A COMPETITIVE STUDY OF A GEOTHERMAL HEAT PUMP EQUIPPED WITH AN INTERMEDIATE ECONOMIZER FOR R134a AND R513a WORKING FLUIDS

by

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At temperatures below 60 °C, the best way to use geothermal sources for heating is to use heat pumps. A heat pump can provide air conditioning for a residential, commercial, etc., all year round by heating in winter and cooling in summer using a low temperature source. Also, a heat pump can be used for water distillation through evaporation. The ground source heat pump with a high COP and low temperature thermal energy sources is one of the best technologies for using RES. In the present study, the effects of changing ambient temperature and soil temperature on a heat pump's overall COP and energy efficiency are investigated using a simulated geothermal heat pump with an economizer. The system's thermodynamic simulation is first performed in the engineering equation solver software for R134a and R513a working fluids. The exergy destruction of different components for both working fluids was calculated and displayed as a figure. The COP of the heat pump for R134a working fluid is equal to 3.916, equal to 3.729 for R513a working fluid, which indicates that R134a fluid has about 5% better performance. The COP of the system for R134a working fluid is equal to 3.662, which is equal to 3.504 for R513a working fluid, which indicates that R134a fluid has about 4.5% better performance.

Key words: *exergoenvironmental, exergy destruction, geothermal, heat pump, R134a, R513a*

Introduction

In recent years, cooling and heating have become one of the most important parts of energy consumption in buildings, so in tropical areas, cooling and cold regions, heating has tak-

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en an important part of energy consumption. Even in regions and countries that do not face the energy consumption crisis in buildings, a diversification strategy is the most common solution reduce dependence on a specific energy source [1]. Human's growing need for energy and the subsequent use of fossil fuels has led to many problems, among which we can mention the environmental issues caused by using fossil fuels and the finiteness of these resources. Such problems have made using renewable resources such as solar, wind, geothermal and recycled energy even more developed [2]. The growing need to improve energy efficiency and sustainable energy systems has increased the use of geothermal heat pumps. Geothermal heat pumps are a sustainable alternative to heat or cool buildings and homes in most climates [3]. A geothermal heat pump uses energy stored in the ground to transfer heat for heating or cooling in buildings. Geothermal heat pumps can be used effectively in many weather conditions to provide space heating, cooling and domestic hot water. The use of geothermal heat pumps is not only optimal, but these pumps require little cost for repairs and maintenance and can be expected to be used efficiently for up to 20 years [4].

Examining different arrangements of geothermal heat pumps and the importance of their working fluids has always been of great importance to researchers' works. Pishkariahmadabad [5] studied a geothermal heat pump for heating with an economizer. Then they evaluated the effects of variations in various parameters like evaporator pressure, condensation pressure, and intermediate pressure on the heat pump and total COP and energy efficiency. Using engineering equation solver (EES) software, Zhao *et al.* [6] created a heat pump with an intermediate economizer and a ground source energy drive. Then, for five working fluids, including R134a, R12, R152a, R1234yf, and R1234ze(E), the impact of altering various parameters, such as the pressure of various cycle components, the degree of superheating at the evaporator outlet, and the degree of subcooling at the condenser outlet, is examined. Three indicative technical parameters for paralleled-loop exhaust air heat pump R and D, including nominal heating-cooling capacity, maximum required fresh air to return air ratio, and system energy efficiency grades, were calculated and summarized according to the demand for indoor thermal comforts using statistic method based on the simulation results by Ji *et al.* [7] for a typical rural ultra-low energy building in five different climatic regions of China. Yamankaradeniz [8] examined the Turkish climate-related performance of a dual-tank heat pump system with Sun assistance. Software for transient system modelling was used to model and simulate this system and its parts. The device was made to provide a restaurant with domestic hot water and heat and cool the water throughout the year.

The application of Second law analysis to the construction of energy systems has significantly increased in recent times [9]. A plan based on the first law of thermodynamics only looks at the system's energy balance. Still, the Second law of thermodynamics offers a better perspective of the system's effectiveness. An effective instrument for designing, improving, and raising the efficiency of energy systems is exergy analysis. This kind of analysis can be beneficial in determining the primary cause of irreversibility and minimizing the production of entropy in different processes [10]. Exergy analysis is a proper thermodynamic technique for using the mass and energy conservation equations and the second law of thermodynamics for constructing and analyzing heating systems, according to Dincer and Rosen's studies [11]. He bought the equipment as well. The system's thermodynamics are optimized as irreversibility is reduced [12, 13].

The subject of this article is a comparative study of the performance of two working fluids, R134a and R513a, for a geothermal heat pump equipped with an economizer in heating mode in winter. All heat pump processes have been thoroughly investigated, and various exergy indices have been analyzed and calculated in all components and the system. The novelty

of the present work is first to examine the effects of changes in ambient temperature and soil temperature on the performance quality of the proposed heat pump and then calculate the exergy destruction of various components of the system and the exergeoenvironmental analysis of heat. Pump system using the results of exergy analysis, the general goals of this research are:
- Designing a geothermal heat pump system equipped with an economizer with R134a and R513A working fluids in the winter operating mode and simulating the current system in the EES software.
- A competitive investigation of the effect of changes in ambient temperature and soil temperature on the COP and exergy efficiency of the heat pump and the whole system
- Calculation of exergy destruction of different system components for every working fluid.
- Exergeoenvironmental analysis of the proposed heat pump system for every working fluid.

The present work simulates a geothermal heat pump equipped with an intermediate economizer in the EES software. Two R134a and R513a fluids are used as working fluids of the proposed system. In the next step, the output results from the energy and exergy analyses and the exergoenvironmental analyses of the two fluids are compared. Then, the effects of soil and ambient temperature changes on system design parameters for both proposed fluids are studied. Finally, the exergy destruction of different components for both working fluids is calculated and displayed as a diagram.

System modelling

Figure 1 shows the schematic diagram of the studied geothermal heat pump system. As stated, the ground is a good source of heating in the cold months of the year, and the ground's heat can be used to provide heating for buildings and can also be used optimally for cooling in the hot months of the year. The mechanical process of this device is that in the cooling phase, the hot air inside the room enters the device through the suction device, and after cooling, it is blown into the room. Inside the device, heat is transferred to the refrigerant. After the refrigerant passes through the relevant (refrigeration cycle), the heat is transferred to the water inside the ground coil, which a two-pipe exchanger installs inside the polyethylene pipes. On the contrary, in the heating phase, the heat required by the building is supplied by using the heat in the ground. Calculating the length of the pipe, designing the lay-out, and optimal arrangement of polyethylene pipes inside the ground is the most critical factor in increasing the efficiency of the geothermal heat pump system and reducing its installation cost. Figure 2 also shows the flowchart for solving the studied problem.

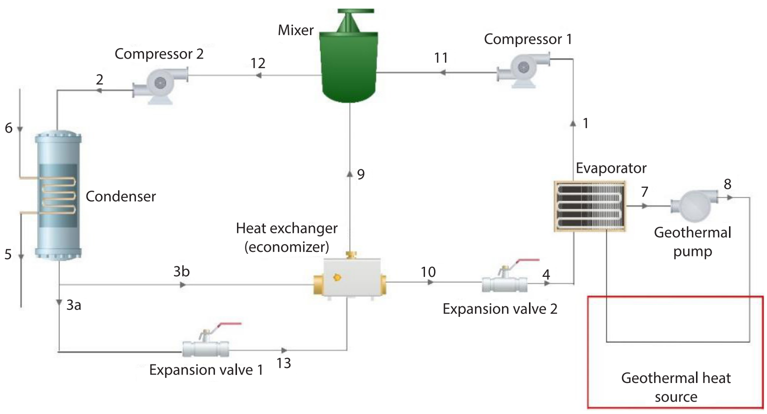

Figure 1. Cycle of the geothermal heat pump, shown schematically

To model the system, the following hypotheses are taken into account:

- The system is working in a steady-state mode.
- Any pressure loss in the components' heat exchangers and interface pipes is negligible.
- The isotropic efficiency of Compressors 1 and 2 and the ground source pump is 80%, 80%, and 90%, respectively [14].
- All the pumps and compressors are considered adiabatic.
- A mixture of water and alcohol with a bulk proportion of 30% alcohol in water and a freezing value of –13.1 °C makes up the geothermal fluid.
- Condenser pressure, which falls between 1100-3000 kPa, is the cycle's highest pressure and evaporator pressure, which falls between 350-500 kPa, is the cycle's lowest pressure.
- Points 5 and 6 consider the hot water's input and exit temperatures, of 35 °C and 30 °C. The intake and exit geothermal fluid temperatures are 7 °C and 11 °C, respectively, at Points 7 and 8. Points 7 and 8 always have temperatures lower than the earth's average temperature to effectively transport heat from the earth to the fluid.

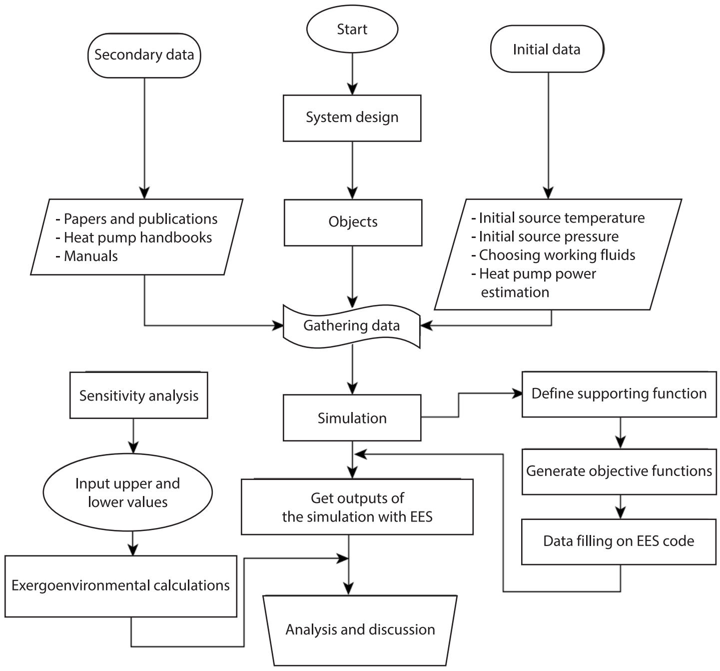

Figure 2. Flowchart of the problem solving process

The heat exchanger's outlet temperature is assumed to be 5 °C below the additional cooling. In the heating cycle, the water and antifreeze or refrigerant solution (which flows in the pipes to receive heat from the soil) enter the heat pump. In some systems that use underground-water or hot springs, this fluid enters the geothermal heat exchanger and gives its heat to the refrigerant.

In the evaporator, the heat causes the refrigerant to boil and turning it into low temperature steam. The return valve directs the refrigerant in a vapor state to the compressor. Then the steam is compressed, and its volume decreases. This decrease in volume increases the temperature of the refrigerant vapor. The next returnable valve directs the hot gas to the condenser. The refrigerant heat is given to water or air to provide the required heating by circulating through the internal duct system of the environment. After the refrigerant loses its heat in the condenser, it enters a pressure expansion valve where the pressure and temperature drop sharply. After that, the refrigerant re-enters the first heat exchanger to start the cycle again.

The cooling cycle is actually the opposite of the heating cycle of the heat pump. In this case, the refrigerant flow direction is changed using a four-way valve. Refrigerant receives heat from the internal environment and transfers it to the evaporator. Then the heat is transferred to the outside in the condenser by going through the reverse heating cycle.

Exergy loss shows the amount of exergy loss. This index is an absolute value that cannot be used to calculate the exergy consumption efficiency of various energy components or processes. Exergy efficiency shows the effect of the components and the whole geothermal heat pump system, and by using it, it is possible to achieve the effect of improving the efficiency of the components and the whole heat pump system on the amount of exergy consumption. At the same time, this index cannot show the ratio between the components and the whole geothermal heat pump system in exergy loss, so it cannot be used to determine which component in the geothermal heat pump system consumes less.

Governing equations

With constant flow conditions and the assumption of each cycle component as the control volume, the energy balance and the irreversibility relationship in each of the components are written [15]:

$$\dot{Q} - \dot{W} + \sum \dot{m}_{\text{in}} h_{\text{in}} + \sum \dot{m}_{\text{out}} h_{\text{out}} = 0 \tag{1}$$

$$\dot{I} = \dot{m} T_0 \frac{\mathrm{d}s_{\text{total}}}{\mathrm{d}t} \tag{2}$$

where $Q^{\cdot}$, $\dot{W}$, $\dot{m}$, and h are the rate of heat exchanged, work exchanged, mass-flow rate, and specific enthalpy, respectively, the subscripts in and out are related to the input flows to the control volume and the output flows from the control volume, respectively. In eq. (2), are irreversibili-ty, ambient temperature, and specific entropy, respectively:

$$\dot{I} = \dot{m} T_0 \left(\sum s_{\text{out}} - \sum s_{\text{in}} + \frac{\mathrm{d}s_{\text{system}}}{\mathrm{d}t} + \sum \frac{q_k}{T_k} \right) \tag{3}$$

where T_k is the temperature of the heat source and q_k – the amount of heat transfer per unit of mass between the heat source and the working fluid, in the case that the system reaches a steady-state, it becomes $\mathrm{d}s_{\text{system}}/\mathrm{d}t = 0$.

The internal irreversibilities in the system components are calculated using the $\sum s$ term and the external irreversibilities are calculated using the $\sum q/t$ term. Also, in this research, for simplicity, internal irreversibilities corresponding to pressure drop in system components such as heat exchangers and pipes are omitted. The aforementioned equations are written separately for each cycle component to obtain work and heat exchange and irreversibility. Also, to obtain exergy efficiency and exergy destruction fraction in different components of the cycle [15]:

$$\eta_{\text{ex,i}} = \frac{\dot{E}x_{\text{desired,out}}}{\dot{E}x_{\text{used}}} \tag{4}$$

$$y_{\text{ex,i}} = \frac{\dot{I}_{\text{i}}}{\dot{I}_{\text{tot}}} \tag{5}$$

where η_{ex} is the exergy efficiency, $\dot{E}x_{\text{desired,out}}$ – the exergy corresponding to the desired output currents, and $\dot{E}x_{\text{used}}$ – the exergy corresponding to the input supply currents, and in eq. (5), y_{ex} is the exergy destruction fraction [16]:

$$\dot{m}_{\text{ref}} = \frac{\dot{Q}_{\text{cond}}}{h_2 - h_3} \tag{6}$$

$$\dot{m}_{\text{sup}} = \dot{m}_{\text{ref}} \frac{h_{10} - h_3}{h_{13} + h_{10} - h_3 - h_9} \tag{7}$$

$$\dot{m}_{\text{main}} = \dot{m}_{\text{ref}} \frac{h_9 - h_{13}}{h_3 + h_9 - h_{13} - h_{10}} \tag{8}$$

where $\dot{m}_{\text{ref}}$, $\dot{m}_{\text{sup}}$, and $\dot{m}_{\text{main}}$ are mass-flow rate through the condenser, expansion valve 1, and evaporator, respectively [16], and $\dot{Q}_{\text{cond}}$ is the rate of heat exchanged in the condenser:

$$\dot{Q}_{\text{cond}} = \dot{m}_2 h_2 - \dot{m}_2 h_3 \tag{9}$$

In the present study, the system capacity is assumed to be 100 kW ($\dot{Q}_{\text{cond}} = 100$ kW). The work of the geothermal pump is obtained from [17]:

$$\dot{W}_{\text{pump,gl}} = \frac{\dot{V}_{\text{w,gl}} \Delta P_{\text{w,gl}}}{\eta_{\text{pump}}} \tag{10}$$

where $\dot{V}_{\text{w,gl}}$ is the volume flow rate of the fluid in the geothermal ring, η_{pump} – the isentropic efficiency of the pump, and $\Delta P_{\text{w,gl}}$ – the pressure drop of the fluid in the geothermal ring, which is obtained:

$$\Delta P_{\text{w,gl}} = f\left(\frac{l_{\text{gl}} \rho_{\text{w,gl}} V_{\text{w,gl}}^2}{2d_{\text{i,gl}}}\right) \tag{11}$$

$$f = \left(0.79 \ln \text{Re}_{\text{w,gl}} - 1.64\right)^{-2} \tag{12}$$

where $\rho_{\text{w,gl}}$ is the fluid density, $V_{\text{w,gl}}$ – is the fluid velocity, and $\text{Re}_{\text{w,gl}}$ – the Reynolds number of the fluid inside the geothermal exchanger, as well as $d_{\text{i,gl}}$ and l_{gl}, respectively, are the internal diameter and length of the geothermal tube, and the length of the geothermal tube is calculated:

$$l_{\text{gl}} = \frac{\left(\dfrac{\dot{Q}_{\text{eva}}}{N_b}\right)}{T_{\text{s}} - T_{\text{w,gl}}} \left(\frac{1}{\pi d_{\text{i,gl}} h_{\text{w,gl}}} + \frac{\ln\left(\dfrac{d_{\text{o,gl}}}{d_{\text{i,gl}}}\right)}{2k_{\text{gl}}} + \frac{F}{U_{\text{s}} d_{\text{o,gl}}} \right) \tag{13}$$

where $T_{\text{w,gl}}$ is the average temperature of the fluid inside the geothermal tube, N_b and $d_{\text{o,gl}}$ are the number and outer diameter of U-shaped geothermal tubes placed in parallel, $h_{\text{w,gl}}$ – the heat transfer of the fluid displacement inside the geothermal tube, and k_{gl} – the heat transfer coefficient of the tube. Geothermal and F is the ratio of the number of hours of maximum load to the total number of hours required for heating in the ventilated space. Also, T_{s} and U_{s} are soil temperature and soil heat transfer coefficient, respectively, and $\dot{Q}_{\text{eva}}$ is the rate of heat exchanged in the evaporator. Finally, the COP and exergy efficiency of the heat pump and the whole system are obtained from [18]:

$$COP_{\text{hp}} = \frac{\dot{Q}_{\text{cond}}}{\dot{W}_{\text{comp1}} + \dot{W}_{\text{comp2}}} \tag{14}$$

$$COP_{\text{tot}} = \frac{\dot{Q}_{\text{cond}}}{\dot{W}_{\text{comp1}} + \dot{W}_{\text{comp2}} + \dot{W}_{\text{pump,gl}}} \tag{15}$$

$$\eta_{\text{ex,hp}} = \frac{Ex_{\text{in,cond}} - Ex_{\text{out,cond}}}{\dot{W}_{\text{comp1}} + \dot{W}_{\text{comp2}}} \tag{16}$$

$$\eta_{\text{ex,tot}} = \frac{Ex_{\text{in,cond}} - Ex_{\text{out,cond}}}{\dot{W}_{\text{comp1}} + \dot{W}_{\text{comp2}} + \dot{W}_{\text{pump,gl}}} \tag{17}$$

The work of Compressors 1 and 2 is obtained:

$$W_{\text{comp1}} = \dot{m}_{\text{main}} \left(h_{11} - h_1 \right) \tag{18}$$

$$W_{\text{comp2}} = \dot{m}_{\text{ref}} \left(h_2 - h_{12} \right) \tag{19}$$

Exergoenvironmental analysis is used to assess system performance from an environmental point of view. Based on evaluations of the thermodynamic and ambient conditions, the analyses' guiding principles are formed. This is how the exergoenvironment factor can be expressed [19]:

$$f_{\text{ei}} = \frac{\dot{E}x_{\text{tot.des}}}{\sum \dot{E}x_{\text{in}}} \tag{20}$$

where subscripts in and tot.des are the stand for system input and total exergy destruction, respectively, while $\dot{E}x$ is the exergy rate. The exergy stability factor can be determined [19]:

$$f_{\text{es}} = \frac{\dot{E}x_{\text{tot.des}}}{\dot{E}x_{\text{tot.out}} + \dot{E}x_{\text{tot.des}} + 1} \tag{21}$$

where the subscript tot.out is the system's total energy destruction. As a result, the total exergy destruction rate is directly correlated with both the exergoenvironment factor and the exergy stability factor. This equation yields the exergoenvironmental effect coefficient [19]:

$$C_{\text{ei}} = \frac{1}{\eta_{\text{ex}}} \tag{22}$$

where η_{ex} is the system's exergy efficiency. Efficacy factor for environmental damage is calculated [19]:

$$\theta_{\text{ei}} = f_{\text{ei}} C_{\text{ei}} \tag{23}$$

However, the impact of exergoenviornmental enhancement can be estimated [19]:

$$\theta_{\text{eii}} = \frac{1}{\theta_{\text{ei}}} \tag{24}$$

Working fluids and initial conditions

As working fluids for the investigation, R134a and R513a have been chosen. Table 1 displays the working fluids' parameters. The following elements can be listed as some of the most significant ones that affect the choice of fluids: normal boiling point, critical temperature, and critical pressure. Each refrigerant's formula and chemical name are included in the table below, obtained from ANSI/ASHRAE Standard 34-2010: Standard 34-2010, Designation and Safety Classification of Refrigerants. The primary input variables for the current heat pump simulation are listed in tab. 2.

Table 1. Properties of the refrigerants [20-23]

Refrigerant	Molecular formula	Molecular weight [kgkmol^{-1}]	Normal boiling point [°C]	Critical temperature [°C]	Critical pressure [MPa]	Safety class	ODP	GWP
R134a	CH_2FCF_3	102	–26.1	101.1	4.059	A1	0	1300
R513a	$CF_3CF = CH_2/CF_3CH_2F$	108.4	–29.60	96.5	3.76	A1	0	573

Since their introduction about ten years ago, there has been discussion about the suitability of global warming potentials (GWP) as indexes representing the relative warming effects of emissions of various GHG. A particular mass of GHG GWP measures how much is thought to be contributed to global warming. A relative scale is used to relate a gas to a similar mass of CO_2 [24, 25]. Compared to the same quantity for CFC-11, ozone depletion parameter (ODP) is the shift in globally averaged column ozone caused by an exciting compound at a steady-state per unit mass emission rate [26].

Table 2. Initial conditions used for the simulation process in the software

Sign	Definition	Amount	Reference
P_0 [kPa]	Ambient Pressure	101	[5, 27, 28]
T_0 [°C]	Ambient Temperature	5	[5, 27, 28]
P_1 [kPa]	Evaporator Pressure	200	[5, 27, 28]
P_2 [kPa]	Condenser Pressure	400	[5, 27, 28]
P_9 [kPa]	Heat exchanger temperature	1200	[5, 27, 28]
T_{soil} [°C]	Soil Temperature	15	[5, 27, 28]
η_{comp1}	Compressor 1 efficiency	0.8	[5, 27, 28]
η_{comp2}	Compressor 2 efficiency	0.8	[5, 27, 28]
$\eta_{geopump}$	Geothermal pump efficiency	0.9	[5, 27, 28]
T_5 [°C]	Heated water temperature	35	[5, 27, 28]
T_6 [°C]	Entrance water temperature	30	[5, 27, 28]
T_7 [°C]	Geothermal pump entrance temperature	7	[5, 27, 28]
T_8 [°C]	Geothermal pump exit temperature	11	[5, 27, 28]
$\dot{Q}_{cond}$ [kW]	Condenser capacity	100	[5, 27, 28]
U_s [Wm^{-2}K^{-1}]	Soil heat transfer coefficient	12	[29]
F	Ratio of the number of hours of maximum load to the total number of hours	0.4	[29]
k_{gl} [Wm^{-1}K^{-1}]	Heat transfer coefficient of the tube	0.45	[17]
d_i [m]	Internal diameter	0.0218	[17]
d_o [m]	Outer diameter	0.0267	[17]
V [ms^{-1}]	Fluid speed inside the geothermal tube	1	[17]

Results and discussion

The EES software simulates all formulae for mass and energy conservation in parts and irreversibility equations in various parts of the heat pump cycle. This program has a subset of various fluid properties that can simulate the cycle (in this case, R134a and R513A). To analyze the parameter in the basic working mode, the cycle's high, medium, and low pressures are considered 1200 kPa, 400 kPa, and 200 kPa, respectively. It is supposed that the condenser output subcooling and the evaporator output overheating are both 5 °C. Tables 3(a) and 3(b) display the simulation results and values for the system's different points' mass-flow rate, pressure, temperature, enthalpy, and entropy.

Table 3(a). Properties of different states in basic mode (R134a)

State	Working fluid	Mass-flow rate [kgs^{-1}]	Pressure [KPa]	Temperature [°C]	Enthalpy [kJkg^{-1}]	Entropy [kJkg^{-1}]
1	R134a	0.5101	200	–5.093	248.7	0.9538
2	R134a	0.5343	1200	66.99	297.3	0.9843
3	R134a	0.5343	1200	41.29	110.2	0.4004
4	R134a	0.5101	200	–10.09	102.7	0.399
5	Water	4.782	250	35	146.8	0.5049
6	Water	4.782	250	30	125.9	0.4365
7	Geo-fluid	4.432	250	7	28.47	–
8	Geo-fluid	4.432	250	11	45.27	–
9	R134a	0.02421	400	21.1	266.9	0.9662
10	R134a	0.5101	1200	36.29	102.7	0.3766
11	R134a	0.5101	400	21.1	266.9	0.9662
12	R134a	0.5343	400	21.1	266.9	0.9662
13	R134a	0.02421	400	8.91	110.2	0.4115

Table 3(b). Properties of different states in basic mode (R513a)

State	Working fluid	Mass-flow rate [kgs^{-1}]	Pressure [kPa]	Temperature [°C]	Enthalpy [kJkg^{-1}]	Entropy [kJkg^{-1}]
1	R513a	0.5863	200	–8.236	372.1	1.664
2	R513a	0.6183	1200	58.41	416.3	1.692
3	R513a	0.6183	1200	39.35	254.6	1.184
4	R513a	0.5863	200	–13.24	247.3	1.184
5	Water	4.786	250	35	146.9	0.505
6	Water	4.786	250	30	126	0.4367
7	Geo-fluid	4.356	250	7	28.47	–
8	Geo-fluid	4.356	250	11	45.27	–
9	R513a	0.03204	400	15.77	388.8	1.676
10	R513a	0.5863	1200	34.35	247.3	1.16
11	R513a	0.5863	400	15.77	388.8	1.676
12	R513a	0.6183	400	15.77	388.8	1.676
13	R513a	0.03204	400	6.137	254.6	1.196

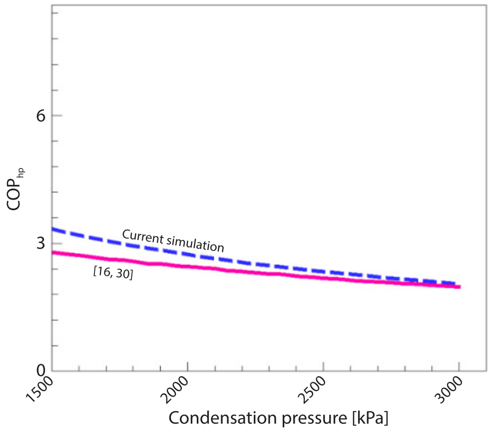

Figure 3. Comparison of the results of the present work with the results of [16, 30]; the effect of condensation pressure changes on the heat pump coefficient of performance

Figure 3 shows the simulation validation of the present work. To validate the results, the effects of high pressure variations on the heat pump's coefficient of performance with the same input cases have been compared with those of Self *et al.* [16, 30] The standard deviation of the results is adequate.

The modelling outputs for the system under study are shown in tab. 4. The heat pump's COP and energy efficiency for the total system is estimated based on the findings of tab. 4. In addition, environmental parameters including, exergoenvironment factor, exergy stability factor, exergoenvironmental impact coefficient, environmental damage effectiveness factor, and exergoenvironmental impact improvement factor, are also calculated and listed in the table.

Table 4. System output results

Parameters	Definition	Values for R134a	Values for R513a
m_{sup} [kgs^{-1}]	Mass-flow rate through expansion valve 1	0.02421	0.03204
m_{main} [kgs^{-1}]	Mass-flow rate through the evaporator	0.5101	0.5863
W_{comp1} [kW]	Compressor 1 work	9.26	9.815
W_{comp2} [kW]	Compressor 2 work	16.28	17
COP_{hp}	Coefficient of performance of heat pump	3.916	3.729
COP_{tot}	Coefficient of performance of total system	3.663	3.504
$\eta_{ex,hp}$	Exergy efficiency of heat pump	0.5184	0.4686
$\eta_{ex,total}$	Exergy efficiency of total system	0.485	0.4402
f_{ei}	Exergoenvironment factor	0.6566	0.6674
f_{es}	Exergy stability factor	0.4807	0.5252
C_{ei}	Exergoenvironmental impact coefficient	2.062	2.272
θ_{ei}	Environmental damage effectiveness factor	1.354	1.516
θ_{eii}	Exergoenvironmental impact improvement factor	0.7386	0.6595

The COP of the heat pump for R134a working fluid is equal to 3.916, equal to 3.729 for R513a working fluid, which indicates that R134a fluid has about 5% better performance.

The COP of the system for R134a working fluid is equal to 3.662, which is equal to 3.504 for R513a working fluid, which indicates that R134a fluid has about 4.5% better performance. The heat pump's exergy efficacy for the R134a working fluid is 51.84%, compared to 46.86% for the R513a working fluid, meaning that the R134a fluid performs about 10.63% better. The exergy efficiency of the system for the R134a working fluid is 48.5%, compared to 44.02% for the R513a working fluid, indicating a performance improvement of about 10.18% for the R134a fluid. The lower values of the exergoenvironment and exergy stability factors are preferable, according to equations 20 and 21, because these indices directly correlate with the

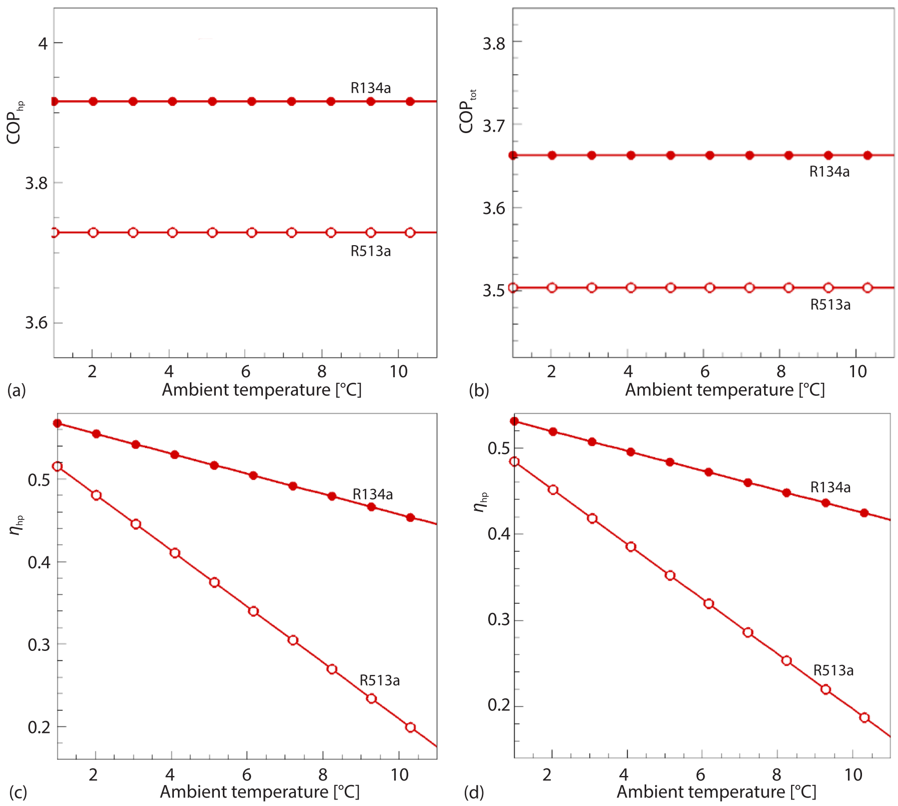

Figure 4. Changes in (a) COP_{hp}, (b) COP_{tot}, (c) heat pump exergy efficiency, and (d) total exergy efficiency relative to ambient temperature changes for the working fluids under study

overall exergy destruction rate. The exergoenvironment factor for the R134a working fluid is equal to 0.6566, which is equal to 0.6674 for the R513a working fluid, indicating that the R134a fluid performs better by about 1.64%. The exergy stability factor for the R134a working fluid is 0.4807%, which is 0.5252 for the R513a working fluid, which indicates that the R134a fluid has a better performance of about 9.26%.

Figures 4(a)-4(d) show the effects of ambient temperature changes on the COP of the heat pump, the COP of the total system, the exergy efficiency of the heat pump and exergy efficiency of the total system for two working fluids, respectively. According to figs. 4(a) and 4(b), changes in ambient temperature do not affect the COP of the heat pump and the COP of the total system for both working fluids. The reasons for this are the constant value of $\dot{Q}_{cond}$ on the one hand and the lack of influence of the work of the geothermal pump and compressors from the ambient temperature on the other hand. But the case is different regarding the effect of ambient temperature on exergy efficiency. As the ambient temperature increases, both the heat pump's exergy efficiency and the total system's exergy efficiency begin to decrease with a relatively large slope. As the ambient temperature increases from 1-11°C, the exergy efficiency of the heat pump for R134a decreases from 56.72-44.51%, and the exergy efficiency of the entire system decreases from 53.07-41.64%. About R513a, the exergy efficiency of the heat pump decreases from 51.55-17.55%, and the exergy efficiency of the total system decreases from 53-41.5%. Therefore, increasing the ambient temperature in winter will have a negative effect on the pump performance from the exergy point of view. Figures 5(a)-5(d) show the effects of soil temperature changes on the COP of the heat pump, the COP of the total system, the exergy efficiency of the heat pump and exergy efficiency of the total system for two working fluids, respectively. According to the figures, changes in soil temperature do not affect the COP and exergy efficiency of the heat pump for both working fluids.

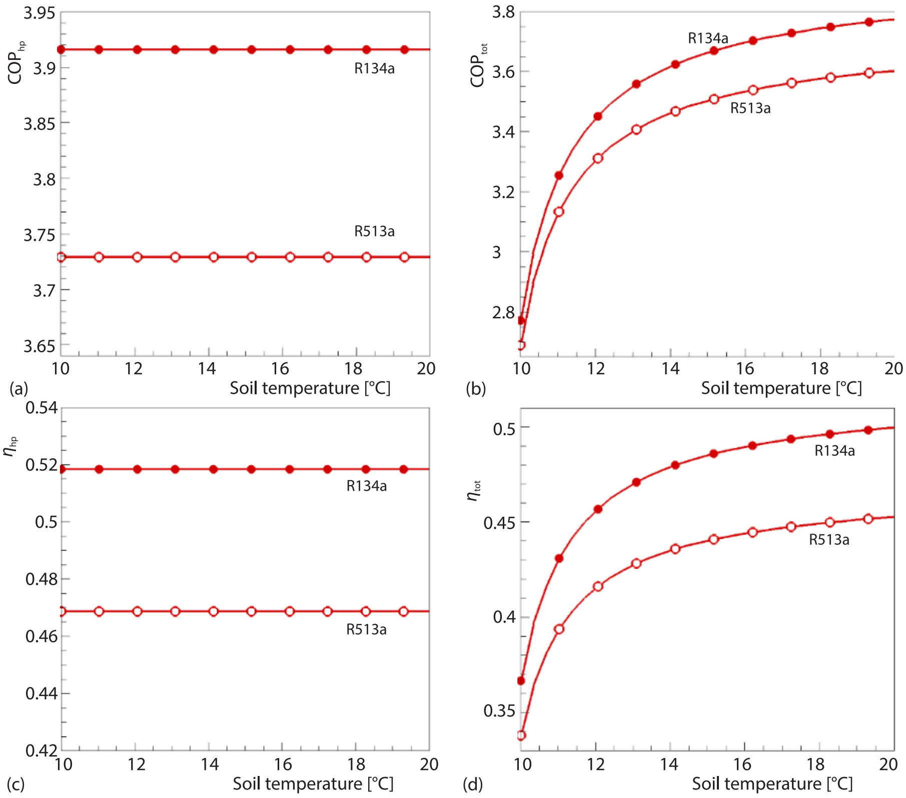

Figure 5. Changes in (a) COP_{hp}, (b) COP_{tot}, (c) heat pump exergy efficiency, and (d) total exergy efficiency relative to soil temperature changes for the working fluids under study

Table 5. Exergy destruction of different system components for the studied working fluids

Component	R134a	R513a
Evaporator	5.374	6.213
Condenser	4.242	3.569
Compressor 1	1.763	1.902
Compressor 2	2.684	2.873
Valve 1	0.07462	0.1039
Valve 2	3.18	3.847
Heat exchanger (economizer)	0.3525	0.419
Mixer	0	0
Total exergy destruction	17.67	18.93

Table 5 shows the exergy destruction of different system components for the studied working fluids. Total exergy destruction for R134a working fluid is equal to 17.67 kW and for R513a working fluid is equal to 18.93 kW. Both operating modes have the highest amount of exergy destruction in the evaporator.

Conclusion

The present work simulated a geothermal heat pump equipped with an intermediate economizer in the EES software. Two R134a and R513a fluids were used as working fluids of the proposed system. The output results from the energy and exergy analyses and the exergo-environmental analyses of the two fluids were compared. Then, the effects of soil and ambient temperature changes on system design parameters for both proposed fluids were studied. Finally, the exergy destruction of different components for both working fluids was calculated and displayed as a figure. The COP of the heat pump for R134a working fluid is equal to 3.916, equal to 3.729 for R513a working fluid, which indicates that R134a fluid has about 5% better performance. The COP of the system for R134a working fluid is equal to 3.662, which is equal to 3.504 for R513a working fluid, which indicates that R134a fluid has about 4.5% better performance. The heat pump's exergy efficacy for the R134a working fluid is 51.84%, compared to 46.86% for the R513a working fluid, meaning the R134a fluid performs about 10.63% better. The exergy efficiency of the system for the R134a working fluid is 48.5%, compared to 44.02% for the R513a working fluid, indicating a performance improvement of about 10.18% for the R134a fluid.

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Contribution

Yashar Aryanfar and Soheil Mohtaram contributed equally to this work and should be regarded as co-first authors.

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