Energy Analysis and Comparison of Advanced Vapour Compression Heat Pump Arrangements

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Abstract

Reductions in fossil fuel use and increases in system efficiency are required to make space heating more environmentally benign. Ground loop heat pumps offer an alternative heating solution that is more environmentally benign than conventional methods. Past studies of these heat pumps have usually focused on basic system arrangements, but new advanced systems are being developed. Here, energy analyses and comparisons are reported for conventional and advanced heat pump arrangements including a basic vapour compression cycle, a vapour compression cycle including electric motor cooling, and a vapour compression cycle with an economizer. The results show that heat pump designs with an economizer have potential for increased coefficients of performance (COPs). A cycle with motor cooling is found to have the same heat pump COP and lower system COP compared to the basic cycle.

1 Introduction

Energy is often utilized in the form of electricity and heat, usually produced using fossil fuels, which comprise most of the global energy supply. However, fossil fuels are finite resources and their combustion is harmful to the environment and contributes to climate change through greenhouse gas emissions. Demand for energy is increasing and future fossil fuel shortages are predicted (Ediger et al. 2007). Fossil fuel depletion along with pollutant emissions and global warming are important factors affecting the degree to which energy systems are sustainable and environmentally benign (Hammond 2000). Such concerns have motivated efforts to reduce society's dependence on fossil fuels, by reducing demand and substituting alternative energy sources that are more environmentally benign than fossil fuels yet economically viable.

Beyond fossil fuels, the Earth's crust stores a large amount of thermal energy. Geothermal energy systems are relatively benign environmentally, with the emissions much lower than for conventional fossil fueled systems (Office of Energy Efficiency and Renewable Energy 2004; Wu 2009). Low (i.e., near environmental) temperature geothermal resources are abundant and can be utilized in most locations around the world. Heat pumps (HPs) extract low temperature thermal energy and raise its temperature to provide an environmentally and economically advantageous option for space heating.

Most reports on ground source heat pumps focus on use and operation, with the majority for devices that utilize the basic vapour compression cycle. Comparisons of ground- and air-source heat pumps are common. Energy and exergy analyses have been carried out for the basic heat pump arrangement, using simulation and experimental methods. For instance, Hepbasli and Balta (2007) use energy and exergy approaches and experimental data to assess the performance of a heat pump system using low temperature geothermal resources, including determination of the COP and identification of the locations of the greatest irreversibilities within the system. Akdemir and Hepbasli (2004) thermodynamically analyze ground source heat pump systems with a U-bend ground heat exchanger for district heating purposes, derive energy and exergy relations and apply them to a geothermal heat pump (GHP) system providing heat. Healy and Ugursal (1997) investigate computationally the effect of various system parameters on GHP performance with a horizontal loop arrangement, and compare economically a GHP with a conventional heating/cooling system and an air source heat pump. Kara (2007) experimentally determines the performance of a GHP system in heating mode in the city of Erzurum, Turkey, while Wang et al. (2008) investigate the effects of compressor and motor cooling on a heat pump system. Ma and Chai (2004) develop an alternative heat pump cycle that incorporates an economizer into the vapour compression cycle with two compression processes between the condenser and the evaporator. Ma and Zhao (2008) continue the work of Ma and Chai (2004) by experimentally comparing the new heat pump cycle with a similar cycle that employs a flash tank with vapour separation and two compression processes.

The previous studies suggest that assessments of advanced heat pump arrangements, including a HP cycle with motor cooling and a HP cycle using an economizer, and comparisons to basic vapour compression cycles, are lacking. Application of advanced HP cycles within geothermal heat pumps and the effect on the ground loop heat exchanger also has not been adequately explored. The present study aims to improve the understanding of advanced vapor compression cycles and their use in geothermal applications. This study seeks insights into the operation of these systems and describes their benefits and challenges. The research aims to provide a foundation for more advanced transient simulations using comprehensive building simulation methods. Economic aspects of the systems are not investigated here.

2 Approach and Methodology

We carry out a preliminary analysis to compare advanced geothermal heat pump cycles, and utilize a non-transient approach for simplicity and clarity (although comprehensive building simulation methods are expected to be used in subsequent investigations). Energy analyses are performed for three ground source heat pump arrangements with identical heating loads: 1) a basic vapour compression cycle, 2) a vapour compression cycle including electric motor cooling, and 3) a vapour compression cycle with an economizer. There are two loops within each system: the heat pump cycle and the ground loop heat exchanger. The ground loop heat exchanger is also commonly referred to as the ground loop (GL). The arrangement and analysis for the ground loop is identical for each of the three systems.

Mass and energy balances are developed for each system, for steady state with steady flow conditions. Some state conditions are assumed common to all systems. A model and analysis tool is thereby developed for each system, using Engineering Equation Solver (EES) software. The heat pump systems are compared for static operating assumptions. Heat pump and system COPs and ground loop characteristics are evaluated and compared.

Several general simplifying assumptions are made: pressure drops within heat pumps and elevation changes are negligible, and all processes are adiabatic. Also, assumptions specific to the heat pump and ground loop parts of the systems are made, as noted below.

Heat pump cycle assumptions and data

Natural Resources Canada (2003) presents an average heating load of 100 kW for commercial and small institutional buildings within their document “Survey 2000, Commercial and Institutional Building Energy Use: Summary Report”; this heating load is used here for all system arrangements. Many other system parameters are determined from a set of initial simulations performed to determine suitable parameter ranges (see Table 1).

Table 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

Ground loop assumptions and data

The ground loop fluid utilized in this study is a glycol solution of water and propylene glycol, with 30% propylene glycol by mass. The average ground temperature is 9.3°C, bases on data for Ottawa, Ontario (Environment Canada 2011). For a constant ground temperature the pipe depth is set to 100 m (Dowlatabadi & Hanova 2007). The piping is standard high density polyethylene DN32 PN10 with a nominal diameter of 32 mm, an inner diameter of 26.2 mm, and a pressure rating of 10 bar (CANMET Energy Technology Centre 2002). The pipe material is assumed smooth, noting the equivalent roughness value for new commercial polyethylene pipe is 0 (Cengel 2007), and to have a thermal conductivity of 0.375 W/m·K based on an average from various sources (ISCO Industries 2011). The borehole diameter is 101.6 mm (4 inches) (Omer 2008), and the grout thermal conductivity is 1.56 W/m·K (CETCO Drilling Products 2009). The evaporator inlet and outlet temperatures for the ground loop are 7.3°C and 1.3ºC respectively, and the flow rate through an individual parallel loop is 0.3 kg/s.

3 System Configurations

System 1

System 1 (Figure 1) is a basic heat pump cycle 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. The refrigerant exits the evaporator at state 2 as superheated vapour and enters the compressor, exiting as a high pressure superheated vapour at state 3. The refrigerant enters the condenser and thermal energy is extracted and supplied to the space. Exiting the condenser (state 4), the refrigerant is a subcooled liquid. This arrangement is widely utilized due to its simplicity and ease of design (Omer 2008).

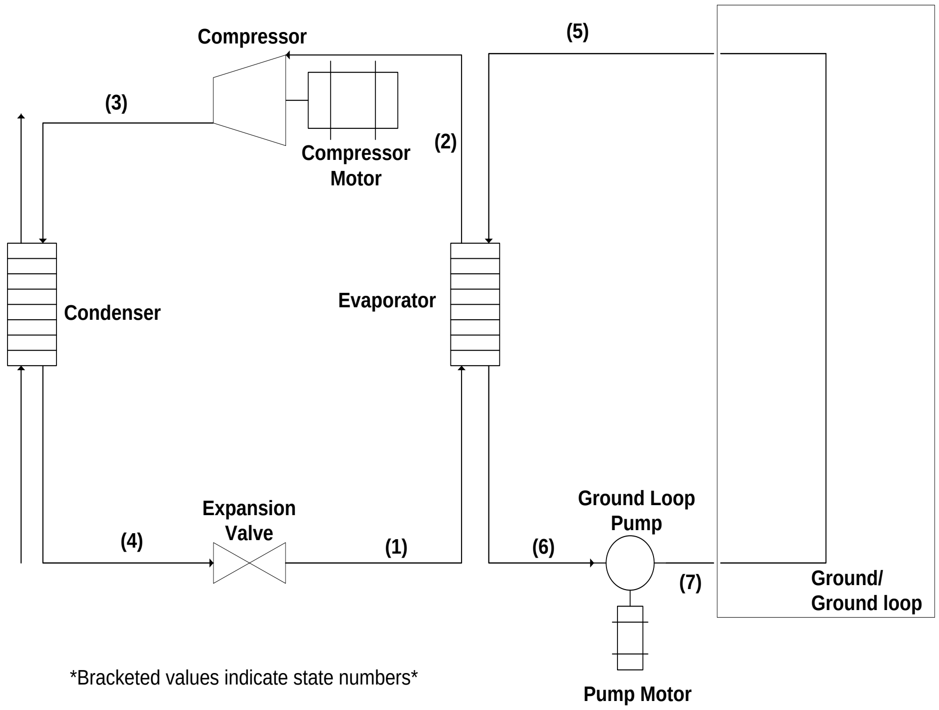

Figure 1: System diagram of heat pump system 1

System 2

System 2 (Figure 2) is identical to system 1 except for a modified flow path used for motor cooling. The new path directs the refrigerant to the compressor motor where it comes in contact with the electric motor assembly, heating it.

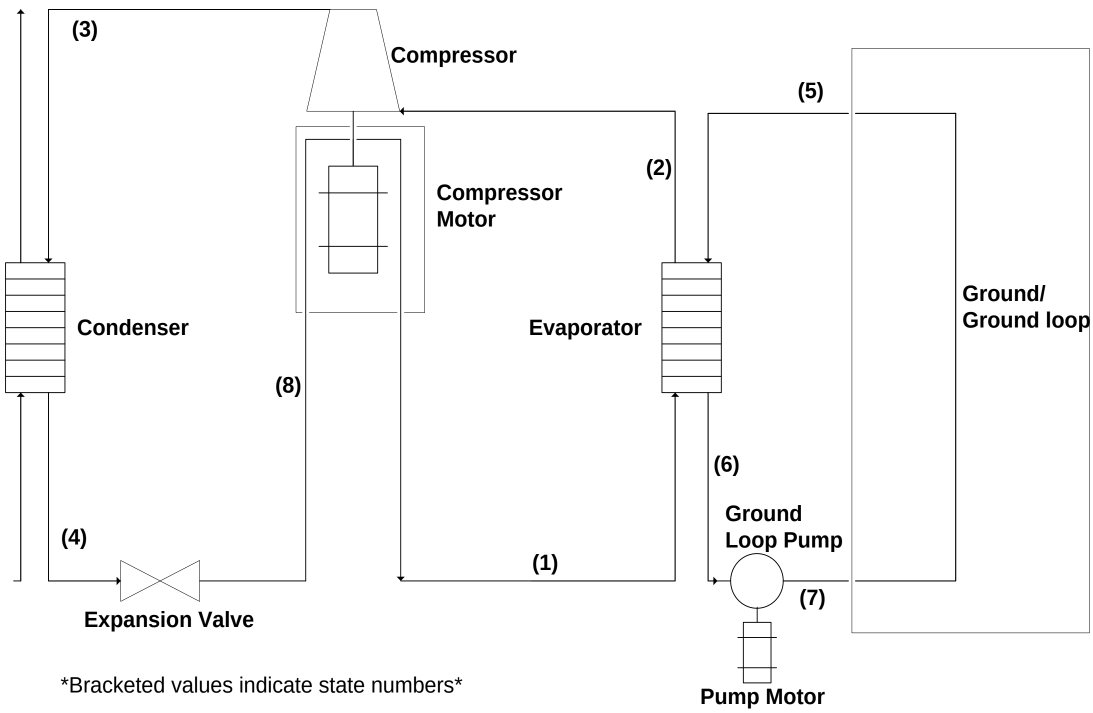

Figure 2: System diagram of heat pump system 2

System 3

System 3 (Figure 3) is a heat pump system with an economizer, as described by Ma and Chai (2004). In the economizer heat is transferred between two refrigerant flows. The new flow path (the supplementary circuit) includes an expansion valve that allows the flow to exist at an intermediate pressure between the evaporator and condenser pressures. Heat is transferred from the main refrigerant circuit to the supplementary circuit, and the main flow then expands to the evaporator pressure. The supplementary circuit flows from the economizer to the compressor through a supplementary inlet. Ma and Zhao (2008) treat the compression process as quasi two-stage compression with an intermediate mixing chamber.

Ground loop heat exchanger

The ground loop heat exchanger draws heat from the ground and supplies it to the heat pump during operation. All systems considered here utilize the same ground loop arrangement. We examine HP units with a commercial heating load and use a U-tube vertical borehole arrangement since CANMET Energy Technology Centre (2002) indicates this is best for large heating applications. The borehole design consists of a flow path through the evaporator, a pump, and multiple parallel loops. A cool glycol solution flows from the evaporator to the pump, where its pressure is increased to overcome the pressure losses within the ground loop. The flow splits through a header into parallel loops, which receive heat from the surrounding ground. The mixture streams exit the parallel loops and joins to a single warm stream which enters the evaporator where heat is transferred to the refrigerant in the HP cycle.

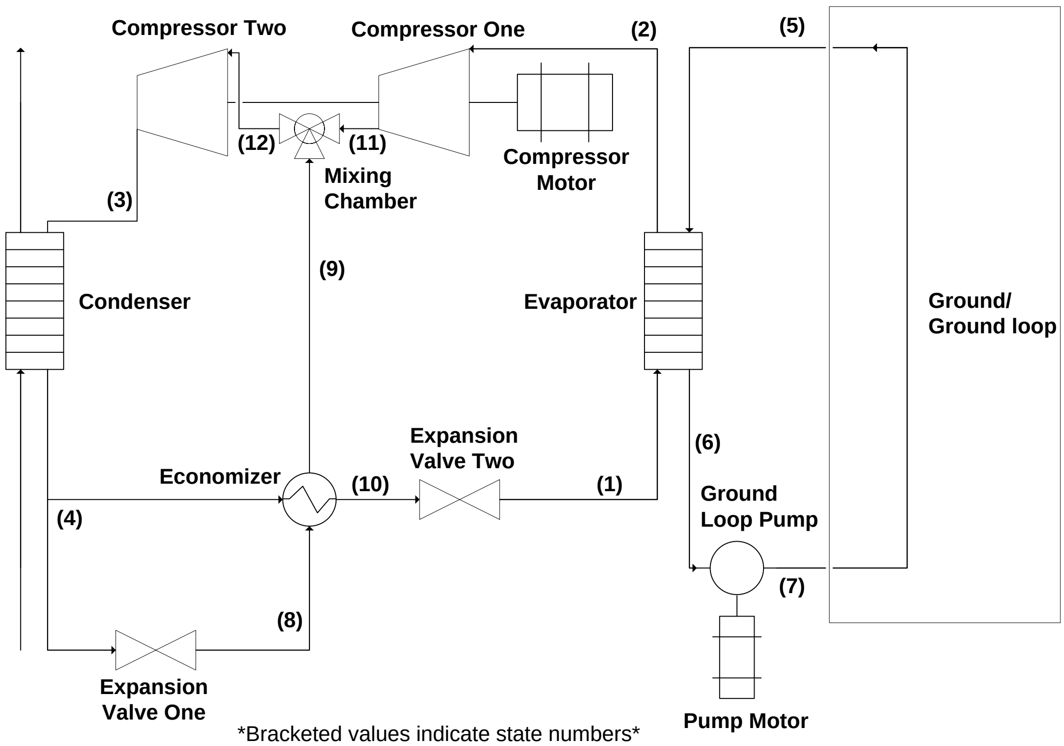

Figure 3: System diagram of heat pump system 3

4 Analysis

Heat pump system 1

Refrigerant enters the inlet of the compressor at state 2. The temperature is the saturation temperature plus an assumed degree of superheating, as presented by Ma and Zhao (2008). The refrigerant is then compressed to a higher pressure at state 3. The superheated refrigerant exiting the compressor enters the condenser where it condenses at constant pressure and becomes a pressurized liquid. The heat extracted from the refrigerant for space heating is set to the building load. The temperature at state 4 is taken as the saturation temperature minus an assumed degree of subcooling (Ma and Zhao 2008). Using the specific enthalpy at states 3 and 4, the mass flow rate of refrigerant is found with a condenser energy balance:

$$\dot{Q}_{load} = \dot{m}_{ref}(h_3 - h_4) \tag{1}$$

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. Flow through the evaporator also occurs without pressure drop. We assume isenthalpic behaviour through the expansion valve.

The heat transfer rate ($\dot{Q}_{evap}$) from ground loop to refrigerant via the evaporator is:

$$\dot{Q}_{evap} = \dot{m}_{ref}(h_2 - h_1) \tag{2}$$

where h_1 and h_2 are the specific enthalpies at states 1 and 2, respectively. Here, the heat transfer rate to the refrigerant is the rate of heat removal from the GL glycol solution through the evaporator. The compressor power ($\dot{W}_{comp}$) is expressible as:

$$\dot{W}_{comp} = \dot{m}_{ref}(h_3 - h_2) \tag{3}$$

The compressor is driven by an electric motor for which the electrical power consumption ($\dot{E}_{comp\ motor}$) is:

$$\dot{E}_{comp\ motor} = \frac{\dot{W}_{comp}}{\eta_{EM}} \tag{4}$$

where η_{EM} is the efficiency of the electric motor.

The heat pump coefficient of performance (COP_{HP}), without motor and pump work, is

$$COP_{HP} = \frac{\dot{Q}_{load}}{\dot{W}_{comp}} \tag{5}$$

where $\dot{Q}_{load}$ is the building heating load. The system coefficient of performance COP_{System} accounts for electrical energy consumption by the compressor and pump motors $\dot{E}_{motor,total}$:

$$COP_{System} = \frac{\dot{Q}_{load}}{\dot{E}_{motor,total}} \tag{6}$$

where

$$\dot{E}_{motor,total} = \dot{E}_{comp\ motor} + \dot{E}_{pump\ motor} \tag{7}$$

where $\dot{E}_{pump\ motor,}$ is the rate of electrical energy consumption of the pump motor.

Heat pump system 2

The analysis the heat pump system with motor cooling (system 2) is similar to that of system 1, except that state 8 is added and the conditions differ at state 1. The conditions at state 8 are found as for state 1 in system 1. We assume no pressure drop is between states 8 and 1 and an isenthalpic expansion valve. The available waste energy from the compressor motor is the difference between the compressor power and the electrical power consumption of the motor:

$$\dot{E}_{motor\,waste} = \dot{E}_{comp\,motor} - \dot{W}_{comp} \tag{8}$$

where $\dot{E}_{motor\,waste}$ is the rate waste energy associated with the compressor motor. It is assumed that all waste energy from the electric motor is converted to thermal energy and transferred to the refrigerant. That is,

$$\dot{E}_{motor\,waste} = \dot{m}_{ref}(h_1 - h_8) \tag{9}$$

The expressions for coefficient of performance for system 2 are the same as for system 1.

Heat pump system 3

In system 3, the refrigerant exiting the evaporator is compressed to an intermediate pressure. The refrigerant exiting compressor 1 enters the mixing chamber with the refrigerant from the economizer in the supplementary circuit. An energy balance across the mixer yields:

$$\dot{m}_{ref}h_{12} = \dot{m}_{main}h_{11} + \dot{m}_{supp}h_9 \tag{10}$$

where $\dot{m}_{main}$ is the mass flow rate of refrigerant through the evaporator, and $\dot{m}_{supp}$ is the mass flow rate through the supplementary circuit. The conditions at state 9 are set to those at state 11, following the approach presented by Ma and Chai (2004) and Ma and Zhao (2008).

The main and supplementary flow rates are fractions of the total refrigerant flow rate through the condenser. The ratio *X* of the flow rate of the main circuit to the total refrigerant flow rate through the condenser is defined as:

$$X = \frac{\dot{m}_{main}}{\dot{m}_{ref}} \tag{11}$$

Then equation 10 becomes

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

or, excluding the total refrigerant flow rate,

$$h_{12} = Xh_{11} + (1 - X)h_9 \tag{13}$$

This equation allows the enthalpy value at state 12 to be calculated without the use of the refrigerant flow rate. The value of *X* is calculated via an energy balance for the economizer:

$$X\dot{m}_{ref}(h_4 - h_{10}) = (1 - X)\dot{m}_{ref}(h_9 - h_8) \tag{14}$$

Solving for *X* yields:

$$X = \frac{h_8 - h_9}{h_4 + h_9 - h_8 - h_{10}} \tag{15}$$

At the condenser exit the refrigerant splits: one flow feeds the main circuit and the other the supplementary circuit. The former flow enters the economizer where heat is extracted from the refrigerant resulting in a higher degree of subcooling at state 10 (Ma and Zhao 2008). We assume an intermediate pressure and state 9 temperature, which must allow the refrigerant at state 12 to exist as a saturated or superheated vapour after mixing with the flow at state 11. With two compression processes the compressor work rate is:

$$\dot{W}_{comp} = \dot{W}_{comp,1} + \dot{W}_{comp,2} \tag{16}$$

where

$$\dot{W}_{comp,1} = \dot{m}_{main}(h_{11} - h_2) \tag{17}$$

$$\dot{W}_{comp,2} = \dot{m}_{ref}(h_3 - h_{12}) \tag{18}$$

where $\dot{W}_{comp,1}$ and $\dot{W}_{comp,2}$ are rate of work required by compressors 1 and 2 respectively. The heat pump and system coefficients of performance are as for system 1.

Ground loop heat exchanger analysis

The GL consists of a mixture of water and glycol, which is treated as ideal mixture. The mass fraction of glycol is set to have an appropriate freezing point, which is required to be below that of the ground temperature and heat pump evaporator inlet temperature (Plastic Institute 2009). The pressure within the evaporator is assumed constant. The temperature at state 6 is assumed noting it must remain above the freezing temperature of the GL fluid. The mass flow rate of the glycol solution is determined with an energy balance across the evaporator:

$$\dot{m}_{ref}(h_2 - h_1) = \dot{m}_{GL,Evap}(h_5 - h_6) \tag{19}$$

where $\dot{m}_{GL,Evap}$ is the glycol solution mass flow rate through the evaporator.

The flow through the evaporator is 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 to promote heat transfer. The number of parallel pipes (n_{pl}) is determined as:

$$n_{pl} = \frac{\dot{m}_{GL,Evap}}{\dot{m}_{pl}} \tag{20}$$

where $\dot{m}_{pl}$ is the flow rate though an individual parallel loop.

The pressure at state 7 is estimated using the pressure drop in the GL 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:

$$\Delta P_{pl} = f\,\frac{l_{pl}}{D_i}\,\frac{\rho_{5,7}\,{V_{avg}}^2}{2} \tag{21}$$

where f is the friction factor, l_{pl} 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 a pipe.

The length of the GL heat exchanger is estimated through heat transfer analysis, with the assumption that the outer surface of the grout is at the temperature of the ground, and the ground is an infinite heat source. Within vertical loop systems two pipes are contained in a single borehole and common borehole diameters exist within industry. The radius r_3 is estimated through the assumption that there is no thermal interaction between the pipes. The maximum and minimum grout radius from the pipe center line are calculated and the average is used as an estimate of r_3.

The heat transferred to the ground loop fluid per unit length ($\dot{Q}_{ht}$) is calculated as

$$\dot{Q}_{ht} = \frac{T_{ground} - T_{fluid}}{R_{total}} \tag{22}$$

where T_{ground} denotes the ground temperature, T_{fluid} is the mean fluid temperature between states 5 and 7, and R_{total} the total thermal resistance of the vertical pipe arrangement. The length of a single parallel loop is then determined:

$$l_{pl} = \frac{\dot{Q}_{pl}}{\dot{Q}_{ht}} \tag{23}$$

where $\dot{Q}_{pl}$ is the heat transfer rate to one parallel pipe. The heat transfer rate to a parallel pipe is expressible as follows:

$$\dot{Q}_{pl} = \frac{\dot{Q}_{GL,actual}}{n_{pl}} \tag{24}$$

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:

$$\dot{Q}_{GL,actual} = \dot{Q}_{evap} - \dot{W}_{pump} \tag{25}$$

where $\dot{W}_{pump}$ is the pump work rate, calculated as:

$$\dot{W}_{pump} = n_{pl}\dot{m}_{pl}w_{pump,actual} \tag{26}$$

where $w_{pump,actual}$ is the specific pump work. Here, the pump is not assumed isentropic, so:

$$\eta_{pump} = \frac{w_{pump,ideal}}{w_{pump,actual}} \tag{27}$$

The isentropic efficiency of the pump is assumed for calculation of actual pump work rate. The ideal specific pump work is found using the pressure difference across the pump:

$$w_{pump,ideal} = \nu_6(P_7 - P_6) \tag{28}$$

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

$$P_7 = P_5 + \Delta P_{pl} \tag{29}$$

The conditions at state 7 are required 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. An iterative approach is necessary in order to obtain the conditions at state 7. Then, the pump work rate is obtained, and the electrical power required from the motor for pump operation is found as

$$\dot{E}_{pump\ motor} = \frac{\dot{W}_{pump}}{\eta_{EM}} \tag{30}$$

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.

5 Results

The systems are compared, in terms of compressor and pump work rate, rate of heat transfer from the ground loop system, overall ground loop length, heat pump COP and overall system COP. Tables 2, 3, and 4 illustrate the conditions at each state for systems 1, 2, and 3, respectively. The tables contain a combination of values derived from the given assumptions as well as those calculated through analysis. The main aspects of each heat pump's operation are compared in Table 5.

Table 2: State conditions for system 1, following states identified in Figure 1

State	Fluid	Temperature (°C)	Pressure (kPa)	Quality	Specific enthalpy (kJ/kg)	Mass flow rate (kg/s)
1	Refrigerant	-10.09	200	0.33	99.92	0.51
2	Refrigerant	-5.09	200	1	248.7	0.51
3	Refrigerant	61.16	1000	1	294.6	0.51
4	Refrigerant	34.37	1000	0	99.92	0.51
5	Glycol Solution	7.3	150	0	170.9	3.76
6	Glycol Solution	1.3	150	0	150.6	3.76
7	Glycol Solution	2.11	223.6	0	150.6	3.76

Table 3: State conditions for system 2, following states identified in Figure 2

State	Fluid	Temperature (ºC)	Pressure (kPa)	Quality	Specific enthalpy (kJ/kg)	Mass flow rate (kg/s)
1	Refrigerant	-10.09	200	0.35	111.4	0.51
2	Refrigerant	-5.09	200	1	248.7	0.51
3	Refrigerant	61.16	1000	1	294.6	0.51
4	Refrigerant	34.37	1000	0	99.92	0.51
5	Glycol Solution	7.3	150	0	170.9	3.47
6	Glycol Solution	1.3	150	0	150.6	3.47
7	Glycol Solution	2.11	223.6	0	150.6	3.47
8	Refrigerant	-10.09	200	0.33	99.92	0.51

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

State	Fluid	Temperature (ºC)	Pressure (kPa)	Quality	Specific enthalpy (kJ/kg)	Mass flow rate (kg/s)
1	Refrigerant	-10.09	200	0.26	92.66	0.49
2	Refrigerant	-5.09	200	1	248.7	0.49
3	Refrigerant	61.86	1000	1	295.3	0.51
4	Refrigerant	34.37	1000	0	99.92	0.51
5	Glycol Solution	7.3	150	0	170.9	3.76
6	Glycol Solution	1.3	150	0	150.6	3.76
7	Glycol Solution	2.11	200	0	150.6	3.76
8	Refrigerant	8.91	400	0.19	99.92	0.02
9	Refrigerant	22.41	400	1	268.1	0.02
10	Refrigerant	29.37	1000	0	92.66	0.49
11	Refrigerant	22.41	400	1	268.1	0.49
12	Refrigerant	22.41	400	1	268.1	0.51

Table 5: 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)	0.298	0.275	0.298
Pump motor energy consumption rate (kW)	0.372	0.344	0.373
Rate of heat supplied from GL system (kW)	76.43	70.54	76.55
Total GL length (m)	3346	3088	3352
$\boldsymbol{COP_{HP}}$	**4.2**	**4.2**	**4.3**
$\boldsymbol{COP_{System}}$	**3.4**	**3.4**	**3.4**

System 3 is observed to have the highest heat pump and system coefficient of performance, with a COP_{HP} of 4.3 and a COP_{system} of 3.4. The higher COP_{HP} value can be attributed to lower compressor work compared to the other two systems. With an increased COP, system 3 places the highest heat rate requirement on the ground loop, which increases the GL length. As a result the pump work increases due to the increased glycol solution mass flow rate and pressure drop in the loop. The higher pump work reduces COP_{system} for system 3, and overall the COP_{system} of each system is essentially identical.

The COP_{HP} for system 3 is greater than that of system 1 by 2.4%. 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. From Tables 2, 3, and 4 it can be seen that the refrigerant temperature at the inlet of the condenser in system 3 is slightly higher than that of systems 1 and 2 with the same pressure ratio. If the refrigerant temperature at the exit of the compressor of each system were assumed equal it would be seen that system 3 would have a lower pressure ratio between the evaporator and condenser, which would result in lower compressor work rate and further increase in COP_{HP}.

Systems 1 and 2 have the same compressor work rates and therefore the same COP_{HP}. Within system 1 the entire thermal requirement of the system is accommodated by the ground loop. In system 2 a portion of the thermal requirement is covered through recovery of waste heat generated by the compressor electric motor. 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 in the heat requirement rate from the ground means that there is a reduction in the required GL length of approximately 260 meters (8%) for the same heating load. System 2 has a shorter GL length resulting in reduced pump power requirements, which should theoretically increase COP_{system}.

From the results shown in Table 5 it can be seen that the compressor work rate is significantly larger than the pump work rate, where the pump work rate for each system is only about 1% of the total power consumed by the systems. Having compression contribute such a large amount to the total energy consumption renders the effect of varying pump work rate on COP_{system}, through different GL lengths, minor. Through simulation it is found that COP_{system} for systems 2 and 3 increase by 0.09% and 0.5%, respectively, compared to system 1. These increases are sufficiently small that COP_{system} can be considered to be the same for each case.

The GL length can be measured in terms of length per unit heat supplied 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 values coincide with the vertical GL sizing guidelines 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, illustrated by CANMET Energy Technology Centre (2002). Kara (2007) calculates a length of 33.6 m/kW for the basic vapour compression cycle with similar operating conditions.

During the early stages of analysis it was found 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, which requires the pump to have a very high pressure outlet that approaches the upper limit of the pipe material. Employing multiple parallel loops allows for the total pressure drop in the system to be distributed between each loop.

6 Discussion

The comparison of the three systems shows that a heat pump system with the economizer arrangement (system 3) has the highest heat pump and system COPs. The inclusion of an economizer reduces compressor work, contributing directly to higher heat pump COP values. Alternatively the economizer arrangement has the highest GL and pump work requirements.

The heat pump COP values for the basic vapour compression cycle (system 1) and motor cooling (system 2) are identical through having the same compressor work rates. The system COP of the heat pump arrangement with motor cooling is the same as that of the basic vapour compression cycle since the impact of varying pump work rate on system COP is small. Compared to the basic vapour compression cycle the use of motor cooling reduces the heat required from the ground loop, which leads to a decrease in GL length and pump work.

If there is abundant space available for the installation of the ground loop it is favourable to install an economizer based heat pump as it provides the greatest performance. The use of an economizer increases the COP values, which means that it has the lowest energy input to provide an identical degree space heating.

When a heat pump system is designed for a location where limited land is available, the motor cooling arrangement would provide the greatest benefits. Although the heat pump COP value of this system is lower than for the economizer arrangement, the benefits of reduced ground loop length are necessary in some system designs.

To develop a high performance system that does not have large ground loop requirements we suggest a combination of the economizer arrangement and the motor cooling arrangement. This type of setup would allow for the compressor work rate and possibly the ground loop length to be reduced over the basic vapour compression cycle. Compared to the economizer arrangement the ground loop length would be reduced, which would result in a reduction in pump work rate and increase in system COP. Alternatively the combination of these systems would increase system complexity. Further studies into the type of arrangement would be required to determine its feasibility.

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 approximately 1.0% of the total mechanical work input to the system, with the economizer arrangement requiring the greatest contribution of pump work and the motor cooling arrangement requiring the lowest due to an increase and decrease in GL requirements, respectively.

7 Conclusions

Three heat pump designs (basic vapour compression cycle, a vapour compression cycle including electric motor cooling, and a vapour compression cycle with an economizer) are analyzed for identical heating loads and pressure ratios, and compared in terms of COP and ground loop requirements. The main conclusions follow:

- Heat pump designs that include an economizer have the greatest potential for high efficiency, based on the heat pump COPs attained in the present study.
- A heat pump system that utilizes motor cooling can reduce ground loop requirements without sacrificing performance relative to the basic vapour compression cycle.
- Heat pump COP and ground loop length are directly related.
- When the heat pump COP increases the required ground loop length also increases.
- The ground loop pump can be excluded from the analysis, at least during preliminary studies, of heat pump systems with a little inaccuracy.

This study is the first step in analyzing and developing detailed simulations that reflect system operation and feasibility for various situations. Future work is expected to involve transient assessments to allow for enhanced understanding of these systems and the associated benefits they offer.

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9 Nomenclature

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_i	Specific enthalpy at state *i*, kJ/kg
l_{pl}	Length of a single parallel loop, m
$\dot{m}_{GL,Evap}$	Mass flow rate of glycol solution through the evaporator, kg/s
$\dot{m}_{pl}$	Mass flow rate of glycol solution 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, kg/s
n_{pl}	Number of parallel loops within ground loop
P_i	Pressure at state *i*, kPa
$\dot{Q}_{evap}$	Heat input of evaporator to refrigerant, kW
$\dot{Q}_{GL,actual}$	Actual rate of heat transfer required from the ground, kW
$\dot{Q}_{pl}$	Rate of heat transfer to each parallel loop, kW
$\dot{Q}_{ht}$	Rate of heat transfer from ground to glycol solution fluid, kW
$\dot{Q}_{load}$	Heating load of building, kW/m
$\dot{Q}_{waste\ heat}$	Rate of heat transfer from the compressor motor to the refrigerant, kW
R_{total}	Total thermal resistance, °C/W
T_{ground}	Ground temperature, ºC
T_{fluid}	Average fluid temperature between states 7 and 5, ºC
V_{avg}	Average fluid velocity through a parallel ground loop, m/s
ν_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 specific work, kJ/kg
$w_{pump,ideal}$	Ideal pump specific work, kJ/kg
X	Ratio of mass flow rates in the main and supplementary circuits
ΔP_{pl}	Pressure drop within a parallel loop, kPa
η_{EM}	Electric motor efficiency
η_{pump}	Pump isentropic efficiency
ρ_i	Density at state i, kg/m^3

Subscripts

avg	Average
comp	Compressor
EM	Electric motor
evap	Evaporator
GL	Ground loop

HP	Heat pump
ht	Heat transfer
pl	Parallel loop
ref	Refrigerant
supp	Supplimentary

Acronyms

COP	Coefficient of performance
GHP	Geothermal heat pump
GL	Ground loop
HP	Heat pump

10 References

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