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Parametric study of a vertically configured ground source heat pump system

Habtamu B. Madessa *, Torger B., Pål F. Bye, Erlend A.

Oslo and Akershus University College of Applied Science, Department of Civil Engineering and Energy Technology, Postboks 4, St. Olavs plass,0130 Oslo, Norway

Abstract

Ground source heat pumps are being common in western countries in order to reduce the primary energy consumption and the corresponding greenhouse gas emissions associated with heating and cooling of buildings. In this paper, a parametric study was conducted to investigate the performance of a ground source heat pump configured with a vertical ground loop. Mathematical models were developed for the different components of the heat pump, and EES was used to analyse the performance of the heat pump considering the influence of the ground's depth, mass flow rates of brine, types of working fluids, and the dead state condition of the environment. The results revealed that the working fluids performed differently for the various operating parameters, and importantly selection of refrigerants as a working fluid for the ground source heat pump needs through thermodynamics analysis.

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Keywords: Ground source heat pump; Mathematical modelling; Exergy; Parametric analyses; Working fluids

1. Introduction

Buildings consume a large amount of thermal energy for proper functioning of the heating, cooling, ventilation, and hot water production systems. In Norway, about 40% of the total energy use goes to building operations [1]. In addition, as an energy-intensive system, buildings are also environmental pollutants and contribute nearly 33 % of the

* Corresponding author. Tel.: +4745004121
E-mail address: Habtamu-Bayera.Madessa@hioa.no

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greenhouse gas emissions [2,3]. Numerous studies have been conducted to find ways to reduce buildings' energy consumption and increase the utilization of renewable energy resources.

Energy from the ground is clean and freely available renewable energy that can be utilized for space heating and hot water production in buildings. A ground source heat pump (GSHP) consists of a ground loop (primary loop), a heat pump loop, and a secondary loop system that utilizes low-grade heat from the ground, increases the quality of the energy, and transfers it into building. The ground loop transfers heat from the ground borehole to the circulating fluid and supplies the collected heat to the heat pump loop. The pipes in the ground borehole can be buried vertically or horizontally. However, GSHP systems with vertical pipe arrangements in the ground are most widely used in European countries for space heating and cooling. In the Scandinavian countries, for example, the majority of the ground loops are vertically configured [4], and globally, this type of installation has increased dramatically over the past two decades [5]. A vapor compression heat pump, which consists of an evaporator, compressor, condenser, and expansion device, increases the energy quality of the collected heat from the ground and delivers the heat to the secondary loop. The secondary loop represents the application area with an interior heat distribution system that supplies heat to a building via a buffer tank.

Energy and exergy performance analysis of any given system is based on the laws of thermodynamics. The first law of thermodynamics helps analyse the energy performance, while the second law of thermodynamics is used to investigate the exergy efficiency of the system and its components. The coefficient of performance (COP) is an efficiency term that is most widely used to indicate the performance of a heat pump and is defined as the ratio of the heat delivered by the heat pump to the electrical energy expended at its compressor. Owing to stable heat sources and heat sink temperatures, GSHPs generally have higher COPs [6].

This paper presents the energy and exergy performance of a ground source heat pump coupled with a vertical ground loop pipes for heating application. The study investigates the influence of the ground depth, the mass flow rate of brine solution, types of refrigerants used as a working fluid, the heat delivery temperature at the condenser of the heat pump, and the dead state of the environment.

Nomenclature

A	area [m^2]
c_p	specific heat capacity at constant pressure [J/kg K]
COP	coefficient of performance [-]
d	tube diameter [m]
D	hydraulic diameter [m]
Exd	exergy destruction [W]
f	friction factor [-]
G	mass flux [kg/s m^2]
h	specific enthalpy [kJ/kg]
hc	convective heat transfer coefficient [W/m^2 K]
k	thermal conductivity [W/ m K]
L	length [m]
$\dot{m}$	mass flow [kg/s]
Pr	Prandtl number [-]
Q	rate of heat transfer [W]
Re	reynolds number [-]
T	temperature [^{o} C]

U overall heat transfer coefficient [W/m^2 K]
W work [kJ]
x quality [-]

Greek symbols

Ψ flow exergy [kJ/kg]
η thermal efficiency [-]
μ dynamic viscosity [m^2/s]

Subscripts

b brine
c compressor
con condenser
ev evaporator
g ground
i inside
is isentropic
o outside
p pump
tot total
w water
wf working fluid
0 dead state

2. Mathematical modelling

As shown in Fig. 1, the GSHP system consists of a ground borehole, an evaporator, a compressor, a condenser, an expansion valve, and pumps at the primary and secondary cycles. The brine solution, at state point 5*, is pumped to extract heat from the ground, and the brine at state point 6 is used to superheat the working fluid (from state point 4 to state point 1) of the heat pump in the evaporator. The working fluid is further compressed at a higher temperature and pressure in a compressor before it rejects its heat in the condenser (state point 2). The rejected heat in the condenser, depending on the energy quality, can be utilized for space heating or hot water in a building. As the working fluid flows through the expansion valve (state point 3), the fluid's pressure and temperature drop to the evaporator pressure (state point 4). The low-pressure and low-temperature working fluid that enters the evaporator takes heat from the brine, and the cycle repeat again.

The ground loop consists of a U-turn vertical bore heat exchanger, inserted vertically into the ground, and a circulating pump (pump 2). The heat exchanger is made from high-density polyethylene material, with thermal conductivity of 0.35 W/(m*K) and an internal and external diameter of 0.032 m and 0.04 m, respectively. The loop is filled with 35% concentration brine (an ethyl-ethanol and water mixture). In this work, the ground temperature at a depth of 30 m assumed to be 1–2 °C higher than the mean annual outdoor air temperature and increases linearly with about 1–3 °C per 100 m beyond the 30 m depth. The ground loop consists of a U-turn vertical bore heat exchanger, inserted vertically into the ground, and a circulating pump (pump 2). The heat exchanger is made from high-density polyethylene material, with thermal conductivity of 0.35 W/(m*K) and an internal and external diameter of 0.032 m and 0.04 m, respectively.

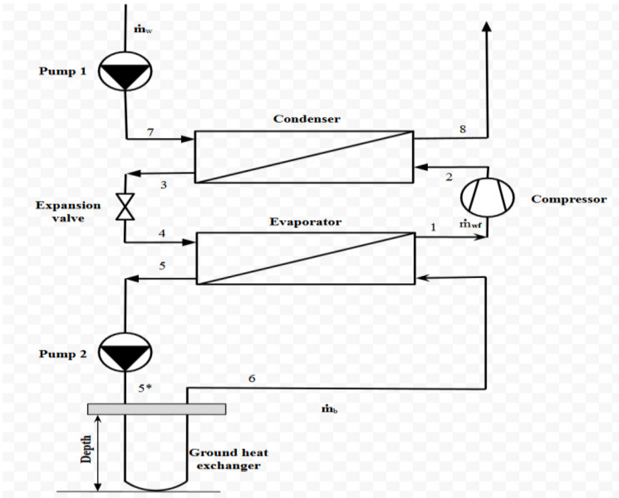


Fig.1. Ground source heat pump showing its different components and state points of the working fluids.

The loop is filled with 35% concentration brine (an ethyl-ethanol and water mixture). In this work, the ground temperature at a depth of 30 m assumed to be 1–2 °C higher than the mean annual outdoor air temperature and increases linearly with about 1–3 °C per 100 m beyond the 30 m depth.The mathematical representations of each component in the system are formulated at steady state conditions based on the mass balance, energy balance, and exergy balance.

Ground source:

$$\dot{m}_{5^*} = \dot{m}_6 = \dot{m}_b \tag{1}$$

$$\dot{Q}_g = \dot{m}_{brine}\, c_{p,b}\,(T_6 - T_{5^*}) \tag{2}$$

$$\dot{E}_{xd} = \dot{m}_b\,(\psi_{5^*} - \psi_6) + \dot{Q}_g(1 - \frac{T_o}{T_g}) \tag{3}$$

Pump at the primary cycle:

$$\dot{W}_p = \frac{\dot{m}_{wf}\,(h_{5^*} - h_5)}{\eta_p} \tag{4}$$

$$\eta_p = \frac{h_{5^*,is} - h_5}{h_{5^*} - h_5} \tag{5}$$

Evaporator:

$$\dot{m}_4 = \dot{m}_1 = \dot{m}_{wf} \tag{6}$$

$$\dot{Q}_{ev} = \dot{m}_{wf}\,(h_1 - h_4) = \dot{m}_{brine}\,(h_6 - h_5) \tag{7}$$

$$\dot{E}_{xd} = \dot{m}_{wf}\,(\psi_4 - \psi_1) + \dot{m}_{brine}\,(\psi_{5^*} - \psi_6) \tag{8}$$

Compressor:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_{wf} \tag{9}$$

$$\dot{W}_c = \frac{\dot{m}_{wf}\,(h_{2s} - h_1)}{\eta_c} = \dot{m}_{wf}\,(h_2 - h_1) \tag{10}$$

Condenser:

$$\dot{m}_2 = \dot{m}_3 = \dot{m}_{wf} \tag{11}$$

$$\dot{Q}_{co} = \dot{m}_{wf}\,(h_2 - h_3) = \dot{m}_w\,(h_8 - h_7) \tag{12}$$

$$\dot{E}_{xd} = \dot{m}_{wf}\,(\psi_2 - \psi_3) + \dot{m}_w\,(\psi_7 - \psi_8) \tag{13}$$

Expansion valve:

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_{wf} \tag{14}$$

$$h_3 = h_4 \tag{15}$$

$$\dot{E}_{xd} = \dot{m}_{wf}\,(\psi_3 - \psi_4) \tag{16}$$

Pump at the secondary cycle:

$$W_p = \dot{m}_w\,(h_{8^*} - h_8) \tag{17}$$

$$\eta_p = \frac{h_{8^*,is} - h_8}{h_{8^*} - h_8} \tag{18}$$

The heat transfer area of the condenser is calculated using the logarithmic mean temperature difference (LMTD) method. In the calculation, it is assumed that the condenser is a double-pipe heat exchanger, and for computational purposes, the length of the condenser tubes is divided into an equal number of small elements so that the temperature and the vapor quality of each elements are assumed to be constant for the single phase and the two phases, respectively. The thermo-physical properties of each element are then evaluated for the pressure and either temperature or vapor quality.

$$\dot{Q}_{j,wf} = \dot{m}_{wf}(h_{wf,j+1} - h_{wf,j}) \tag{19}$$

$$\dot{Q}_{j,w} = \dot{m}_b(h_{w,j+1} - h_{w,j}) \tag{20}$$

$$\dot{Q}_{j,wf} = \dot{Q}_{j,w} \tag{21}$$

$$\dot{Q}_{j,wf} = U_{con} A_{con}\, LMTD \tag{22}$$

$$LMTD = \frac{(T_{wf,j+1} - T_{w,j+1}) - (T_{wf,j} - T_{w,j})}{ln\left[\frac{(T_{wf,j+1} - T_{w,j+1})}{(T_{wf,j} - T_{w,j})}\right]} \tag{23}$$

$$\frac{1}{UA_{con}} = \frac{1}{h_{c,b} A_b} + \frac{\ln(\frac{d_o}{d_i})}{2\ \pi\ L\, k} + \frac{1}{h_{c,wf} A_{wf}} \tag{24}$$

Moreover the heat transfer process in the in the condenser yields two phases for the refrigerant. Thus, the heat transfer coefficients of the single-phase water are approximated based on Petukhov's correlations, as shown in equation (25) while the heat transfer coefficient for the two-phase refrigerant is approximated by Gungor and Winterton [7] as defined in equation (27).

$$h_{c,w} = \frac{k}{D}\left[\frac{\frac{f}{8} * Re * Pr}{12.7\left(\frac{f}{8}\right)^{0.5}\left(\mathrm{Pr}^{\frac{2}{3}} - 1\right) + 1.07}\right] \quad (25)$$

$$f = \frac{1}{(1.82 \log Re_b - 1.64)^2} \quad (26)$$

$$h_{c,wf} = 0.023\left[G(1-x)\frac{d}{\mu}\right]^{0.8} Pr^{0.4} \ \frac{k}{d}\left\{(1-x)^{0.8} + \frac{3.8 * x^{0.76} \ (1-x)^{0.04}}{Pr^{0.38}}\right\} \quad (27)$$

At the primary loop, the heat transfer is also modelled with the LMTD method, and the brine exit temperature from the ground loop is calculated by [8]:

$$T_6 = T_s - (T_s - T_5) \exp(-1/ (\dot{m}_{brine} \ c_{p,brin} \ R_{tot}) \quad (28)$$

The second law efficiency of the system is based on the exergy destruction:

$$\eta_{II} = \frac{(1 - \frac{T_0}{T_H})\dot{Q}_{con}}{\dot{W}_{tot}} \quad (29)$$

The numerical calculations were performed using Engineering Equation Solver (EES) software. The software has a thermo-physical properties databank for different types of working fluids and is a useful tool for solving thermodynamics, and heat transfer problems.. The following general assumptions are considered in the simulation:

- The system operates at steady state conditions, and changes in kinetic and potential energy are negligible.
- Heat losses to the environment for each of the system's components are negligible.
- The pinch temperature difference at the evaporator and the condenser is 5 K.
- The flow is assumed to be incompressible, and the isentropic efficiency of the pumps and the compressor is 85%.
- Constant thermal conductivity and thermal diffusivity of the ground.

In the simulation, it is also assumed that the brine inlet temperature to the ground is 2 °C, and the operating pressure for the evaporator is kept at 103 kPa. The working fluid is subcooled by 5 °C at the condenser and superheated by 5 °C at the evaporator. The pinch point temperature difference for the evaporator and the condenser is considered to be 5 °C.

3. Results and discussion

3.1. Effect of the ground depth

In order to investigate the effect of the depth on the exit temperature of the brine and energy collected from the ground, a simulation was carried out with the mass flow of the brine ranging between 0.08 and 0.2 kg/s, by assuming that the soil temperature is equal to the pipe's outer temperature. As shown in Fig. 2, the exit temperature decreases exponentially with increasing the mass flow rate of the brine. At a mass flow rate of 1 kg/s, a brine exit temperature is between 4 °C and 6 °C and absorbed energy is between 8 kW and 16 kW can be obtained.

The simulation results support that the deeper the ground heat source, the more energy is collected with a higher exit temperature from the ground. However, the temperature difference among the three depths decreases as the mass flow rate increases. The heat extraction rate increases with the increasing mass flow rate until a certain point. For example, for the 100 m depth, a mass flow rate of about 2kg/s gives the optimal result.

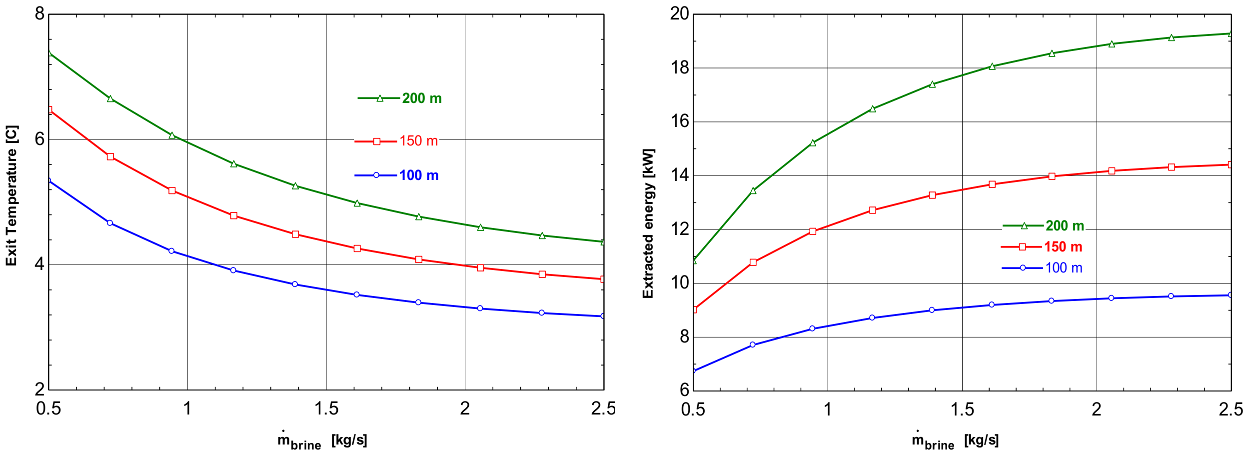


Fig. 2. Exit temperature of the brine (left) and the extracted energy from the ground at different depths of the ground and mass flow rates of the brine (right).

3.2. Effect of the evaporator temperature and working fluids

End-use temperature at the given evaporator temperature is an important parameter for a study of the performance of a heat pump. In this study, for different evaporator temperatures, the effectiveness of the GSHP for three different types of end-use applications were studied using four different types of refrigerants. The first case involved a space heating system with a radiator with a supply and return temperature of 60 °C and 40 °C, respectively (called the 60/40 case). The second case is floor heating of a building space with a supply temperature of 40 °C and a return temperature of 30 (called the 40/30 case). The third case is a hot water production system with cold water inlet temperature of 5 °C and an exit temperature of 60 °C (called the 60/5 case).The refrigerants used in the GSHP have different thermodynamics properties as shown in Table 1. EES has detailed data about the working fluids, including the brine solution.

Table 1. Thermodynamic properties of the working fluids.

Working fluid	Normal boiling point [°C]	Critical temperature [°C]
R-134a	–26.1	101.06
R-290	–42.1	96.675
R-600a	–11.7	134.67
R-717	–33.3	132.25

As shown in fig. 3 and fig.4, for a fixed condenser temperature, as the evaporation temperature increases from –10 °C to 0 °C, the heat pump's COP increases sharply for all refrigerants.

This is mainly due to the heat transfer at the evaporator. All refrigerants showed nearly the same performance (COP) to deliver energy for the floor heating system, with an increase of COP by 1 for the entire evaporator temperature range. However, all the refrigerants showed better performance for the hot water production application, i.e., heating cold water from 5 °C to 60 °C. Comparing the 60/40 case and 60/5, the difference in the COP revealed that the inlet temperature of the water at the condenser has a significant impact on the performance of the heat pump.

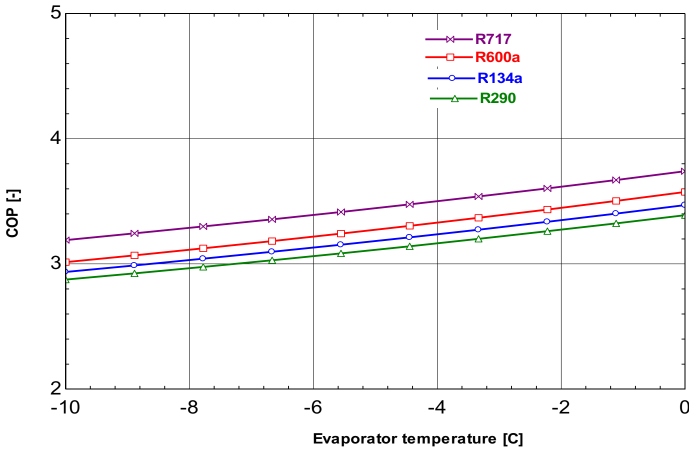
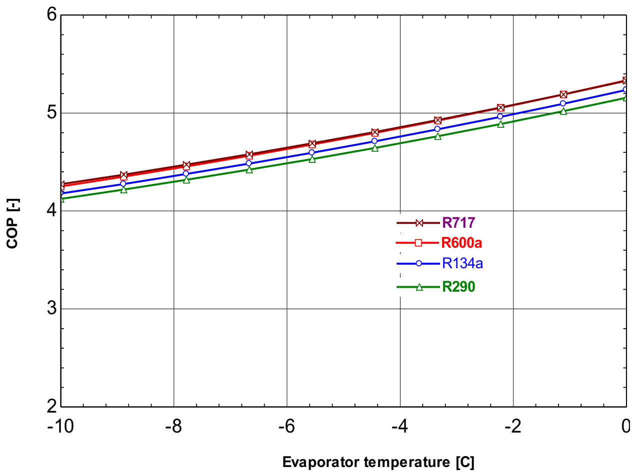


Fig. 3. COP vs. evaporative temperature for different working fluids: 60/40 case.

This corresponds to the amount of energy rejected from the condenser that is higher for hot water production than for the heating systems. Therefore, the effect of the condensing temperature and water inlet temperature on the system performance is obvious.

As shown in Fig. 4 (left), the COP of the heat pump for hot water production varies between 4 and 4.5 regardless of the working medium. Moreover, comparing the four refrigerants, R717 performed the best and R290 the worst.



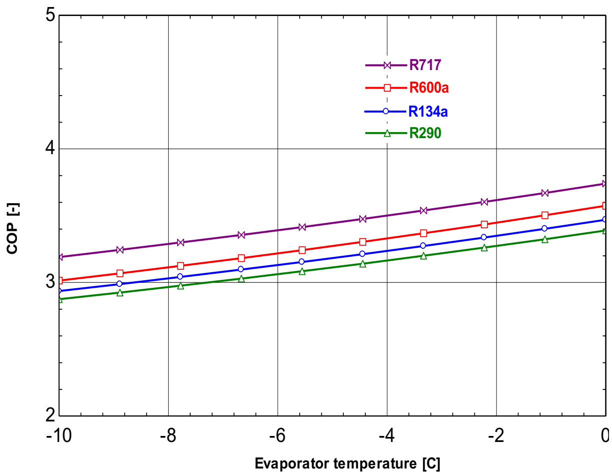


Fig. 4. COP vs. evaporative temperature for different working fluids: 40/30 case (left) and 5/60 case (right).

An additional investigation of the impacts of evaporator temperatures and working fluids on the condenser heat transfer area is shown in Fig. 5 for the 60/40 case. It has been observed that for a given working fluid, the evaporation temperature has no impact on the condenser area. This implies that the evaporation temperature does not have an impact on the LMTD of the condenser. However, comparing the four refrigerants, refrigerant R600a requires a large heat transfer area (2.2 m^2), and R717 requires a small heat transfer area (1.6 m^2) to transfer the same amount of heat, which showed about 27 % heat transfer area difference between the two working media. This is related to the heat of evaporation of the working fluids.

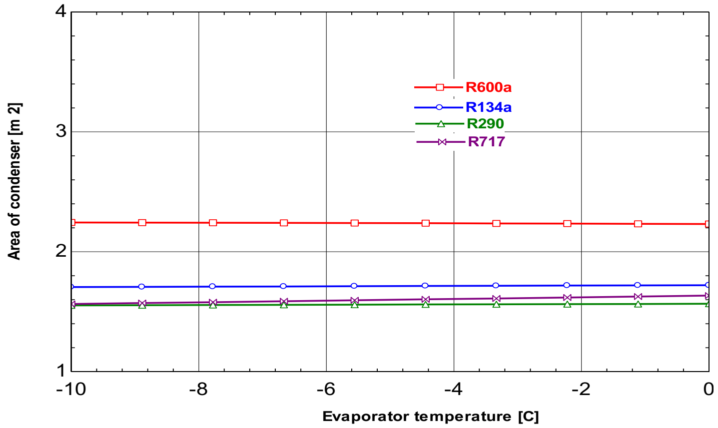


Fig. 5. Condenser area vs. evaporative temperature for different working fluids, for the 60/40 case.

3.3. Effect of the dead state temperature

Exergy represents work potential from the available energy relative to a reference state condition, called the dead state.

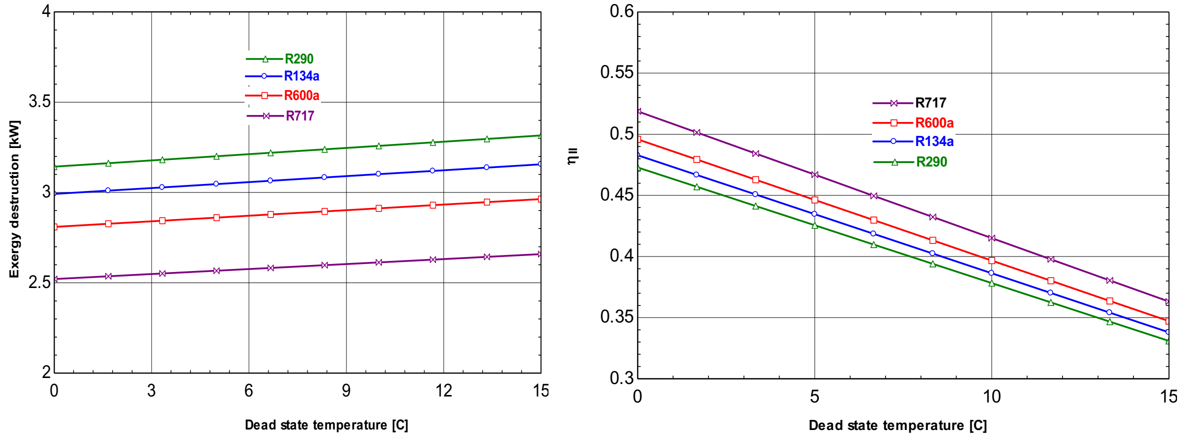


Fig. 6. Exergy destruction vs. dead state temperature (left) and second law efficiency vs. dead state temperature (right).

At the dead state, exergy is zero. The result depicted in fig. 6 shows the exergy destruction and exergy efficiencies of the GSHP system as a function of the dead state temperature range, 0 °C to 15 °C. The exergetic efficiencies of the system decrease due to the increase in exergy destruction following the rise in the temperature difference between the dead state temperature and the system temperature. For example, for refrigerant R717 the exergy efficiency decreased from 52 % to 37 % for the given dead state temperature range. This result shows that the dead state temperature has a significant impact on the performance of the system. The results also indicate that GSHP systems performed better for all refrigerants at a 0 °C dead state temperature with exergy destruction between 2.5 kW and 3.4 kW.

4. Conclusion

In this study, a numerical study was performed in order to investigate the performance of a ground source heat pump (GSHP) with various working fluids (R-134a, R-290, R-600a, and R-717) and end-use application areas. Different parameters were investigated, and the major findings are summarized as follows:

- The depth of the ground affects the evaporator temperature, which in turn affects the COP of the GSHP system.
- The four refrigerants used as working fluid for the GSHP revealed different performances, though all showed the same trends for the different parameters. R-717 showed the best performance among the investigated working fluids.
- All the working fluids show best performance for floor heating application.
- Selection of refrigerants as a working fluid for GSHP needs through thermodynamics analysis.

It should be noted that this work involves assumptions, particularly with the variation of ground temperature at different depth of the ground. Therefore, all the assumptions should be considered carefully for future work. Further analysis with validation of the results, as well as including additional parameters such as CO_2 as a working fluid, varying the pressure of the condenser, and varying the pinch temperature difference will be included in future work.

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