrieved from http://www.ipenz.org.nz/ipenz/forms/pdfs/IPENZTransactions41-RiseofGeothermaHeatPumps.pdf.

Chen, F., & Richman, R. (2020). The effect of shallow depth horizontal ground loop clearance on the heat loss of a single family residential dwelling. Journal of Building Engineering, 29. https://doi.org/10.1016/j.jobe.2020.101185

Hepbasli, A., Akdemir, O., & Hancioglu, E. (2003). Experimental study of a closed loop vertical ground source heat pump system. In / Energy Conversion and Management (Vol. 44). www.elsevier.com/locate/enconman

Michopoulos, A., Voulgari, V., Tsikaloudaki, A., & Zachariadis, T. (2016). Evaluation of ground source heat pump systems for residential buildings in warm Mediterranean regions: the example of Cyprus. Energy Efficiency, 9(6), 1421–1436. https://doi.org/10.1007/s12053-016-9431-1

Oladayo Oladeji, by. (2021). Assessment Of District Energy Integration Between Buildings.

J. Talbert, "Efficient energy modeling: a low carbon source energy assessment of proposed building interconnections based on emerging market modeling tools," Master's, Mechanical & Industrial Engineering, Montana State University, Bozeman, Montana, 2014.

J. Hays, "Energy modeling development and calibration for a mini district energy loop assessment comprised of a set of campus buildings," Master's, Mechanical & Industrial Engineering, Montana State University, Bozeman, Montana, 2019.

ASHRAE,2010. 2020 ASHRAE Handbook, Heating, Ventilating and Air Conditioning Applications

Tian, X., Mao, R., Pei, P., Wu, H., Ma, H., Hu, C., & Zhang, Z. (2022). Experimental study on temperature control optimization of ground source heat pump horizontal headers. Energy and Buildings, 277. https://doi.org/10.1016/j.enbuild.2022.112541

86-1, N. R. (n.d.). Closed-Loop/Ground-Source Heat Pump Systems Installation Guide.

Slagle, R. (n.d.). Electronics Cooling. Retrieved from https://www.electronics-cooling.com/2017/07/vapor-compression-cooling-works/.

U.S. Green Building Council USGBC. (n.d.). Retrieved from Projects: https://www.usgbc.org/projects/norm-asbjornson-hall

Gamage, K. (n.d.). Numerical Methodology for Feasibility Analysis of Ground Source Heat Pumps. *Research Gate*.