Enhancing Geothermal Heat Pump Systems with Parametric Performance Analyses

by

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CERTIFICATE OF APPROVAL

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The undersigned certify that the student has presented **[his/her]** thesis, that the thesis is acceptable in form and content and that a satisfactory knowledge of the field covered by the thesis was demonstrated by the candidate through an oral examination. They recommend this thesis to the Office of Graduate Studies for acceptance.

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Abstract

Parametric performance analyses and comparison of a basic geothermal heat pump, a heat pump cycle with motor cooling/refrigerant preheating, and a heat pump cycle utilizing an economizer with respect to first law is conducted through simulation. Changing compressor, pump, and motor efficiency, along with condenser pressure, evaporator pressure, degree of subcooling at the condenser exit and degree of superheating at the evaporator exit is investigated. Economizer arrangements yield the highest coefficient of performance and resilience to change in COP with variation in evaporator pressure, and degree of superheating and subcooling. The basic vapor compression and motor cooling/refrigerant preheating systems have the lowest COP throughout and greatest resilience to variation in compressor efficiency, motor efficiency and condenser pressure. Motor cooling/refrigerant preheating and economizers have advantages over basic vapor compression cycles. Motor cooling reduces ground loop heat exchanger length with similar COP, and economizers allow for an increase in COP compared to the basic cycle.

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Table of Contents

List of Figures

List of Tables

Nomenclature

Symbols

A_c	Cross sectional area, m^2
c_p	Specific heat at constant pressure, kJ/kg·K
COP_{HP}	Coefficient of performance for heat pump cycle
COP_{System}	Coefficient of performance for entire system
D_i	Inner diameter of ground loop piping, m
$\dot{E}_{motor,comp}$	Electrical power consumed by compressor motor, kW
$\dot{E}_{motor,pump}$	Electrical power consumed by pump motor, kW
$\dot{E}_{motor,total}$	Sum of the electrical power consumed by compressor and pump motors, kW
$\dot{E}_{motor,waste}$	Rate of waste energy available for transfer from compressor motor, kW
f	Friction factor
h_{conv}	Convection heat transfer coefficient, $W/m^2 \cdot {}^oC$
h_f	Specific enthalpy at saturated state, kJ/kg
h_{fg}	Specific enthalpy difference between saturated vapor and saturated liquid, kJ/kg
h_i	Specific enthalpy at state *i*, kJ/kg
h_{ia}	Actual specific enthalpy at state *i*, kJ/kg
h_{is}	Ideal specific enthalpy at state *i*, kJ/kg
h_l	Specific enthalpy of species *l* in brine mixture, kJ/kg
h_m	Specific enthalpy of brine mixture, kJ/kg
k_{brine}	Thermal conductivity of brine fluid, W/m·ºC
k_{grout}	Thermal conductivity of grout material, W/m·ºC
k_{pipe}	Thermal conductivity of pipe material, W/m·ºC
$l_{parallel}$	Length of a single parallel loop, m
mf_l	Mass fraction of species *l* in brine mixture
$\dot{m}_{GL,Evaporator}$	Mass flow rate of brine through the evaporator, kg/s

$\dot{m}_{GL,parallel}$	Mass flow rate of brine through a parallel loop, kg/s
$\dot{m}_{main}$	Mass flow rate through the main circuit in system 3, kg/s
$\dot{m}_{ref}$	Mass flow rate of refrigerant through condenser, kg/s
$\dot{m}_{supp}$	Mass flow rate of refrigerant through the supplementary circuit in system 3, kg/s
Nu	Nusselt number
η_{comp}	Compressor isentropic efficiency
η_{EM}	Electric motor efficiency
n_{pl}	Number of parallel loops within ground loop
η_{pump}	Pump isentropic efficiency
ρ	Density, kg/m^3
P	Pressure at state *i*, kPa
$\Delta P_{parallel}$	Pressure drop within a parallel loop, kPa
Pr	Prandtl number
$\dot{Q}_{evap}$	Heat input of evaporator to refrigerant, kW
$\dot{Q}_{GL,actual}$	Actual rate of heat transfer required from the ground, kW
$\dot{Q}_{GL,parallel}$	Rate of heat transfer to a single parallel loop, kW
$\dot{Q}_{ht}$	Rate of heat transfer from ground to brine fluid, kW
$\dot{Q}_{load}$	Heating load of building, kW
$\dot{Q}_{waste\ heat}$	Rate of heat transfer from the compressor motor to the refrigerant, kW
Re	Reynolds number
R_i	Thermal resistance of section *i*, °C/W
r_i	Radius from center of ground loop piping of section *i*, m
R_{total}	Total thermal resistance, °C/W
$T_{\infty 1}$	Temperature of outer grout wall, ºC
$T_{\infty 2}$	Average fluid temperature between states 7 and 5, ºC
T_i	Temperature at state *i*, °C
μ	Dynamic viscosity, kg/m·s
$V_{avg,parallel}$	Average fluid velocity through a parallel ground loop, m/s
v_i	Specific volume at state *i*, m^3/kg

$\dot{W}_{comp}$	Power consumption of compressor, kW
$\dot{W}_{comp,i}$	Power consumption of compressor i, kW
$\dot{W}_{pump}$	Power consumption of pump, kW
$w_{pump,actual}$	Actual pump work, kJ/kg
$w_{pump,ideal}$	Ideal pump work, kJ/kg
X	Ratio of mass flow rates in the main and supplementary circuits within system three
x	Quality

Acronyms

GHP	Geothermal heat pump
GL	Ground loop
GSHP	Ground source heat pump
HP	Heat pump

Chapter 1: Introduction and Objectives

1.1 Introduction

In this day and age there is an ever increasing demand for energy. Energy is commonly utilized for the creation of electricity and heat. One of the major sources of producing these two necessities is fossil fuels. Fossil fuels such as coal, natural gas, furnace oil, gasoline, diesel, and kerosene provide the required energy to meet the demand for much of the world. Even though a large portion of the energy demand is met by burning fossil fuel, it comes at a cost. A major problem of using fossil fuels is the pollution that is introduced into the environment through the combustion of these fuels. Burning fossil fuels is thought by many to contribute to the rising amount of greenhouse gases on a global scale. Another problem with using fossil fuels that must be addressed is the idea that there is only a finite amount of resources that society can exploit. Even though there are different perspectives on how long we can continue to use these fuels until there is a worldwide shortage, the fact that there will be a shortage at some point in the future is predicted [1]. Hammond [2] argues that fossil fuel depletion along with green house gas emissions are two of the most important factors to be considered for sustainable and environmentally benign energy systems.

With these facts in mind it can be seen that a key path to reduce the amount of polluting emissions and the risk of a fossil fuel shortage is to reduce our dependency on fossil fuels. This means reducing the amount of fuel that we use per capita and/or replacing fossil fuels with alternative energy sources. Recognizing this has created opportunity for new technology to emerge in the last 60 years. Alternative resources should preferably be more environmentally friendly and economical than conventional fossil fuel in order to allow wide scale application.

The earth is a source and reservoir of abundant energy which extends past the idea that it contains fossil fuel. The earth's crust is a large source and storage medium of thermal energy. The thermal energy contained within the ground is referred to as geothermal,

which literally means Earth and heat. The U.S. Department of Energy [3] has done studies into the availability of geothermal energy and has concluded that the energy contained within geothermal resources is abundant and actually exceeds fossil fuel resources in terms of the amount of energy that is potentially available. Along with this, geothermal systems are typically more environmentally friendly than conventional energy systems utilizing fossil fuels. The degree of green house gas emissions associated with geothermal systems is a fraction of what is produced through the use of conventional fossil fuel energy systems for both electrical and heating applications [3, 4].

There are currently three prime ways to utilize geothermal energy. These include: electricity production through power plants, direct use of heat, and geothermal heat pumps. Electricity production, direct use, and geothermal heat pumps use high, moderate, and low temperatures, respectively. High temperature geothermal resources are usually classified as having a temperature of greater than 150ºC [4, 5]. Moderate temperature resources are classified as having temperatures between 90ºC and 150ºC [4, 5]. Low temperature resources refer to those that have a temperature below 90ºC [4, 5]. High and medium temperature resources are usually the product of thermal streams that are produced by the molten core of the earth. Thermal energy flows from deep within the earth and collects in areas of water, or rock. Low temperature resources are mostly created through the collection of solar energy within the ground. Energy from the sun strikes the ground where it is absorbed and stored in the soil. The energy can be extracted from the soil at reasonable depths for use in heat pump systems [4].

Figure 1.1 shows a multitude of applications that coincide with different possible uses for the thermal energy.

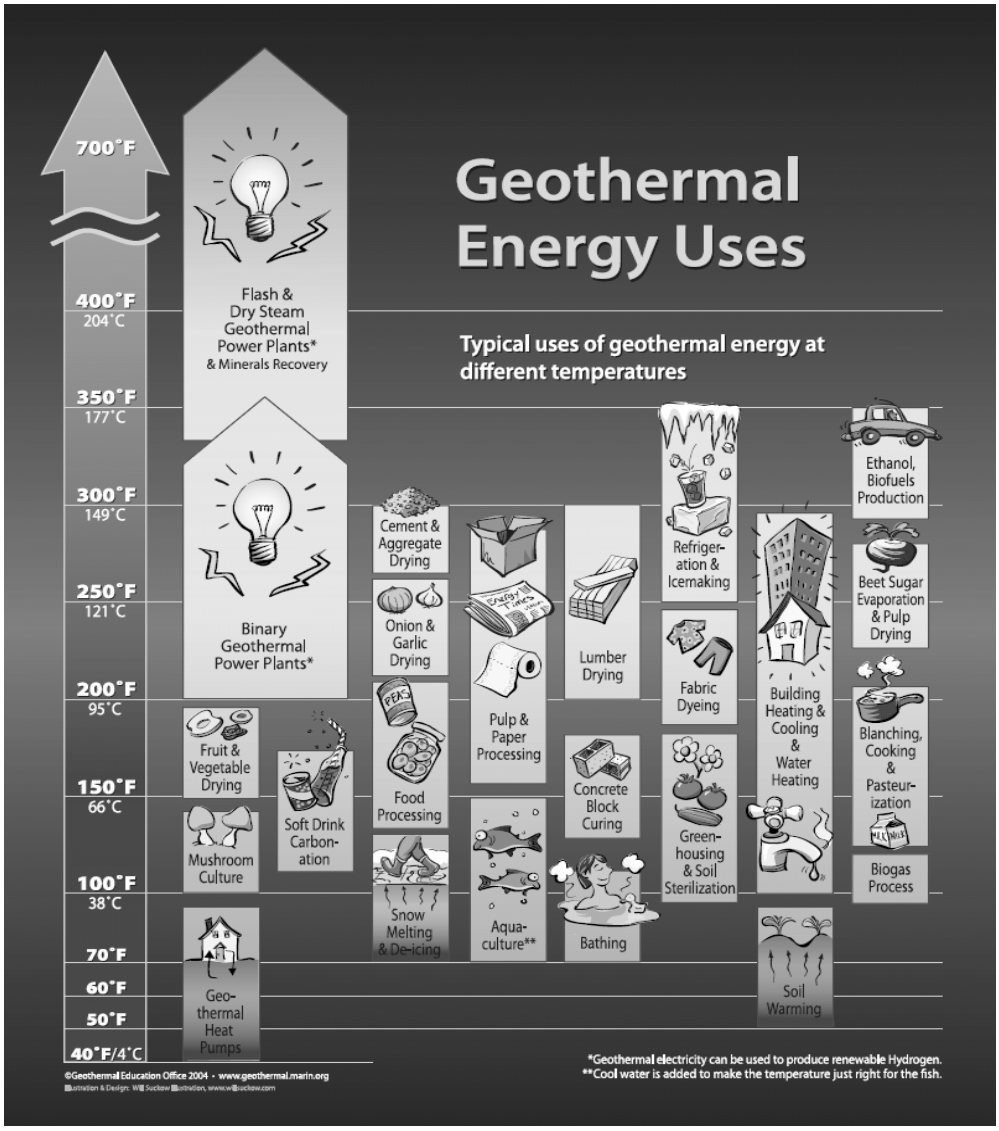

Figure 1.1: Sample of geothermal applications for various ground temperatures [3]

Since high and medium temperature resources are fed by thermal energy flowing from the earth's core, the depth at which they are found is often substantial. In order for a high or medium temperature resource to be utilized it must be close enough to the earth's surface to be economically feasible to reach. At great depths the drilling cost can be substantial and commonly makes the use of such resources economically unfeasible.

On the contrary, low temperature geothermal resources are abundant and can be utilized in almost any location around the world. The energy is supplied by the sun to the ground, where it is absorbed and stored as low temperature thermal energy at relatively shallow depths compared to high and medium temperature resources. The energy can be extracted at low depths with relative ease compared to the other two geothermal resource types. The best way to utilize this low temperature resource is through the use of heat pumps. A heat pump has the ability of extracting low temperature heat from a medium and upgrading it to a high temperature energy flow for practical use. When heat pumps are combined with thermal ground resources they create a system that is more environmentally benign and efficient compared to conventional space heating technology.

Since low temperature resources are abundant and easy to exploit there is increasing interest in the design and implementation of geothermal heat pumps. Over the years, many studies have been performed and reported on such systems, in part to provide an improved understanding of system utilization and performance. Nevertheless, there are areas in the available publications in which information is not provided, especially in terms of the scope and possible applications of the information presented. Thus additional research is needed.

A vast number of publications offer general guidelines to assist in designing and implementing geothermal heat pump systems. Often these guidelines utilize “rules of thumb” and are intended to provide individuals with a tool for geothermal heat pump design. Use of these rules allow for a system that usually works, but does not offer much insight into the optimization aspects for each individual system arrangement and operating conditions. Using these techniques may also introduce inaccuracies into the system design where the resulting system may act differently than originally intended; for example the application temperature (temperature of the refrigerant at the compressor exit) could be different in real life from what is needed by the heat distribution system, or the ground loop length may be too long or short where the systems will either not have enough area to allow for the proper amount of heat transfer between the system and the

ground or the initial cost could be unnecessarily increased by having more piping then required [6].

Much literature is available in the field of geothermal heat pumps. For the most part it is found that the studies conducted in the past have done so under static conditions. The studies reported show a system arrangement, either theoretical or experimental, and present the analysis of the systems but typically show the results under static conditions. Limited effort is devoted to studying how varying operating conditions affects system performance and other characteristics of the systems.

Heat pump technology has been in commercial use for just over sixty years with the first successful commercial project installed in the Commonwealth Building (Portland, Oregon) in 1946 [7]. For the most part compression type heat pumps have been at the forefront of system design and are widely utilized in operating systems today. Past studies with regard to compression heat pumps have usually been performed on the basic system arrangement utilizing the basic compression cycle. New, more advanced systems are being developed that still use compression within the heat pump cycle but have implemented different design features that are not used within basic systems. Examples of these systems include the system presented by Wang et al. [8] where the basic vapour compression cycle includes a method of cooling the compressor motor using the refrigerant, which allows the refrigerant to be heated before it encounters the heat exchanger connected to the ground loop. Another advanced system is presented by Ma and Chai [9] that includes an economizer arrangement that creates a main and supplementary refrigerant flow path and the inclusion of two compression processes. Advanced systems are often assessed only on a basic level, with a basic thermodynamic analysis performed for static conditions.

The above observations suggest research is needed to develop more detailed models and analysis methods in order to provide a better understanding of heat pump design. In particular, refined methods are needed that provide better accuracy than “rule of thumb” approaches. Parametric analyses are also needed to help improve performance and

efficiency, given the limited studies that consider variations in system parameters. Investigations into how variations in different system parameters affect heat pump system operation and performance would enhance knowledge and would also provide a better understanding of the validity of available “rules of thumb” that are presented throughout literature. Since limited information is available about geothermal heat pumps employing advanced vapour compression cycles, models for these systems along with relevant parametric analyses would also be beneficial for improving the understanding of such systems, thereby, contributing to better designs of ground source heat pump systems.

To help address the needs related to geothermal heat pump design and operation as outlined above, parametric investigations using sensitivity analyses relating to heat pump components and operating parameters are carried out in this thesis, considering the basic vapor compression heat pump cycle as well as the advanced arrangements presented by Wang et al. [8] and Ma and Chai [9].

1.2 Objectives

It is apparent that limited work has been reported in the literature in the last 10 years with regard to thermodynamic analyses of advanced vapor compression heat pump cycle arrangements in conjunction with geothermal systems. Analyses investigating how various operating conditions within these heat pump arrangements affect the performance and ground loop requirements are needed. Comparisons between advanced vapor compression arrangements and the basic vapor compression cycle have been performed but with limited scope, where the systems have been compared under static operating conditions.

The present work attempts to contribute to the understanding of advanced systems through comparisons of the advanced systems with the basic vapor compression cycle with respect to heating. Parametric assessments using sensitivity analyses are also carried out, including an investigation into which operating parameters affect each system the

most. Such an investigation will enhance understanding of the operation and limitations of advanced systems as well as contribute to a better understanding of the basic system arrangements. Comparisons and parametric analyses will also allow for the identification of parameters that are important in the design and application of such systems.

The objective of the present study is to model, analyze, and compare advanced geothermal heat pump arrangements and the typical basic vapor compression arrangement, for heating purposes, with varying operating conditions through first law analysis. To the best of the author's knowledge the work reported here has not been published previously.

1.3 Scope of Present Work

First law analysis is performed for three different ground source heat pump arrangements. The analysis includes investigations into how varying compressor, pump, and motor efficiency affect the system performance. Variations in operating conditions are investigated, with the inclusion of condenser and evaporator pressure, and the degree of subcooling and superheating at the condenser and evaporator exit.

The heat pump systems investigated include a basic vapor compression cycle, vapor compression cycle including electric motor cooling presented by Wang et al. [8], and vapor compression cycle with an economizer presented by Ma and Chai [9]. All systems use R-134a as the refrigerant. The two advanced systems were chosen for the study because of their lack of maturity and coverage within literature. The systems will be analyzed and compared under common operating conditions in an attempt to determine their sensitivities to changes involved with each system for different scenarios.

The systems are analyzed assuming that there is one centralized heat pump system supplying a space with the required heat. This is explained by Natural Resources Canada [6] as the most basic system arrangement for heating. Other arrangements exist where

multiple heat pump units are contained within a building that extract and release heat to a main circuit that is connected to a central earth connection. The multiple heat pump arrangement is used to allow for cooling and heating of different spaces within the same building. Since this study is restricted to heating purposes the single heat pump arrangement is adopted.

The model for each system is composed of two different loops including the heat pump cycle and ground loop heat exchanger, commonly referred to as the ground loop. The arrangement and analysis process for the ground loop is identical for application to each of the three systems. A vertical borehole ground loop connection is utilized within the study. The brine in the GL is assumed to be a water/propylene glycol mixture.

The created models are analyzed using computer simulation. Each system model is input into computer software that determines the thermodynamic properties of the fluids involved in the systems. The software also allows important performance and ground loop characteristics to be evaluated for varying operating parameters. The software specifically utilized is Engineering Equation Solver.

It is understood that for a complete understanding of the practicality of a system arrangement, economical and environmental analyses are required [10]. The present study does not explore these aspects of system design, but rather concentrates on the operation and performance aspects of the systems. Furthermore, a second law analysis, which would provide a more comprehensive assessment, is not performed.

1.4 Outline of Thesis

The following is a general outline of the present study:

- General background of geothermal heat pump units with the inclusion of literature review of recent work (Chap. 2)
- Approach and methodology used for the investigation involved in present work with the inclusion of assumptions that are utilized throughout the study (Chap. 3)
- Descriptions and illustrations of the various systems involved in the present study (Chap. 4)
- Analysis developed for the investigation including an outline of equations utilized (Chap. 5)
- Description of simulation and software (Chap. 6)
- Results for each investigation with discussion of the main points of interest (Chap. 7)
- Summary of the main results that have been found within the study (Chap. 8)
- General conclusions that can be drawn from the study given the main results (Chap. 8)
- Recommendations into how the study could be used, areas of improvements, and future work (Chap. 8)

Chapter 2: Background

The concept of heat pumps has been understood since the 1800's and applications of heat pump systems have existed for over 60 years [7]. Basic compression heat pumps operate on the principle that heat can be moved from one medium to another at a higher temperature with the input of energy. Much like a refrigerator, thermal energy can be extracted from a cool location, upgraded to a higher temperature, and moved to a warmer area where it can be released. In a heat pump the energy is output in the form of useable heat, which is often used to provide a comfortable environment within a contained space. An important trait of a heat pump system is its ability to provide more thermal energy to a building than the energy input required to run the unit [11].

Geothermal heat pumps (GHPs) are also known as Ground Source Heat Pumps (GSHP), earth energy systems, GeoExchange heat pumps, ground-coupled heat pumps, earth-coupled heat pumps, ground-source systems, among other variations [12, 13]. GHP systems use the earth, at depths from less than one meter to over 150 m below the surface, as their heat source [14, 15, 16].

GHP systems are comprised of three main components that work together to supply heat to a building as can be seen in Figure 2.1.

The components include:

1. *Geothermal heat pump unit*- The device that moves heat between the building and the earth.
2. *Earth connection*- The heat exchanger loop used to extract thermal energy from the ground for use in the heat pump unit.
3. *Interior heat distribution system*- The systems used to condition and distribute the heat throughout the space [17].

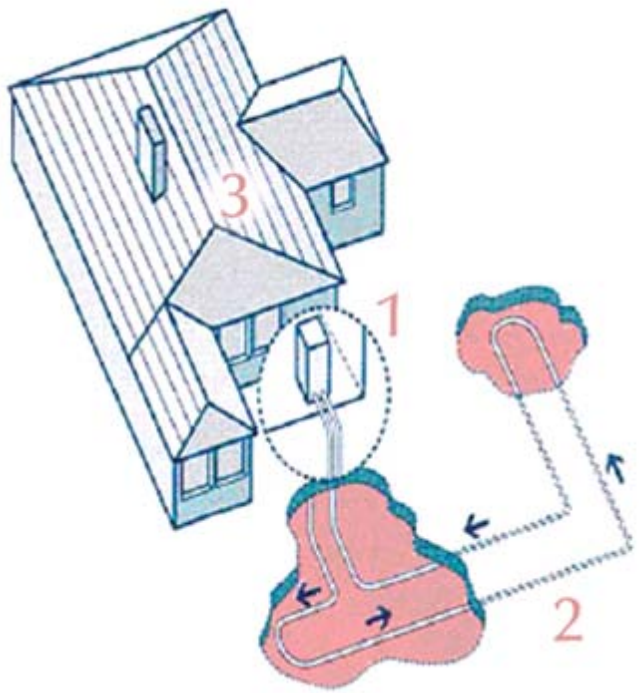

Figure 2.1: Main components of a GHP system [17]

2.1 Heat Pump Systems

The heat pump system can be thought of as the heart of GHP systems. It is the essential part of the entire system that allows the use of ground heat to be utilized for heating buildings. Fundamentally a heat pump is an apparatus which moves heat from a low-temperature location to a high temperature location with the contribution of work input, thus allowing for a supply of usable heat. In modern heat pumps the work is supplied by means of electricity. Even though electricity is supplied to these systems, the system does not convert electricity directly to heat. The electricity is used to run components in the system to provide the necessary work for the concentration and transport of thermal energy [4, 11]. After use, the electricity is dissipated in the form of heat.

Heat pumps use a vapor-compression refrigeration cycle, like the one used in common household fridges. The fluid used in heat pump systems can be of a variety of refrigerants, but the actual refrigerant used depends of the overall characteristics and

requirements of the GHP system. The ability of refrigerants to boil, at low temperatures, into a vapor and then condense back into a liquid is the main trait that allows heat pumps to operate. Simple thermodynamics explain that all refrigerants have a recognized relationship between boiling temperature and pressure. Heat pumps drive the heat flow between the heated space and the earth connection by controlling pressure and temperature by means of compression and expansion [11, 17].

There are five major components incorporated into a GHP unit. These are all housed in one casing and include; a compressor, expansion valve, reversing valve, and 2 heat exchangers. There are various other smaller components and accessories such as fans, piping, and controls, but the five listed above are the essential components in the design of a heat pump system (Figure 2.2) [12, 18]. The heat exchangers include one between the ground loop and heat pump (evaporator), along with one between the heat pump and the space (condenser).

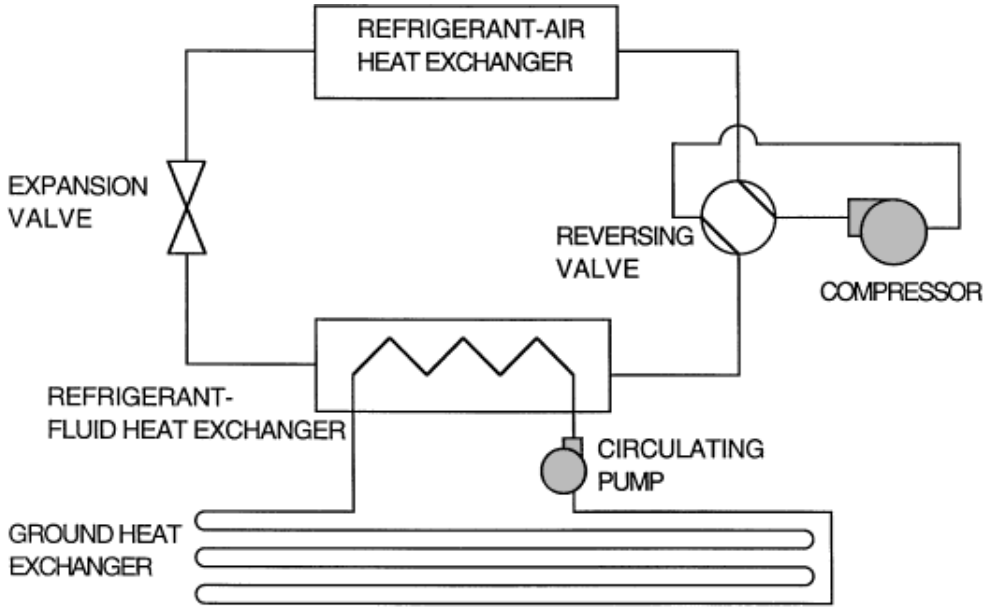

Figure 2.2: Basic layout of heat pump unit [19]

Heat pumps used for heating, which extract heat from the earth, operate in the following fashion [17]:

1. Heat is absorbed in the earth loop and transported to the evaporator, as seen in Figures 2.3 and 2.4.
2. Inside the heat pump unit cold refrigerant, mainly in the liquid state, enters the evaporator. The temperature of the refrigerant is lower than the fluid from the earth connection so heat moves from the earth connection loop to the refrigerant. The heat causes the refrigerant to boil and become a low pressure vapor; the temperature only increases slightly.
3. The cool vapor is then fed into an electrically-driven compressor, where the pressure is increased, resulting in increased temperature. The product is high temperature, high pressure vapor.
4. The high temperature vapor is then fed into the condenser. At this point, the refrigerant is at a higher temperature than the building that it is heating, inducing heat transfer from the refrigerant to the building. As the heat is transferred, the temperature of the refrigerant drops and it condenses back into a high pressure, high temperature liquid.
5. The hot liquid is then fed through an expansion valve that reduces the pressure of the refrigerant, resulting in the temperature to drop drastically. The refrigerant is back to its original state and is fed into the evaporator to start the process over again.

An example of the basic T-s diagram of the process is illustrated in Figure 2.3

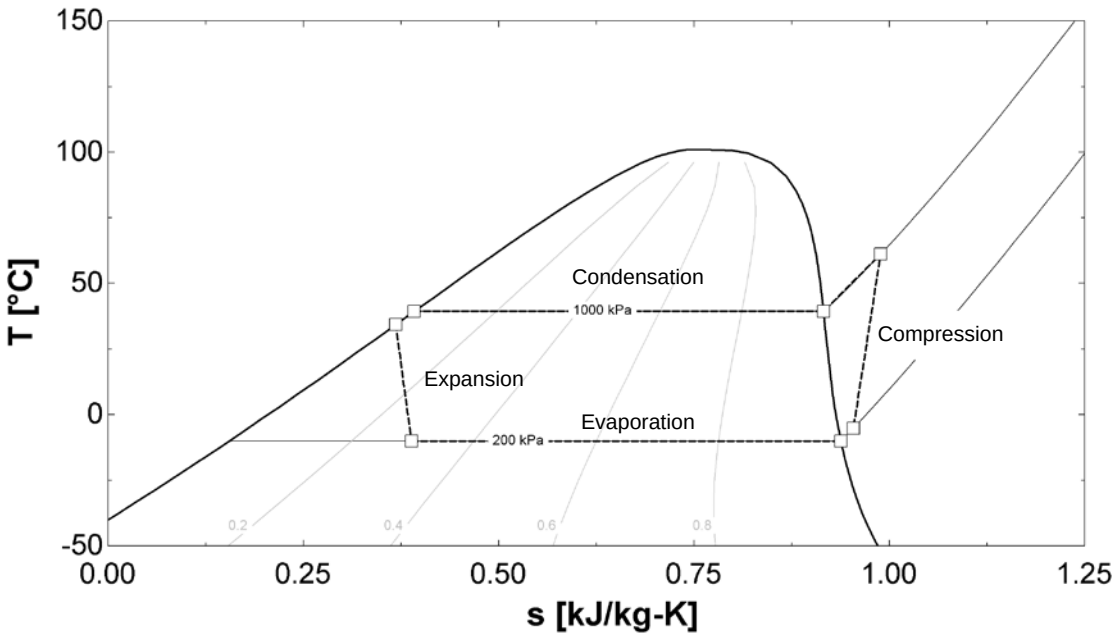

Figure 2.3: T-s diagram of basic heat pump cycle

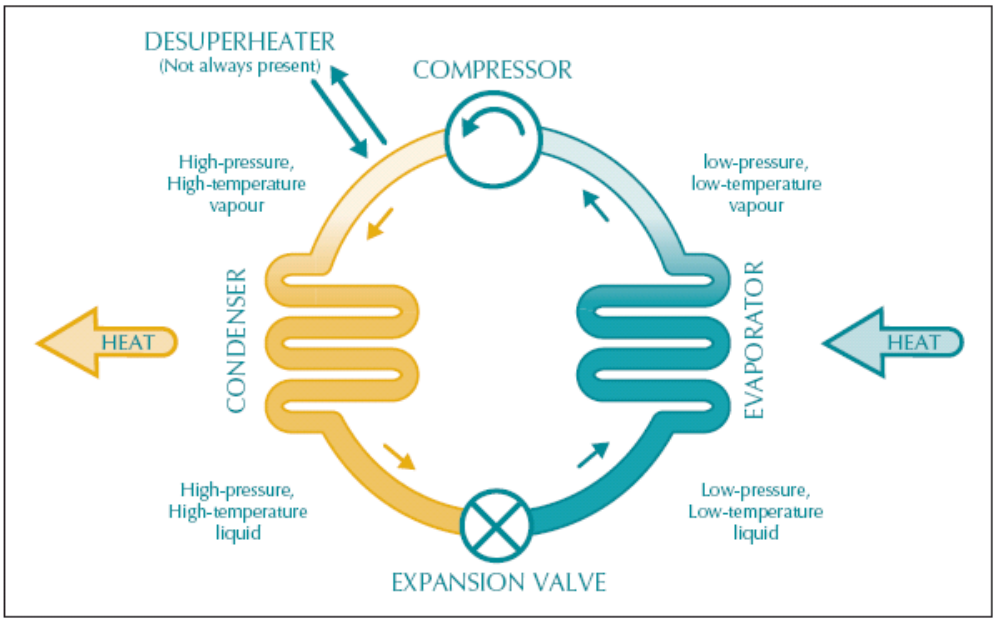

Figure 2.4: Diagram of basic heat pump cycle [17]

Many heat pump systems have a cooling mode that removes thermal energy from a space. When in cooling mode a reversing valve is used to move the fluid in the opposite direction around the cycle. The purposes of the heat exchangers are reversed, where the heat exchanger between the earth connection and refrigerant becomes the condenser and the heat exchanger between the refrigerant and building becomes the evaporator [11, 17].

Figure 2.4 shows the inclusion of a desuperheater. A desuperheater is an auxiliary heat exchanger that is used to provide heat to a hot water tank. Located at the compressor exit it transfers excess heat from the compressed gas to water that circulates a hot water tank. This can greatly reduce or eliminate the energy that would usually be required for hot water production [17].

When technological devices are discussed the scale of merit is usually in terms of efficiency. Efficiency is the ratio of energy output and the amount of energy input to a device expressed in percent. When heat pumps are discussed the concept of efficiency does not hold the same significance as with other devices. Since geothermal heat pumps deliver more heat output compared to the required energy input the systems appear to have efficiencies greater than 100%. The term coefficient of performance (COP) is therefore used when analyzing and comparing the performance of heat pumps against each other or to other heating technologies. Like efficiency, it is the ratio of output energy to input energy for a device but it is not translated into percent. In the case of heat pumps this is the ratio of heat output to the electrical power consumption [12]. Detailed description of COP and its definition is outlined in Chapter 5. Geothermal heat pumps typically have a COP ranging from 3 to 6 [3, 4, 6]. The range of COP is created by different earth connection setups, system sizes, earth characteristics, installation depths, and local climates among other performance affecting related characteristics [20].

As mentioned above one of the three major components of a geothermal heat pump system is the heat distribution system. This is the system that moves the heat throughout the building that is being supplied by the heat pump. There are two main types of

distribution systems, which use either air or water as the medium for moving the heat. In geothermal heat pumps these are referred to as water to air and water to water systems. Currently the most common system type is water to air in North America. Water to air describes heat pump systems that get their heat from fluid in the ground loop and transfers the heat to air inside the building. This is done by having an air coil, which is heated by the condenser of the heat pump unit, and a fan that blows air across to increase its temperature. The air is moved throughout the house and enters different rooms through air vents, very much the same as a conventional forced air furnace [17, 20].

Water to water systems are also commonly known as hydronic systems. Similar to the water to air system, heat is extracted from the ground loop. The heat is then distributed throughout the building using water as the carrier. The system works by pumping water through the condenser of the heat pump unit in order to extract the heat. The water is then pumped around the house delivering the heat to the spaces by way of in-floor radiant heating setups, radiators, or localized air coils heated by the water. These systems allow for heating with the use of low temperatures. In-floor radiant heating has a typical design temperature range of 18-22°C [11, 17, 20, 21].

Hybrid systems exist as well, where there is a combination of both these types of distribution methods to allow for high flexibility of system and improved control of space temperature.

Table 2-1: Design temperatures for different heat distribution methods [21]

System type	Indoor design range
100% radiant floor	18-21ºC
Mixed radiant floor/forced air	20-22°C
Baseboard	20-22ºC

2.2 Earth Connections

When the source of heat is considered there are two main types of heat pumps, which are used in heating applications: air source heat pumps and geothermal heat pumps. Air source heat pumps use the ambient air outside as a heat source where geothermal heat pumps use the surrounding ground.

The ambient air has a very high temperature variation throughout the year, even on a daily basis. The opposite is true within the ground at the same location. The temperature below ground does not change significantly over the course of a day or even a year, except very near the surface. On a daily basis, the temperature will fluctuate significantly in depths between 0.3 m and 0.8 m; below that depth, the temperature variations decrease [22]. For the most part, changes cannot be seen on a day to day basis, but are more pronounced between seasons.

GHPs utilize the fact that, below the surface, the ground has a relatively constant temperature, warmer than the air during winter and cooler than the air during summer. This means that the ground temperature remains closer to the desired temperature inside a building. Air source heat pumps are subject to large differences between the ambient air and desired building temperatures. When there is a large variance between the inside and outside temperature, more work is required to provide the same amount of heating, reducing the COP [18]. If the temperature difference is too large, a heat pump will not be able to move heat from one location to another, rendering it inoperable.

The ground has a constant temperature at a certain depth at a particular location. At shallow depths, the surface environment interacts with the soil either adding heat to it or removing it. As the depth increases, the interaction is limited. The depths at which temperatures stabilize indicate the interface at which seasonal influences are become less significant than the heat flowing to the surface from deep within the earth [4]. Even though the movement of heat from the deep within the earth allows for constant temperature at greater depths the performance and COP are independent of the actual heat transfer intensity of a particular gradient [22].

Figure 2.5 shows the variation of ground temperature with increasing depth for Ottawa, Canada. It can be seen that as the depth increases, the outlying warm and cold temperatures begin to converge. Depending on factors like incoming solar radiation, snow cover, air temperature, precipitation and thermal properties of soils, there exists a depth at which there is little or no temperature variation. In Canada depths below 10 m usually have constant annual temperatures, as seen in Figure 2.6. As the depth increases the temperature line flattens, indicating constant temperature [22].

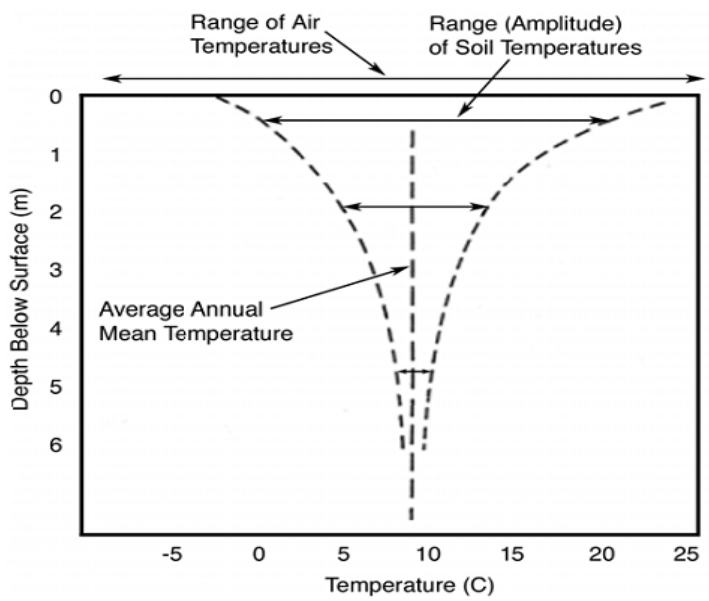

Figure 2.5: Variance in ground temperature with depth [22]

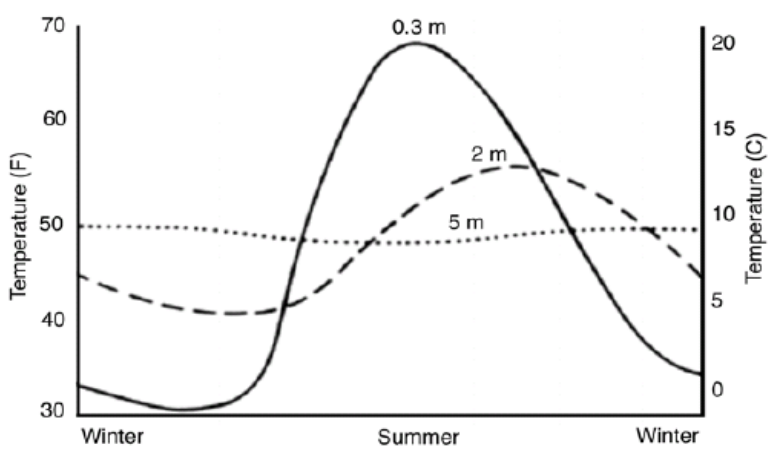

Figure 2.6: Annual ground temperature range for different depths [22]

Essentially, an earth connection is a collection of pipes through which fluid moves, absorbing heat from the ground, which is then fed back to the heat pump unit. The earth connection in a GHP is the component that separates this technology from conventional heat pump systems. Two main categories of ground loops in common use are double loop and single loop configurations.

2.2.1 Double Loop

A double loop configuration is the most common system type. It involves an earth connection that is separate from the heat pump unit. Heat is transferred to the refrigerant via a heat exchanger that has water or a water/antifreeze (propylene glycol, denatured alcohol or methanol) solution entering. The water or water/antifreeze mixture is circulated through tubing from the heat pump unit to the earth, where it absorbs heat and is returned to the heat pump. The use of high grade pipe is not usually necessary as these systems typically operate under low pressure [23]. The standard for the pipe materials used by installers at the moment are high density polyethylene and polypropylene [12].

There are two subtypes that coincide with the double loop configurations. These include closed loop systems and open loop systems.

2.2.1.1 Closed Loop Systems

Closed loop systems are the most commonly used around the world at present. The liquid is always enclosed in a circulating loop and there is no direct interaction between the earth and the heat transfer fluid. The only interaction that occurs is the heat transfer across the piping material from the earth to the fluid.

At this time there are four common classes of closed loop heat exchange systems: vertical closed loop, horizontal closed loop, spiral closed loop, and closed pond loop configurations.

- ***Vertical Closed Loop-*** In a vertical closed loop heat exchange configuration there is a loop field that consists of an array of vertically oriented pipes through which the heat transport fluid flows. A hole is bored into the ground, typically ranging in depth from 45 m to 75 m for most applications but it can be over 150 m for larger industrial purposes [14, 15, 16]. Pairs of pipes are fed into the hole, where they are connected at the bottom by a U-shaped connector (Figure 2.7). This allows for the heat transfer fluid to absorb heat from the ground within a length of pipe double the borehole depth [16]. When the pipes are fed into the boreholes there can be space between the pipes and the outer wall of the hole. In order to create sufficient means of heat transfer, the holes are filled with a pumpable material such as a bentonite grout allowing for increased heat transfer characteristics between the pipe and the earth [12].

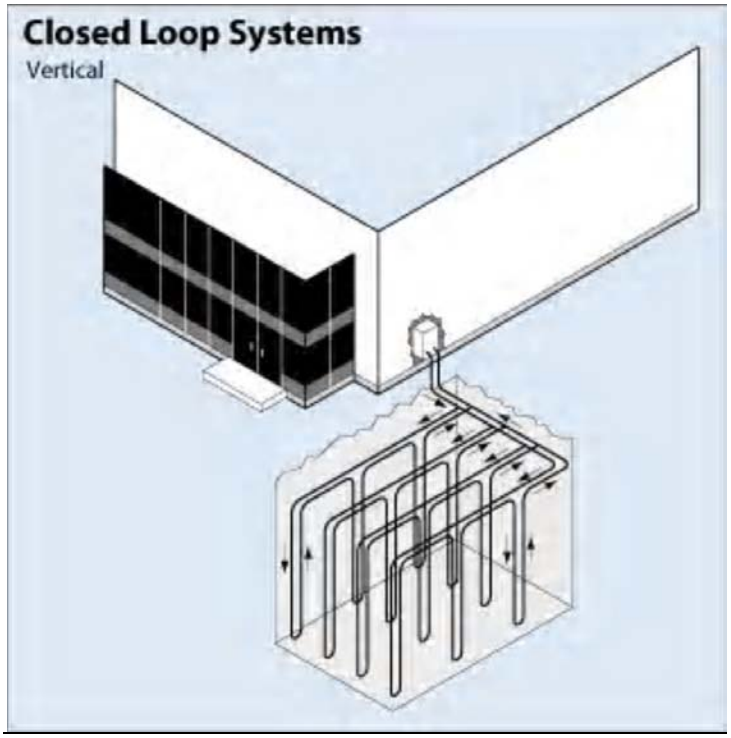

Figure 2.7: Diagram of a vertical closed loop heat exchange system [16]

The borehole diameter is approximately 4 inches for a typical system arrangement and multiple boreholes can be utilized. The actual number of boreholes is dependent on the amount of heat required by the building it is supplying. For a typical residential application the spacing between boreholes is around 5m to 6m in order to prevent one borehole drawing heat from the ground that would otherwise be used by adjacent boreholes [16]. In order to have multiple boreholes a manifold or distribution system must be used to assure that there is equal flow distribution between all boreholes. Manifolds can be housed inside the building or can be buried in trenches above the loop field [12].

The main purpose of a vertical loop configuration is to reduce the surface area that the system occupies. This makes vertical loop setups advantageous for properties where land is limited. Another incentive for these systems is that drilling does not disturb the landscape as much as trenching, allowing for relatively clean installation [12, 16].

Having the piping run deep into the ground ensures the ground temperature stays relatively constant year round. This means that less length of pipe is required in order to meet the capacity during the high and low ambient temperature extremes throughout the year, and the performance of the heat pump systems remain relatively constant [12].

The disadvantage of using a vertical system is the installation cost. The cost of drilling is normally high when compared to horizontal trenching, used for other loop types, which raises the installation costs of utilizing vertical borehole arrangement. With the increased installation cost, the economic feasibility is more challenging for a vertical borehole system in comparison to the other systems for small installations. High performance vertical loop systems are usually found to be most economical for larger applications [12].

- ***Horizontal Closed Loop-*** Horizontal closed loop heat exchange systems tend to be the most common earth connection in areas where there is ample amount of area for the system to be installed. A horizontal system entails a ground loop that is laid out horizontally or close to parallel to the surface within backfilled trenches.

  Horizontal trenches are dug below the frost line, in areas that encounter frost, and pipe is laid within the trenches. The depth of the trenches usually does not exceed more than a couple meters below the surface. Having the loops at these depths means that most of the thermal recharge to the soil comes from the solar radiation as opposed to heat flow from deep within the earth. An increased amount of piping is required, compared to borehole installations, in order to supply and reject enough thermal energy during seasonal extremes. A major restricting factor arises from solar recharge; the surface above the system should not be located under a substantial barrier, such as thick driveways and buildings. Placing the system in such a location hinders the ability of the ground to be recharged by solar radiation [12].

  The arrangement of the loops can vary depending on the capacity required and the actual amount of land available. The three most common configurations involve a basic loop (Figure 2.8), a series loop where the piping is fed back and forth (Figure 2.9), and setup with the piping in parallel (Figure 2.10). All layouts perform the same tasks, but the basic layout requires a substantial amount of land to incorporate sufficient surface area for heat transfer. The series and parallel setups are more practical with the reduced land requirement. The series piping layout is the most common [12]. Configurations can become more complex with the integration of both series and parallel loops allowing these systems to be very diverse and compatible with numerous applications.

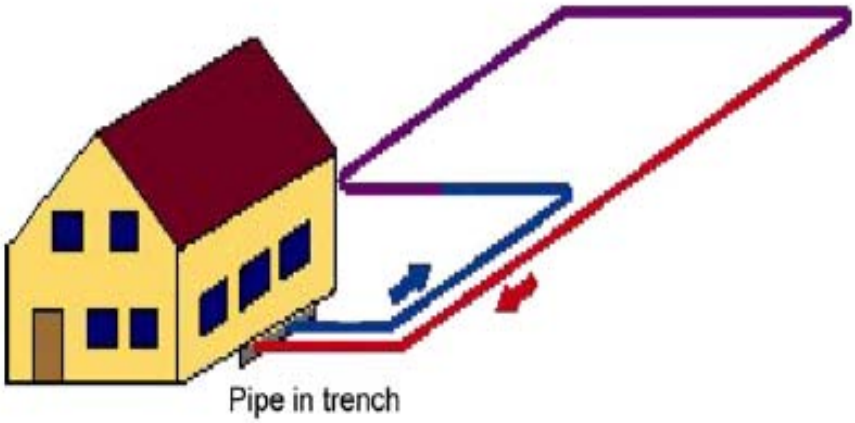

Figure 2.8: Basic horizontal loop configuration [12]

Figure 2.9: Horizontal loop with looped piping in series [12]

Figure 2.10: Horizontal loop with piping in parallel [12]

In general, the cost of digging a trench is less than drilling, and trenching allows the installation process to be flexible. Horizontal setups are normally the most cost effective heat exchange system as long as there is enough land to house them [12].

Aside from the fact that these systems require large areas, they are also subject to potential variations in heat transfer characteristics. Since the systems are shallow there are a number of factors that can change the thermal properties of the soil throughout the year, including: rainfall, snowfall, vegetation growth and shade. As mentioned before, there is variation in the temperature of the soil depending on depth. These systems require a water/antifreeze mix in order to protect against freezing. The added antifreeze causes more pump power to be required to move the fluid through the loop. In general, horizontal loop setups are for low and medium load applications, such as residential housing.

- ***Closed Spiral Loop-*** Spiral loop ground loops are very similar to conventional horizontal loops, mostly in the sense of their horizontal orientation. With a spiral loop configuration the pipes actually overlay one another. The piping is the same as other systems but it is unrolled in circular loops in the trench as seen in Figure 2.11. The end of each spiral loop is complimented with a straight return pipe that leads back to the heat pump unit [12, 16].

Figure 2.11: Picture of slinky loop installation [23]

Spiral loop setups are used in areas where insufficient space is available for a conventional horizontal loop system. Overall the trench required by these systems is a fraction of what is required by horizontal loops. Reducing trench area results in an increase of piping to supply the same load; the amount of piping for a spiral setup is substantially higher than for a horizontal system [12].

Another variation of the spiral-loop system involves placing the loops upright in narrow vertical trenches, so that the spirals stand on end. This is known as the vertical-loop layout. The main advantage to using vertical-loop layout is a reduced horizontal area requirement for installation. This decreases the surface area required beyond the contemporary horizontal loop. These installations also allow for the use of different trenching equipment which may prove more economical [12].

In areas where trenching would be a substantial portion of the cost of a GHP system, spiral loop setups can reduce the installation costs, while the opposite can be true in areas where material costs are high [16]. Even though there is the

potential that the system might cost more due to materials compared to horizontal loops, the overall cost can still be lower than vertical setups involving drilling. Also, properties that are too small to house adequate horizontal setups can still use a GHP with the installation of a spiral-loop earth connection [12].

Aside from the increased amount of piping needed, all the disadvantages are the same as horizontal setups including variations in heat transfer characteristics and substantial land area requirement. It is also noted that greater pumping requirements are associated with closed spiral arrangements compared to horizontal loops due to the added pipe length, which also translates into lowered system COP.

- ***Closed Pond Loop-*** Closed pond loops are not common closed loop heat exchange systems. Pond loops are very similar to spiral-loop systems with the difference being pond loops are submerged in a body of water versus being buried in soil. Piping is arranged in the same fashion as spiral-loop configurations, where the piping is coiled, and is attached to frame work then sunk to the bottom of an adequately sized pond or water source as seen in Figure 2.12.

Figure 2.12: Closed pond loop being installed [21]

The heat transfer loop should be able to be sunk to at least 1.8 m below the surface of the water. It has to be assured that sufficient thermal mass is maintained even in times of low water conditions and prolonged draught. The depth also ensures that the temperature never drops below the freezing point of water. Rivers are not ideal for this application due to their unpredictable nature; flooding and draught in rivers can cause system damage. Moving debris is also a hazard when rivers are considered [12, 16].

For most cases the frame work, with piping, is sunk to the minimum required depth using concrete anchors. These serve a dual purpose as they also support the framework above the bottom of the pond by about 23 cm to 46 cm. The space allows for convective flow around the piping [16].

Pond loops are beginning to gain popularity. Pond loop setups could potentially require the least amount of piping of all closed loop systems. The superior heat transfer characteristics involved with submerging the piping is the main reason. This yields the lowest piping cost of all the systems, along with the lowest installation cost, due to the exclusion of drilling or trenching [12].

Aside from the requirements set by water body size the main disadvantage involved with this system is the limited use of water for other purposes, such as boating and fishing.

2.2.1.2 Open loop systems

Open loop heat exchange systems have a direct interaction with the earth, where the heat transfer fluid is not contained within piping configurations. Open-loop systems use local groundwater or surface water, such as lakes and ponds, as a direct heat transfer medium. The extracted water is fed to the heat exchanger of the heat pump unit, supplying the required heat input, and then discharged back into the water table, stream or lake, or on the ground as irrigation [12].

There are 3 common setups for open loop systems: systems consisting primarily of extraction wells, extraction and reinjection wells, or surface water systems. The most common open loop setup involves both the extraction and reinjection well, as seen in Figure 2.13.

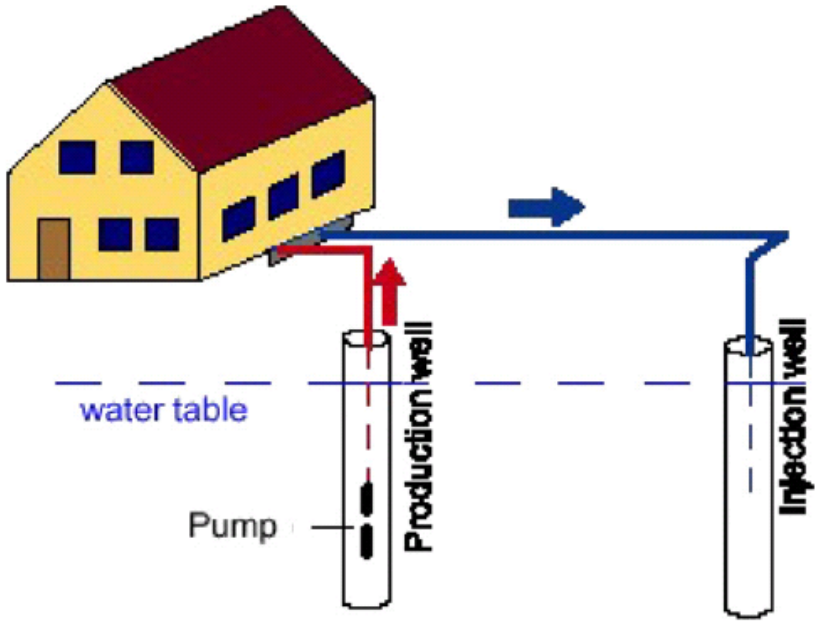

Figure 2.13: Open loop heat exchange system with production and injection wells [12]

Water is extracted from the production well, which is drilled into the local water table, and used in the heat pump via a heat exchanger. The water is then injected back into the same water table some distance from the production well. The distance between the wells

is required to allow for adequate heat transfer from the ground to the water between the wells [12]. The injection well could be excluded with the application of open drainage where the water could be released into a stream, river, lake, pond, ditch, or drainage tile. This is essentially the least expensive and easiest way to discard the used water but this requires that the source of water be of high capacity with little draw down in order to provide prolonged use [18].

As a rule of thumb, generally the flow rate of water required into the heat pump unit is between 5.7-11.4 liters per minute per kW of heating capacity, which translates into a large amount of water over time. Water resource regulations govern such a large quantity of water being extracted and must be explored before implementing GHP systems with open loop configuration [12].

Water quality is a significant factor in these systems; the water is required to be clean for successful installations. The heat exchanger between the heat exchange loop and the heat pump unit is subject to corrosion, fouling, and scaling. In order to avoid these phenomena water should have fairly neutral chemistry; minerals such as iron should be in low content to limit their effects on the heat pump unit [12, 16].

The most important benefit associated with open loop setups is that the water temperature remains constant. Also, since the water is being extracted and directly used the temperature of the fluid entering the heat pump unit is typically higher than what would be observed by fluid involved with closed loop setups. The increase has the effect of increasing the overall COP of the heat pump system, because the temperature difference between the water and inside space is reduced, and increased heat can be extracted [12, 22]. Depending on the employed water extraction method, open loop systems can have the highest pumping loads of all systems, but the overall COP still remains high.

These systems tend to require less drilling than vertical closed loop systems, allowing them to have a lower initial cost. Open systems also tend to be of a simpler design than other earth connections, allowing open loop configurations to be more economical than

closed loop systems designed for the same capacity. The high COP also means that the amount of energy they require, aside from the heat input from the water, is reduced and operating costs are lowered [12].

The significant challenge for open-loop systems is that they are subject to local, provincial, and federal clean water and surface water codes and regulations. In areas where strict regulations exist, installations of open loops systems may not be permitted. Also the water might be subject to changes with seasons, and in extreme cases water availability might be limited or non-existent. Wells may also require maintenance, increasing the work and or cost seen by the user [12].

2.2.2 Single Loop

Single loop configurations are better known as direct exchange systems. In these systems refrigerant, which is used in the heat pump unit, is moved though the heat exchanger pipes. The arrangement of the pipes is the same as in a horizontal closed loop configuration. The refrigerant that is collecting the heat from the earth is the same as the refrigerant that is used in the heat pump unit itself. This means that there is no heat exchanger between the heat collection loop and the heat pump unit, and the heat that is absorbed from the ground is directly used in heat pump system. The heat exchange loop thus becomes the evaporator for the heat pump [12].

This system avoids one heat exchanger and the pump that circulates the fluid through the ground loop piping. The refrigerant is moved via the compressor located in a heat pump unit; the compressor size can be increased slightly to accommodate the increased volume of the evaporator (ground loop). By eliminating the heat exchanger and the heat transfer loop pump, the overall COP of GHP can be increased [22]. Another advantage involves the piping material. Copper piping is commonly used in these systems due to its superior heat transfer characteristics. Thus shorter and smaller pipes are required to supply sufficient heat transfer, possibly reducing initial costs.

The main disadvantage to direct exchange systems is that the system is pressurized, necessitating the use of high quality, durable piping [22]. Pressurization also increases the probability of rupturing if substantial above ground force is applied to the piping area, or if the system is running abnormally. When the piping system is damaged it is likely that the entire system will need to be exposed (dug up) for repairs. Another disadvantage is the increased volume of the evaporator, resulting in a greater quantity of refrigerant, which is more expensive than water and antifreeze. Sufficient knowledge of refrigeration cycles is required for designing and installing these types of units. Since the heat pump is not sealed in these cases the thermodynamic calculations of evaporator size, required heat, flow rate, and many other parameters need to be carefully determined in order to have a properly working geothermal heating system [12].

These types of systems are gaining popularity around the world as people try to exploit higher geothermal heat pump COP's. Some countries (e.g., France and Austria) are exploring direct exchange units that have systems with direct evaporators coupled with direct condensers for applications such as floor heating [12].

2.3 Global Status

The main advantage of geothermal heat pumps is their ability to utilize soil and ground water temperatures typically between 5°C and 30°C. This temperature range can be found in almost all countries of the world, meaning that geothermal heat pumps are essentially applicable worldwide [20].

At present, GHPs are considered one of the fastest growing applications of renewable energy globally. As of 2004 there were about 30 countries that were using these heat pump systems in significant numbers. Continental North America and Europe are currently the leaders in terms of growth of the technology. The leading countries include the U.S., Sweden, Germany, Switzerland, Canada and Austria. Table 2-2 outlines the leading countries and their installed capacities. Other countries where this technology is

gaining popularity include France, The Netherlands, China, Japan, Russia, UK, Norway, Denmark, Ireland, Australia, Poland, Romania, Turkey, Korea, Italy, Argentina, Chile, Iran, the UK and Norway [20].

As of 2004 the worldwide installed GHP capacity was around 12 GW_{th}, which required an annual energy usage of 20 TWh. Since 1994 the annual growth rate for geothermal heat pumps has been about 10%, meaning that today there are close to a 1.7 million applications around the world [17].

In the past the growth of geothermal heat pumps has been relatively slow compared to the progression of other renewable and conventional energy technologies. This is due to several reasons: system designs are not standardized, the capital cost is significant compared to other heating systems, there is a shortage of people that are knowledgeable in the installation of such systems, government policies sometimes do not permit or encourage adoption of the technology, and economies of scale and scope are rarely exploited [22]. These issues are starting to be resolved and there is a growing acceptance of the technology. An increasing number of countries are beginning to develop GHP programs [20].

Table 2-2: Leading countries using geothermal heat pumps as of 2004 [20]

Country	Installed Thermal Capacity (MW_{th})	Annual Energy Use (GWh)	Number of GHP installations
U.S.	6300	6300	600,000
Sweden	2000	8000	200,000
Germany	560	840	40,000
Switzerland	440	660	25,000
Canada	435	300	36,000
Australia	275	370	23,000

With growing acceptance and use of geothermal heat pump systems, research is increasingly needed to improve the performance of the technology and develop systems to work in a wide range of applications.

2.4 Recent Studies Relevant to Present Work

The literature on ground source heat pumps appears to be limited to the basic aspects of use and operation. The majority of the literature in this field corresponds to heat pump systems that utilize the basic vapor compression cycle. Comparisons between ground- and air-source heat pumps are common in previous studies. Ground source heat pumps have been used in residential applications for decades, and currently commercial and industrial large scale systems have begun to be employed around the world. Extensive energy and exergy analyses have been carried out for the basic heat pump arrangement, using both simulation and experimental methods. However, there are a limited number of studies performed that investigate the effect of changing various operating conditions of a ground source heat pump. Also, an even smaller amount of work has been performed on advanced ground source heat pump systems with variations in the heat pump cycle components and arrangements, and how they compare to the basic vapor compression cycle.

Several studies have been reported that are relevant to and coincide with the present work:

- Hepbasli and Tolga Balta [19] present an experimental model to determine the performance of a heat pump system using low temperature geothermal resources. Energy and exergy analyses are presented, including determination of the COP and identification of the locations of the greatest irreversibilites within the system.

- Hepbasli and Akdemir [24, 25] also undertake a thermodynamic analysis of ground source heat pump systems, with a U-bend ground heat exchanger for district heating purposes. Energy and exergy relations are derived and applied to a GSHP system providing heat.

- Healy and Ugursal [18] investigate the effect of various system parameters on GSHP performance with a horizontal loop arrangement using a computer model. Parameters

considered include ground loop (GL) size, GL depth, heat pump capacity, heat transfer fluid, heat transfer fluid flow rate, GL pipe size, horizontal pipe spacing of GL and soil type. The study indicates that performance of a GSHP is affected by a number of ground loop parameters. A comparative economic evaluation is also carried out to assess the feasibility of using a GSHP in place of a conventional heating/cooling system and an air source heat pump.

- Kulcar et al. [26] describe the economics of exploiting heat from low-temperature geothermal sources for high-temperature heating of buildings using a heat pump. The presented economic calculation shows the system considered is capable of economically exploiting low-temperature geothermal energy for district heating of buildings and that the study of large scale applications is merited.

- Kara [27] presents an experimental study to determine the performance of a ground source heat pump (GSHP) system in heating mode in the city of Erzurum, Turkey. The GSHP system has a single U-tube ground heat exchanger (GHE) made of polyethylene pipe. Illustrated is an experimental performance evaluation of a GSHP in with a ground loop connection similar to that of the present work. Thus the results of Kara [27] provide a data source for model development and validation in this thesis.

- Wang et al. [8] investigate the effects of compressor and motor cooling, where the heat is transferred to the refrigerant for preheating, in a heat pump system. The use of different refrigerants is explored for motor cooling arrangement in the study.

- Ma and Chai [9] develop an improved heat pump cycle that incorporates an economizer into the vapour compression cycle with two compression processes between the condenser and evaporator. An optimization of the improved system is performed with a basic comparison to the conventional heat pump system. Details regarding model development for these systems and results are applicable to the present work, providing data for validation.

- Ma and Zhao [28] continued the work of Ma and Chai [9] by experimentally comparing the improved heat pump cycle with a similar cycle, which employs a flash tank with vapour separation and the inclusion of two compression processes. The study provides further information into the method of analysis for the heat pump system with the economizer, and also provides a data source for validation in the current thesis.

Chapter 3: Approach/Methodology and Assumptions

3.1 Approach and Methodology

First law analysis is performed for three different ground source heat pump arrangements. The heat pump systems under investigation include a basic vapor compression cycle, a vapor compression cycle including electric motor cooling, and a vapor compression cycle with an economizer, designated as systems 1, 2, and 3, respectively. System descriptions are presented in Chapter 4.

The systems are analyzed assuming that there is one centralized heat pump system that supplies the entire space with the required heat. This is explained by Natural Resources Canada [6] as the most basic system arrangement for heating a large space. Other arrangements exist where multiple heat pump units are contained within a building that feed and feed off of a main earth connection. The multiple heat pump arrangement is used to allow for cooling and heating of different spaces within the same building. Since this study is restricted to heating purposes, the single heat pump arrangement is adopted.

There are essentially two different loops within the complete system including the heat pump cycle and ground loop heat exchanger, commonly referred to as the ground loop. The arrangement and analysis process for the ground loop is identical for its application to each of the three systems.

A series of balance equations for mass and energy flows are developed for each system. A number of state conditions are assumed to be common to all systems and are described in Section 3.2. It is also assumed that a constant heating load is present throughout the entire study. With a variety of assumed conditions, energy and mass balance equations are arranged in such a fashion as to utilize the assumed values for the calculation of key characteristics, for example COP. The equations and analysis of each system and the ground loop are presented in Chapter 5.

Using the derived equations for each system, a model is created using computer software. Engineering Equation Solver (EES) is used to analyze each system with the given set of parameters. The equations created from the balance equations and assumptions are input for the software with the assumptions attached to the appropriate equations. The simulation analyzes the system and allows important parameters to be calculated. Parametric studies are also performed within EES when a parameter is changed over a specified range. Further discussion on the simulation and software can be found in Chapter 6.

In terms of the analysis, the systems are compared for static operating assumptions. Heat pump and system COP are compared, as are ground loop characteristics. The comparisons allow for a better understanding of the operation of each system before considering parameter variations.

Each analysis in the present study is performed with a common set of assumptions for each system arrangement. When the investigations are carried out with a variation of a particular operating condition all preset assumptions are applied, aside from the assumption originally set for the condition, being varied. When a condition is varied it is done so over a practical range to ensure the results created are valid for the actual application. The range for each variable is either found through literature or confined by the operation of the system itself, i.e., the range is limited in order to allow for proper system operation.

In the analyses, the effects of varying parameter values on heat pump and system COP are evaluated. The effect on the ground loop length and other system characteristics are also analyzed. Trends are identified and described within each analysis, and the responses of the different systems are compared. The sensitivity to a particular change is compared for all systems using parametric studies. To understand how the different systems respond to changes, reasons why the systems react in a particular way are explored.

3.2 Assumptions

Many general assumptions are made, in addition to those noted earlier, to simplify the present work and are listed below:

- Pressure drops are neglected within the heat pump unit
- All processes are treated as adiabatic
- Changes in elevation are negligible
- The system operates at steady state, with steady flow conditions

Additionally, there are assumptions specific to the heat pump and ground loop parts of the systems, and these are noted in the following subsections.

3.2.1 Heat Pump Cycle Assumptions and Data

The required heating load for the system is set as the basis for the design of all the systems. Natural Resources Canada illustrates an average heating load of 100kW for commercial and small institutional buildings in their 2003 summary report of Commercial and Institutional Building Energy Use [29]; this heating load is used for all system arrangements and analyses.

Many of the other system parameters are determined from a set of initial simulations performed to determine suitable parameter ranges. The parameters of the heat pump systems are outlined in Table 3-1.

Table 3-1: Determined heat pump system assumptions

Parameter	Setting
Refrigerant	R134a
Condenser pressure	1000 kPa
Evaporator pressure	200 kPa
Intermediate pressure*	400 kPa
Degree of subcooling	5°C
Degree of superheating	5°C
Extra degree of subcooling*	5°C
Compressor efficiency	75%
Pump efficiency	90%
Electric motor efficiency	80%

*Specific to system 3 only

3.2.2 Ground Loop Assumptions and Data

The ground loop fluid utilized in this study is a mixture of water and propylene glycol, which is specified as a suitable anti-freeze by Ochsner [30] in his guide for planning and installing ground source heat pumps. Both Natural Resources Canada [6] and Ochsner [30] specify an appropriate concentration of propylene glycol of between 15-40%, by mass; 30% is used for the present study.

The average ground temperature is found using Canada's National Climate Data and Information Archive [31]; data shows the ground temperatures several meters below the surface, for much of Canada, are relatively constant. For this study a mean ground temperature of 9.3°C is utilized which is the mean ground temperature well below the surface for Ottawa, Ontario, Canada.

Standard piping is used for ground source heat pumps and includes high density polyethylene DN32 PN10, which has a nominal diameter of 32 mm with a pressure rating of 10 bar and an inner diameter of 26.2 mm [2, 6, 32].

The pipe material allows for the assumption of a smooth pipe interior [33]. Polyethylene can be assumed to be smooth when considering internal flow since the equivalent roughness value for new commercial polyethylene pipe is 0 as defined by Cengel [34]. Under the smooth pipe assumption the calculated pressure drop is less than the pressure drop that would exist within real applications.

The thermal conductivity of the pipe material is assumed as an average from various sources and is set at 0.375 W/m·K [35, 36]. In order to allow for the constant ground temperature the pipe depth is set to 100m [22].

The borehole diameter is set at 4 inches or 101.6 mm [12] and the grout thermal conductivity taken as 1.56 W/m·K [37].

Other system parameters are determined from a set of initial simulations performed to determine suitable parameter ranges. The parameters of the heat pump systems are outlined in Table 3-2.

Table 3-2: Determined ground loop assumptions

Parameter	**Setting**
Evaporator inlet temperature (GL side)	7.3°C
Evaporator outlet temperature (GL side)	1.3ºC
Flow rate through individual parallel GL*	0.3 kg/s

* Note the flow rate was determined to allow for the proper flow regime within the GL piping; transitional or turbulent flow is desired within these systems [6, 29].

Chapter 4: System Configurations

This chapter describes each system considered in this study. The operation of each system is also explained.

For this study there are two loops within a complete system: the heat pump cycle and ground loop heat exchanger loop. The arrangement and analysis for the ground loop is the same for each of the three systems being explored.

4.1 Description of Heat Pump System 1

The first heat pump system considered utilizes a basic vapor compression cycle. This arrangement is widely utilized due to its simplicity and ease of design [12, 38]. As seen in Figure 4.1, the heat pump cycle consists of an evaporator, compressor, condenser and expansion valve, coupled with an electric motor for the compressor and a ground loop with a pump.

Refrigerant at state 1 enters the evaporator where thermal energy is transferred to it from the ground loop. At state 2, the refrigerant exits the evaporator as super heated vapor and enters the inlet of the compressor. The pressure is increased to the condenser pressure and the refrigerant exits the compressor as a high pressure super heated vapor at state 3. The refrigerant enters the condenser and thermal energy is extracted from the refrigerant and supplied to the space. At the state 4 (exit of the condenser), the refrigerant is a subcooled liquid and enters an expansion valve that reduces the pressure to that of the evaporator.

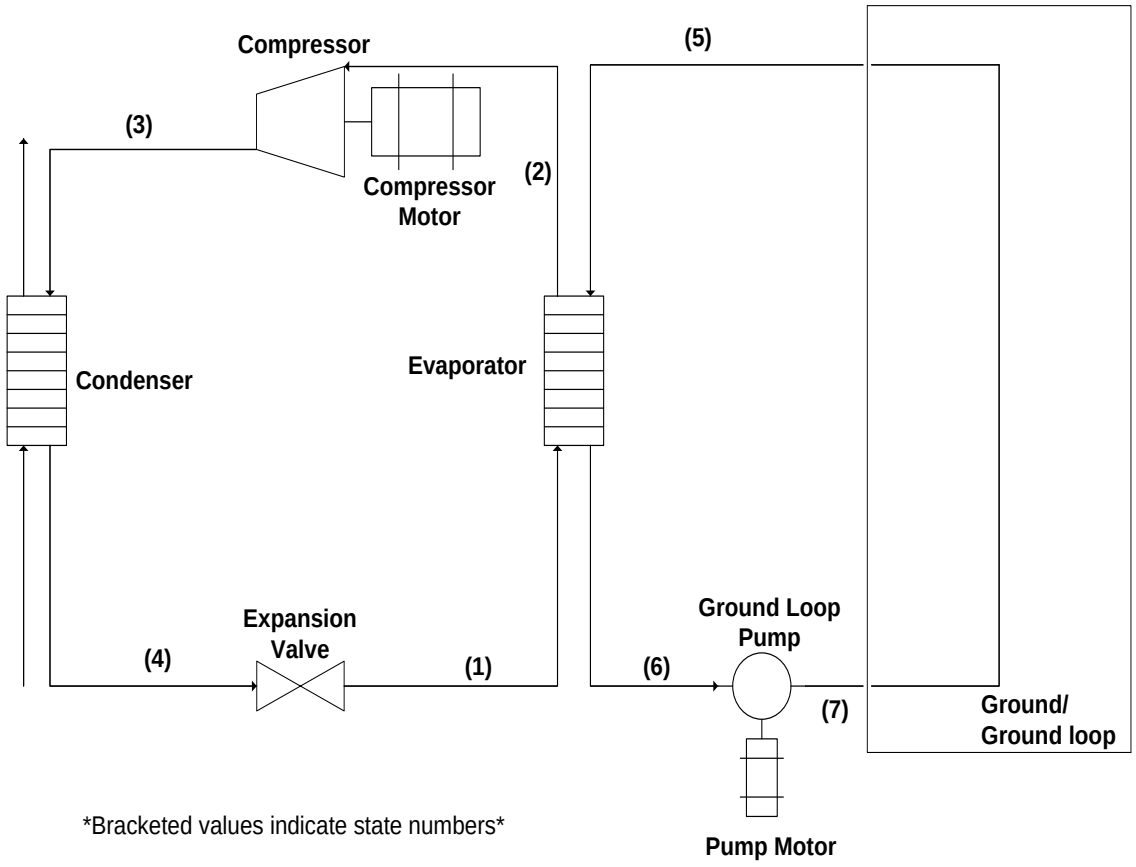

Figure 4.1: System diagram of heat pump system 1

4.2 Description of Heat Pump System 2

The second configuration considered is shown in Figure 4.2. It is identical to system 1 except for a modified flow path used for motor cooling. Essentially the new path directs the refrigerant to the compressor motor where it comes in contact with the electric motor assembly. During regular operation a portion of electrical energy supplied to the electric motor is wasted. The energy lost within the motor assembly would mostly appear in the form of thermal energy and kinetic energy, i.e. vibrations. The wasted energy that appears in the form of heat is extracted from the motor by the refrigerant flow through it. The result is an increase in the energy content of the refrigerant at the evaporator inlet.

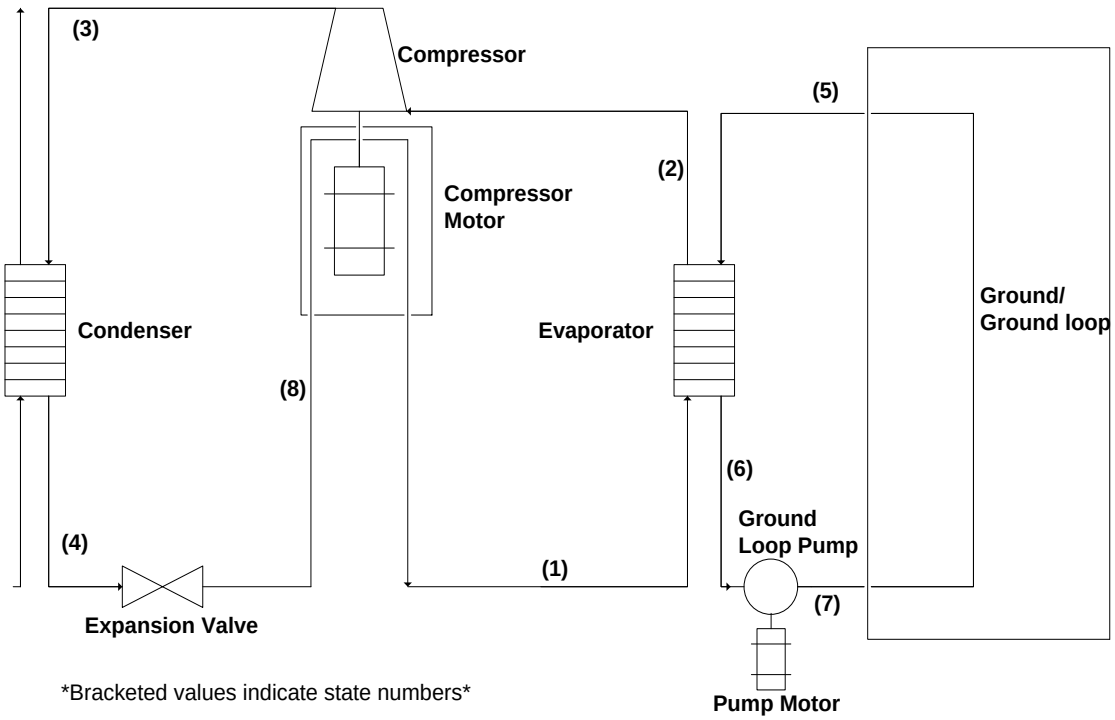

Figure 4.2: System diagram of heat pump system 2

4.3 Description of Heat Pump System 3

System 3 is a heat pump system with an economizer, and is described by Ma and Chai [28]. The system (Figure 4.3) is more complex then the basic HP arrangement. In the economizer, also known as an internal heat exchanger, heat is transferred between two refrigerant flows. The new flow path, called the supplementary circuit, includes an expansion valve that allows the flow to exist at an intermediate pressure that is set between the evaporator and condenser pressure. Heat is extracted from the main refrigerant circuit by the supplementary circuit, and the main flow passes through an expansion valve lowering its pressure to the evaporator pressure. The supplementary circuit flows from the economizer to the compression process through a supplementary inlet. For ease of analysis, Ma and Chai [28] as well as Ma and Zhao [9] suggest interpreting the compression process as quasi two-stage compression with an intermediate mixing chamber.

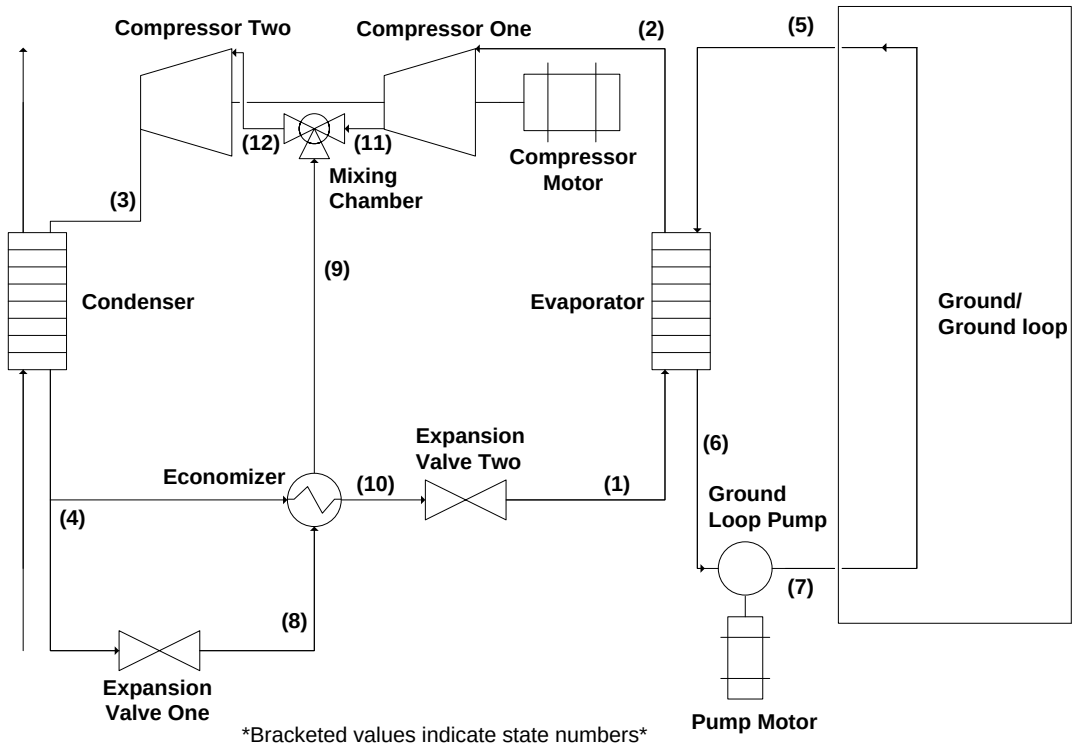

Figure 4.3: System diagram of heat pump system 3

4.4 Ground Loop Heat Exchanger Description

The ground heat exchanger connects the heat pump units and the ground. This heat exchanger draws heat from the ground and supplies it to the HP during operation. All of the systems considered here utilize the same ground loop arrangement. The study considers HP units on a commercial scale. Natural Resources Canada [6] suggests a vertical borehole with a U-tube arrangement is best for heating and cooling applications so this arrangement is adopted here [6], as illustrated in Figure 4.4.

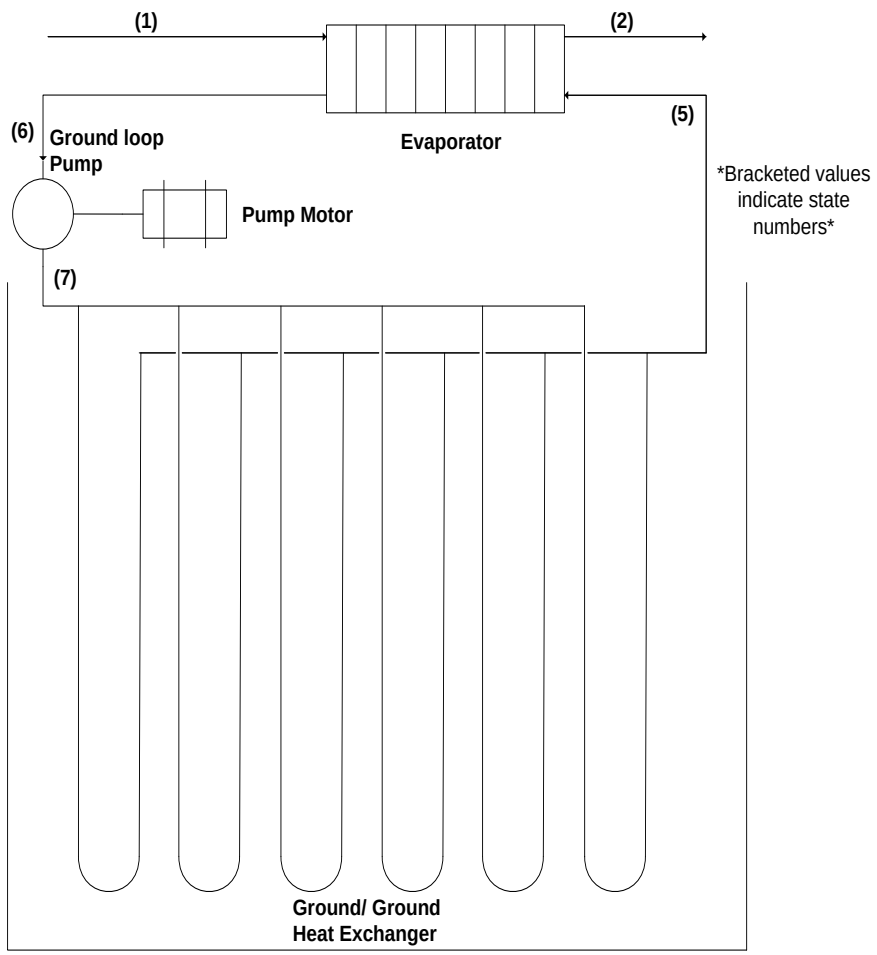

Figure 4.4: Basic layout of ground loop

The borehole design consists of a main flow through the evaporator, a single pump, and multiple parallel loops. A cool water/glycol (brine) mixture flows from the evaporator to the pump, where the pressure is increased to the required level. The brine flow is then split into the parallel loops, and absorbs heat from the surrounding ground. The entirety of the brine exits the parallel loops and joins to form a single warm stream which enters the evaporator in order to transfer heat to the refrigerant in the HP cycle.

Chapter 5: Analysis

The analyses performed on the systems considered and in the parametric studies are explained in this chapter. Included are the equations utilized. The analysis material is presented in four subsections: one for each of the heat pump cycles and one for the ground loop. The analysis of the ground loop is separated because it is identical for each heat pump system.

5.1 Analysis of Heat Pump System 1

Refrigerant enters the inlet of the compressor at state 2. The pressure at state 2 is assumed and temperature is taken as the saturation temperature plus an assumed degree of superheating, in degrees Celsius, as presented by Ma and Zhao [9] and Fu et al. [39]. The temperature at this state is restricted to be below that of state 5 (evaporator inlet for the GL). The refrigerant is then compressed to a higher pressure at state 3.

The ideal enthalpy value at state 3 is determined assuming isentropic compression, where the ideal specific entropy at state 3 is equal to the actual specific entropy at state 2 ($s_2 = s_{3s}$), and a preset pressure at the compressor exit. For an actual compression process, the specific enthalpy at state 3 is calculated using the compressor efficiency as follows:

$$\eta_{comp} = \frac{(h_{3s} - h_2)}{(h_3 - h_2)} \tag{5.1}$$

where h_{3s} denotes the ideal specific enthalpy for an isentropic process, h_2 and $h_3$are the actual specific enthalpy at states 2 and 3, and η_{comp} is the isentropic efficiency of the compressor. The isentropic compressor efficiency and the pressure at state 3 are given. The specific enthalpy at state 2 and temperature at state 3 are found using thermodynamic property tables.

The superheated refrigerant exiting the compressor enters the condenser where it condenses at constant pressure and becomes a pressurized liquid. The amount of heat extracted from the refrigerant for space heating is set to meet the building load.

The temperature at state 4 is taken as the saturation temperature minus an assumed degree of subcooling, in degrees Celsius, as suggested by Ma and Zhao [9] and Fu et al. [39]. Specific enthalpy and entropy values are obtained from thermodynamic property tables. Using the conditions at states 3 and 4, the mass flow rate of refrigerant can be found by using an energy rate balance for the condenser:

$$\dot{Q}_{load} = \dot{m}_{ref}(h_3 - h_4) \tag{5.2}$$

where $\dot{Q}_{load}$ is the specified heating load, $\dot{m}_{ref}$ is the refrigerant mass flow rate, and h_3 and h_4 are the specific enthalpy for states 3 and 4 respectively.

Similar to the condenser, flow through the evaporator occurs with zero pressure drop, i.e., the pressures at states 1 and 2 are equal. Assuming isenthalpic pressure drop across the expansion valve ($h_4 = h_1$) the temperature of the refrigerant at state 1 is obtained using thermodynamic property tables. The quality of the refrigerant is evaluated as follows:

$$x = \frac{h_1 - h_f}{h_{fg}} \tag{5.3}$$

where x is the quality of the refrigerant, h_1 is the specific enthalpy at state 1, h_f is the enthalpy of saturated liquid refrigerant and h_{fg} is the enthalpy difference between saturated liquid and saturated vapor refrigerant coinciding with the pressure at state 1.

The rate of heat transfer from the ground loop to the refrigerant through the evaporator is calculated as follows:

$$\dot{Q}_{evap} = \dot{m}_{ref}(h_2 - h_1) \tag{5.4}$$

where $\dot{Q}_{evap}$is the rate of heat transfer to the refrigerant, $\dot{m}_{ref}$ is the mass flow rate of refrigerant, and h_1 and h_2 are the specific enthalpy at states 1 and 2, respectively.
The rate of heat transfer to the refrigerant is equal to the rate of heat removal from the GL brine through the evaporator. $\dot{Q}_{evap}$ is used for the analysis of the ground loop heat exchanger (the methodology for the ground loop analysis is presented in section 5.4).

The rate of work required by the compressor is calculated using the mass flow rate and change in enthalpy across the compressor:

$$\dot{W}_{comp} = \dot{m}_{ref}(h_3 - h_2) \tag{5.5}$$

where $\dot{W}_{comp}$ is the rate of work supplied to the compressor, $\dot{m}_{ref}$ is the mass flow rate of refrigerant through the compressor, and h_2 and h_3 are the specific enthalpy at states 2 and 3 respectively.

An electric motor, which provides mechanical work to the compressor, is included in the system arrangement. Using the calculated compressor work rate and an assumed motor efficiency the rate of electrical energy consumption by the compressor motor is calculated as follows:

$$\dot{E}_{comp,motor} = \frac{\dot{W}_{comp}}{\eta_{EM}} \tag{5.6}$$

where $\dot{E}_{comp,motor}$ is the electrical energy consumed by the electric motor, $\dot{W}_{comp}$ is the rate of work required for the compression process, and η_{EM} is the efficiency of the electric motor.

The performance of HP units in heating applications is conventionally measured as the coefficient of performance (COP). For this system arrangement, the COP follows the basic definition:

$$COP_{HP} = \frac{\dot{Q}_{load}}{\dot{W}_{comp}} \tag{5.7}$$

Where COP_{HP} is the coefficient of performance of the heat pump cycle with the exclusion of motor and pump work, $\dot{Q}_{load}$ is the building heating load, and $\dot{W}_{comp}$ is the rate of work required for the compression process.

The COP can be expressed in an alternate form, which is used to verify the analysis results:

$$COP_{HP} = \frac{\dot{Q}_{load}}{\dot{Q}_{load} - \dot{Q}_{Evap}} \tag{5.8}$$

where COP_{HP} is the coefficient of performance of heat pump cycle with the exclusion of motor and pump work, $\dot{Q}_{load}$ is the building heating load, and $\dot{Q}_{Evap}$ is the rate of heat transfer to the refrigerant through the evaporator.

The system COP is similar to the heat pump COP (COP_{HP}). The difference arises in the work term in the denominator of the expression. Instead of employing the compressor work rate the electrical energy supply rate is utilized. The system COP is defined as

$$COP_{System} = \frac{\dot{Q}_{load}}{\dot{E}_{motor,total}} \tag{5.9}$$

where COP_{System} is the coefficient of performance for the entire system, $\dot{Q}_{load}$ is the building heating load, and $\dot{E}_{motor,total}$ is the total rate of electrical energy consumed by both the compressor and pump motor.

The total rate of electrical consumption is defined as

$$\dot{E}_{motor,total} = \dot{E}_{comp,motor} + \dot{E}_{motor,pump} \tag{5.10}$$

where $\dot{E}_{motor,total}$ is the total rate of electrical energy consumed, $\dot{E}_{comp,motor}$ is the rate of electrical energy consumption of the compressor motor, and $\dot{E}_{motor,pump}$ is the rate of electrical energy consumption of the pump motor. The method of calculating $\dot{E}_{motor,pump}$ is illustrated in Section 5.4

5.2 Heat Pump System 2 Analysis

The method of analyzing the heat pump system with motor cooling is similar to that of system 1. The only differences are an additional state (state 8) and the conditions at state 1.

The conditions at state 8 are found in the same manner as state 1 for system 1. It is assumed that there is zero pressure drop between states 8 and 1. Utilizing isenthalpic pressure drop across the expansion valve ($h_4 = h_8$) the temperature of the refrigerant is found in thermodynamic tables for state 8. The quality of the refrigerant is determined as

$$x = \frac{h_8 - h_f}{h_{fg}} \tag{5.11}$$

where x is the quality of the refrigerant, h_8 is the specific enthalpy at state 8, h_f is the enthalpy of saturated liquid refrigerant and h_{fg} is the enthalpy difference between saturated liquid and saturated vapor refrigerant coinciding with the pressure at state 8.

The available waste energy from the compressor motor is the difference between the compressor work rate and the electrical energy consumption rate of the motor:

$$\dot{E}_{motor,waste} = \dot{E}_{comp,motor} - \dot{W}_{comp} \tag{5.12}$$

where $\dot{E}_{motor,waste}$ is the rate waste energy associated with the compressor motor, $\dot{E}_{comp,motor}$ is the rate of electrical energy consumption for the motor, and $\dot{W}_{comp}$ is the rate of work required by the compressor.

For the developed model it is assumed that all waste energy from the electric motor is converted to thermal energy which is transferred to the refrigerant. The transferred heat increases the specific enthalpy of the refrigerant from the inlet to outlet of the motor assembly, that is,

$$\dot{Q}_{waste\ heat} = \dot{m}_{ref}(h_1 - h_8), \tag{5.13}$$

where

$$\dot{Q}_{waste\ heat} = \dot{E}_{motor,waste} \tag{5.14}$$

As before the rate of heat transfer required from the GL, through the evaporator, is calculated using the conditions at states 1 and 2 along with the flow rate of refrigerant, using Equation 5.4.

The definition of heat pump efficiency is the same as system 1. When motor cooling is incorporated the alternate definition, used for verification, of heat pump COP is slightly modified. For this definition the heat supply rate to the refrigerant from the motor is included in the equation:

$$COP_{HP} = \frac{\dot{Q}_{load}}{\dot{Q}_{load} - (\dot{Q}_{evap} + \dot{Q}_{waste\ heat})} \tag{5.15}$$

The heat supply rate by the motor is included in the COP expression due to the fact that, similar to the ground loop, it supplies heat to the refrigerant before it is compressed.

COP_{System} is defined the same as for system 1 (Equation 5.9).

5.3 Heat Pump System 3 Analysis

Same as previous systems, the refrigerant enters the compressor at state 2, where the pressure is assumed and the temperature is again taken as the saturation temperature plus an assumed degree of superheating, in degrees Celsius. The refrigerant is compressed to an intermediate pressure (state 11).

The ideal enthalpy at state 11 is estimated assuming isentropic compression ($s_2 = s_{11s}$.) with a set pressure. For actual compression, the specific enthalpy at state 11 is calculated using the compressor efficiency:

$$\eta_{comp} = \frac{(h_{11s} - h_2)}{(h_{11} - h_2)} \tag{5.16}$$

where h_{11s} denotes the ideal specific enthalpy for an isentropic process, h_2 and $h_{11}$are the actual specific enthalpy at states 2 and 11, and η_{comp} is the isentropic efficiency of the compressor. The isentropic compressor efficiency and the pressure at state 11 are specified. The specific enthalpy at state 2 and temperature at state 11 are found using thermodynamic property tables.

The refrigerant at the exit of compressor 1 enters the mixing chamber with the refrigerant from the economizer in the supplementary circuit. An energy balance across the mixer yields

$$\dot{m}_{ref} h_{12} = \dot{m}_{main} h_{11} + \dot{m}_{supp} h_9 \tag{5.17}$$

where $\dot{m}_{ref}$ is the total mass flow rate of the refrigerant, $\dot{m}_{main}$ is the mass flow rate of refrigerant through the evaporator, $\dot{m}_{supp}$ is the mass flow rate through the supplementary circuit, and h_{12}, h_{11}, and h_9 are the specific enthalpy at states 12, 11, and 9 respectively.

For simplicity, the conditions at state 9 are set equal to those at state 11, following that the approach proposed by Ma and Chai [28] as well as Ma and Zhao [9]. The mass flow rates for state 11 and state 9 are the main and supplementary flow rates within the system. The two flow rates are fractions of the total refrigerant flow rate through the condenser. The variable X is used to define the ratio of the flow rate of the main circuit to the total refrigerant flow rate through the condenser:

$$X = \frac{\dot{m}_{main}}{\dot{m}_{ref}} \tag{5.18}$$

The flow rates through the regular and supplementary loop are then represented as

$$\dot{m}_{main} = X\dot{m}_{ref} \tag{5.19}$$

$$\dot{m}_{supp} = (1 - X)\dot{m}_{ref} \tag{5.20}$$

Equations 5.19 and 5.20 are combined with equation 5.17 to obtain an equation that is represented in terms of the total mass flow rate through the condenser. The resulting equation is

$$\dot{m}_{ref}h_{12} = X\dot{m}_{ref}h_{11} + (1 - X)\dot{m}_{ref}h_9 \tag{5.21}$$

which can also be written excluding the total refrigerant flow rate as

$$h_{12} = Xh_{11} + (1 - X)h_9 \tag{5.22}$$

This form of the equation allows the enthalpy value at state 12 to be calculated without the use of the refrigerant flow rate. The conditions at state 3 are used to evaluate the flow rate within the system, as for the previous systems, but the conditions at state 3 are found using the conditions at state 12.

The value of X is calculated through energy balance for the economizer, which involves the same flows as the mixing chamber:

$$\dot{m}_{main}h_4 + \dot{m}_{supp}h_8 = \dot{m}_{main}h_{10} + \dot{m}_{supp}h_9 \tag{5.23}$$

where $\dot{m}_{main}$ is the mass flow rate of refrigerant through the evaporator, $\dot{m}_{supp}$ is the mass flow rate through the supplementary circuit, and h_4, h_8, h_9, and h_{10} are the specific enthalpy at states 4, 8, 9, and 10 respectively.

The balance can be rearranged and Equations 5.19 and 5.20 introduced to obtain a balance in terms of X and $\dot{m}_{ref}$. :

$$X\dot{m}_{ref}(h_4 - h_{10}) = (1 - X)\dot{m}_{ref}(h_9 - h_8) \tag{5.24}$$

Solving for X yields

$$X = \frac{h_8 - h_9}{h_9 - h_8 - h_4 + h_{10}} \tag{5.25}$$

The parameter X is found and used to solve for the enthalpy at state 12, where the enthalpy values are found with the conditions and assumptions outlined below.

State 4 is determined, assuming a degree of subcooling. At the condenser exit the refrigerant is split into two separate flows, one which feeds the main circuit and the other the supplementary circuit. The flow in the main circuit enters the economizer where heat is extracted from the refrigerant resulting in a higher degree of subcooling at state 10 [9, 28]. The extra degree subcooling beyond state 4 is set as a degree of temperature difference between states 10 and 4 where

$$T_{10} = T_4 - \ degree\ of\ extra\ subcooling \tag{5.26}$$

With an assumed intermediate pressure the properties at state 8 are determined through assuming isenthalpic expansion through the expansion valve. The temperature is found using thermodynamic tables.

The temperature at state 9 is assumed and the enthalpy is found using thermodynamic tables. The assumed temperature must allow for the refrigerant at state 12 to exist as a saturated or superheated vapour after mixing with the flow at state 11.

As for the second compression processes, the ideal enthalpy at state 3 is found through first assuming isentropic compression ($s_{12} = s_{3s}$) and a set pressure. For real compression, the specific enthalpy at state 3 is calculated as follows:

$$\eta_{comp} = \frac{(h_{3s} - h_{12})}{(h_3 - h_{12})} \tag{5.27}$$

where h_{3s} denotes the ideal specific enthalpy at state 3 for an isentropic process, h_3 and h_{12} are the actual specific enthalpy at states 3 and 12, and η_{comp} is the isentropic efficiency of the compressor. The isentropic compressor efficiency and the pressure at state 3 are specified. The specific enthalpy at state 12 and temperature at state 3 are found using thermodynamic property tables.

As for the previous systems the mass flow rate of refrigerant through the condenser is found using the calculated enthalpies at states 3 and 4, and the load as expressed by Equation 5.2. The total flow rate is used to find the main and supplementary flow rates through the use of the ratio X.

The heat transfer rate that is required from the GL for this system arrangement is determined as

$$\dot{Q}_{evap} = \dot{m}_{main}(h_2 - h_1) \tag{5.28}$$

Since there are two compression processes the compressor work rate required is the sum of required work rates from both compressors:

$$\dot{W}_{comp} = \dot{W}_{comp,1} + \dot{W}_{comp,2} \tag{5.29}$$

where

$$\dot{W}_{comp,1} = \dot{m}_{main}(h_{11} - h_2) \tag{5.30}$$

and

$$\dot{W}_{comp,2} = \dot{m}_{ref}(h_3 - h_{12}) \tag{5.31}$$

$\dot{W}_{comp,1}$ and $\dot{W}_{comp,2}$ are rate of work required by compressor one and two respectively.

COP definitions for the HP and the system remain the same as Equations 5.7-5.9.

5.4 Ground Loop Heat Exchanger Analysis

Initially the pressure at state 5 is set and the temperature at state 5 is determined as 2°C below the ground temperature. The GL consists of a mixture of water and glycol. Treating this as an ideal mixture, the enthalpy at state 5 is found as follows:

$$h_m = \sum_{i=1}^{k} mf_i \cdot h_i \tag{5.32}$$

where h_m is the specific enthalpy of the solution, mf_i is the mass fraction of species i, and h_i is the enthalpy of species i in the mixture at the given conditions. The mass fraction of glycol is pre-determined in order to provide the ground loop fluid with a reasonable freezing point. The freezing point temperature is required to be below that of the coldest ground temperature and the temperature at the inlet of the evaporator on the HP side [33].

The pressure across the GL side of the evaporator is assumed constant. The temperature at state 6 is assumed with the constraint that it must remain above the freezing temperature of the GL fluid. Using the same method as state 5 the enthalpy at state 6 is determined. The mass flow rate of brine is calculated through an energy balance across the evaporator, and coincides with the calculated flow rate and enthalpies on the HP side of the heat exchanger. The mass flow rate is then divided between a series of parallel loops. The mass flow rate through an individual parallel loop is set to allow for a desired flow regime and also establishes the number of parallel loops required, as follows:

$$n_{pl} = \frac{\dot{m}_{GL,Evaporator}}{\dot{m}_{GL,parallel}} \tag{5.33}$$

where n_{pl} is the determined number of parallel loops, $\dot{m}_{GL,Evaporator}$ is the flow rate across the evaporator between states 5 and 6, and $\dot{m}_{GL,parallel}$ is the flow rate though an individual parallel loop.

The pressure at state 7 is estimated from the pressure drop within the GL system and the enthalpy is found using the required pump work to counter the pressure loss. The pressure loss for an individual parallel loop is found as

$$\Delta P_{parallel} = f\ \frac{l_{parallel}}{D_i}\ \frac{\rho_{5,7}\, {V_{avg}}^2}{2} \tag{5.34}$$

where f is the friction factor, $l_{parallel}$ is the length of a single parallel loop, D_i is the inner diameter of the pipe, $\rho_{5,7}$ is the density of the fluid at the average temperature between states 5 and the estimated state 7, and V_{avg} is the average fluid velocity in the piping.

It is required to identify the conditions at state 7 in order to find the pressure drop but these are initially unknown. To begin, the temperature at state 7 is assumed to be that of state 6 in order to estimate the fluid properties between state 5 and 7.

A standard pipe size for the GL plumbing is selected and the velocity of the fluid inside a parallel loop is found using:

$$V_{avg,parallel} = \frac{\dot{m}_{GL,parallel}}{\rho_{5,7}\ A_c} \tag{5.35}$$

where $\dot{m}_{GL,parallel}$ is the mass flow rate of fluid through a single parallel loop, A_c is the cross sectional area of the piping, and $\rho_{5,7}$ is the density of the fluid at the average temperature between state 5 and the estimated state 7.

The Reynolds number of the flow through a parallel loop is found as

$$Re = \frac{\rho_{5,7}\ V_{avg,parallel}\ D_i}{\mu_{5,7}} \tag{5.36}$$

where *Re* is the Reynolds number, $\mu_{5,7}$ is the estimated dynamic viscosity, D_i is the inner diameter of the pipe, $\rho_{5,7}$ is the density of the fluid at the average temperature between states 5 and the estimated state 7, and V_{avg} is the average fluid velocity in the piping.

The pipe material, polyethylene, can be assumed to be smooth when considering internal flow as defined by Cengel [34]. With this consideration the friction factor of the pipe can be determined as follows:

$$f = \begin{cases} \dfrac{64}{Re} & for\ Re \leq 2300 \quad (5.37a) \\ (0.790 \ln Re - 1.64)^{-2} & for\ Re > 2300 \quad (5.37b) \end{cases}$$

Different equations are used to evaluate the friction factor for laminar flow ($Re \leq 2300$) and transitional/turbulent flow ($Re > 2300$). The equation for laminar flow is a modified version of the Darcy-Wiesbach Friction Factor given by Cengel [34]. The equation used for transitional and turbulent flow (Equation 5.37 b) is the *first Petukhov equation* [34] which defines the friction factor for turbulent flow within smooth pipes.

The length of the GL heat exchanger is estimated through heat transfer analysis. Under the assumption that the outer surface of the grout is at the temperature of the ground, and the ground is an infinite heat source, a cross sectional schematic of the piping arrangement and resistive network is developed (see Figure 5.1).

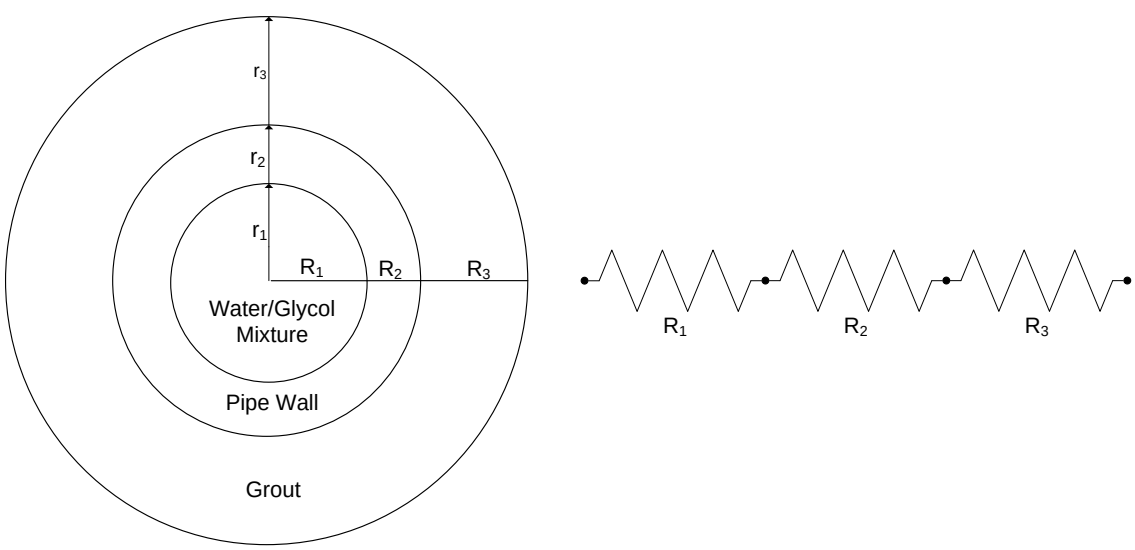

Figure 5.1: Cross-section of pipe configuration/resistive analysis arrangement

The thermal resistances of the layers in Figure 5.1 are as follows:

$$R_1 = \frac{1}{h_{conv}(2\pi\, r_1 l)} \tag{5.38}$$

$$R_2 = \frac{\ln\,(r_2 - r_1)}{2\pi\, k_{pipe}\, l} \tag{5.39}$$

$$R_3 = \frac{\ln\,(r_3 - r_2)}{2\pi\, k_{grout}\, l} \tag{5.40}$$

where R_1 is the convection thermal resistance between the pipe wall and the GL fluid, R_2 is the conductive resistance through the pipe wall, and R_3 is the conductive resistance through the grout encasing the pipe. The terms r_1, r_2, and r_3 correspond to the radius of each surface from the pipe center line, while r_1 and r_2 are set by the standardized sizing typical GL piping.

The terms k_{pipe} and k_{grout} are the thermal conductivity of the pipe wall and grout respectively and are found through literature with specification indicated within section 3.2.2. Also, h_{conv} is the heat transfer coefficient between the GL fluid and the pipe inner wall, and is calculated assuming forced internal convection with constant surface temperature.

Within vertical loop systems two pipes are contained in a single borehole and common borehole diameters exist within industry. Figure 5.2 shows a diagram of a borehole cross-section. The radius r_3 is estimated through the assumption that there is no thermal interaction between the pipes. Figure 5.2 illustrates the basic cross-section of a borehole with pipes installed. The maximum and minimum grout radius from the pipe center line are calculated with trigonometry. The average of the maximum and minimum is used as an estimate of r_3.

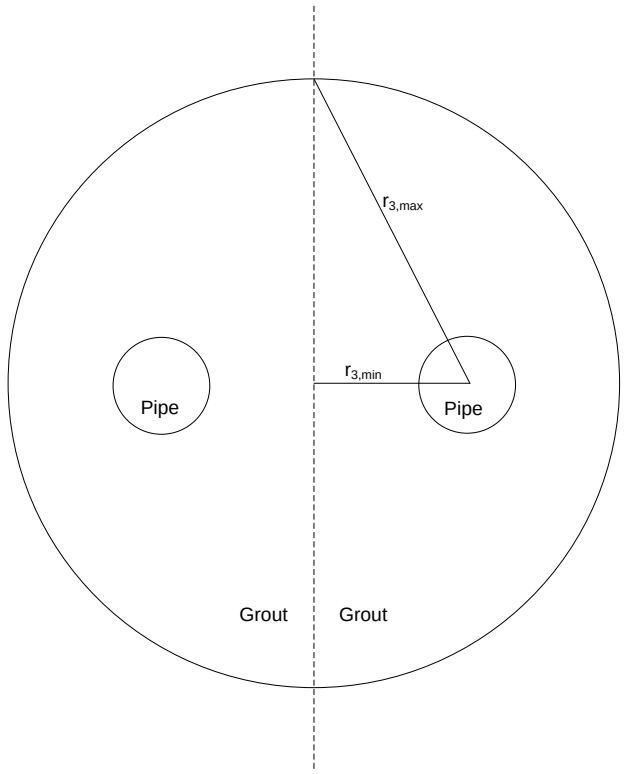

Figure 5.2: Cross-section of a single borehole

To find h_{conv} the Nusselt number is first calculated as follows:

$$Nu = \begin{cases} 3.66 & for\ Re \leq 2300 \quad (5.41a) \\ \dfrac{\left(f/_8\right)(Re - 1000)Pr_{5,7}}{1 + 12.7\left(f/_8\right)^{0.5}\left(Pr_{5,7}{}^{\frac{2}{3}} - 1\right)} & for\ 2300 < Re < 10000 \quad (5.41b) \\ 0.023Re^{0.8}Pr_{5,7}{}^{n} & for\ Re \leq 10000 \quad (5.41c) \end{cases}$$

where Nu is the Nusselt number, f is the friction factor, *Re* is the Reynolds number, and $Pr_{5,7}$ is the Prandtl number corresponding to the conditions between states 5 and 7. Equations 5.41 (b) and (c) are known as the *second Petukhov equation* and the *Dittus-Boelter equation,* respectively. They are used to determine the Nusselt number with a high degree of accuracy with respect to the Reynolds number [34]. Within Equation 5.41c, n is set to 0.4 for heating as specified by Cengel [34].

Using the Nusselt number the heat transfer coefficient is calculated using the equation:

$$Nu = \frac{h_{conv}\, k_{brine}}{D_i} \tag{5.42}$$

where k_{brine} is the thermal conductivity of the GL brine for the estimated average temperature between states 5 and 7.

The equations for thermal resistance (Equations 5.38-5.40) are solved on a per unit length basis. From the arrangement of the heat transfer mediums, the resistances are combined in series and summed to provide a total resistance per unit length of pipe:

$$R_{total} = R_1 + R_2 + R_3 \tag{5.43}$$

The heat transferred to the ground loop fluid per unit length is calculated as

$$\dot{Q}_{ht} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} \tag{5.44}$$

where $T_{\infty 1}$ denotes the temperature on the outer wall of the grout (ground temperature) and $T_{\infty 2}$ the mean fluid temperature between states 5 and 7.

The length of a single parallel loop is related to the rate of heat transfer that is required from the individual loop and the rate of heat that can be transferred, per meter length. The following equation represents this correlation:

$$l_{parallel} = \frac{\dot{Q}_{GL,parallel}}{\dot{Q}_{ht}} \tag{5.45}$$

The length is directly related to the heat transfer rate required from a single parallel loop. This rate is linked to the total heat rate that is required from the entire ground loop and the number of parallel loops, as follows:

$$\dot{Q}_{GL,parallel} = \frac{\dot{Q}_{GL,actual}}{n_{pl}} \tag{5.46}$$

where $\dot{Q}_{GL,actual}$ is the actual heat transfer rate required from the ground. When pump work is taken into account the heat rate required from the ground differs from the rate required by the HP through the evaporator. The following energy rate balance for the GL system is utilized:

$$\dot{Q}_{GL,actual} = \dot{Q}_{evap} - \dot{W}_{pump} \tag{5.47}$$

The total pump work rate is a function of the work required per unit mass to overcome the pressure drop within a parallel loop, the mass flow rate through a loop and the number of loops within the ground heat exchanger system. This relationship can be written as follows:

$$\dot{W}_{pump} = n_{pl}\dot{m}_{GL,parallel} \;\; w_{pump,actual} \tag{5.48}$$

where $\dot{W}_{pump}$ is the rate of work required, n_{pl} is the number of parallel loops, $\dot{m}_{GL,parallel}$ is the flow rate through a single parallel loop, and $W_{pump,actual}$ is the specific pump work.

For this model the pump is not assumed isentropic, but instead has an isentropic efficiency defined as

$$\eta_{pump} = \frac{w_{pump,ideal}}{W_{pump,actual}} \tag{5.49}$$

The isentropic efficiency of the pump is needed to find the actual pump work rate required. The actual specific enthalpy value at state 7 is found using an expanded version of the efficiency:

$$\eta_{pump} = \frac{w_{pump,ideal}}{\mathrm{h}_{7\mathrm{a}} - \mathrm{h}_6} \tag{5.50}$$

The ideal pump work is the amount of work required, assuming that the pumping process occurs isentropically. The ideal specific pump work per unit mass is found using the pressure difference across the pump:

$$w_{pump,ideal} = v_6(P_7 - P_6) \tag{5.51}$$

where P_7 is found through the use of the pressure drop that occurs in a parallel loop using:

$$P_7 = P_5 + \Delta P_{parallel} \tag{5.52}$$

The arrangement of the system and the method for analysis of this model requires numerous equations be solved simultaneously. The equations and solutions are

interrelated and utilize the values previously described. Equations 5.36, and 5.45 through 5.52 are solved concurrently.

Finally a new temperature for state 7 is calculated using the heat transfer rate required and the mass flow rate of a parallel loop, along with an estimated specific heat assuming the original average temperature between states 5 and 7. The relation can be expressed as

$$\dot{Q}_{GL,parallel} = \dot{m}_{parallel}\, c_{p,5,7}(T_5 - T_{7,new}) \tag{5.53}$$

where $\dot{Q}_{GL,parallel}$ is the heat transfer rate occurring within a parallel loop, $\dot{m}_{parallel}$ is the mass flow rate through a parallel loop, $c_{p,5,7}$ is the specific heat given the mean temperature between states 5 and 7, T_5 is the preset temperature at state 5, and $T_{7,new}$ is the estimated temperature at state 7. The new temperature estimated for state 7 is then used to repeat the above analysis in an iterative manner until a better estimate of T_7 is found. The calculated length and pressure loss of the GL provide an accurate estimate.

Once the iteration is complete, a good approximation of pump work rate is obtained. The electrical power required from the motor for pump operation is found as

$$\dot{E}_{motor,pump} = \frac{\dot{W}_{pump}}{\eta_{EM}} \tag{5.54}$$

where $\dot{E}_{motor,pump}$ is the rate of electrical energy consumed by the electric motor, $\dot{W}_{pump}$ is the rate of pump work required within the GL, and η_{EM} is the motor efficiency.

Chapter 6: Simulation

To simulate the performance of the 3 systems and to test their sensitivity to parametric variations, computer code is developed. The programs save considerable time, especially for iterative calculations. Programs have been developed by the author using Engineering Equation Solver (EES) to simulate the heat pump systems, while allowing for different operating conditions and parameters to be varied parametrically. The code structure is based on the systems, models, assumptions, and analyses described in Chapters 3-5. EES includes built in libraries of thermodynamic data tables for many common substances, including water and a range of refrigerants and antifreezes. EES accesses these tables and interpolates for the necessary thermodynamic information. One limitation of EES is an incomplete set of thermodynamic properties for water/anti-freeze mixtures. In determining values for a water/anti-freeze mixture, the mixture is assumed ideal, as described in Section 3.4. Nonetheless, EES contains a limited amount of information for water/anti-freeze solutions through the use its internal "BRINEPROP" function, which provides the freezing point, density, specific heat, thermal conductivity, dynamic viscosity and Prandtl number for a known antifreeze mixture concentration and temperature.

The simulation program allows the user to vary inputs to the systems and observe the effect on outputs. EES allows for parametric studies to be performed within the program itself. Parametric tables are developed within the program and the results are used by graphing programs to display the trends in various ways. A sample of the program code can be found in the Appendix .

The simulation allows the entire system to adjust to a variation in a parameter. Within the system, various parameters and variables are interrelated. When a parameter changes so do various other variables. For example, the temperatures at the exit of the condenser and evaporator are dependent on the pressure at those locations, even though the degrees of subcooling and superheating remain constant. This variation is accomplished by not over-specifying the system operating conditions. The original conditions are assumed to allow

the entire system to react to the change of a single variable in an attempt to simulate real operation behavior.

The simulation program is verified in several ways. Originally the system arrangements are assumed to be ideal with process efficiencies of 100%. The values for such configurations coincide with the values obtained though hand calculations. To confirm that the simulation correctly analyzes a system with irreversibilities, predicted values for each of the systems are again compared with results calculated by hand, and found to be in agreement. Through these tests, the simulation code was deemed verified.

In order to validate the simulations, they are applied to a system presented in an experimental study by Hapbasli [24]. The system analyzed in Hapbasli's study is a basic vapour compression cycle that utilizes a vertical borehole ground connection. R-22 is used as the refrigerant and the brine is a water/ethyl glycol mixture with 10% glycol by mass. When the system parameters from Hapbasli's study are applied to the developed simulation, the results are found to be almost identical in terms of HP and system COPs, with differences in the COP values from the simulation and those found of Hapbasli being about 1%. This difference is deemed acceptable for validation of the simulation. Comparison of system parameters presented by Hapbasli and those created through simulation are illustrated in Tables 6-1 and 6-2.

Table 6-1: Comparison of state conditions between those from simulation and from the alternative study

	Temperature (°C)		Pressure (kPa)		Enthalpy (kJ/kg)		Entropy (kJ/kg K)	
Description	**PS***	**AS****	**PS***	**AS****	**PS***	**AS****	**PS***	**AS****
Evaporator inlet	-11.12	-10.8	341	341	250.7	251.15	1.195	1.196
Evaporator exit	-6.0	-6.0	341	341	404.2	404.6	1.781	1.782
Condenser inlet	99.3	99.2	1911	1911	464.4	464.3	1.822	1.822
Condenser exit	40.77	40.8	1911	1911	250.7	251.15	1.169	1.170

*PS represents the simulation program presented in the present study
**AS represents the alternative study utilized for validation

Table 6-2: Comparison of performance characteristics of simulation program and alternative study

	Results from current study simulation	Results from alternative study
Compressor power (kW)	1.2	1.5
Refrigerant flow rate (kg/s)	0.199	0.02
COP_{HP}	2.84	2.85

Chapter 7: Results and Discussion

A series of different investigations are performed. A basic comparison is first developed where the systems are compared to each other when the original conditions and assumptions are held fixed. Further investigation is performed regarding the components that consume power within the system. The effect of changing efficiency of these components is explored. Different operating conditions specific to the heat pump cycle are also explored. The effect on performance and ground loop requirements of variations in condenser and evaporator pressure along with the degree of superheating and subcooling at the outlet of the evaporator and condenser, respectively, are also explored.

For the studies the initial conditions are held constant while one parameter is varied over an appropriate range. The ranges are selected so that the systems can theoretically operate, and to cover the ranges described in the literature. The choice of range for a study is explained in its section.

7.1 Basic comparison

A comparison is presented of the systems considered, which includes specification of all state conditions for each system. The main aspects are compared; including compressor and pump work rate, rate of heat transfer from the ground loop system, overall ground loop length, and performance. The performance of each system is compared in terms of heat pump COP and overall system COP. Tables 7-1, 7-2, and 7-3 illustrate the conditions at each state for systems 1 to 3, respectively; values contained within the tables are a combination of values derived from the given assumptions as well as those calculated through analysis. Comparison of the main aspects of each heat pump's operation is shown in Table 7-4.

Table 7-1: State conditions for system 1, following states identified in Figure 4-1 and 4-4

State	Fluid	Temperature (ºC)	Pressure (kPa)	Quality	Specific enthalpy (kJ/kg)	Mass flow rate (kg/s)
1	Refrigerant	-10.09	200	0.2985	99.92	0.5136
2	Refrigerant	-5.093	200	1	248.7	0.5136
3	Refrigerant	61.16	1000	1	294.6	0.5136
4	Refrigerant	34.37	1000	0	99.92	0.5136
5	Brine	7.3	150	0	170.9	3.758
6	Brine	1.3	150	0	150.6	3.758
7	Brine	2.111	223.6	0	150.6	3.758

Table 7-2: State conditions for system 2, following states identified in Figure 4-2 and 4-4

State	Fluid	Temperature (ºC)	Pressure (kPa)	Quality	Specific enthalpy (kJ/kg)	Mass flow rate (kg/s)
1	Refrigerant	-10.09	200	0.3542	111.4	0.5136
2	Refrigerant	-5.093	200	1	248.7	0.5136
3	Refrigerant	61.16	1000	1	294.6	0.5136
4	Refrigerant	34.37	1000	0	99.92	0.5136
5	Brine	7.3	150	0	170.9	3.468
6	Brine	1.3	150	0	150.6	3.468
7	Brine	2.111	223.6	0	150.6	3.468
8	Refrigerant	-10.09	200	0.2985	99.92	0.5136

Table 7-3: State conditions for system 3, following states identified in Figure 4-3 and 4-4

State	Fluid	Temperature (ºC)	Pressure (kPa)	Quality	Specific enthalpy (kJ/kg)	Mass flow rate (kg/s)
1	Refrigerant	-10.09	200	0.2632	92.66	0.4905
2	Refrigerant	-5.093	200	1	248.7	0.4905
3	Refrigerant	61.86	1000	1	295.3	0.5117
4	Refrigerant	34.37	1000	0	99.92	0.5117
5	Brine	7.3	150	0	170.9	3.764
6	Brine	1.3	150	0	150.6	3.764
7	Brine	2.111	200	0	150.6	3.764
8	Refrigerant	8.91	400	0.1878	99.92	0.02119
9	Refrigerant	22.41	400	1	268.1	0.02119
10	Refrigerant	29.37	1000	0	92.66	0.4905
11	Refrigerant	22.41	400	1	268.1	0.4905
12	Refrigerant	22.41	400	1	268.1	0.5117

Table 7-4: Summary of heat pump performance and ground loop characteristics

Calculated characteristics	System 1	System 2	System 3
Pressure ratio	5	5	5
Compressor work rate (kW)	23.57	23.57	23.45
Compressor motor energy consumption rate (kW)	29.46	29.46	29.31
Pump work rate (kW)	0.2978	0.2749	0.2983
Pump motor energy consumption rate (kW)	0.3723	0.3436	0.3729
Rate of heat supplied from GL system (kW)	76.43	70.54	76.55
Total GL length	3346	3088	3352
COP_{HP}	**4.242**	**4.242**	**4.265**
COP_{System}	**3.352**	**3.355**	**3.369**

One of the most important performance measures for each system is its system COP. Of the 3 systems, system 3 is observed to have the highest COPS, with a HP COP of 4.265 and a system COP of 3.369. The high COPs are mostly due to the fact that the compressor work is lower for this unit compared to the other two. System 3 also requires the most heat from the ground for operation, causing the GL length to be the longest. The increased GL length results in the system having the highest pump work due to the coinciding pressure drop in the loop. The increase in pump work reduces the system COP but this effect is relatively insignificant because of the comparably higher compressor work and HP COP.

Compared to system 1, system 3 exhibits a HP COP and a system COP that are higher by 0.54% and 0.54%, respectively. As mentioned earlier, the increase is due to the fact that the compressor work is lower. In system 3 not all the refrigerant, which flows through the condenser, is compressed from the lowest to highest pressure. The system allows for a fraction of the refrigerant to be compressed from an intermediate pressure to high pressure, reducing the work required by the first compressor.

System 3 is found to be able to produce the same compressor exit temperature with a lowered pressure ratio between the condenser and evaporator pressure, which translates into a further reduction in compressor work if the compressor exit temperature is set to that of system 1 and 2.

Both systems 1 and 2 are seen to require the same compressor work and have the same HP COP of 4.242. The difference in the systems is that all the required heat in system 1 is extracted from the ground, where system 2 allows for some of the heat to be extracted from the electric motor as waste heat; thermal energy is extracted from the ground for system 1 at a rate of 76.43 kW compared to only 70.54 kW for system 2. The reduction of the heat transfer rate from the ground translates directly into a reduction in the required GL length of almost 260 meters (8%) for the same heating load. A shorter length leads to reduced pump power requirement through a decrease in total flow through the pump within the GL. Less pump work translates into an increase in system COP, following Equation 5.7.

The GL length can be measured in terms of length per kW of heat supply rate to the building space. Systems 1, 2 and 3 exhibit values of 33.46 m/kW, 30.88 m/kW and 33.52 m/kW, respectively. The calculated values coincide with the vertical GL sizing guidelines of Natural Resources Canada [6], which suggests a range of 17 to 39 m of vertical loop length per kW heating load for GL layouts utilizing a nominal pipe size between 25.4 and 50.8 mm. Kara [27] calculates a length of 33.6 m/kW for the basic vapor compression cycle with similar operating conditions.

In Table 7-4 it can be seen that the difference is about 8% between heat input rate, GL length and pump work rate between system 1 and 2. This energy saving is not directly observed for the system COP; the system with the motor cooling only has a system COP increase of about 0.09%. Through Equations 5.9 and 5.10 it can be seen that pump work has little influence on the COP due to the large difference in power required by compressor compared to the pump.

Note that during the early stages of analysis it was noticed that multiple parallel loops are required for a system that involves high heat rates from the GL. Originally a single loop was employed and it was found that the pressure drop that occurs with the corresponding length of the pipe is substantial. In order to overcome the pressure drop the pump requires a very high pressure outlet that approaches the upper pressure limit of the pipe itself. Employing multiple parallel loops allows for the total pressure drop in the system to be distributed between each loop, resulting in a lower pressure at state 7.

7.2 System Component Analysis

This section investigates and compares the effects of varying the parameters of certain heat pump component characteristics. Most of the original conditions are held fixed while one operating parameter is varied over an appropriate range. The condenser and evaporator pressure for each system remain the same as those used in the basic comparison. Compressor, pump and electric motor efficiencies are investigated.

7.2.1 Compressor Efficiency Analysis

The compressor efficiency is varied from 65% to 100% for each of the three heat pump systems. The range outlines the efficiencies stated by Cengel et al. [33] for low to high efficiency compressors.

The results are presented in Figure 7.1. In that figure an overall trend can be seen in which the HP COP increases with increasing compressor efficiency. This result is expected since the compressor work decreases as the efficiency increases, according to Equations 5.1, 5.16, and 5.27. The trend appears to be close but is not exactly linear. The trend is created through a non-linear relationship between compressor efficiency and the required rate of compressor work within the system models. This relationship is created through changes in the specific enthalpy at the exit of the compression process as well as the mass flow rate within the heat pump systems.

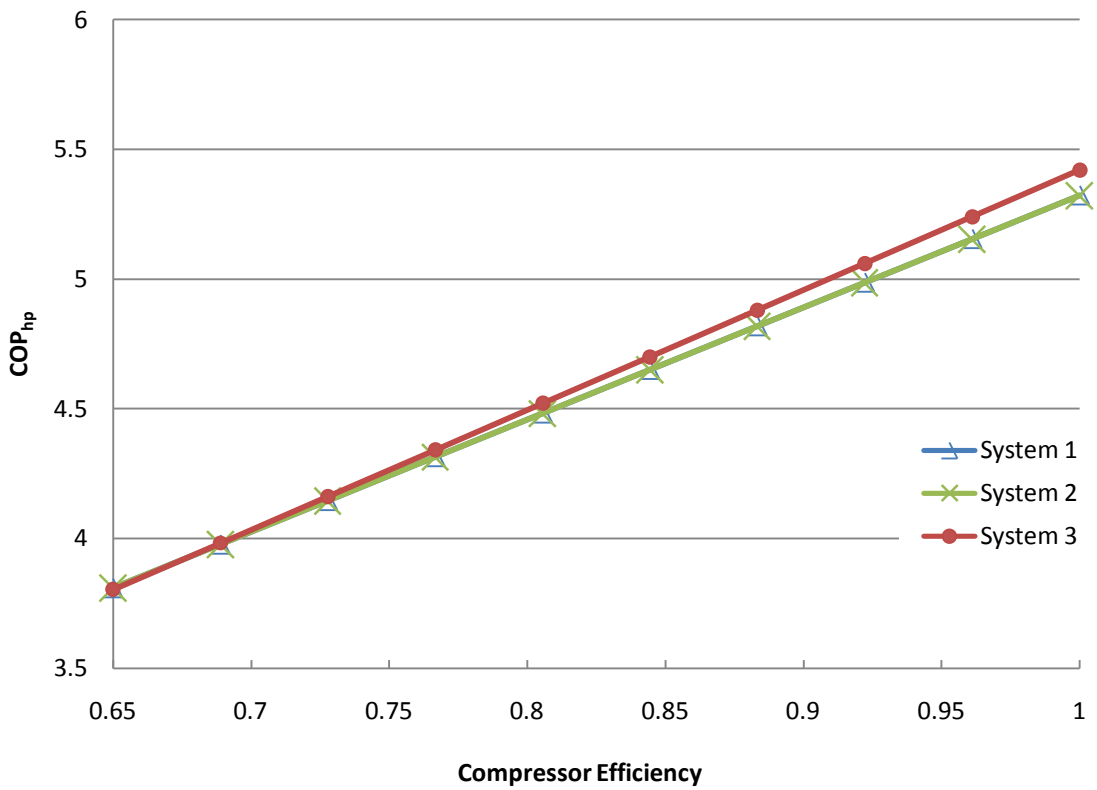

Figure 7.1: Effect of changing compressor efficiency on heat pump COP for each system

As the compressor efficiency increases, systems 1 and 2 both exhibit identical COPs and trends. The main difference between the systems is the heat extracted from the ground loop for each system. Operating conditions at the compressor inlet and exit are identical, so both systems have identical responses when the efficiency is varied.

Comparing systems 1 and 3 shows that the slopes of the linear relationships differ. The COP of system 3 increases more rapidly with variation in compressor efficiency compared to the other systems. The differences in COP can be seen in Table 7-5.

Table 7-5: Comparison of heat pump COP with varying compressor efficiency

	COP_{HP}	
Compressor efficiency	**System 1***	**System 3**
0.65	3.81	3.80
0.69	3.98	3.98
0.73	4.14	4.16
0.77	4.32	4.34
0.81	4.48	4.52
0.84	4.65	4.70
0.88	4.82	4.88
0.92	4.98	5.06
0.96	5.15	5.24
1	5.32	5.42
Range**	1.51	1.62

*Systems 1 and 2 have identical calculated COP_{HP} values

**Range is the difference in calculated COP_{HP} values coinciding to the highest and lowest motor efficiency

Table 7-5 shows the difference in HP COP from the lowest and highest compressor efficiency for both systems 1 and 3. System 3 has the largest change in HP COP of the 3 systems, which indicates that system 3 is the most sensitive to variations in compressor efficiency, followed by systems 1 and 2.

It is thought that this response stems from the design and operation of system 3. System 3 includes two stages of compression; each stage contributes to the total amount of compressor work for the HP. For compressor one, the specific entropy at state 11 is the same as that at state 2 when assuming isentropic compression. The entropy at state 2 is used to determine the ideal specific enthalpy after compression which is utilized with the compressor efficiency to find the actual enthalpy at state 11. As the compressor efficiency

is varied the ideal enthalpy at state 11 does not change and but the actual enthalpy does and is a function of compressor efficiency only.

In the developed model, the conditions at state 12 are identical to those of state 11. Since the conditions between states 11 and 12 are identical the conditions at state 12 are also a function of compressor efficiency with respect to compressor one. As the compressor efficiency increases the actual specific enthalpy and entropy at state 11, and in turn state 12, decreases.

When compressor two is considered the conditions at the inlet (State 12) change with compressor efficiency, unlike compressor one which has static inlet conditions over the entire range of efficiencies. Since the inlet conditions are changing for compressor two the ideal exit conditions also change with compressor efficiency. When the efficiency equation for compressor two (Equation 5.27) is reviewed it can be seen that both compressor efficiency and ideal enthalpy change over the range of compressor efficiencies. Having both variables changing within Equation 5.27 creates a situation where the actual specific enthalpy at state 3 is reduced past what the specific enthalpy would be if the inlet conditions were held constant and only the efficiency was varied. The specific entropy and enthalpy for the specified compressor efficiencies are illustrated in Table 7-6 for systems 1 and 3.

The final result is a low specific enthalpy at state 3 for system 3. Equation 5.31 shows that the rate of work required by compressor two is dependent on the actual enthalpy at the exit of the compressor. The specific enthalpy at the exit of this compressor is reduced more rapidly with increasing compressor efficiency compared to systems 1 and 2, which translates into reduced work rate requirement for the compression process as well. The lowered rate of work required in system 3 contributes to the divergence of the calculated COP between the systems.

Table 7-6: Ideal specific enthalpy and entropy values at State 3

	System 1*			System 3		
Compressor efficiency	s_{3s} (kJ/kg K)	h_{3s} (kJ/kg)	h_3 (kJ/kg)	s_{3s} (kJ/kg K)	h_{3s} (kJ/kg)	h_3 (kJ/kg)
0.65	0.9538	283.1	301.7	0.9803	291.8	303
0.69	0.9538	283.1	298.7	0.9761	290.4	299.8
0.73	0.9538	283.1	296	0.9723	289.2	296.9
0.77	0.9538	283.1	293.6	0.9689	288.1	294.3
0.81	0.9538	283.1	291.4	0.9658	287	291.9
0.84	0.9538	283.1	289.5	0.9629	286.1	289.8
0.88	0.9538	283.1	287.7	0.9604	285.3	287.9
0.92	0.9538	283.1	286	0.958	284.5	286.2
0.96	0.9538	283.1	284.5	0.9558	283.8	284.6
1	0.9538	283.1	283.1	0.9538	283.1	283.1

* Systems 1 and 2 have identical calculated COP_{HP} values solutions

It is also seen that at a very low efficiency the HP COP of system 1 exceeds that of system 3. This observation is also a result of varying ideal specific entropy at the exit of compressor two in system 3. When the efficiency is low it creates a situation where the work required by system 3 exceeds that required by system 1 to achieve the same condenser pressure.

As the efficiency is varied the mass flow rate of the refrigerant through the condenser alters as well. The specific enthalpy at state 3 is the main contributor for the calculation of the flow rate. Figure 7.2 shows that the specific enthalpy at state 3 for systems 1 and 2 are identical. Overall the specific enthalpy with regard to the same state within system 3 exceeds that of system 1 and progressively approaches the values observed for the basic system as efficiency increases.

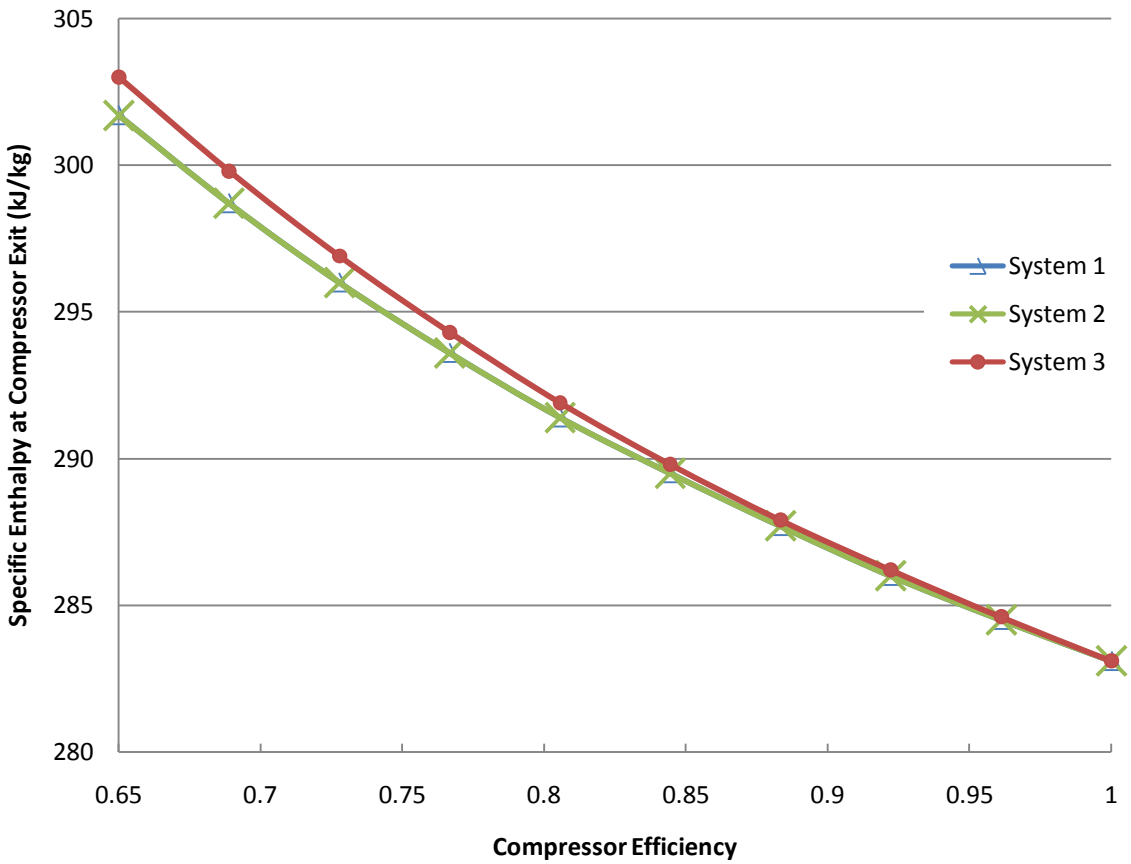

Figure 7.2: Effect of varying compressor efficiency on specific enthalpy at condenser inlet for each system

The trend is inverted when the mass flow rate is considered, as illustrated by Figure 7.3. As the compressor efficiency increases the refrigerant flow rate increases. The trend occurs because the enthalpy at the inlet of the condenser decreases with increasing compressor efficiency; the enthalpy is determined through the isentropic efficiency of ther compressor (Equations 5.1 and 5.27), where increasing the efficiency with static inlet conditions results in a reduation in the calculated enthalpy at the compressor exit. The flow rate for system 3 is always below that of the other two systems until the efficiency of the compressor is 100%, at which point the flow rates are equal.

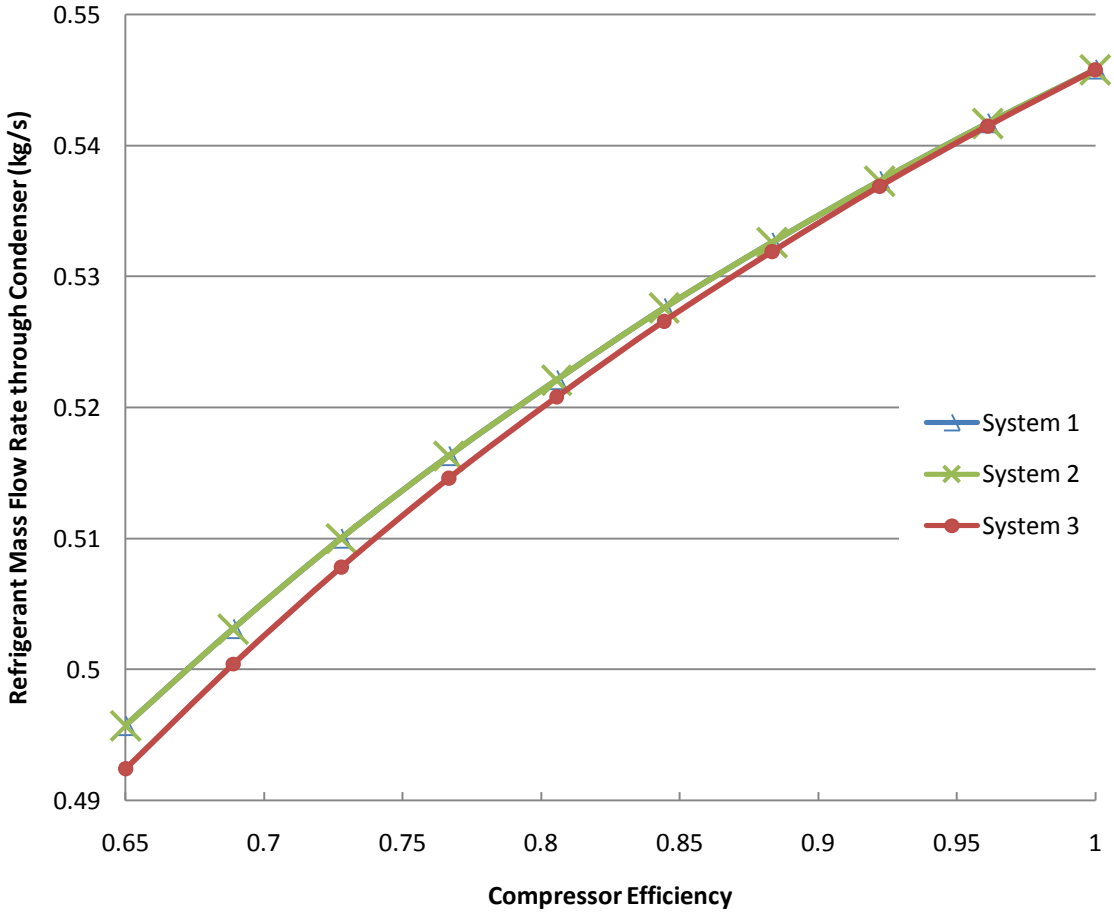

Figure 7.3: Effect of varying compressor efficiency on condenser flow rate for each system

From the first investigation one could assume that when the compressor efficiency is 100%, the compressor work for all the systems would be equal. This is not the case, as seen in Figure 7.4. When the efficiency is set to 100%, the compressor work in system 3 is lower than for the other two systems.

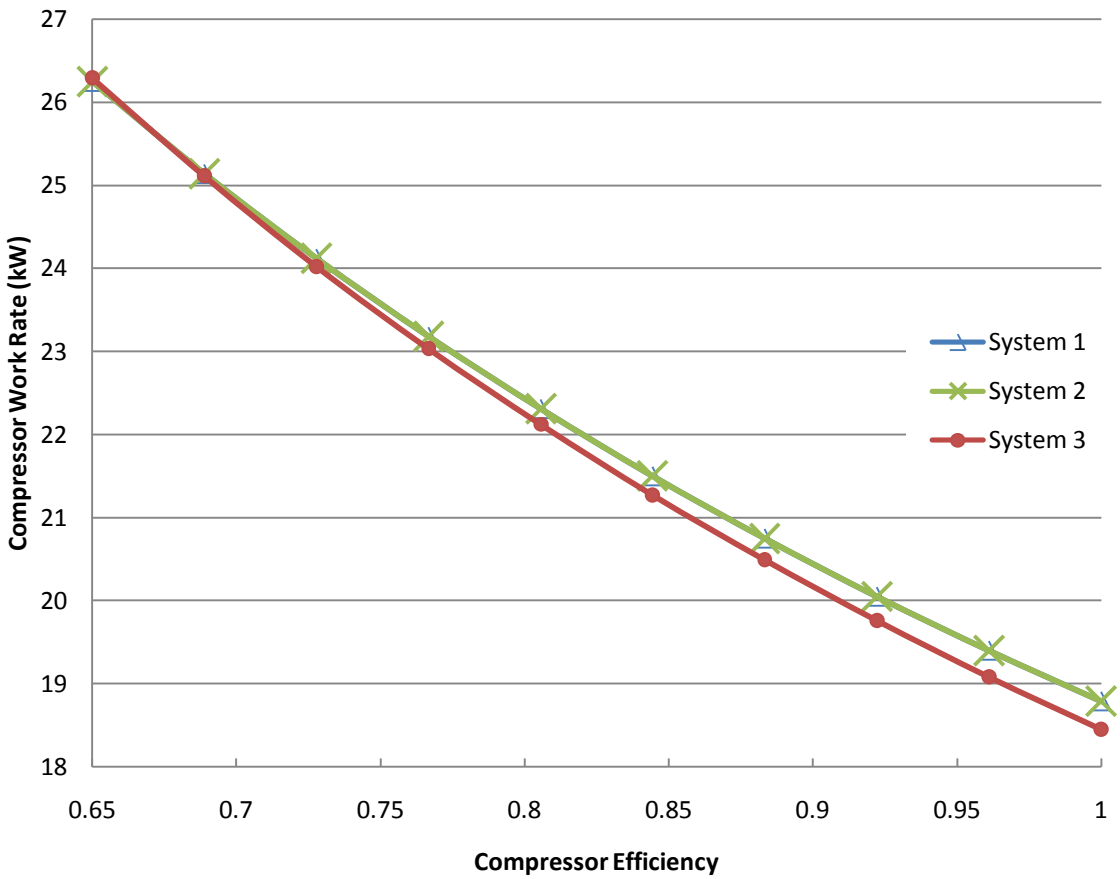

Figure 7.4: Effect of varying compressor efficiency on rate of compressor work required for each system

The trend of decreasing compressor work and increasing mass flow rate through the evaporator seem to be contrdictive. The trend of reducing compressor work with increasing flow rate results from a insignificant change in refrigerant flow rate over the range of compressor efficiencies. The calculated compressor work is mostly effected by the changes in specific enthalpy across the compression stages.

It can also be observed in Figure 7.4 that system 3 is the most sentitive to changes in compressor efficiency as it covers the largest range of COP values. The economizer arrangement, within system 3, is best utilized when high efficiency compressors are available.

Similar trends are observed when system COP is considered, as illustrated in Figure 7.5. As the compressor efficiency increases, a larger difference is seen in system COP between system 3 and the other two systems, comparable to the HP COP trends.

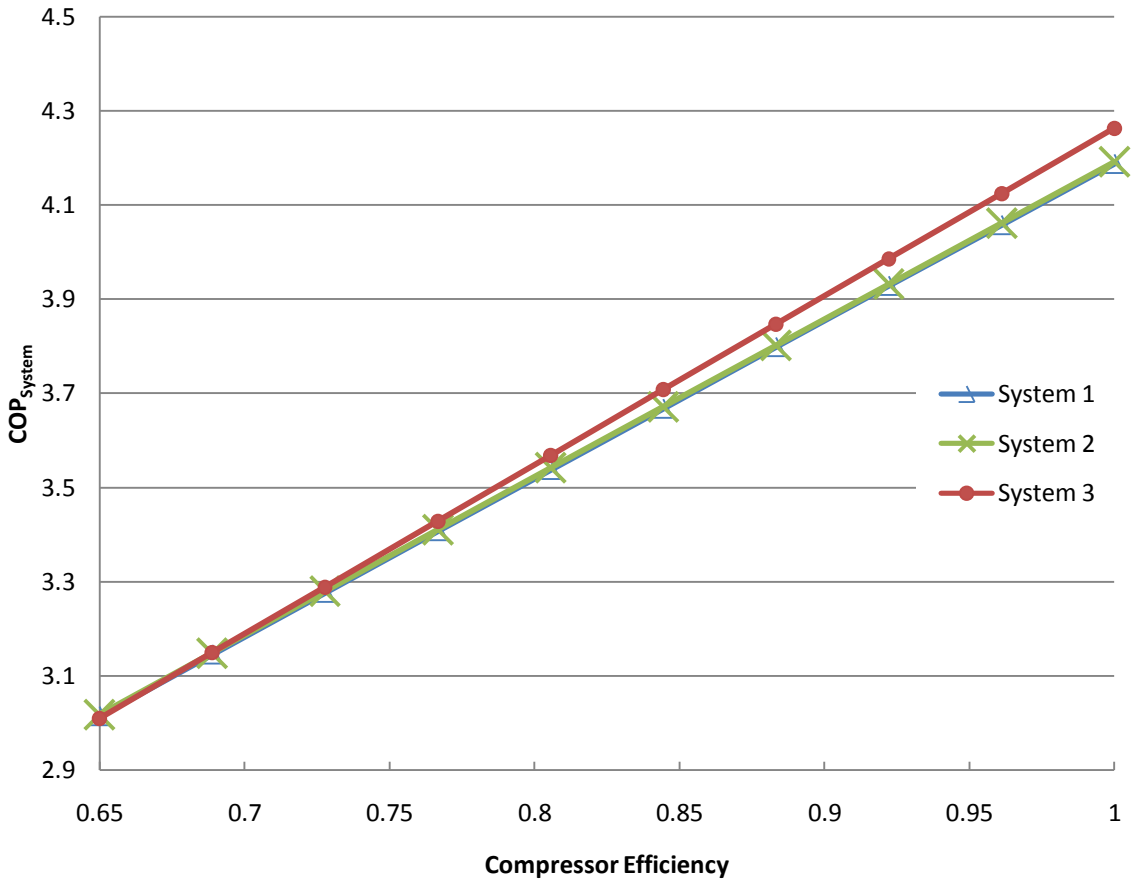

Figure 7.5: Effect of varying compressor efficiency on system COP for each system

Although difficult to see in Figure 7.5, the system COP values for system 2 are slightly higher than for system 1. Despite the fact that different COP values are observed, the slopes of the two established trends are identical. System COP values are compared in Table 7-7.

Table 7-7: Effect on system COP with variation in compressor efficiency

	COP_{System}		
Compressor efficiency	**System 1**	**System 2**	**System 3**
0.65	3.02	3.02	3.01
0.69	3.14	3.15	3.15
0.73	3.28	3.28	3.29
0.77	3.41	3.41	3.43
0.81	3.54	3.54	3.57
0.84	3.67	3.67	3.71
0.88	3.8	3.80	3.85
0.92	3.93	3.93	3.99
0.96	4.06	4.06	4.12
1	4.19	4.19	4.26
Range*	1.17	1.17	1.25

*Range is the difference in calculated COP_{system} values coinciding to the highest and lowest motor efficiency

As the compressor efficiency varies, it directly affects the rate of heat required from the GL. Figures 7.6 and 7.7 show that as the compressor efficiency increases so does the required rate of heat transfer from the GL and in turn the GL length. As compressor efficiency increases less work is provided to the heat pump system; with a preset heating load this translates into an increased demand from the ground loop in order to maintain thermodynamic balances within the HP cycles. It can be seen that system 3 has the largest range in terms of the rate of heat transfer from the GL. This trend is attributable to the fact that system 3 encounters the largest range of calculated COP values.

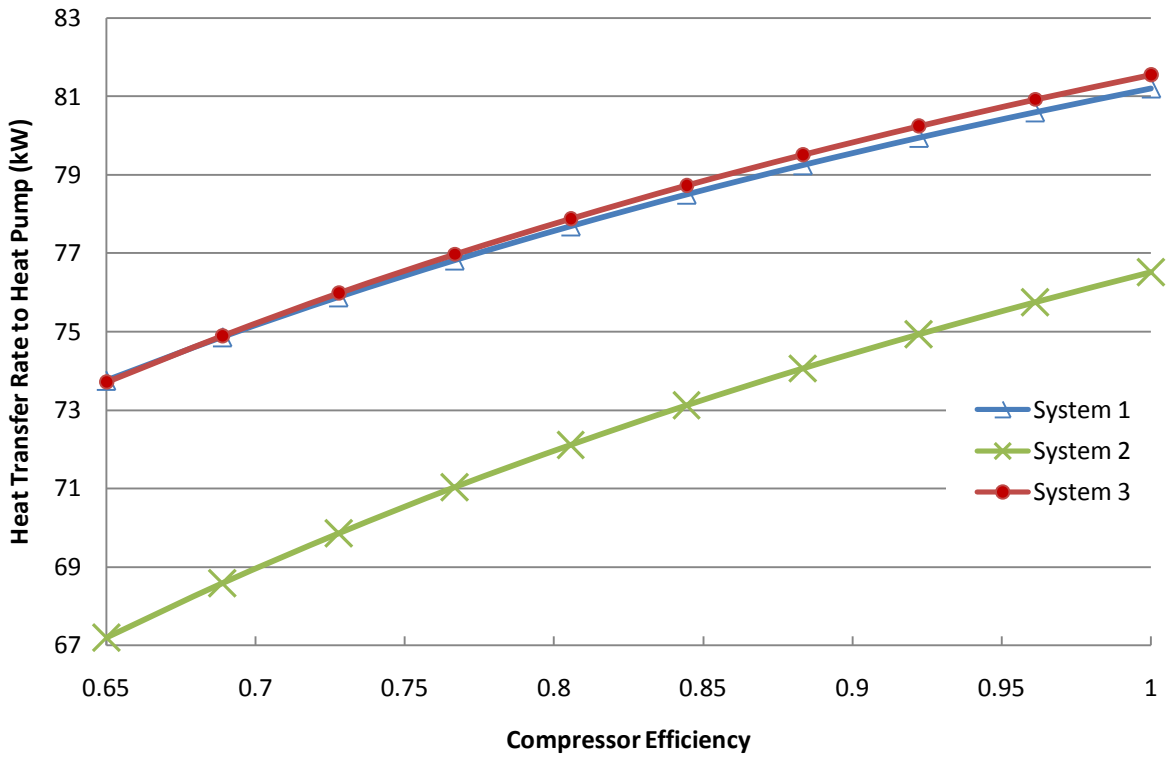

Figure 7.6: Effect of varying compressor efficiency on the rate of heat transfer to the heat pump from the ground loop

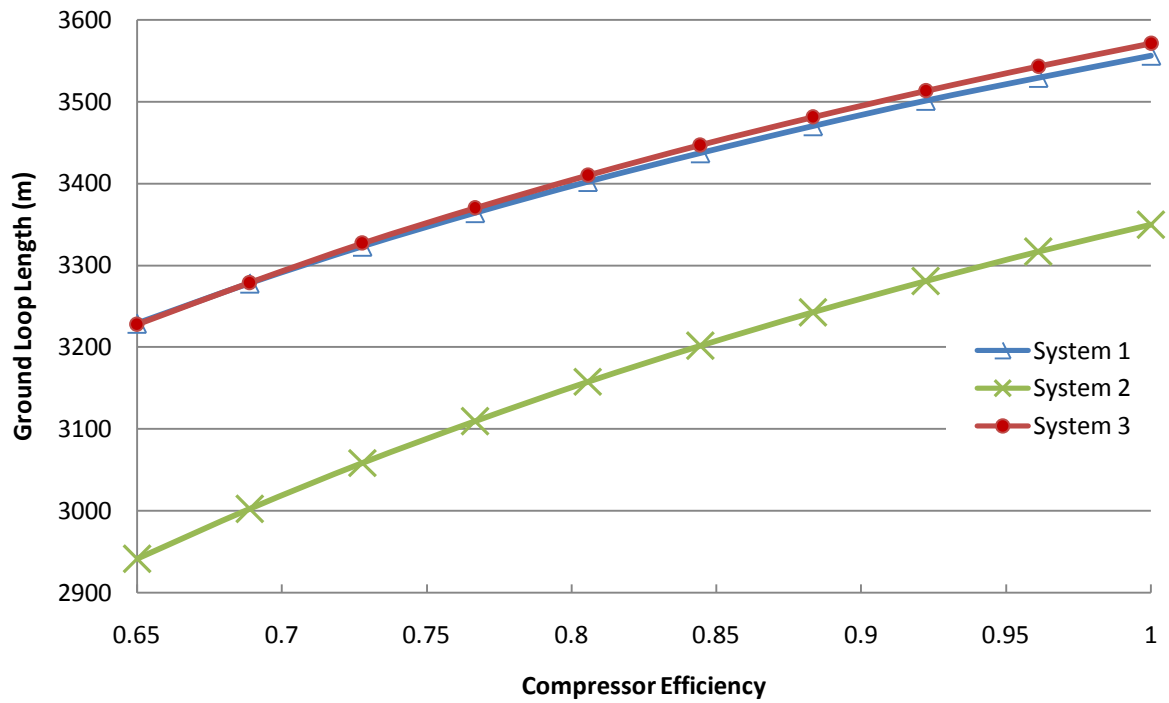

Figure 7.7: Effect of varying compressor efficiency on ground loop length for each system

There is a direct connection between the COP of a heat pump cycle and the GL length that is required for system operation. From the figures presented in this section it can be clearly seen that as the COP increases so does the amount of heat required from the GL and thus GL length. The trend arrises from concept of energy balance and the conservation of energy. Essentually there are three main energy flows with regard to the heat pump cycle; energy flow from the heat pump to the building, energy flow from the GL to the heat pump cycle, and energy flow from the compressor to the heat pump cycle. When the heating load is static, reduction in the amount of energy being supplied to the heat pump cycle from the compressor creates an imbalance in energy flows into and out of the heat pump cycle. In order to allow for the heating load to remain constant more energy is required from the GL. In an investigation of GL sizing for vertical closed loop arrangements, Cane et al. [40] illustrate that as the rate of heat required from the GL increases so does the GL length required. RETScreen International [17] also provides information that suggests that there is a direct relationship between the COP of a heat pump system and the length required for the ground loop, when they describe ground source heat pump project analysis.

7.2.2 Pump Efficiency Analysis

The effect of varying pump efficiency on the three different systems is investigated. All variables from the basic comparison are held constant with the exception of pump efficiency, which is varied from 75% to 100% for each heat pump system, where the values follow those suggested by Cengel et al. [33].

The pump is discretely involved in the ground loop system. When only the pump is considered, the operation of the HP systems, independent of the GL, remains constant. The rate of heat required by the refrigerant through the evaporator remains constant for each system, causing the HP COP for each system to remain unchanged throughout the investigation. System COP can only be considered for comparison of performance. The relationship between pump efficiency and system COP is illustrated in Figure 7.8.

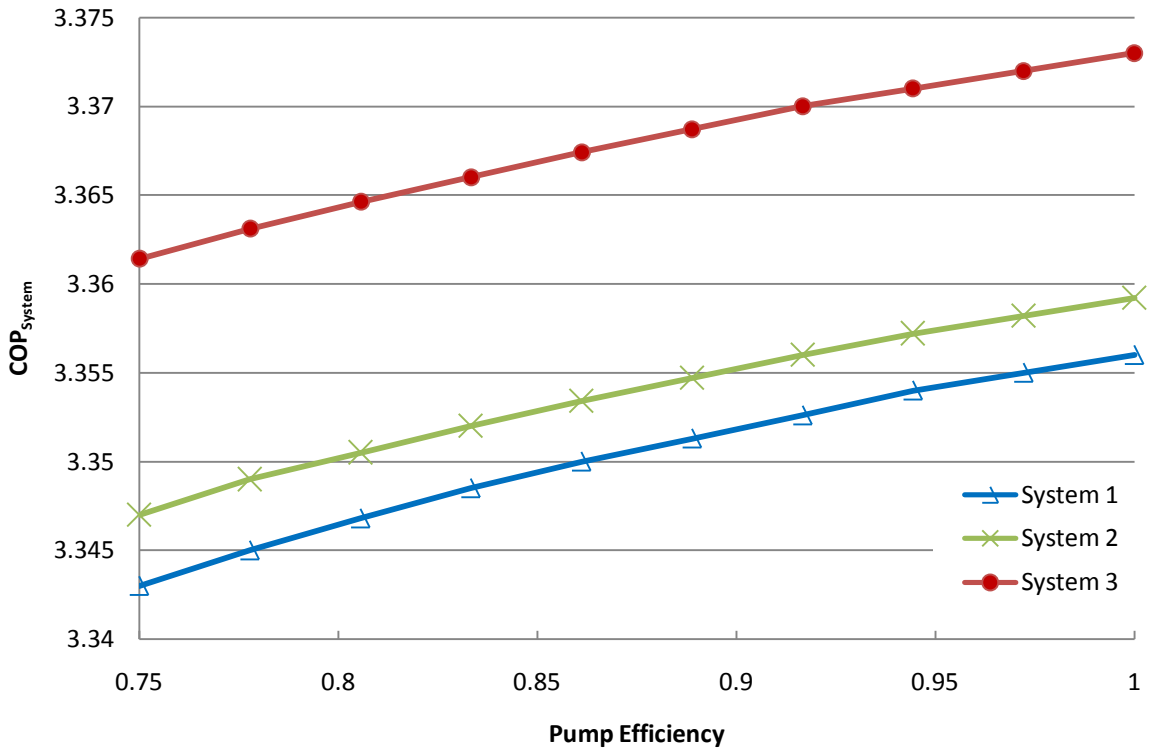

Figure 7.8: Effect of varying pump efficiency on system COP for each system

The system COP for all HP arrangements is found to increase with increasing pump efficiency. Every system has an overall COP change of 0.013 for the range of efficiencies considered. An identical change in system COP for each system suggests that all the systems have the same general response and sensitivity to the change in terms of performance. The magnitude of the change translates into a difference of about 0.09% for each system, which is relatively insignificant.

The increase in system COP is due to a reduction in the pump power within all systems. Figure 7.9 illustrates how the pump power is reduced as pump efficiency increases. It is observed that systems 1 and 3 have the same reduction in pump power (0.089 kW) and system 2 has a reduction of 0.082 kW. Systems 1 and 3 exhibit the largest reductions in pumping power with variation in pump efficiency due to the fact that they have a higher

pump requirements compared to system 2. When high efficiency is applied the degree of power savings for these systems is greater than for system 2.

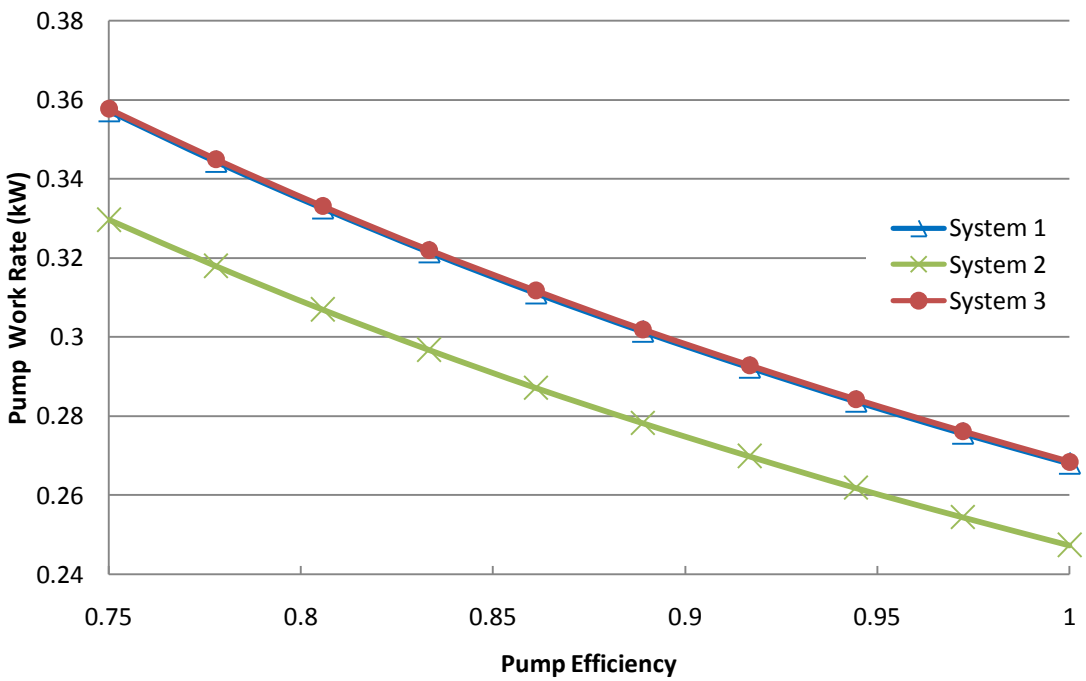

Figure 7.9: Effect of varying pump efficiency on rate of pump work required for each system

The model utilized throughout the study directly relates the pump power and required length, allowing the energy flows into and out of the GL to be equal. As the pump power is reduced more energy must be drawn from the ground in order to supply the heat pumps with the energy they require. Each system experiences the same increase in the ground loop length of about 2 m. This length is relatively insignificant when the entire HP system and total GL length is considered.

7.2.3 Motor Efficiency Analysis

The developed model includes electric motors that are coupled to the compressors and pump within each system. The motors convert electricity to mechanical work for the use in compression and pumping. The motor efficiency defines how well a particular motor converts electrical energy to mechanical energy. For this analysis the motor efficiency is varied from 35% to 100% for each heat pump system. This range is suggested as suitable by Cengel et al. [33].

Similar to the pump analysis, varying the motor efficiency only has an effect on the system COP when considering performance. Figure 7.10 shows the trends with respect to system COP for the three systems. All the systems are seen to exhibit similar, but not identical trends. The trend appears to show a linear relationship between motor efficiency and system COP; which is expected as the electrical energy consumed by the motor has a linear relationship to motor efficiency through Equations 5.6 and 5.54. The rate of electrical energy consumed is reduced linearly, through a linear increase in motor efficiency, resulting in a linear increase in system COP through Equation 5.9.

Table 7-8 lists calculated values for system COP and the range that each system covers.

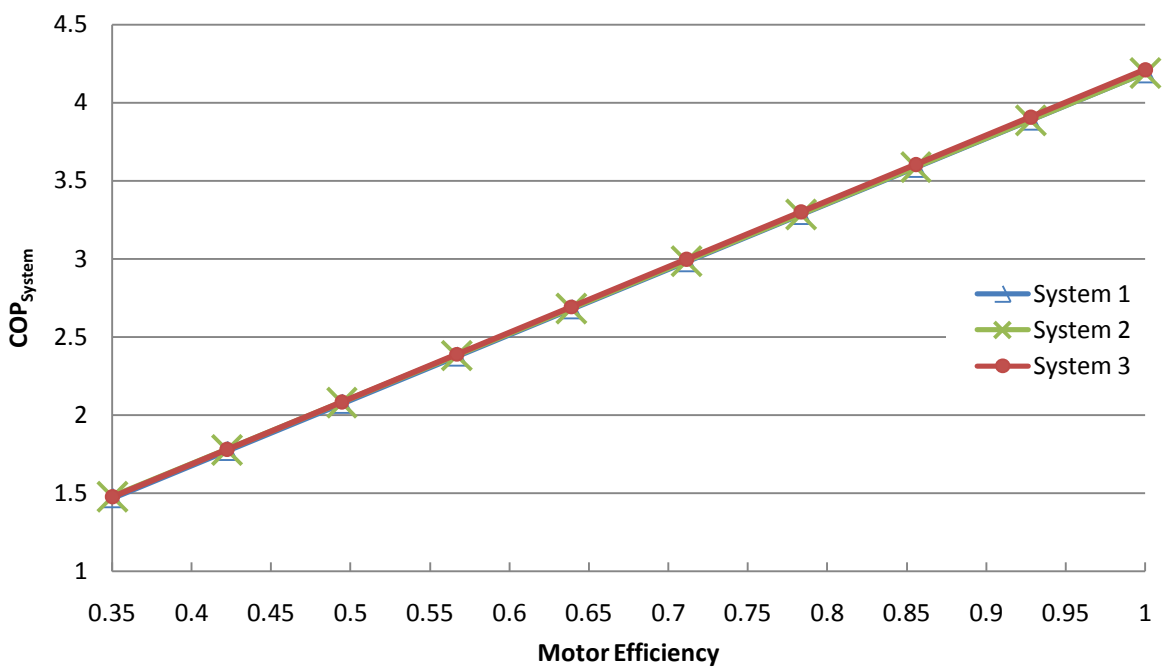

Figure 7.10: Effect of varying motor efficiency on system COP for each system

Table 7-8: System COP values for various motor efficiencies

	COP_{System}		
Motor efficiency	**System 1**	**System 2**	**System 3**
0.35	1.47	1.48	1.47
0.42	1.77	1.78	1.78
0.49	2.07	2.08	2.08
0.56	2.37	2.38	2.39
0.64	2.68	2.68	2.69
0.71	2.98	2.98	3.0
0.78	3.28	3.29	3.3
0.85	3.58	3.59	3.60
0.92	3.89	3.89	3.91
1	4.19	4.19	4.21
Range*	2.72	2.71	2.74

*Range is the difference in calculated COP_{System} values coinciding to the highest and lowest motor efficiency

The calculated range of system COP values for systems 1, 2 and 3 are seen in Table 7.8 to be COP of 2.72, 2.71 and 2.74, respectively, over the selected range of motor efficiencies. System 3 has the largest change in system COP for the given motor efficiencies, suggesting that it is the most sensitive to changes in motor efficiency. During this study, the power required for compression remains constant within each system and is equal to the values calculated in Section 7.1. Compared to systems 1 and 2, system 3 has a more pronounced change in system COP values with varying motor efficiencies because of its lower compressor requirements.

The system COP range for system 2 is the smallest for the given range of motor efficiencies due to the fact that at the lowest motor efficiency the system has the highest COP which stems from lowered GL and pump requirements. When motor efficiency is low more thermal energy is available to the refrigerant through motor cooling, which reduces the energy transfer requirement of the GL as seen in Figure 7.11. As the efficiency increases it is seen that the COP for system 2 approaches that of system 1. This trend is expected due to the fact that when the motor efficiency approaches 100% the

amount of heat extracted from the motor assembly approaches zero. If no heat is supplied to the system by the motor, all of the required heat must come from the ground as seen in Figure 7.11.

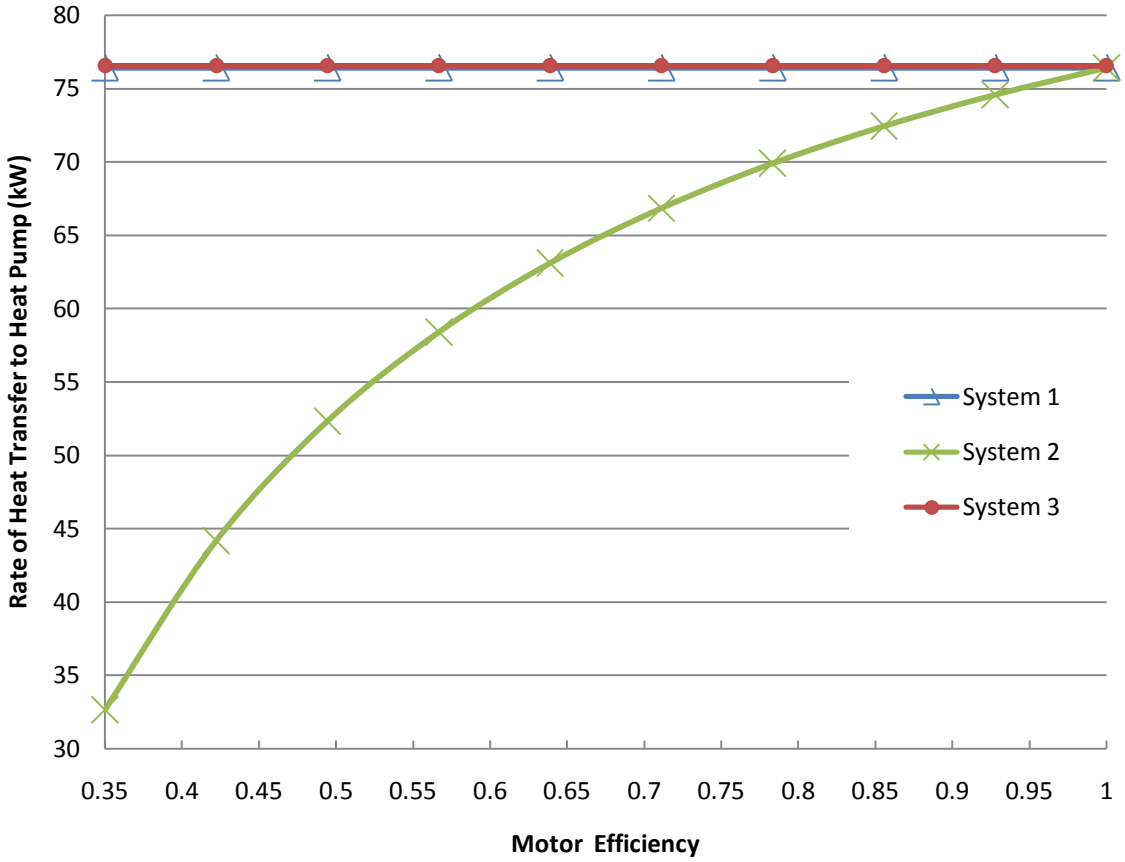

Figure 7.11: Effect of varying motor efficiency on rate of heat transfer to refrigerant through the evaporator for each system

The rate of heat transfer required from the GL system is compared for each system in Figure 7.11. The rates of heat transfer through the evaporators in systems 1 and 3 do not change with efficiency. The rate of heat transfer for system 2 is affected significantly by the motor efficiency, as motor waste heat contributes to the heat requirement of the HP system. Identical trends are found for the GL length and pump power requirement as those found for the heat requirement from the evaporator.

When the GL load is reduced the length and pumping requirements associated also decrease (Figure 7.12), allowing for system 2 to have the highest system COP at low motor efficiencies.

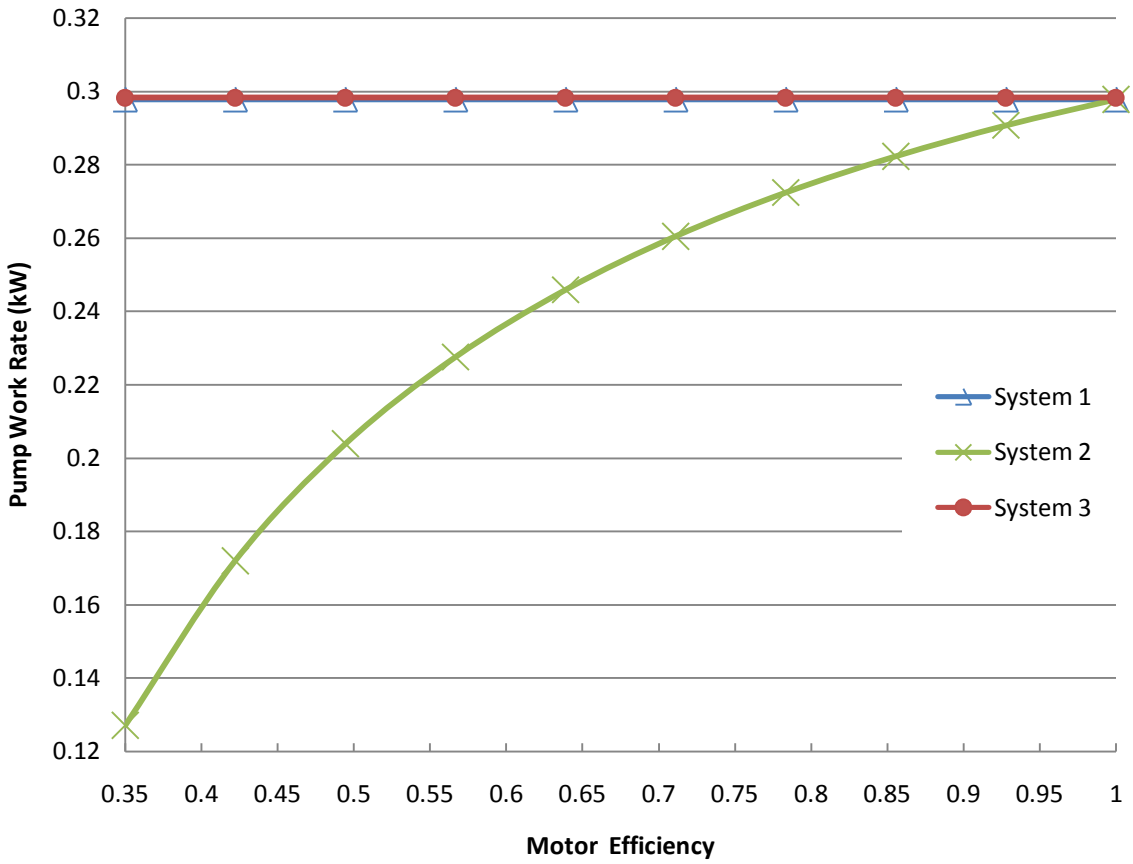

Figure 7.12: Effect of motor efficiency on pump work rate required within the ground loop for each system

Figure 7.13 illustrates the varyiation of the amount of energy consumed by the pump motor with motor efficiency. For systems 1 and 3 the trends are similar, with the amount of energy required by the pump motor decreasing with increasing motor efficiency. This trend is expected since the motor efficiency directly determines, for a given output, the amount of energy it requires, which for this model is rate of pump work. More energy is consumed by a motor with low compared to high efficiency, as illustrated in Equation 5.54.

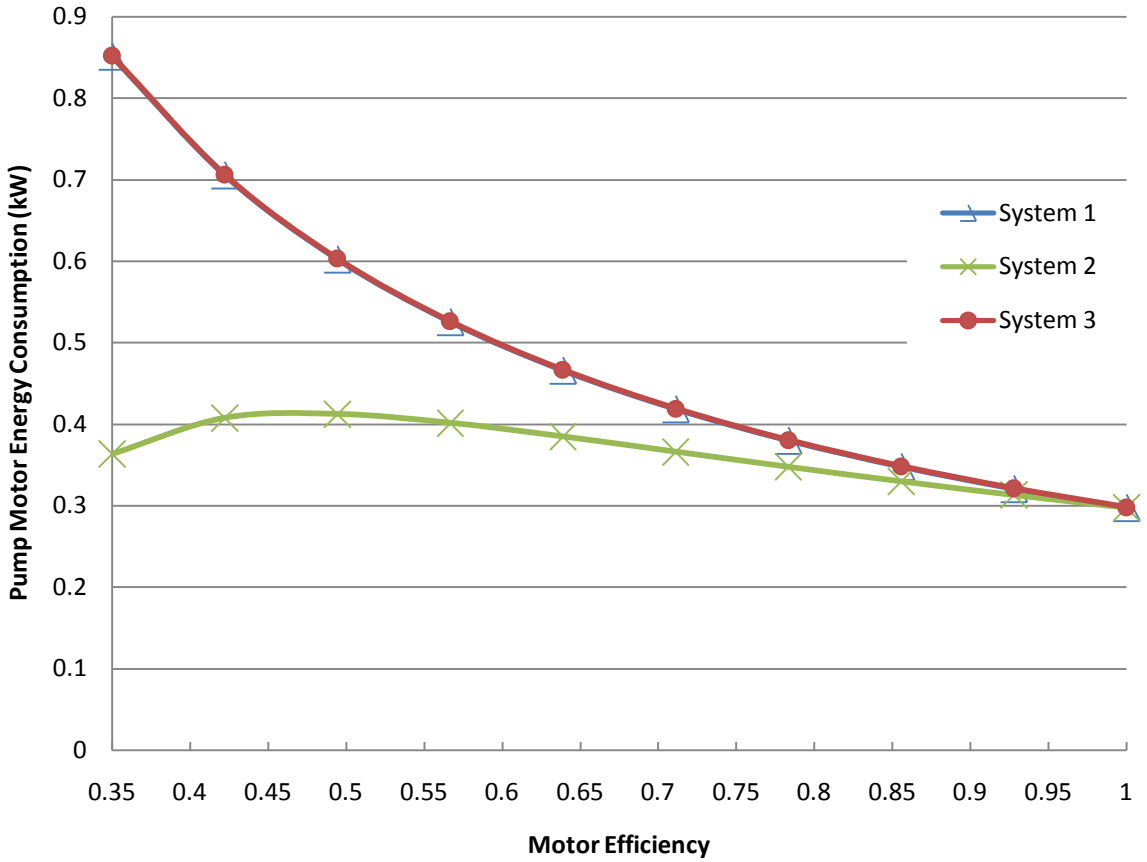

Figure 7.13: Effect of motor efficiency on pump motor work for each system

A strange trend arises for system 2 in the variation of the amount of energy required by the pump motor. Within a certain range of motor efficiencies, the amount of pump motor work increases and peaks before decreasing with an increasing efficiency. The trend is attributable to the changing rate heat transfer from the GL for system 2 in this analysis. When the heat transfer requirement changes so does the GL length, with the variation in the GL length directly affecting the power required by the pump to move the brine around the loop. The pump power also decreases when the GL length is reduced; reducing the requirement of the pump motor to supply pump shaft work. This phenomenon coupled with variation in the pump motor efficiency leads to the trend seen in Figure 7.13 for system 2.

The range of pump motor energy consumption for system 2 remains relatively constant, exhibiting a variation of about 0.11 kW over the entire range of efficiencies compared to about 0.55 kW for the other 2 systems. A relatively constant, low, pump work requirement contributes to an increase in system COP.

The range of typical motor efficiencies considered in the analysis does not take into account motor design. But motors with efficiencies approaching 100% typically employ special methods to increase the efficiency, including motor or winding cooling. Thus, the motor efficiency ranges likely differ for systems with and without motor cooling. Figure 7.14 shows a more realistic representation of the COP trends with respect to a redefined range of motor efficiencies for systems 1 and 3. Motor efficiency is varied from 35 to 85% for these systems, but the efficiency range is maintained at 35 to 100% for system 2.

When more realistic motor efficiency ranges are employed, system 2 exhibits a larger potential for a high system COP (Figure 7.14), since the system COP values extend past those of systems 1 and 3.

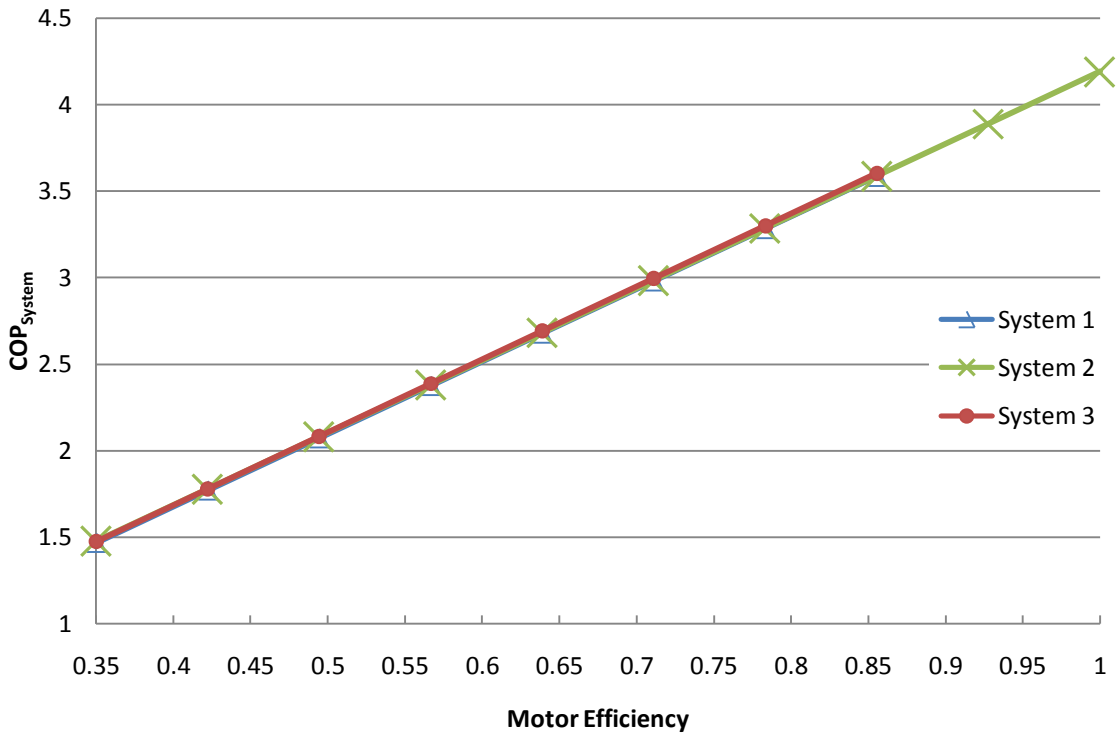

Figure 7.14: Effect of motor efficiency on system COP for each system, considering revised efficiency ranges

In general, the performances of systems 1 and 3 are affected the most by varying motor efficiency since they exhibit the largest change in system COPs. When realistic motor efficiencies are employed for systems 1 and 3, system 2 is found to have the largest range of system COPs. Motor cooling causes system COP not to be affected as drastically due to its reduced effect on the pump motor energy use. When the ground loop is considered only system 2 is affected.

7.3 Operating Conditions Analysis

The impact on the performance of the heat pump units is investigated by varying several operating conditions, including condenser and evaporator pressure as well as the degree of superheating and subcooling at the outlet of the evaporator and evaporator, respectively. System 3 utilizes an intermediate pressure between the evaporator and condenser pressure; the effect of varying this intermediate pressure is also investigated. Throughout this analysis, most of the original operating conditions and assumptions outlined in Section 3.6 are held constant while one operating condition is varied.

7.3.1 Condenser Pressure

A compressor pressure range of 400 to 3,000 kPa is utilized in the analysis of condenser pressure. This range is thought to be appropriate as it covers the range of pressures reported in literature for similar systems. Through review of literature involving working systems and studies it is found that the condenser pressure is usually below 2,500 kPa. Two such systems are illustrated by Hepbasli and Balta [19] and Bergander [41]. An upper limit is set for this analysis at 3,000 kPa to provide a complete understanding of how the system operation varies with compressor pressure.

The models used throughout the study show that the temperature at the condenser exit is dependent on the condenser pressure, where the temperature is set by assuming a degree of subcooling below the saturation temperature of refrigerant at the specified pressure. When the condenser pressure is varied the temperature at this location also changes. It is found that at low condenser pressures the temperatures at the condenser inlet and exit (states 3 and 4 within all system arrangements) become too low for use with the conventional heat distribution systems utilized in building design. Figure 7.15 shows the condenser inlet and exit temperature for pressures between 400 and 3000 kPa. The inlet and outlet condenser temperatures are low for all systems at low condenser pressures, e.g., at 400 kPa the inlet and outlet temperatures for all the systems are about 22°C and 4°C, respectively.

Of all the heat distribution arrangements, hydronic heat distribution systems typically have the lowest design temperature, usually 18-22°C. The inlet and outlet temperatures of the condenser on the building side should not exist below this temperature range [9, 17, 20, 21]. Thus the lowest temperatures at the condenser inlet and exit for the heat pump cycle should be within or above the temperature range for hydronic systems to allow for proper heat transfer across the condenser between the heat pump and the building heat distribution system.

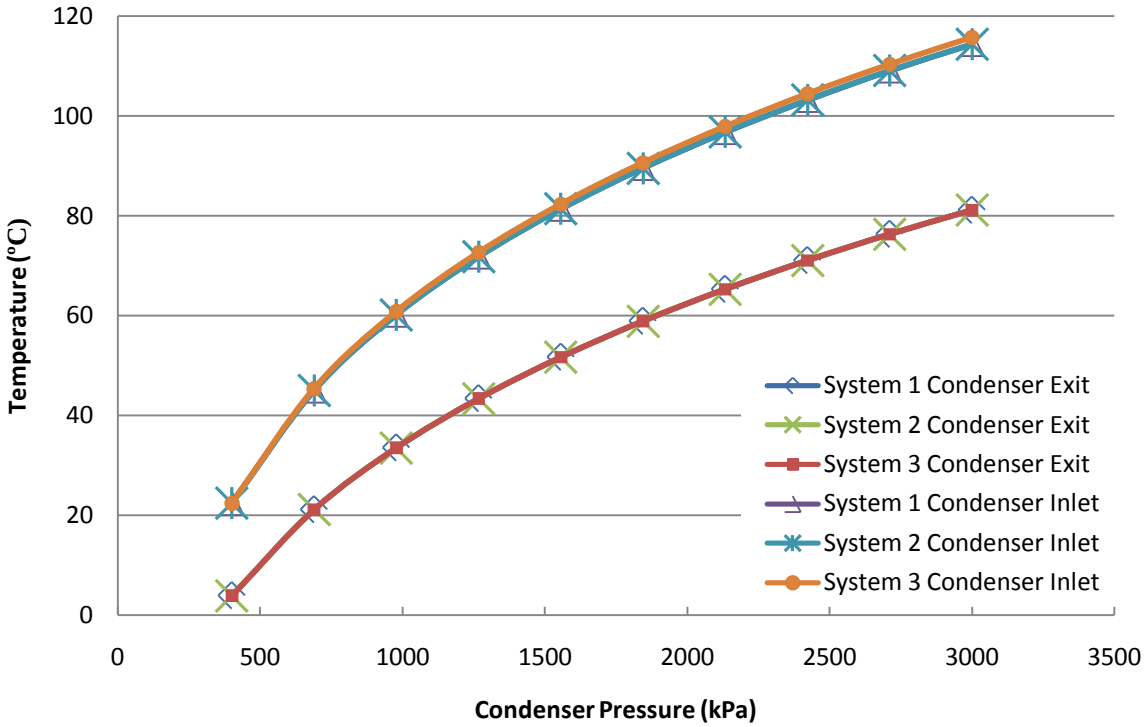

Figure 7.15: Effect of varying condenser pressure on condenser inlet and outlet temperatures for each system

For the pressure range specified above, the investigation considers pressures between 650 kPa and 3000 kPa, thereby allowing appropriate temperatures across the condenser. All other conditions and assumptions specified in Section 3.2 are held constant. For the new range of condenser pressures, the HP and system COPs for all the systems decrease with increasing condenser pressure. All systems follow similar trends as pressure increases;

with a sudden reduction in COP initially followed by a gradual decrease in COP (see Figures 7.16 and 7.17). This trend is thought to be attributable to the compressor power over the range of condenser pressures. Figure 7.18 shows how compressor power varies with condenser pressure. The increase in compressor power as pressure increases is notable at low condenser pressures and levels off at higher pressures. The change in compressor power arises from an increasing difference between the evaporator and condenser pressure, resulting in an increase in specific compressor work. Also increasing condenser pressure raises the temperature at the inlet and exit of condenser. The condenser inlet and exit temperatures vary is different fashions with changing condenser pressure (Figure 7.15). The condenser inlet temperature increases more than the condenser exit temperature over the range of condenser pressures, which creates an increasing difference between these temperatures as the pressure increases. This in turn reduces the refrigerant flow rate through the condenser. These two factors together lead to the trend observed for compressor work, as well as for HP and system COP.

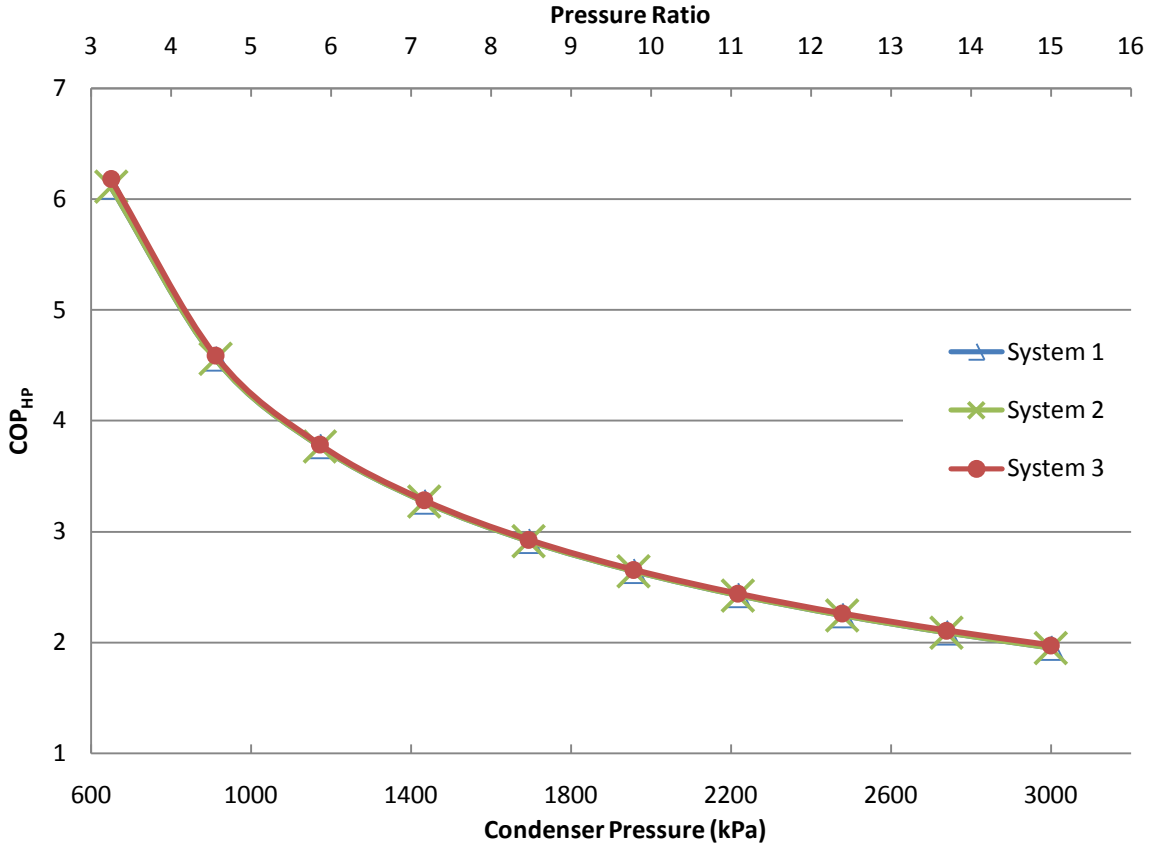

Figure 7.16: Effect of varying condenser pressure on heat pump COP for each system

It can be seen in Figures 7.16 and 7.17 that the COP decreases with increasing pressure ratio, and that system 3 exhibits a slightly higher sensitivity to variation in condenser pressure than the other two systems. As in previous analyses, the HP COPs for systems 1 and 2 are identical. The HP COPs are summarized in Table 7-9.

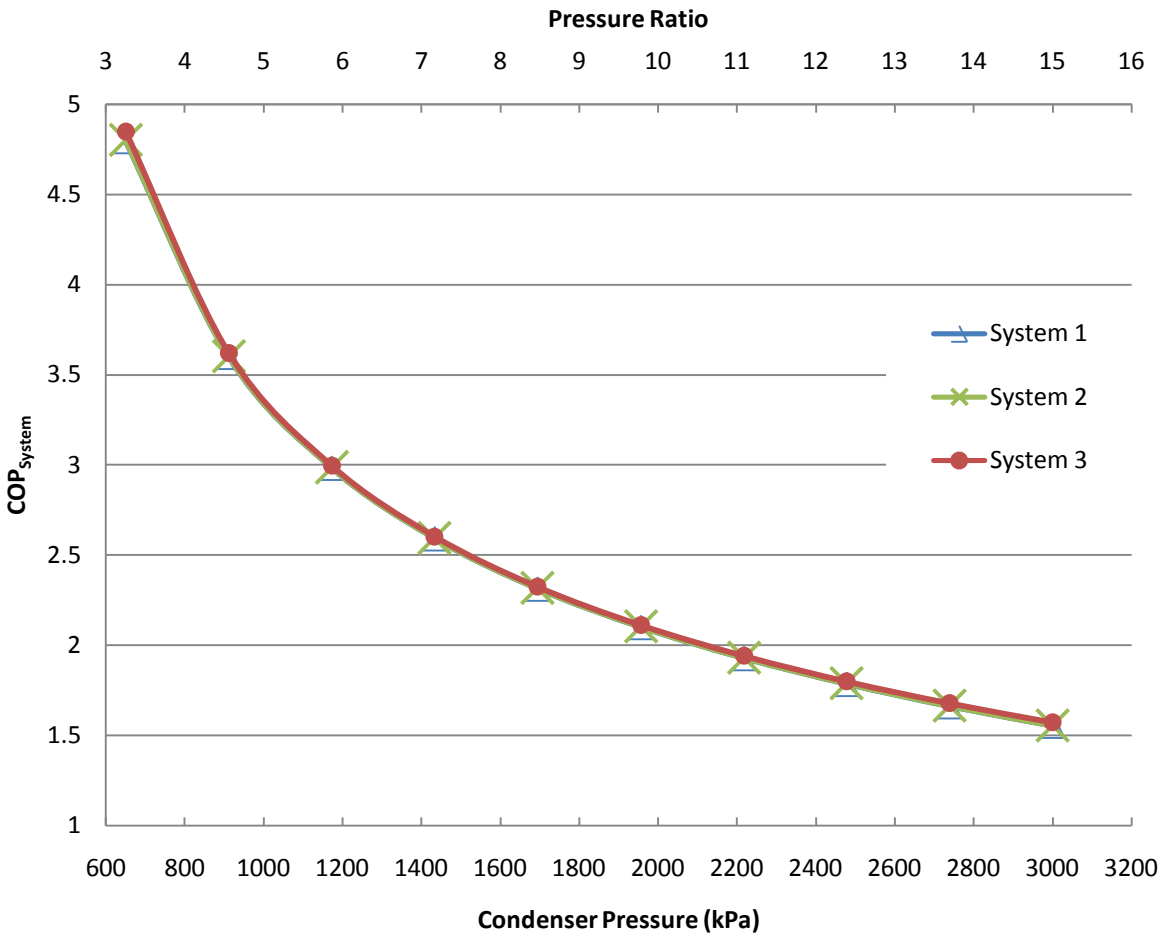

Figure 7.17: Effect of varying condenser pressure on system COP for each system

Table 7-9: Heat pump COPs for a range of condenser pressures

		COP_{HP}	
Condenser pressure (kPa)	**Pressure ratio**	**System 1***	**System 3**
650	3.25	6.113	6.182
911	4.556	4.561	4.589
1172	5.861	3.77	3.786
1433	7.167	3.269	3.282
1694	8.472	2.914	2.927
1956	9.778	2.643	2.656
2217	11.08	2.425	2.44
2478	12.39	2.243	2.26
2739	13.69	2.087	2.107
3000	15	1.949	1.973
Range**		4.18	4.21

* Systems 1 and 2 have identical calculated COP_{HP} values
**Range is the difference in calculated COP_{HP} values coinciding with the highest and lowest condenser pressure

It can be seen in Table 7-9 that systems 1 and 2 have a slightly smaller range of COP than system 3. This effect is caused by the difference in compressor power as seen in Figure 7.18.

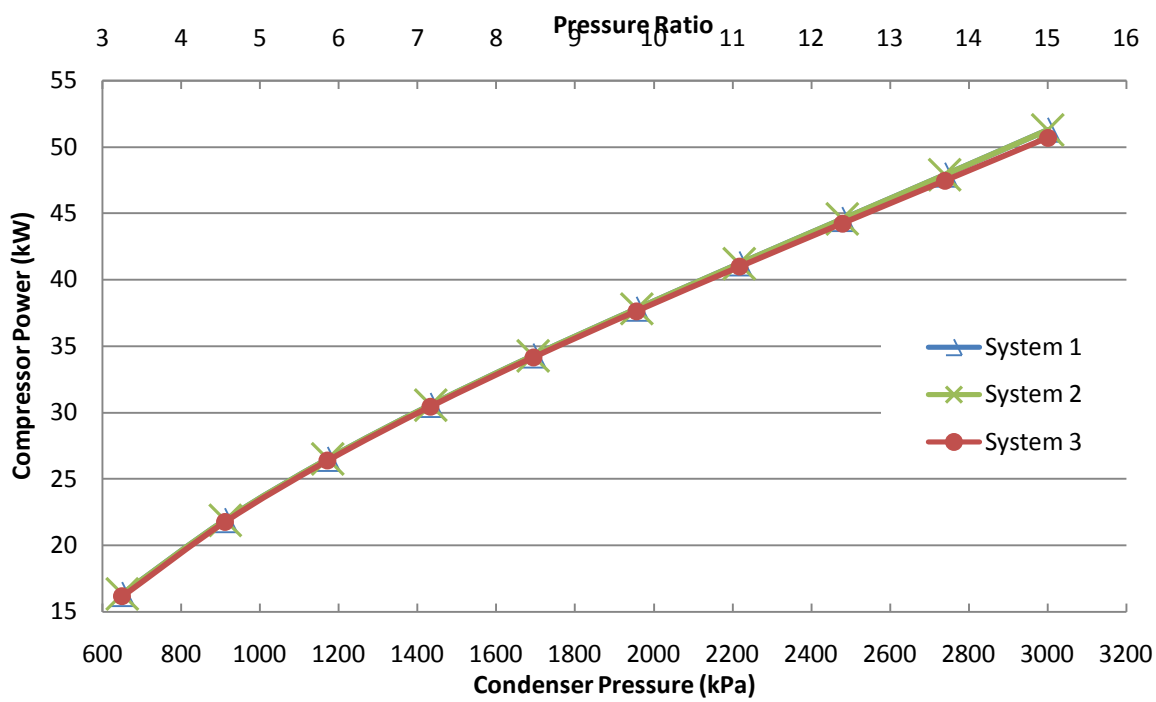

Figure 7.18: Effect of varying condenser pressure on compressor work for each system

It is found that the GL length decreases with increasing condenser pressure (Figure 7.19). This trend is a result of the decrease in calculated HP COP values. When the HP COP decreases, more energy is supplied to the HP cycles through the compression process for the same heating load. For the energy flows in and out of the HP systems to match the rate of heat transfer from the GL is reduced as condenser pressure is increased as seen in Figure 7.20. System 2 is affected the most by changes in condenser pressure. An increase in compressor power creates an associated rise in the rate of compressor motor energy consumption, which is directly associated with an increase of available waste energy from the compressor motor for preheating the refrigerant in system 2. The motor waste heat is transferred to the refrigerant, reducing the heat transfer rate between the GL and the HP, when compared to systems 1 and 3, as the condenser pressure increases.

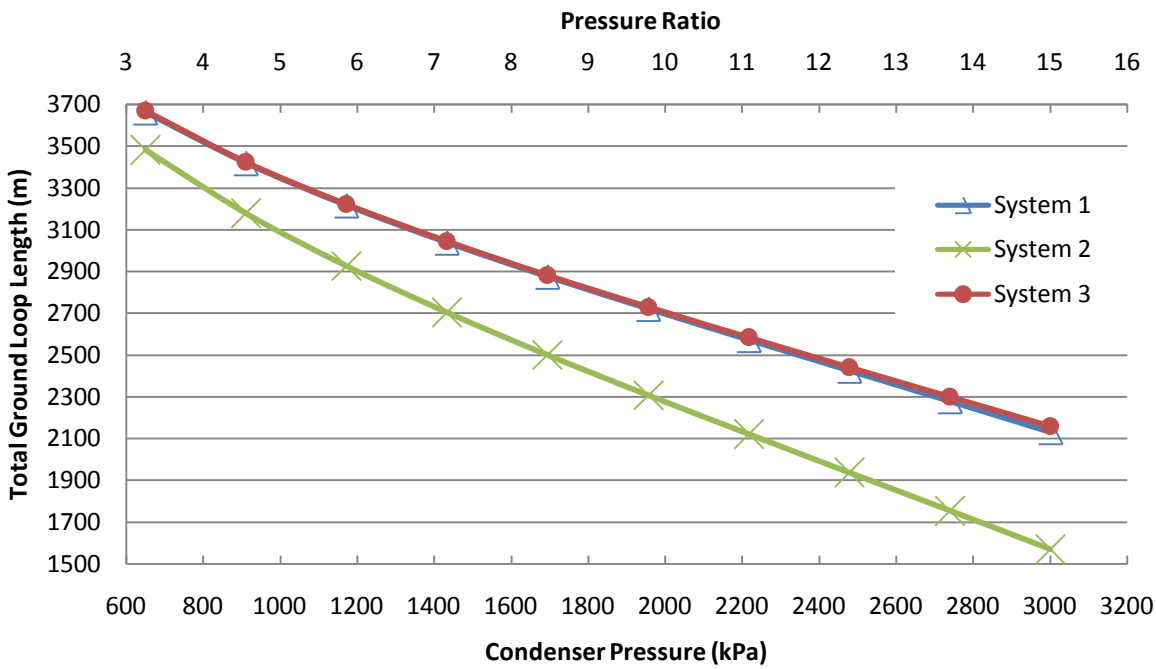

Figure 7.19: Effect of varying condenser pressure on ground loop length for each system

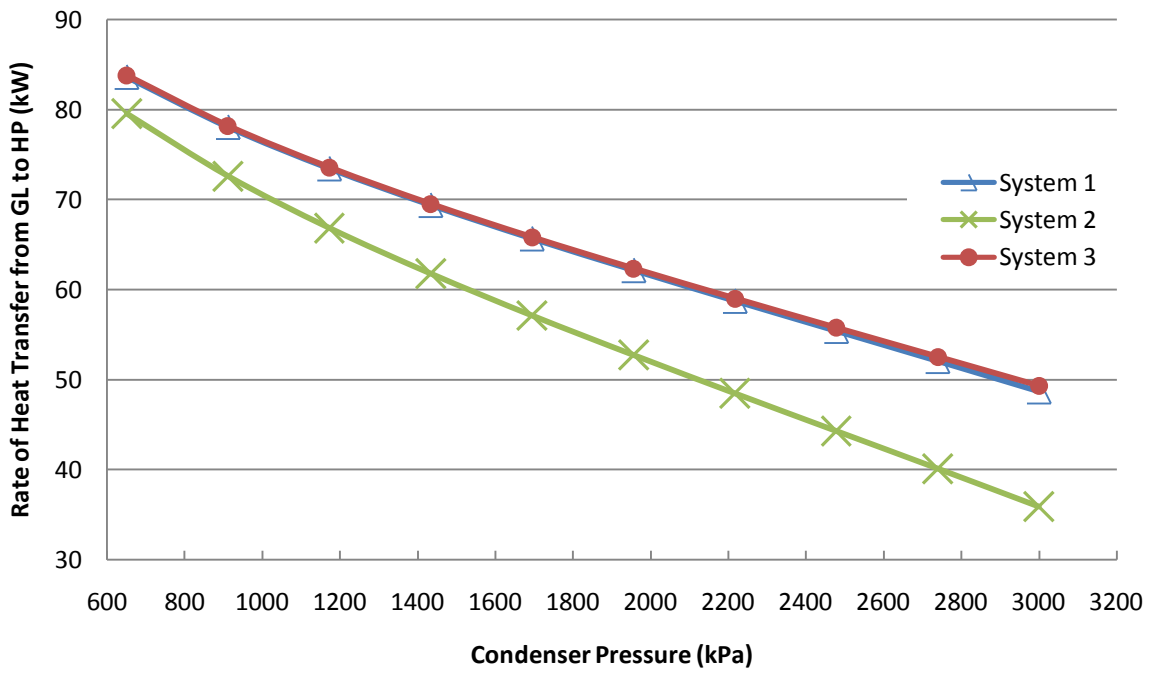

Figure 7.20: Effect of varying condenser pressure on the heat transfer rate from the GL for each system

7.3.2 Evaporator Pressure

The effect of varying evaporator pressure on the systems is considered. The minimum evaporator pressure is set so that the refrigerant temperature at the evaporator inlet is above the freezing temperature of the brine solution in the ground loop (-13.08ºC). The maximum pressure is set so that the refrigerant temperature at the evaporator exit does not exceed the brine temperature at the evaporator inlet and so that the refrigerant evaporator inlet temperature is below that of the brine evaporator exit temperature, to allow for proper heat transfer conditions. An evaporator pressure range of 178 to 305 kPa satisfies theses conditions. All other conditions and assumptions specified in Section 3.2 are held constant.

Figures 7.21 and 7.22 illustrate the variations in HP and system COPs with evaporator pressure. As the evaporator pressure increases it is seen that the HP and system COP increases for all systems. The trends are similar to the ones observed in the condenser pressure analysis. As the pressure ratio between the condenser and evaporator increases, the HP and system COPs for the systems decrease. The difference in HP COP for systems

1 and 3 increases as the evaporator pressure is lowered (as pressure ratio increases), a direct result of the alteration in the fraction of total refrigerant that is fed through the main circuit and, in turn, the evaporator. When the evaporator pressure is increased the specific enthalpy at the exit of compressor one (state 11) decreases. Equation 5.25 indicates that as the specific enthalpy at this point increases the proportion of the flow encountered by the main circuit decreases. The specific enthalpy at the exit of compressor two is higher for system 3 than the other two systems, reducing the refrigerant flow rate through the condenser. When the variation of the fraction of flow rate associated with the main circuit and the change in flow rate are combined, the trend observed in Figures 7.21 and 7.22 results.

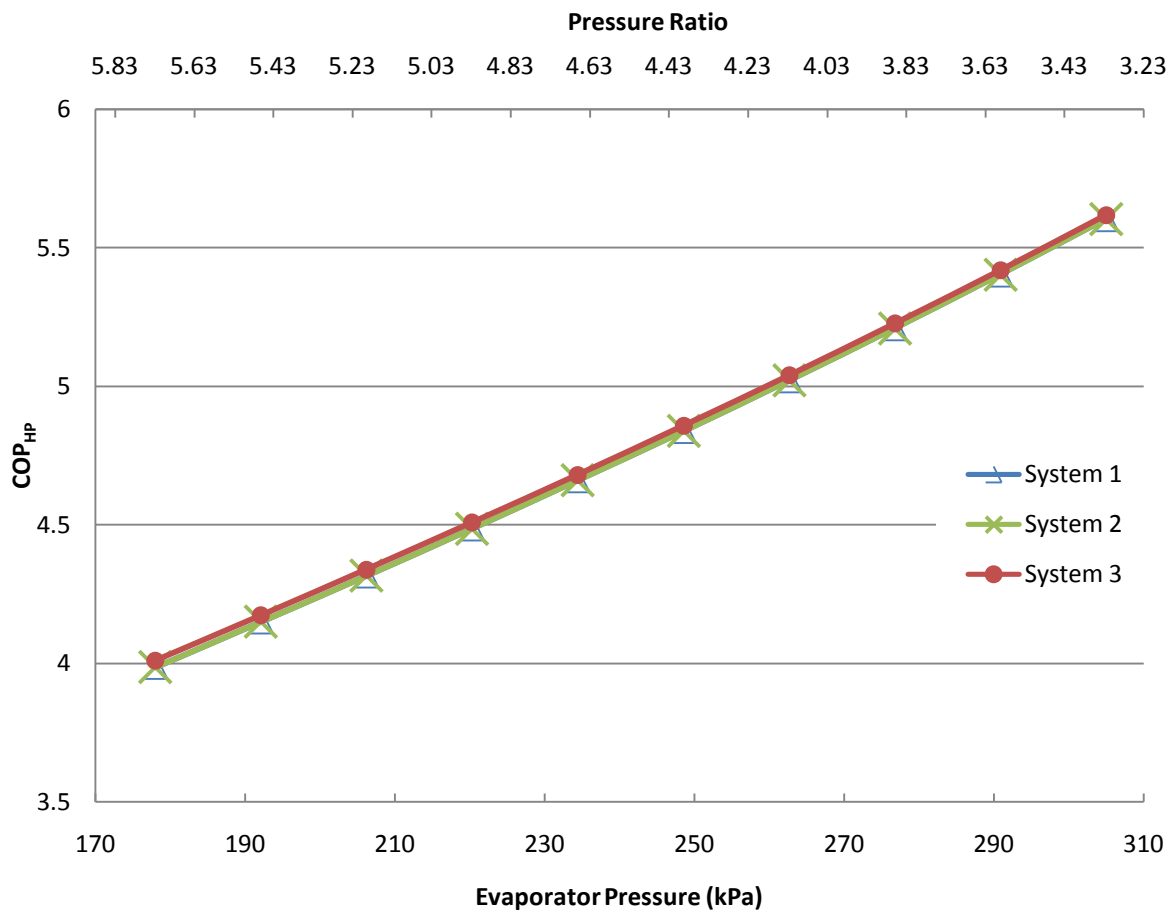

Figure 7.21: Effect of varying evaporator pressure on heat pump COP for each system

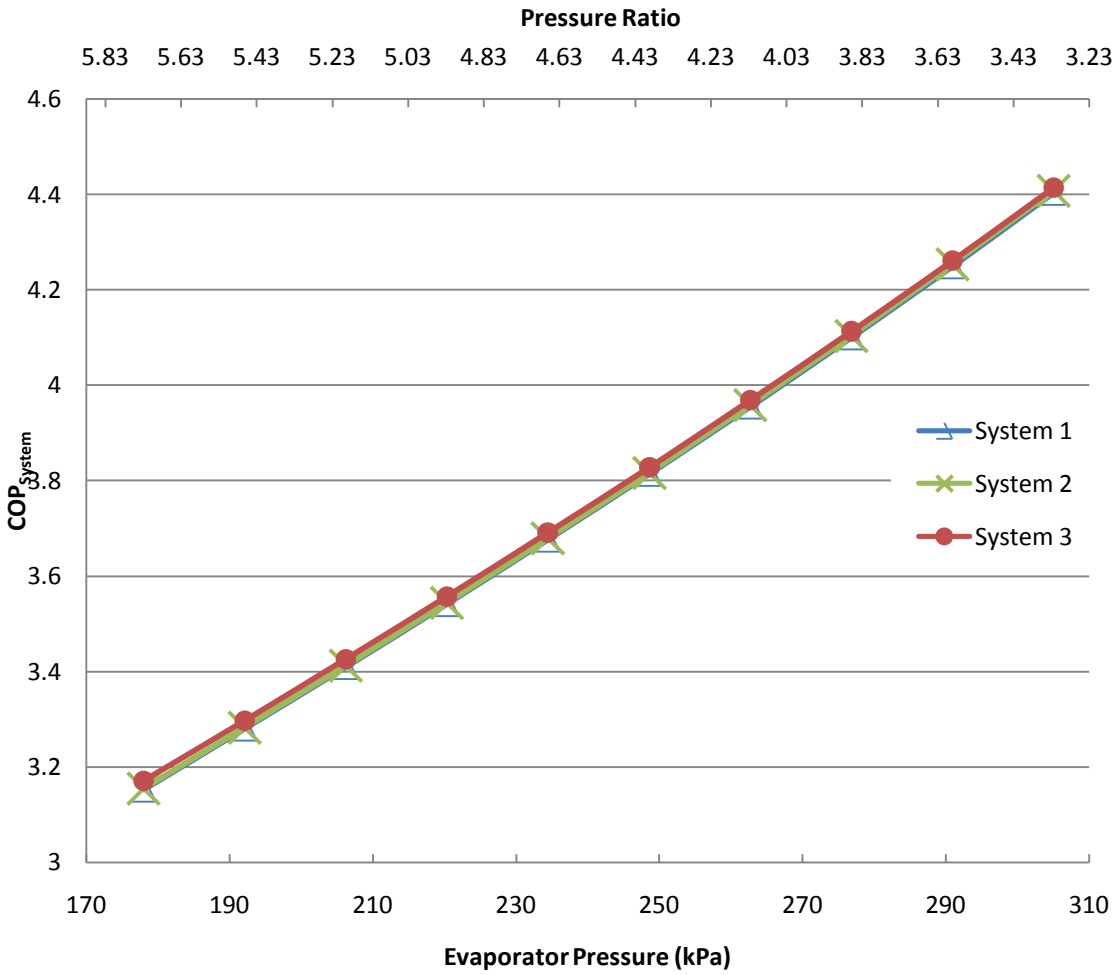

Figure 7.22: Effect of varying evaporator pressure on system COP for each system

It can be seen in Figures 7.21 and 7.22 that COP decreases with decreasing evaporator pressure and increasing pressure ratio, similar to the result observed the condenser pressure analysis (Section 7.3.1). It is apparent that systems 1 and 2 are slightly more sensitive to variations in condenser pressure than system 3. As in the previous analyses, the HP COP for systems 1 and 2 are identical. The HP COPs are listed in Table 7-10.

Table 7-10: Heat pump COPs for various condenser pressures

		COP_{HP}	
Evaporator pressure (kPa)	**Pressure ratio**	**System 1***	**System 3**
178	5.618	3.986	4.01
192.1	5.205	4.15	4.173
206.2	4.849	4.316	4.339
220.3	4.539	4.486	4.508
234.4	4.265	4.66	4.68
248.6	4.023	4.838	4.858
262.7	3.807	5.021	5.039
276.8	3.613	5.209	5.226
290.9	3.438	5.403	5.418
305	3.279	5.603	5.617
Range**		1.62	1.61

* Systems 1 and 2 have identical calculated COP_{HP} values
**Range is the difference in calculated COP_{HP} values coinciding to the highest and lowest evaporator pressure

Table 7-10 illustrates that system 1 and 2 have a larger HP COP range than system 3, but the difference in HP COP ranges between systems 1 and 3 is small (0.01 or 0.62%). So in general systems 1, 2, and 3 have almost the same response to changes in evaporator pressure over the entire range of pressures considered.

The GL length is found to increase with increasing evaporator pressure (see Figure 7.23). As in the analysis of condenser pressure, the ground loop length decreases with increasing pressure ratio between the condenser and evaporator. This trend is a result of HP COP increasing with increasing pressure. When the HP COP of a system increases, less energy is supplied to the HP cycles as compression work for the same heating load. For the energy flows in and out of the HP systems to be equal, the thermal energy extracted from the GL increases as evaporator pressure is increased, as seen in Figure 7.24.

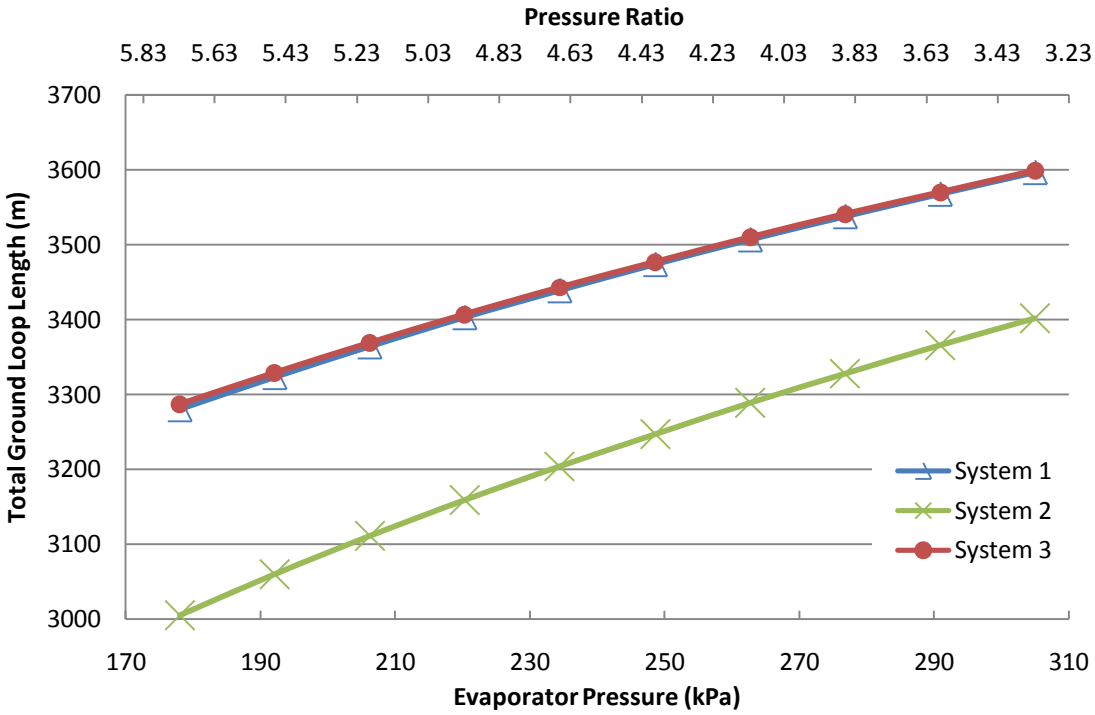

Figure 7.23: Effect of varying evaporator pressure on total GL length for each system

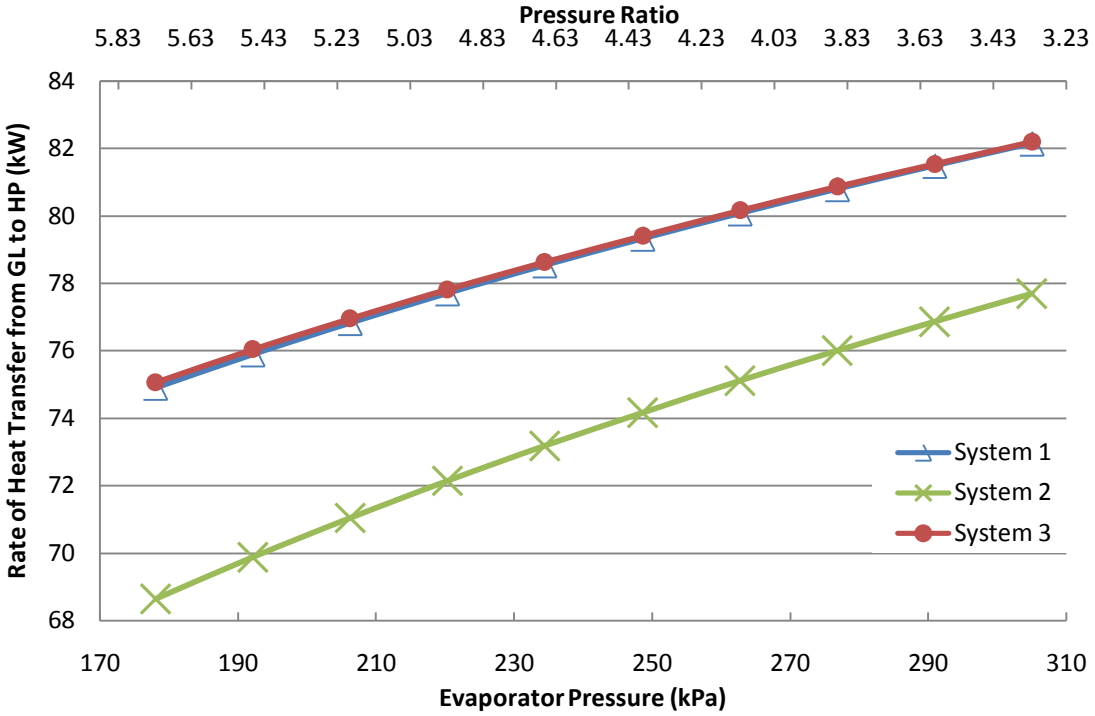

Figure 7.24: Effect of varying evaporator pressure on the heat transfer rate from the GL for each system

7.3.3 Effect of Changing Condenser and Evaporator Pressures for Similar Pressure Ratios

To determine whether the trends developed in the condenser and evaporator pressure analyses are the result of the pressures across the devices or a result of the pressure ratio, the HP COPs from each analysis are compared for a common range of pressure ratios. Data utilized from the condenser pressure analysis (Section 7.3.1) is only over the range of pressure ratios specified in the evaporator analysis (Section 7.3.2). The pressure ratios coincide with varying evaporator pressure and a fixed condenser pressure of 1000 kPa and the pressure ratios corresponding to varying condenser pressure with a fixed evaporator pressure of 200 kPa. The trends with changing pressure ratios, due to changing either condenser or evaporator pressure, are compared in Figures 7.25 and 7.26 for all three systems. As in the previous investigations, the HP COPs for systems 1 and 2 are identical.

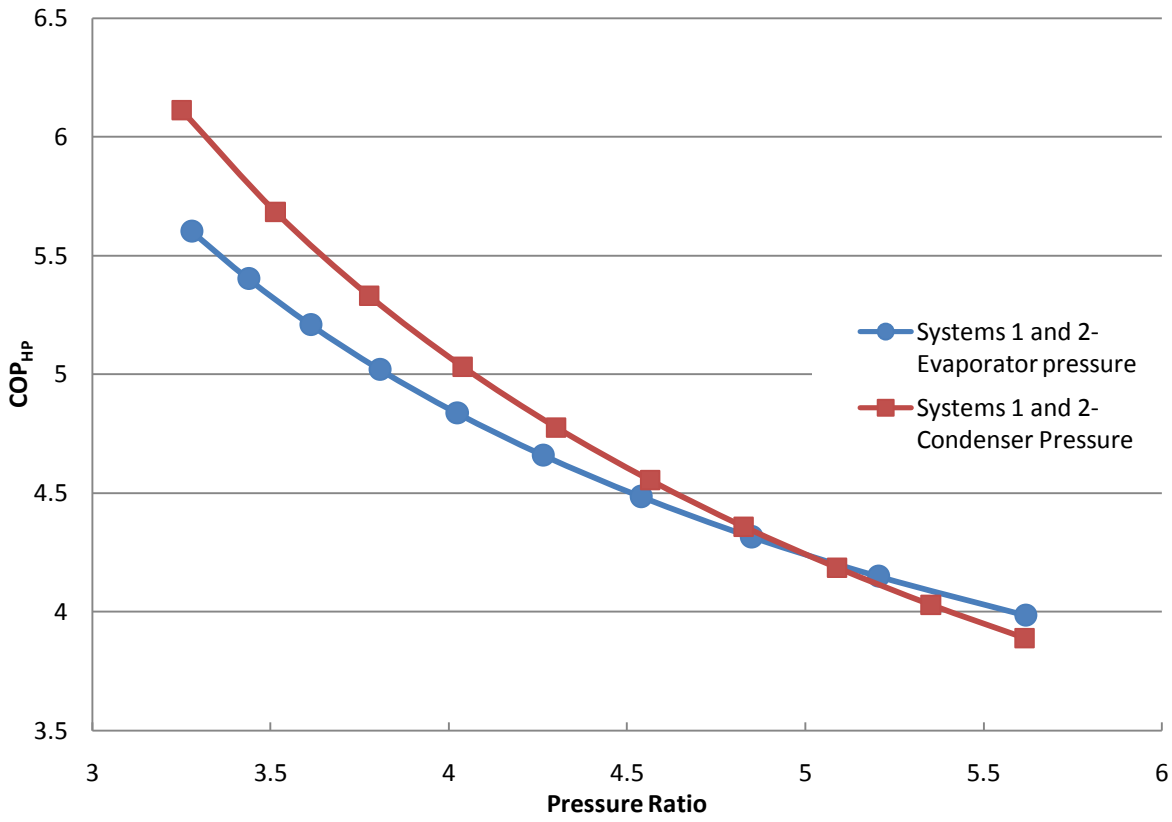

Figure 7.25: Variation of heat pump COPs with condenser and evaporator pressure ratio, for systems 1 and 2

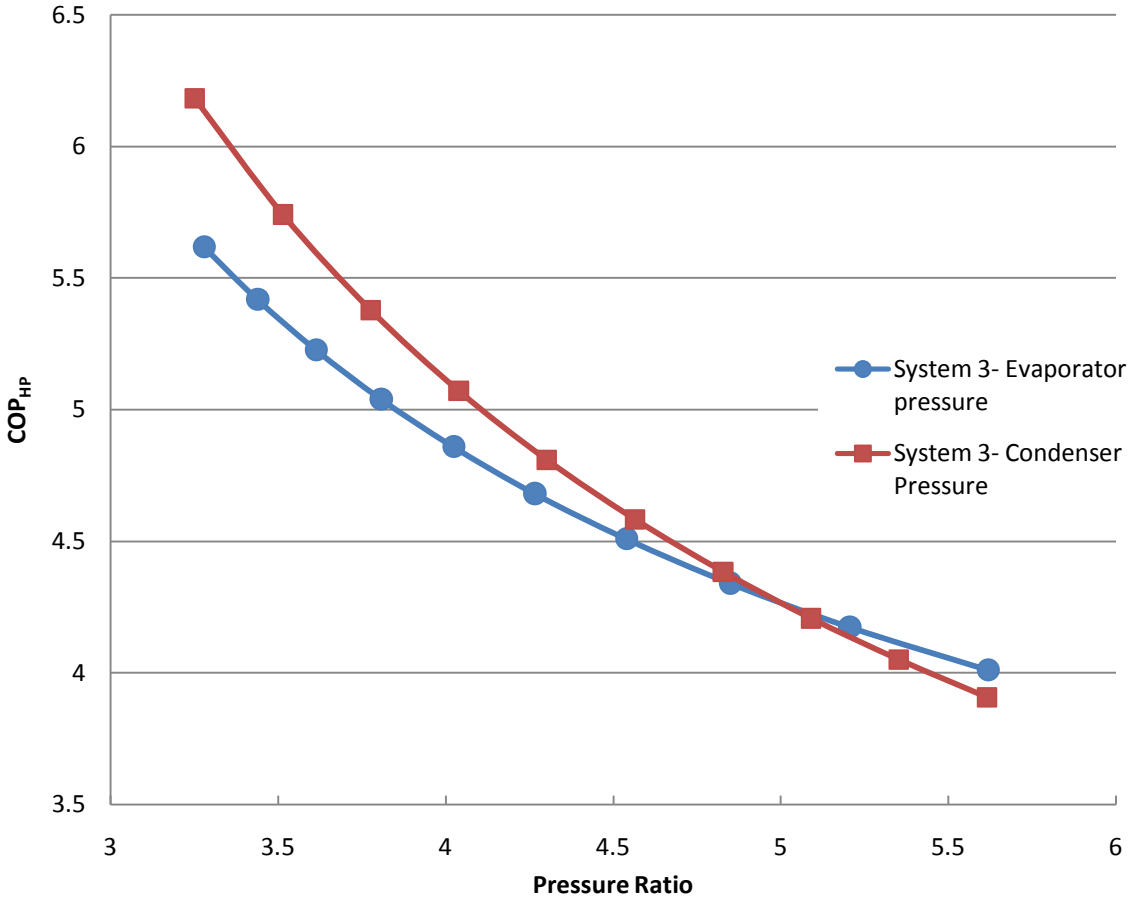

Figure 7.26: Variation of heat pump COPs with condenser and evaporator pressure ratio, for system 3

The HP COPs of these systems are independent of the pressure ratio between the condenser and evaporator. Rather the COPs are more dependent on the actual condenser and evaporator pressures for the systems. Hence, the pressure ratios could be the same for many combinations of condenser and evaporator pressures, but the COP values for each situation will not necessarily be identical. This finding is illustrated in Figures 7.25 and 7.26 by showing the variations in the COPs for common pressure ratios.

For the pressure ratios considered, the HP COPs are found to change the most when condenser pressure rather than evaporator pressure is varied, suggesting that the COPs of

these heat pump arrangements are more dependent on condenser than evaporator pressure.

In both comparisons the two series intersect at the point where the condenser and evaporator pressures are the same.

7.3.4 Intermediate Pressure in System 3

The effect of varying the intermediate pressure in system 3 in terms of performance and design parameters is explored. The analysis is not applicable to systems 1 and 2. In previous analyses the intermediate pressure of the supplementary circuit is fixed. Here, the intermediate pressure is varied between the condenser and evaporator pressure given in the basic analysis. To ensure proper heat transfer for the arrangement of the economizer, the temperature at state 9 must be below that of state 4. Similarly, the temperature at state 8 must be below that of state 10. The temperature at state 9 is equal to the temperature at the exit of compressor one, so the maximum pressure is 535.2 kPa and the conditions for states 8 and 10 are also satisfied at this pressure. The intermediate pressure is therefore varied from 200 kPa to 535.2 kPa.

The HP COPs increase as the intermediate pressure increases (see Figure 7.27). Over the range of intermediate pressures there is a difference of about 1.3% in COP. The trend is attributable to a reduction in compressor power. When the intermediate pressure is increased the flow rate through the main circuit increases, and the flow rate through the supplementary circuit decreases. As the pressure changes, the specific enthalpy at the exit of compressor one increases, causing the fraction of total flow rate in the main circuit to change. Simultaneously the specific enthalpy at the exit of compressor two decreases with increasing intermediate pressure. The combination of these effects leads to the reduction in required rate of compressor work with rising intermediate pressure, causing the HP COP to increase. The trend of rising HP COP with increasing intermediate pressure observed is consistent with the trends found by Ma and Chai [9] for this system arrangement.

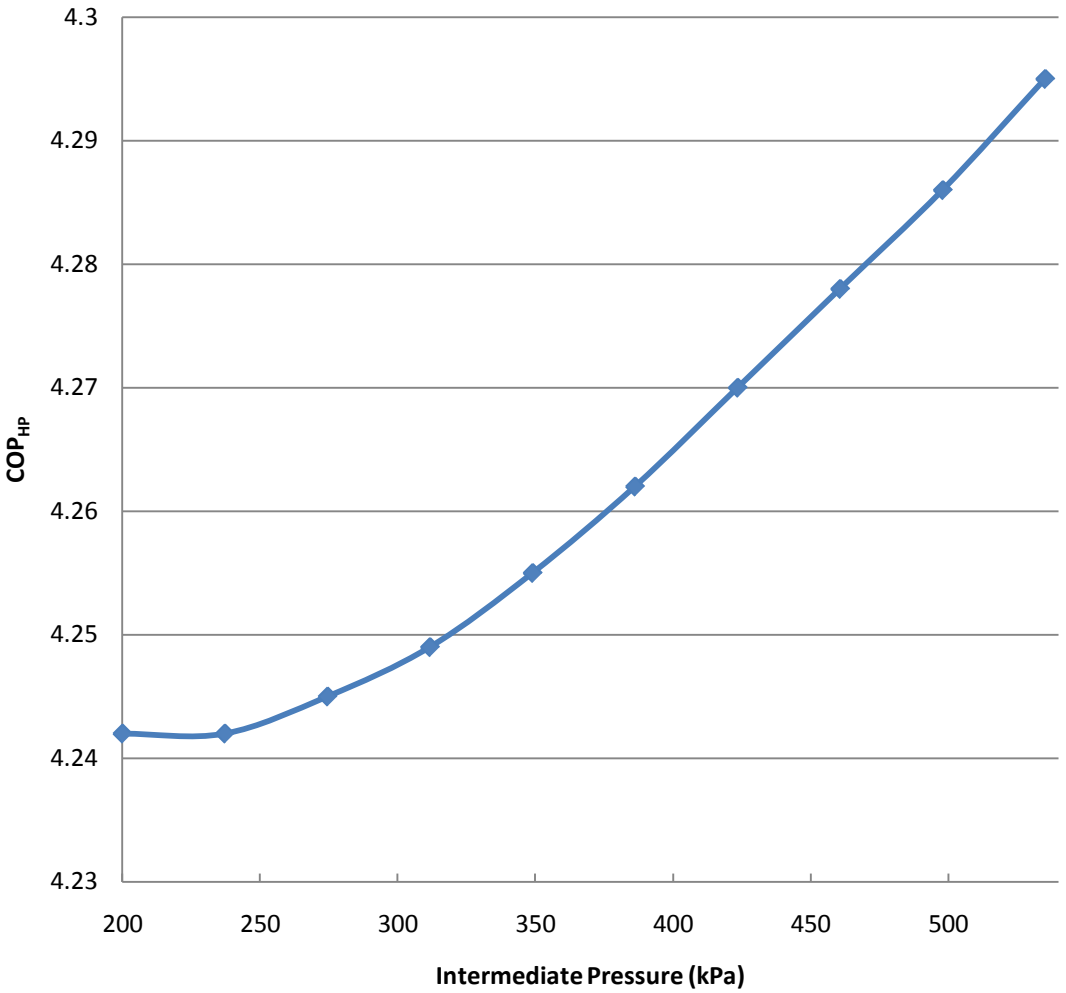

Figure 7.27: Effect of varying intermediate pressure on heat pump COP for system 3

If an increased intermediate pressure was employed in the basic comparison of the systems (Section 7.1) the difference in COPs between system 3 and the other two systems would be greater, with the COP of system 3 becoming increasingly higher with a raised intermediate pressure. It is felt that if all the present analyses were repeated with a higher intermediate pressure, the results would show a greater variation between the performance of system 3 and the other two systems.

7.3.5 Subcooling at Condenser Outlet

In previous analyses the degree of subcooling below saturation at the condenser exit has been fixed. The effect is investigated here for the degree of subcooling at the condenser on the performance and operation of the three systems considered. As mentioned in the investigation of condenser pressure (Section 7.3.1), the temperature at the condenser exit must be limited to a reasonable minimum to allow proper heat transfer to the heat distribution system. To ensure this condition is satisfied, the degree of subcooling is allowed to vary from 0 to 20°C for a fixed condenser pressure of 1000 kPa. All other conditions and assumptions specified in Section 3.2 are maintained.

Both the HP and system COPs are observed to increase almost linearly with increasing degree of subcooling (Figures 7.28 and 7.29). The main cause of this trend is the reduction in mass flow rate through the condenser in all the systems. When the temperature at the condenser exit is increased, the mass flow rate reduces according to Equation 5.2. The variation of the refrigerant flow rate through the condenser with degree of subcooling is nearly linear as illustrated in Figure 7.30.

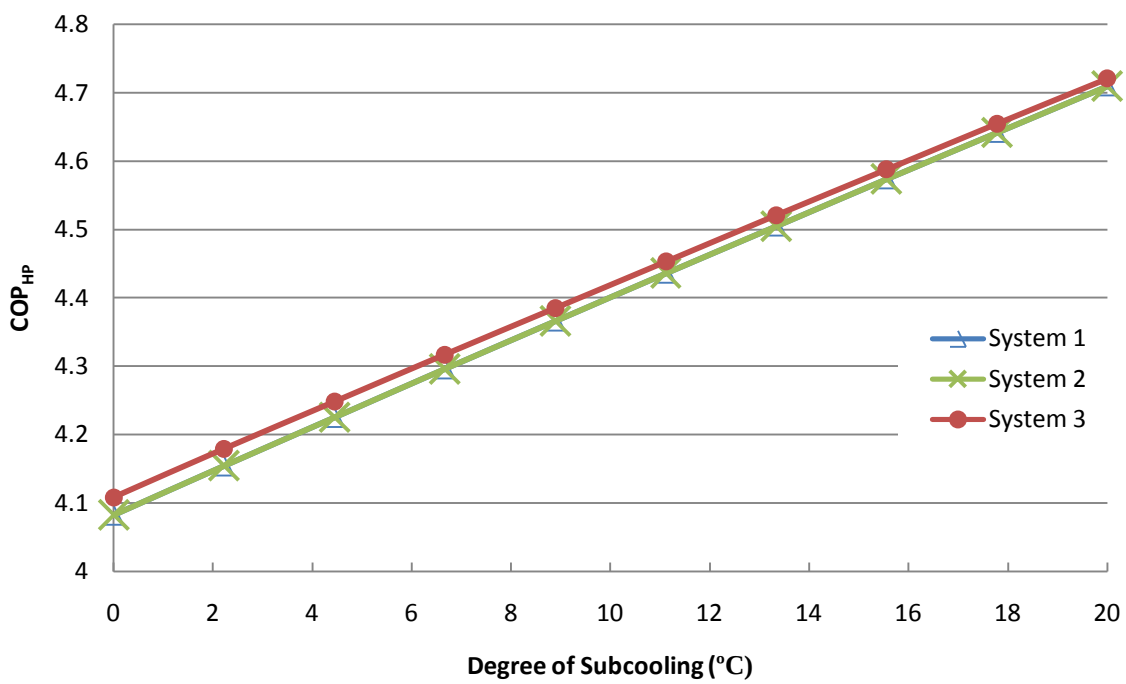

Figure 7.28: Effect of varying degree of subcooling at condenser exit on heat pump COP for each system

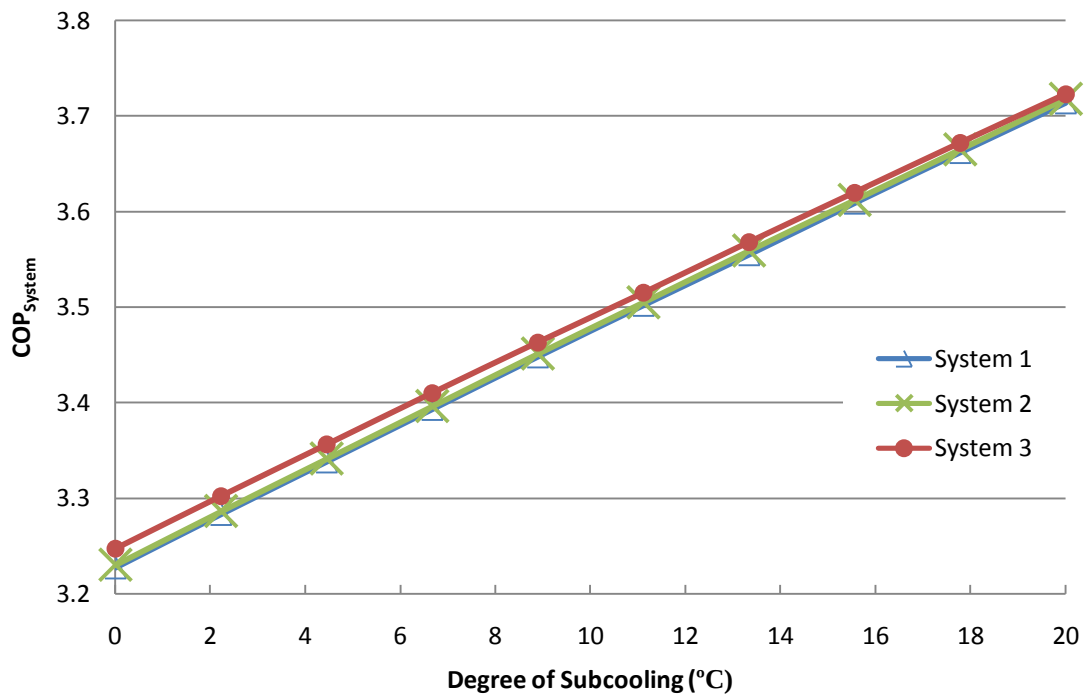

Figure 7.29: Effect of varying degree of subcooling at condenser exit on system COP for each system

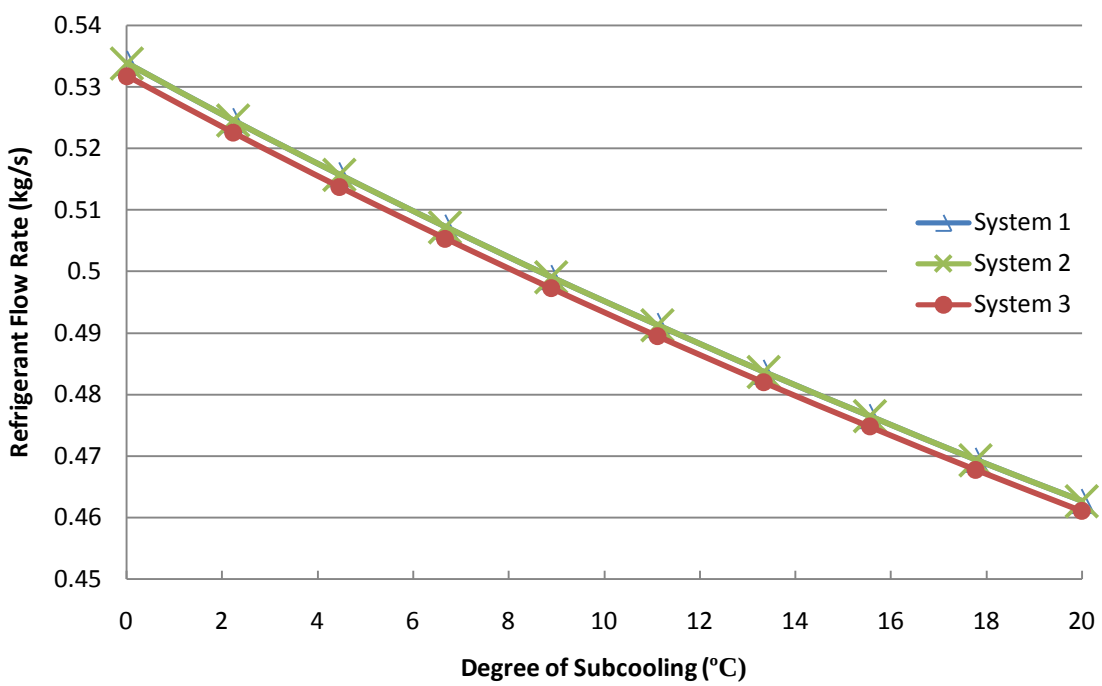

Figure 7.30: Effect of varying degree of subcooling at condenser exit on refrigerant flow rate for each system

Systems 1 and 2 are observed to be the most sensitive to subcooling. The range of COPs for system 3 is the narrowest. For a variation in subcooling of 20ºC, the HP COP changes by 0.63 for systems 1 and 2 and by 0.61 for system 3. On a normalized basis, the HP COP changes by about 0.031 and 0.030 per degree of subcooling for systems 1 and 3, respectively.

The system COP for system 2 exhibits the greatest sensitivity to variations in the amount of subcooling at the condenser exit, but the difference between systems 1 and 2 is small. The HP and system COPs are listed in Table 7-11.

Table 7-11: Comparison of effect of subcooling on heat pump and system COP

		COP_{HP}			COP_{System}		
Degree of subcooling (ºC)	**Temperature at State 4 (ºC)**	**System 1**	**System 2**	**System 3**	**System 1**	**System 2**	**System 3**
0	39.36	4.082	4.082	4.108	3.227	3.23	3.247
2.231	37.14	4.154	4.154	4.179	3.283	3.286	3.302
4.452	34.92	4.225	4.225	4.248	3.338	3.341	3.356
6.673	32.69	4.296	4.296	4.317	3.393	3.396	3.41
8.894	30.47	4.366	4.366	4.385	3.448	3.451	3.463
11.12	28.25	4.436	4.436	4.453	3.502	3.505	3.515
13.34	26.03	4.505	4.505	4.521	3.555	3.559	3.568
15.56	23.81	4.574	4.574	4.588	3.609	3.612	3.62
17.78	21.59	4.642	4.642	4.655	3.662	3.665	3.672
20	19.37	4.71	4.71	4.721	3.714	3.718	3.723
Range*		0.63	0.63	0.61	0.487	0.488	0.476

*Range is the difference in calculated COP values coinciding to the highest and lowest degree of subcooling

System 3 is the least effected by subcooling because of its supplementary circuit. In system 3 the degree of subcooling also affects the conditions at the economizer exit on the main circuit, which in turn changes the refrigerant flow rate through the supplementary loop, through equation 5.25. From Figure 7.31 it can be seen that as subcooling increases the refrigerant flow rate through the main circuit also increases. As

the degree of subcooling increases, the fraction of flow rate passing through the main circuit increases and through the supplementary circuit decreases. When the flow rate through the supplementary loop increases the COP of this system also increases. Therefore, if the degree of subcooling is increased sufficiently the COP of system 3 approaches the COP of system 1.

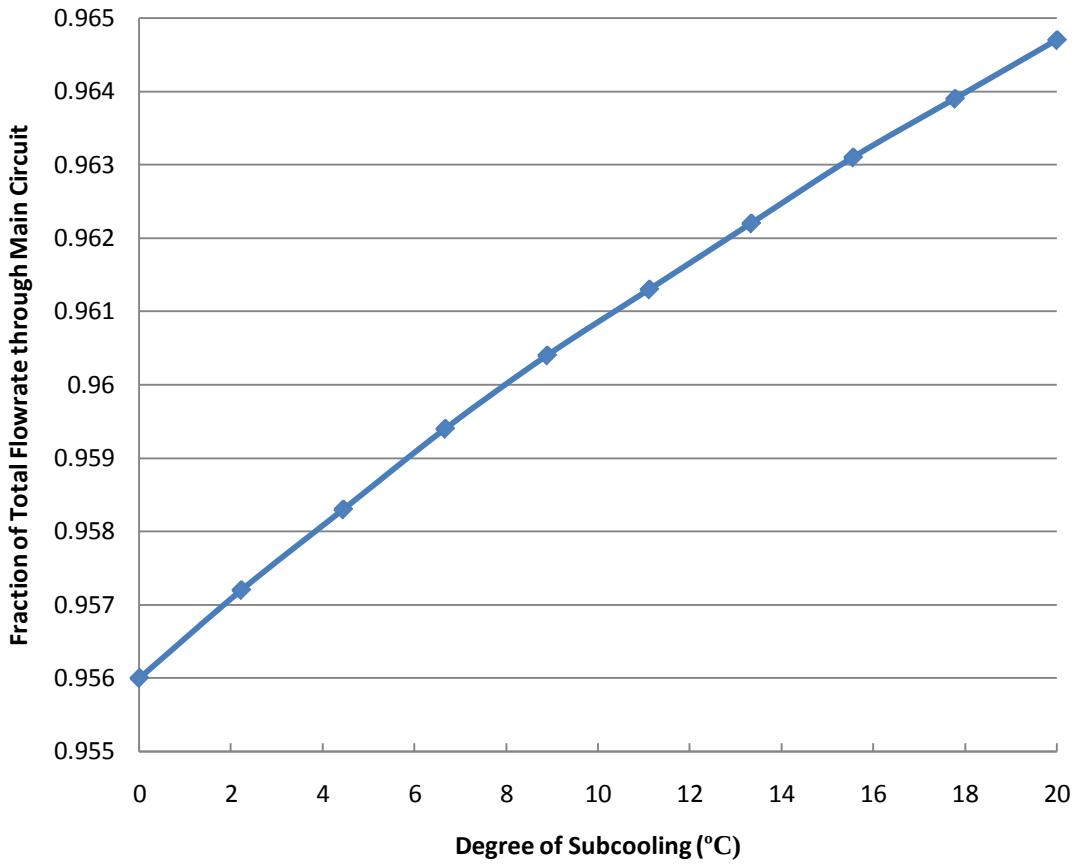

Figure 7.31: Effect of varying degree of subcooling at condenser exit on the amount of flow utilized in the main circuit of system 3

7.3.6 Superheating at Evaporator Outlet

The degree of superheating at the evaporator exit is assumed fixed in the earlier analyses. Here, the effect is investigated of varying the degree of superheating on the performance

and operation of the 3 HP arrangements. The range of superheating is limited to ensure the temperature of the refrigerant leaving the evaporator is below that of the warm brine entering the evaporator, to allow for proper heat transfer. Thus, the degree of superheating employed is between 0 and 17°C.

Like the trends developed in the subcooling assessment, the HP and system COPs increase with increasing degree of superheating, as seen in Figures 7.32 and 7.33. The main difference corresponds to the range of COP values, which are relatively narrow, e.g., for a variation of 17°C, the HP COP changes by 0.036 for systems 1 and 2 and by 0.033 for system 3. On a normalized basis, the HP COP changes by about 0.0021 and 0.0019 per degree of superheating for systems 1 and 3, respectively. The COP values are summarized in Table 7-12.

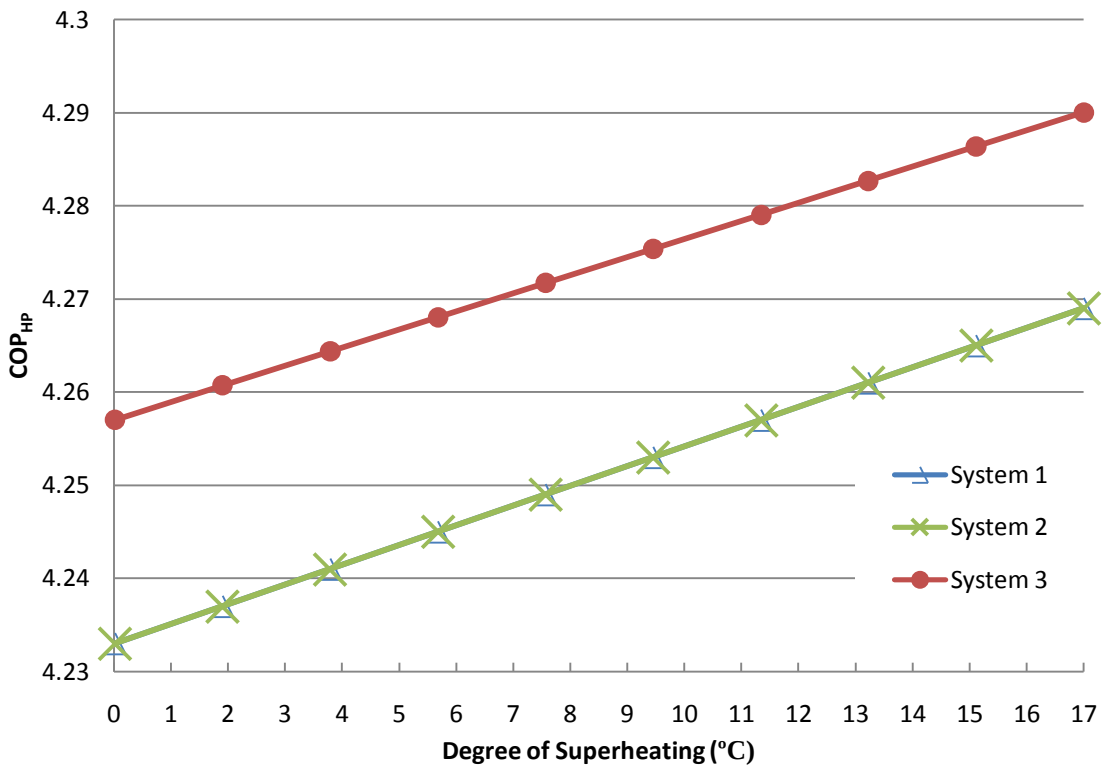

Figure 7.32: Effect of varying the degree of superheating at evaporator exit on heat pump COP for each system

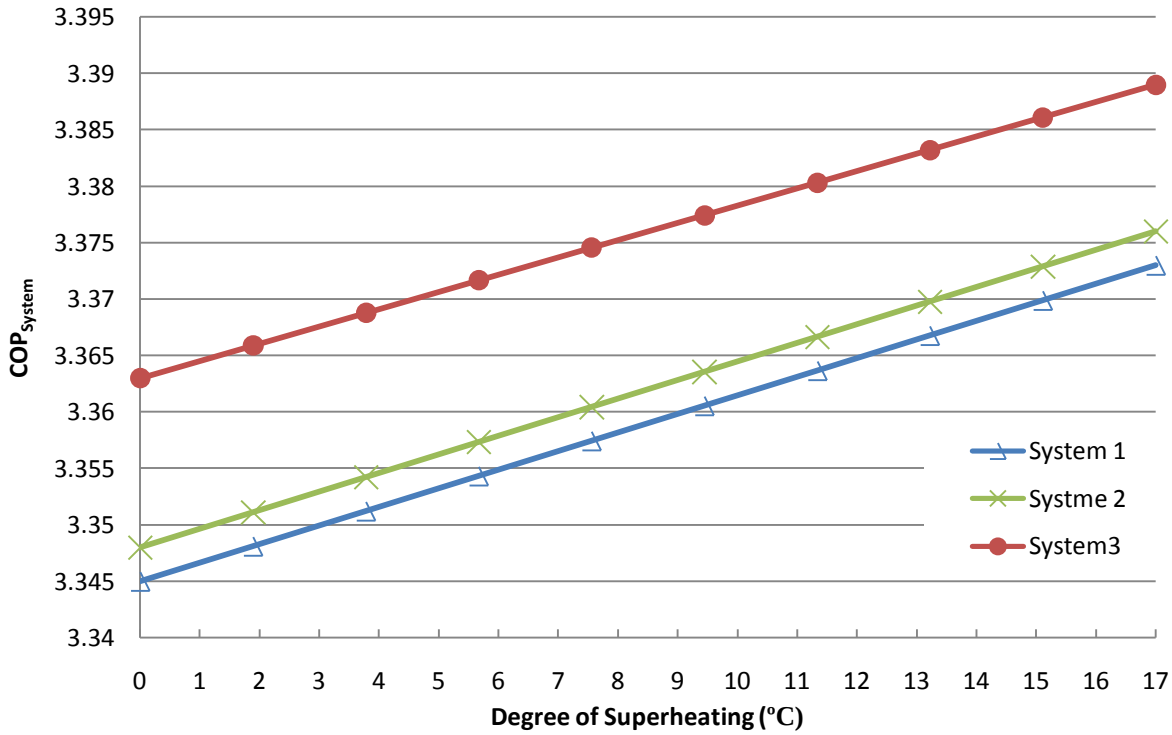

Figure 7.33: Effect of varying the degree of superheating at the evaporator exit on system COP for each system

Table 7-12: Effect of varying superheating on heat pump and system COP

		COP_{HP}			COP_{System}		
Degree of superheating (°C)	**Temperature at State 2 (°C)**	**System 1**	**System 2**	**System 3**	**System 1**	**System 2**	**System 3**
0	-10.09	4.233	4.233	4.257	3.345	3.348	3.363
1.898	-8.196	4.237	4.237	4.261	3.348	3.351	3.366
3.786	-6.308	4.241	4.241	4.264	3.351	3.354	3.369
5.673	-4.42	4.245	4.245	4.268	3.354	3.357	3.372
7.561	-2.532	4.249	4.249	4.272	3.357	3.360	3.375
9.449	-0.6445	4.253	4.253	4.275	3.361	3.364	3.377
11.34	1.243	4.257	4.257	4.279	3.364	3.367	3.380
13.22	3.131	4.261	4.261	4.283	3.367	3.370	3.383
15.11	5.019	4.265	4.265	4.286	3.370	3.373	3.386
17	6.907	4.269	4.269	4.290	3.373	3.376	3.389
Range*		0.036	0.036	0.033	0.028	0.028	0.026

*Range is the difference in calculated COP values coinciding to the highest and lowest degree of superheating

The ranges of HP and system COPs for systems 1 and 2 are greater than that for system 3, suggesting that systems 1 and 2 are affected the most by changing the amount of superheating after evaporation. The difference in trends between the systems is due to the varying division of flow rate through the main and supplementary circuit within system 3, as with the previous analysis.

The COP of all systems increases when the degree of superheating increases. Further, the temperature at the compressor exit increases as the temperature at the compressor inlet increases, as seen in Figure 7.34. When the temperature at the compressor is increased (for a constant heat load), the necessary refrigerant flow rate across the condenser decreases, which results in a decrease in compressor work and an increase in COP.

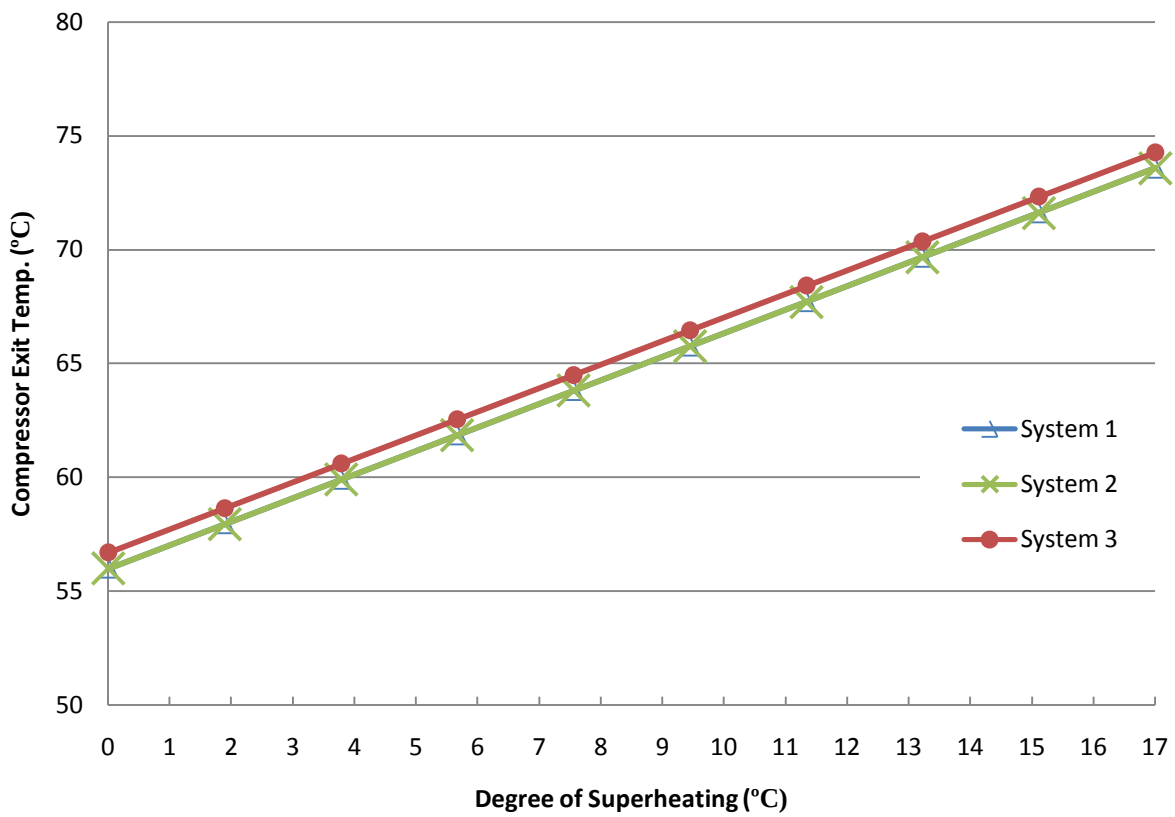

Figure 7.34: Effect of varying the degree of superheating at the evaporator exit on the compressor exit temperature

These trends suggest that it is beneficial to have a high degree of superheating (higher compressor inlet temperature). The main way to achieve greater superheating is to utilize a ground source that has a higher temperature, permitting a broader range for the degree of superheating degrees.

The effect of superheating on the supplementary flow rate within system 3 is shown in Figure 7.35, where the flow rate through the supplementary circuit is seen to decrease with increasing evaporator temperature. This trend is qualitatively similar to that found by Ma and Zhao [28].

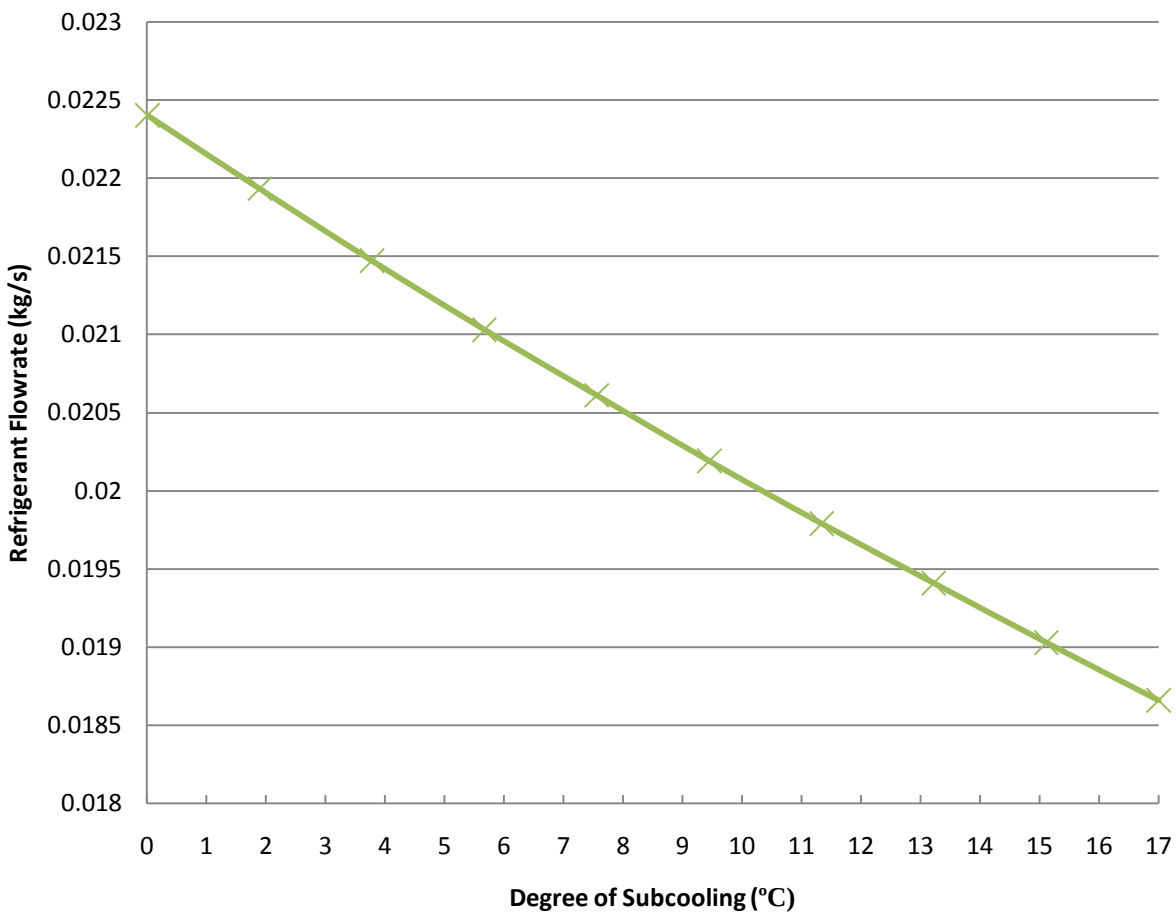

Figure 7.35: Effect of super heating on supplementary circuit flow rate.

Chapter 8: Closure

The closing sections of the present study are presented. Summary of principal results, conclusions, and recommendations are included.

8.1 Summary of Principal Results

The main results and findings for each study performed in the present work are summarized:

- **Basic comparison:** The comparison of the three systems under the original assumptions and operating parameters indicates that system 3 has the highest heat pump and system COPs. The economizer arrangement reduces compressor work for system 3, contributing directly to its higher COPs. Systems 1 and 2 have identical heat pump COPs. The system COP of system 2 is greater than that of system 1 because the use of motor cooling reduces the heat required from the ground loop. This leads to a decrease in GL length and pump work, yielding a system COP above that of the basic system. Furthermore, the amount of pump work required for the systems is minor compared to the compressor work for the heat pump. The pump is responsible for 1.0 to 1.2% of the total mechanical work input to the system, with system 3 requiring the greatest contribution of pump work (1.2%) and system 2 the lowest due to its reduced GL requirement through refrigerant preheating.

- **Compressor Efficiency:** Increasing the compressor efficiency increases the HP and system COPs for all systems, approximately linearly. The heat pump COPs for systems 1 and 2 are identical over the range of compressor efficiencies considered, while the system COP for system 2 is slightly higher than that for system 1. System 3 exhibits the greatest sensitivity to variation in compressor efficiency, with heat pump and system COPs varying by 1.62 and 1.23, respectively, over the range of compressor efficiencies. The corresponding changes for systems 1 and 2 are 1.51 and 1.17, respectively.

- **Pump Efficiency**: Trends are observed in system COP and GL length with pump efficiency, and all systems exhibit the same response. System COP increases as pump efficiency increases, but the change for all the systems is small (about 0.09%). Increasing pump efficiency leads to an increase in GL length, but again this effect minor as the increased GL length for all systems is about 2 m.

- **Motor Efficiency**: System COP and GL requirements vary with pump efficiency. The system COPs for all systems increase approximately linearly with increasing motor efficiency. Systems 1 and 2 exhibit almost identical trends and an overall change in system COP of around 2.72 for the motor efficiency range considered, while the system COP for system 3 changes by 2.74. When a more practical range of motor efficiencies is applied to systems 1 and 3, the change in system COPs become 2.12 and 2.13 for systems 1 and 3, respectively. The inclusion of motor cooling has the potential of producing the highest COP values over the practical range of motor efficiencies. The GL length of systems 1 and 3 does not change with motor efficiency. System 2 responds slightly differently, with the motor energy consumption for the pump in system 2 varying by a small amount (0.11 kW) from the maximum to minimum motor efficiency. Systems 1 and 3 have a variation in pump motor energy consumption of 0.45 kW over the range of motor efficiency.

- **Condenser Pressure**: A range of condenser pressures exists where the pressure is sufficiently high to allow the corresponding temperatures at the inlet and exit of the condenser to be adequate for typical heat distribution systems. The HP and system COPs of each system decreases with increasing condenser pressure. System 1 and 2 have identical HP COPs over the entire range of pressures, and are the least sensitive to condenser pressure (the HP COP range is 4.18). System 3 is the most sensitive to change in condenser pressure, as indicated it exhibiting the largest range in HP COP (4.21). The system COP range is observed to be 3.24, 3.25 and 3.28 for systems 1, 2 and 3, respectively. When the entire system (both

heat pump and ground loop) is considered, system 3 exhibits the greatest sensitivity to condenser pressure, while system 2 demonstrates the greatest reduction in GL length and pump power requirement.

- **Evaporator Pressure**: The evaporator pressure must be high enough to prevent the refrigerant temperature from falling below the lowest temperature in the GL, and low enough to prevent the refrigerant from exceeding the maximum temperature in the GL. As evaporator pressure increases, the HP and system COPs increase for all systems. The COPs for systems 1 and 2 show the greatest sensitivity to variations in evaporator pressure. For systems 1, 2 and 3, respectively, the HP COP range is 1.62, 1.62 and 1.61, while the system COP range is 1.25, 1.25 and 1.24.

- **Condenser/Evaporator Pressure Comparison**: Increasing either the condenser or evaporator pressure increases the COPs of all systems. The COPs are investigated over a similar range of pressure ratios, developed by varying either condenser or evaporator pressure. Discrepancies are noted for the same pressure ratios; the COPs developed by changing condenser and evaporator pressure, independently over the range of common pressure ratios, indicate that the all systems are more sensitive to changes in condenser pressure than evaporator pressure. The trends created through changing condenser and evaporator pressures differ when pressure ratios and COPs are considered, suggesting that COP is dependent on the evaporator and condenser pressures rather than the pressure ratio.

- **Intermediate Pressure**: Increasing the intermediate pressure in system 3 improves system performance, resulting in HP COP increasing. When the intermediate pressure is equal to the evaporator pressure the HP COP is identical to those of systems 1 and 2 in the basic comparison. System 3 thus attains higher COPs when the intermediate pressure is above the evaporator pressure. Systems including an economizer should be operated at the highest possible intermediate

pressure to achieve the greatest increase in COP. However, increasing the intermediate pressure negatively affects GL length, causing it to increase because more energy transfer is required from the ground with increased HP COP.

- **Subcooling at Condenser Outlet**: HP and system COPs increase approximately linearly for all the systems when the degree of subcooling at the condenser exit is increased (temperature is decreased). Systems 1 and 2 exhibit the greatest sensitivity to subcooling, leading to larger changes in both HP and system COPs. The change in HP COP is about 0.031 and 0.030 per degree of subcooling for systems 1 and 3, respectively. HP and system COPs for system 3 are always greater than those for systems 1 and 2. As subcooing is increased, the COPs for all three systems converge. Thus, the lowest condenser exit temperature should be utilized in heat pumps, while still allowing heat transfer to the heat distribution system within the building.

- **Superheating at Evaporator Outlet:** When the degree of superheating increases the HP and system COPs increase approximately linearly for all systems. The change in COP compared change in superheating is minor, with the variations in HP COPs for systems 1 and 3 being about 0.0021 and 0.0019 per degree of superheating, respectively. Systems 1 and 2 are the most sensitive to variations in superheating as they present the largest change in COP per degree of super heating. The effect of superheating is insignificant relative to that of subcooling.

8.2 Conclusions

The main objective of the present study is to investigate with parametric analyses the system characteristics of three geothermal heat pump arrangements that provide space heat for various operating conditions. The parametric studies compare the systems and their responses to variations in selected system parameters. The main conclusions drawn from the present study follow:

- Heat pump designs that include an economizer have the greatest potential for high efficiency, based on the heat pump and system COPs attained in the present study. For every analysis, system 3 attains the highest COPs for the ranges of variables investigated.

- A heat pump system that utilizes motor cooling/refrigerant preheating can reduce ground loop requirements without sacrificing performance relative to the basic vapor compression cycle. Throughout the study, system 2 requires the lowest rate of heat transfer from the ground loop, which reduces the required ground loop length. For every study, the ground loop length of system 2 is less than system 1, while attaining identical heat pump COPs and very similar system COPs.

- The COPs for heat pump designs that utilize an economizer are the least sensitive to variations in evaporator pressure, degree of subcooling and degree of superheating, relative to basic vapor compression cycles and compression cycles with motor cooling. The variations in COPs for system 3 are the smallest over the specified range of parameters. Systems with an economizer also are the most resilient to variations in the parameters cited above, where system performance differs the least relative to the other arrangements.

- Heat pump designs that utilize the basic vapor compression cycle or a compression cycle with motor cooling are the least sensitive to changes in compressor efficiency, motor efficiency and condenser pressure, compared to heat pump cycles with an economizer. The variations in COPs for systems 1 and 2 are

the smallest over the specified range of parameters. The basic heat pump and the heat pump with motor cooling are the most resilient to changes in the parameters considered, with the performance of the systems differing the least relative to the economizer unit.

- The parameter having the greatest effect on COPs for all systems considered is condenser pressure. Systems 1, 2 and 3 exhibit significant changes in both heat pump and system COPs over a given range of condenser pressures. The effect on system COP of the other parameters can be ranked from highest to lowest as motor efficiency, evaporator pressure, compressor efficiency, degree of subcooling, degree of superheating and pump efficiency, for the selected parameter ranges.

- The ground loop pump can be excluded from the analysis of heat pump systems with a little inaccuracy, as it plays a minimal role in system performance. The pump power is determined in the study to be small compared to the compressor power, with the pump energy accounting for about 2% of the total energy consumed.

- COP is dependent on the specific condenser and evaporator pressure within a system rather than the pressure ratio between the condenser and evaporator pressures. It is observed that the COP differs when the condenser and evaporator pressure are varied independently for identical pressure ratios.

- To maximize the performance characteristics of a heat pump system with an economizer, the intermediate pressure should be as high as possible while allowing normal operation of the system. As the intermediate pressure of system 3 increases and with all other operating assumptions remaining fixed, the COPs also increase.

- The heat pump COPs and ground loop length requirements are directly related. When the heat pump COP increases the required ground loop length also increases for all considered systems. Alternatively the length of the ground loop increases as the heat pump COP improves. Within this study system 3 requires the longest GL length, as it typically has the highest COP.

8.3 Recommendations

The results and findings of the present study illustrate how the systems covered react to variations in operating parameters and lead to several conclusions. The following recommendations are made based on the results and findings as well as conclusions. The recommendations relate mainly to utilization of the results and findings in both other studies and industry.

- Since the advanced systems covered in this study are not extensively investigated in literature, the present results and findings enhance the available information on the systems and their operation. Hence, it is recommended that the present work be used as a guide for future studies on these systems.

- The present results and findings, including trends, should also be used to assist design and optimization activities for the systems, to provide insight into improving the performance of heat pump systems that are in operation.

A number of conclusions were drawn from the results and findings of the analyses. Several of the conclusions lead to recommendations related to the design of the systems. These recommendations do not consider economics or environmental impact, as those factors are outside the scope of the present investigation. The recommendations follow:

- If a high efficiency heat pump is desired, an economizer should be adopted.

- When a heat pump system is designed for a location where limited land is available, a heat pump system with motor cooling/refrigerant pre-heating should be employed. Although the COPs of these systems are typically lower than for the economizer-based systems, the benefits of reduced ground loop length are necessary in some system designs.

- If a system operates with variations in the amount of superheating or subcooling, a heat pump system including an economizer should be used. COP Variations for this system are affected the least. Variations in the degree of superheating and subcooling often occur with changes in ground conditions such as temperature, and temperatures and loads corresponding to the space being heated.

- If a system is designed or optimized to provide the highest possible COP, regardless of ground loop length, the condenser and evaporator pressure should be as high as possible. High efficiency compressors, pumps and motors should be employed and the degree of subcooling and superheating maximized, subject to the constraint that the actual parameters must allow for proper system operation and adequate heat transfer within heat exchangers.

- When designing a heat pump system for heating purposes, the operating conditions should be chosen to optimize the COP while accommodating ground loop requirements. For example, a heat pump system with a high COP may not be suitable for implementation if there is limited space for the ground loop installation.

Though the present work illustrates reasonable insight into the operation of the discussed heat pump systems, recommendations to improve the study exist and are as follows:

- It is recommended that actual operational data be obtained for each of the three heat pump arrangements and used to improve the present results. Efforts are made to establish realistic settings for the components and operating conditions for

simulating the systems over reasonable parameter ranges, based primarily on data from literature and colleagues. This approach was necessary because, although attempts were made to obtain actual operational data, industry is often reluctant to provide complete details. Data would provide a better understanding of actual and complete configurations, more accurate results and additional sources of validation. Additional information on systems involving motor cooling and an economizer would be particularly important since data for these 2 types of systems are very sparse in the literature.

- The software utilized for simulation in the present study is somewhat limited. A complete collection of thermodynamic data for water/glycol mixtures does not exist within EES. Because of this the GL fluid was assumed an ideal mixture. To increase the accuracy of the models it is recommended that complete thermodynamic data for such mixtures be obtained and used, either from an external database or through different software that offer a more complete database.

Although the objectives of the present work have been satisfied, a number of opportunities for further work have been identified while completing the work:

- Exergy analysis should be used as a complementary addition to the present study. The present approach utilizes only a first law perspective for the analysis and comparison of the three systems. The use of exergy would allow the determination of the processes having the greatest irreversibilities, as well as the causes and locations of the irreversibilities. Exergy analysis would allow exergy efficiencies to be determined for each system and its components. Such knowledge can be used in designs or retrofits of systems for increasing system performance.

- A heat pump system with motor cooling or an economizer unit provides benefits over the basic vapor compression cycle, as motor cooling reduces GL length and the use of an economizer increases HP performance. The integration of these two

systems into a hybrid system design should be investigated. The methodology utilized in the present study could be adopted.

- Supplementary heating and cooling are sometimes employed in heat pump units. Supplementary heating utilizes an independent heater to provide additional heat to the HP cycle beyond what is gained from the GL, while supplementary cooling uses a typical HP cycle with a cooling tower. Since both concepts are designed to reduce heating or cooling loads on the GL, they could enhance ground source heat pumps systems. Hence, it is recommended that these technologies be investigated, with the systems considered in this thesis, to determine if a more beneficial system can be developed.

- It is recommended that this study be enhanced by adding economic assessments. The three systems involve different arrangements and components and respond differently to the GL length requirement with different operating conditions. These factors affect the cost of a ground source heat pump system. A cost analysis would provide details into whether these particular systems are economic for industrial and other applications. Economic analysis would also provide essential information for use in optimization of the systems. Hence economic considerations could determine which system provides the best efficiency while remaining competitive in the market, and identify areas where cost could be reduced without substantial losses in system performance.

- The study focuses on the provision of space heating. In many climates, space cooling is also required, especially during summer months. To obtain enhanced insights into the advantages of these systems in such climates, the cooling aspect of the systems should be investigated, considering both heating and cooling requirements throughout the year. Such a study would provide insights into the sizing of the HP equipment and GL lengths. Evaluations of how the systems operate and vary with parameters, for cooling and heating, could assist the selection of a system with operating conditions that are most favorable for the

entire year. Also, a study into the variations of system efficiency measures to cooling load variations would help develop a complete understanding of the possible systems and arrangements.

- Within the study a variable X was introduced for system three. This represented the ratio of refrigerant flow through the condenser and main circuit. No optimization of this variable with respect to system performance was performed within the study. It is recommended to perform a study on the optimization of this variable; optimization of the amount of refrigerant flow through the secondary and main circuit within heat pump systems utilizing economizer arrangements.

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Appendix: Program Code Developed for Simulation

The following is a sample of the EES program developed for the presented study. The sample is the basic layout for system 1.

```
FUNCTION Nu#(Re#,Pr_brine_5_6,Friction_factor) "function to determine the Nusselt # for the ground loop
dependant on reynolds number"
IF (Re#<=2300)  THEN  "IF statement for the use of the correct Nu# equation according to the reynolds
number (laminar, turbulent or transitional flow)"
Nu:=3.66  "nusselt # assuming constant surface temperature laminar flow"

ELSE
    IF(Re#>=2300) and (Re#<=10000) THEN

    Nu=((Friction_factor/8)*(Re#-
1000)*Pr_brine_5_6)/(1.07+12.7*((Friction_factor/8)^.5)*((Pr_brine_5_6^(2/3))-1))

    ELSE

    Nu:=0.023*(Re#^0.8)*(Pr_brine_5_6^.4)   "nusselt # assuming constant surface temperature turbulent
flow with the specific power for heating"
    ENDIF
ENDIF
Nu#:=Nu
END


FUNCTION Ff(Re#)      "function to determine the Friction factor for the ground loop dependant on reynolds
number"
IF (Re#<=2300)  THEN

F:=64/Re#
ELSE

F:=(0.790*ln(Re#)-1.64)^(-2)

ENDIF
Ff:=F
END



"***************Heating System*******************"
"Steady operating conditions exist"
"Kenetic and potential changes are negligible"


"*********************************{building Characteristics}*****************************************************"

Load= 100                                                    "heating load in kW"

"****************************************{Heat Pump}*****************************************************"
```

```
"####States####"

"1"
"enthalpy is calculated in the exansion valve"
P[2]=P[1]                                          "no pressure loss across evaporator"
x[1]=Quality(R134a,P=P[1],h=h[1])                  "determination of quality"
T[1]=Temperature(R134a,P=P[1],h=h[1])              "determination of temperature"
s[1]=Entropy(R134a,h=h[1],P=P[1])                  "determination of entropy"


"2"
P[2]=200                                           "set pressure"
T_superheat=5                                      "degree of superheating degrees C"
T[2]=T_sat(R134a,P=P[2])+T_superheat               "actual temperature"
x[2]=quality(R134a,T=T[2],P=P[2])
h[2]=Enthalpy(R134a,T=T[2],P=P[2])                 "determination od enthalpy"
s[2]=Entropy(R134a,T=T[2],P=P[2])                  "determination of entropy"



"3"
P[3]=1000                                          "the pressure must be set as to allow the
refrigerant to be condensed at state 4"
h_s[3]=Enthalpy(R134a,s=s[2],P=P[3])               "assuming isentropic compression"
T[3]=Temperature(R134a,P=P[3],h=h[3])              "determination of temperature"
x[3]=quality(R134a,h=h[3],P=P[3])
s[3]=entropy(R134a,h=h[3],P=P[3])                  "determination of entropy"

"4"
P[4]=P[3]
T_subcool=5                                        "degree of subcooling degrees C"
T[4]=T_sat(R134a,P=P[4])-T_subcool "temperature is set as to be higher then that of the return water from a
radiant floor heating system"
x[4]=Quality(R134a,T=T[4],P=P[4])                  "determination of quality"
h[4]=Enthalpy(R134a,T=T[4],P=P[4])                 "determination of enthalpy"
s[4]=Entropy(R134a,T=T[4],P=P[4])                  "determination of entropy"


"####Devices####"
"Expansion Valve"
h[4]=h[1]                                          "isenthalpic process"

"Electric motor"
eta_motor=.8                                       "electric motor efficiency"
W_dot_motor_comp=W_dot_comp/eta_motor              "power consumed by electric motor"


"Compressor"
eta_comp=.75                                       "compressor isentropic efficicency"
eta_comp=(h_s[3]-h[2])/(h[3]-h[2])
W_dot_comp=m_dot_ref *(h[3]-h[2])                  "compressor work (kW)"

"Condenser"
Q_dot_out=Load
Q_dot_out=m_dot_ref*(h[3]-h[4])                    "heat rejected to space (kW)"

"Evaporator"
"Q_in=h[2]-h[1]"                                   "specific heat entering heat pump (kJ/kg)"
Q_dot_in=m_dot_ref*(h[2]-h[1])                     "heat entering heat pump (kW)"
```

```
PressureRatio_HP=P[3]/P[2]


"*****************************************{Ground Loop}*****************************************************"
antifreeze_concentration=.3
$reference propylene ash                                    "Sets reference state for the propylene glycol
to that of the ASHRAE standard --> sets specicic enthalpy and entropy to 0  for a saturated liquid at - 40
degrees C"


"####States####"

"5"
T[5]=T_ground-2 "temp. of brine exiting ground loop set as 2 degrees below the ground temperature"
p[5]=150
hw[5]=enthalpy(Water,P=P[5],T=T[5])                         "enthalpy of water in mixture"
sw[5]=Entropy(Water,P=P[5],T=T[5])                          "entropy of water"
ha[5]=enthalpy(Propylene,P=P[5],T=T[5])                     "enthalpy of antifreeze"
sa[5]=Entropy(Propylene,P=P[5],T=T[5])                      "entropy of antifreeze"

h[5]=(1-antifreeze_concentration)*hw[5]+antifreeze_concentration*ha[5]
s[5]=(1-antifreeze_concentration)*sw[5]+antifreeze_concentration*sa[5]

"6"
P[6]=P[5]                                                   "pressure at state 6"
T[6]=T[5]-6          "temperature at state 6 with a temperature difference of 3 degrees from HP inlet"
hw[6]=enthalpy(Water,P=P[6],T=T[6])                         "enthalpy of water in mixture"
sw[6]=Entropy(Water,P=P[6],T=T[6])                          "entropy of water"
ha[6]=enthalpy(Propylene,P=P[6],T=T[6])                     "enthalpy of antifreeze"
sa[6]=Entropy(Propylene,P=P[6],T=T[6])                      "entropy of antifreeze"

h[6]=(1-antifreeze_concentration)*hw[6]+antifreeze_concentration*ha[6]
s[6]=(1-antifreeze_concentration)*sw[6]+antifreeze_concentration*sa[6]

CALL BRINEPROP2('PG',antifreeze_concentration*100[%],T[6]:FreezingPt_brine_6, Density_brine_6,
SpecHeat_brine_6, ThermalC_brine_6, DynVisc_brine_6, Pr_brine_6) "determination of brine charateristics
at state 6"


"7"
P[7]=P[5]+delta_P_parallel                                  "operating pressure within ground loop"
sw[7]=Entropy(Water,t=t[7],h=h[7])                          "entropy of water"
sa[7]=Entropy(Propylene,t=t[7],h=h[7])                      "entropy of antifreeze"
s[7]=(1-antifreeze_concentration)*sw[7]+antifreeze_concentration*sa[7]


"####Devices####"
"Ground"
T_ground=9.3 "Ground temperature surrounding GL heat exchanger-->also assumed to be the surface
temperature of the ground heat exhcanger-->National Climate Data and Information Archive for Ottawa"

"Loop"
m_dot_GL_Parallel=.3                                        "mass flowrate through each parallel loop"
n_parallel_loops=m_dot_GL_evap/m_dot_GL_Parallel            "number of parallel loops required to
accomdate the overall mass flow through the evaporator"
Q_dot_GL_Parallel=Q_dot_loop/n_parallel_loops               "amount of energy picked up by each parallel
loop"
```

```
T_GL_mean=(T[7]+T[5])/2                                  "average temperature of fluid in the ground
heat exchanger"
CALL BRINEPROP2('PG',antifreeze_concentration*100[%],T_GL_mean:FreezingPt_brine_5_7,
Density_brine_5_7, SpecHeat_brine_5_7, ThermalC_brine_5_7, DynVisc_brine_5_7, Pr_brine_5_7)
"determination of brine charateristics in ground heat exchanger"

A_c_pipe_inner=(Pi*(Diameter_inner^2))/4                 "internal cross sectional area of ground loop
piping"
V_avg=m_dot_GL_parallel/(Density_brine_5_6*A_c_pipe_inner)    "average fluid velocity"
Re#=(Density_brine_5_6*V_avg*Diameter_inner)/DynVisc_brine_5_6   "reynolds #"

Friction_factor=Ff(Re#)
Nusselt#=Nu#(Re#,Pr_brine_5_6,Friction_factor)
Nusselt#=(h_T_x*Diameter_inner)/(ThermalC_brine_6)       "general definition of nusselt # (solving for
heat transfer coefficient in w/m^2 C)"

R_convection=1/(h_T_x*A_s_inner)
R_Total=R_convection+R_pipe+R_grout                      "Thermal Resistance of ground loop system"
Q_heat_transfer=((T_ground-T_GL_mean)/R_Total)/1000 "Amount of heat that can be transfered per meter
of pipe length (kW)"

Length_parallel=Q_dot_GL_Parallel/Q_heat_transfer        "Length of one parallel loop (m)"
Length_total=Length_parallel*n_parallel_loops            "Total length of pipe in the ground loop (m)"


delta_P_parallel=(Friction_factor*(Length_parallel/Diameter_inner)*((Density_brine_5_6*(V_avg^2))/2))/1000
     "pressure drop within a single parallel loop (kPa)"
delta_P_Total=delta_P_parallel*n_parallel_loops  "Total pressure drop within the entire ground loop"


Q_dot_loop=Q_dot_GL-W_dot_pump   "amount of heat actually extracted from the ground when energy
balance is done including ground loop pump work"
Q_dot_GL_Parallel=m_dot_GL_parallel*SpecHeat_brine_5_6*(T[5]-T[7])


{polyethylene pipe}
depth=100    "borehole depth assumed to be 100 m"
Diameter_outer= 0.032   "1 1/4 inch=.032m ->diameter of ground loop piping 33.8mm or (1.33 inch) another
common size is 19mm (3/4 inche) (commercial earth systems and directed studies report)"
Radius_outer=.5* Diameter_outer
Diameter_inner=.0262   "Pipe inner diameter is set by the thickness of the pipe which coincides with the
pressure rating. --> the pipe used is hard polyehtylene DN 32, PN 10"
Radius_inner=.5* Diameter_inner
ThermalC_pipe=0.375  "Thermal conductivity of the pipe material (w/mK) -->taken from various sources for
hard polyehtylene"

A_s_inner=2*Pi*(1/2*Diameter_inner)                      "inner surface area of piping per meter
length"

R_pipe=(ln(Radius_outer/Radius_inner))/(2*Pi*ThermalC_pipe) "thermal resistance of pipe wall (k/W per unit
length)"


{Grout}
Thickness_grout=0.02506  "Average thickness of grout material incasing the ground loop piping 25.06 mm"
ThermalC_grout=1.56  "thermal conductivity of grout material (W/mK) --> taken from geothermal grout-
enhanced thermally conductive grout"

R_grout=(ln((Radius_outer+Thickness_grout)/Radius_outer))/(2*Pi*ThermalC_grout) "thermal resistance of
grout (k/W per meter)"
```

```
"Evaporator"
Q_dot_GL=Q_dot_in                                   "assuming the evaporator to be 100%
efficient"
Q_dot_GL=m_dot_GL_evap*(h[5]-h[6])
T_Evap_mean=(T[5]+T[6])/2    "close approximation of the average temperature difference of the ground
piping"
CALL BRINEPROP2('PG',antifreeze_concentration*100[%],T_Evap_mean:FreezingPt_brine_5_6,
Density_brine_5_6, SpecHeat_brine_5_6, ThermalC_brine_5_6, DynVisc_brine_5_6, Pr_brine_5_6)
"approximation of the conditions between states 7 and 5-- used to calcuate the ground loop length and
pressure drop, thus pump work and h[7]"

"pump"
eta_pump=.90
W_pump_ideal= (1/Density_brine_6)*(P[7]-P[6])          "specific pump work (kJ/kg) assuming 100%
efficient"
h_s[7]=h[6]+W_pump_ideal                            "assuming isentropic compression"
W_pump=h[7]-h[6]
eta_pump=W_pump_ideal/W_pump

W_dot_pump_parallel=m_dot_GL_Parallel*W_pump        "pump work for each pump supplying a
collection of loops(kW)"
W_dot_pump=n_parallel_loops*W_dot_pump_parallel
W_dot_pump_check= m_dot_GL_evap*W_pump


"Electric motor"
W_dot_motor_pump=W_dot_pump/eta_motor    "power consumed by electric motor to provide the nessary
mechanical work for the pumps"



"*********************************{PERFORMANCE}*****************************************************"
COP_HP_check=Q_dot_out/W_dot_comp                   "heat pump COP"
COP_HP=Q_dot_out/(Q_dot_out-Q_dot_in)               "heat pump COP"

COP_system=Q_dot_out/(W_dot_motor_comp+W_dot_motor_pump)    "system COP"
```