The Derivation of System Parameters for a Ground Source Heat Pump (GSHP)

Unal Camdali*‡, Murat Tunc**, Mehmet Emre Erdem**, Erhan Tuncel*

*Ankara Yildirim Beyazit University, Engineering and Natural Sciences Faculty, Mechanical Engineering Department, 06100, Ankara, Turkey

**Istanbul Bilgi University, Faculty of Engineering and Natural Sciences, Santral-İstanbul Campus, 34060, Istanbul, Turkey

(unalcamdali@gmail.com, tuncmu@gmail.com, emreerdem79@hotmail.com, tuncelrhn@gmail.com)

‡Corresponding Author; Unal Camdali, Ankara Yildirim Beyazit University, Engineering and Natural Sciences Faculty, Mechanical Engineering Department, 06100, Ankara, Turkey, Tel: +90 312 906 22 33, unalcamdali@gmail.com

Received: 21.11.2023 Accepted:28.12.2023

Abstract- As well known the widespread use of nonrenewable energy sources such as fossil fuels cause environmental problems. Ground source heat pumps (GSHP) are highly efficient renewable energy system used for heating and cooling where cool deep ground source is used for higher efficiency.

In this study, a new simulation model has been developed and used for heat pumps for low global warming potential (GWP) refrigerants such as R134a and validated by the experimental data. Also, the system parameters of a GSHP are obtained theoretically for heating and cooling modes of a building, located in Bolu City, Turkey. For this purpose, non-linear equations are determined by considering GSHP models in relevant literatures for heating and cooling capacity of 1.4 and 2.2 kW respectively for a building having dimensions as 4.4 m x 2.2 m x 3.6 m. In this respect, COPs for heating and cooling are calculated and obtained results are depicted. An algorithm is generated to design the R134a refrigerant heat exchanger to maximize the efficiency.

Key words: Ground Source Heat Pump, GSHP; Heat pump, Heating, Bolu, Bolu City, Turkey.

1. Introduction

The consumption of energy in the world is increasing rapidly, especially in developing countries. The depletion of fossil fuels coupled with the increasing demand for energy has increased demand for renewable energy sources. Solar, geothermal etc. energies can be regarded as alternative energy sources. The use of the GSHPs as seen in Fig. 1 has advantages owing to that the temperature of soil is always higher than the temperature of air and ground source temperature variations are very low compared to air source temperature variations [1].

Ground source heat pumps can be used for both heating and cooling purposes. In heating application, the heat transferred from the soil through the brine circuit is transferred to the cooling circuit. In cooling application, the heat transferred from the media to be cooled is transferred into the ground through the brine circuit.

As the applications of GSHPs are very widespread, there are many publications about the ground sources in the literature. Ayyildiz studied and developed a model for a heat pump used in geothermal water [2]. In this study, FORTRAN codes have been generated. Healy and Ugursal evaluated various parameters of a GSHP including the ground heat exchanger depth, brine flow rate etc. [3]. Petit et al. compared a vertical GSHP and another heat pump sourced by air and found that GSHPs are more reasonable than Heat Pumps sourced by air [4]. Zogou et al. studied the effects of climate conditions for the heat pump design. [5]. Piechowski et al, derived a mathematical model for a horizontal GSHP [6]. Noorollahi et al. achieved the heating and cooling powers in a greenhouse for a year in Karaj, Iran [7]. Akbulut et al. studied experimentally and theoretically exergo-environmental, and exergo-economic analyses for wall cooling systems by using a vertical type of GSHP. They found that the efficiencies of energy and exergy for the system were 74.85% and 29.90% respectively [8]. Hepbasli investigated GSHP in İzmir case [9]. In this study, the authors obtained COP of GSHP for cooling condition. Balbay and Esen achieved experimental and numerical studies for temperature variation in pavement and bridge slabs heated by vertical ground-source heat pump. For this purpose, a finite element model has been developed for

numerical investigation [10]. Esen et al. installed a ground-coupled heat pump (GCHP) system and tested in the conditions of Elazıg, Turkey [11]. Hepbasli and Ozgener giving the knowledge about the development of geothermal energy (GE) using in Turkey with its historical development and advantages [12]. There are also the other studies about the GSHPs which can be obtained from the literatures. These studies are related to history of the use of geothermal energy, investigation of the system parameters for the heat pump constructed in Elazığ, energy and exergy analysis for a heat pump, obtaining experimentally the COPs of GSHPs constructed in Bursa and temperature distribution for GSHP etc. [13–17]. Also, Camdali and Tuncel studied the cost of the GSHP by comparing it to other conventional systems [18]. Camdali, et al. give the GSHP simulation model for the system and for this purpose, the codes needed for equations are written in a computer program [19].

Optimization of parameters can be seen in numerous studies for GSHP systems. Michael de Paly gives a new approach for the optimization of extraction energy in shallow geothermal systems by using a linear program [20]. Esen et al. established slinky type heat exchanger (GHE) for a ground source heat pump which is solar assisted. Also, they performed system modelling by using the experiment data and Artificial Neural Network (ANN) and Adaptive Neuro-Fuzzy Inference System (ANFIS) [21]. In the other study, an optimization procedure of borehole heat exchanger fields has been stated in order to estimate the temperature distributions in the subsurface. This study is stating that the system refrigerants and structural parameters have effects on the cycling thermal performance on GSHP [22].

Another study discussing the operating parameters of GSHP for heating and cooling states that the electricity inputs can be decreased by improving the COP of GSHP, thus the operating parameters of a GSHP system can be used for optimization for both heating and cooling modes, using Taguchi Method [23]. Tagushi method can be seen in another study where optimization of vertical ground coupled heat pump (VGCHP) [24]. Taguchi method has been applied to these parameters to optimize coefficient of performance (COP) of the VGCHP system, resulting that depth of borehole becomes the most significant parameter, affecting the COP by 67.77%. There are also recent studies regarding to ground-source heat pumps (GSHPs) performance China [25].

In this study, system parameters of a GSHP are obtained for an experimental house shown in Fig. 2 located in Bolu, Turkey. The heating and cooling powers are 1.4 and 2.2 kW respectively for the area where the building is located. For this purpose, a non-linear model for the whole system is derived and solved to obtain system parameters of the GSHP. A simulation model has been developed and it is used for HPs with low GWP refrigerants and confirmed by using measured data. The effect of soil temperature on the system performance is investigated. The temperature model of the area, where the GSHP has built, is also given in this study due to the effects of soil temperature on the performance of the system. The final design of the heat exchanger has been optimized for each alternative refrigerant using the developed algorithm to obtain the highest performance improvement, which more numerical results can be found next study.

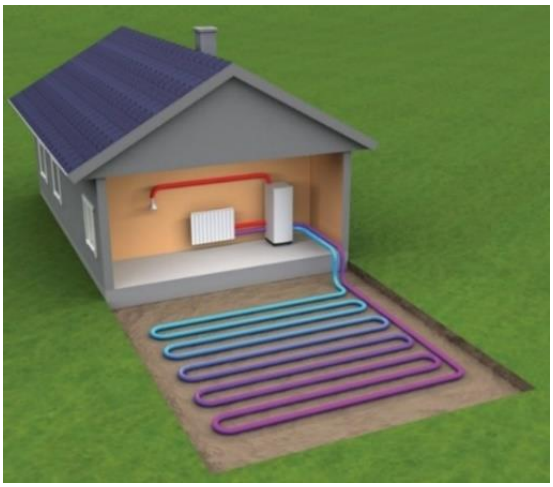


Fig. 1. GSHP application for a house**.**

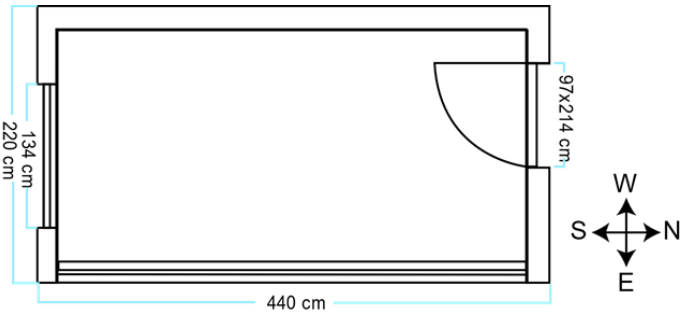


Fig. 2. Heating and cooling area (h= 3.6 m) in Bolu City, Turkey.

2. Ground Temperature Distribution Model

The temperature distribution for the ground source depending on x (depth) and t parameters is written in Equation (1) by using heat conduction principle in transient state [26,27]:

$$T(x,t) = T_a + T_{amp} e^{-x\left(\frac{\pi}{\alpha P}\right)^{1/2}} cos\left(2\pi\frac{t}{P} - x\left(\frac{\pi}{\alpha P}\right)^{1/2}\right) \quad (1)$$

When this formula is applied to measurement data obtained from Turkish State Meteorological Service by using curve fitting toolbox in MATLAB computer program with Ta= 12.95 °C, T_{amp} = 11.69 °C and α = 0.00000068 m^2/s, P(period, seconds)= $365 \times 24 \times 60 \times 60$ [26], Equation (1) can be depicted for Bolu City as given below [27]:

$$T(x,t)= 12.95+ 11.69e^{-x\left(\frac{\pi}{0.00000068*365*24*60*60}\right)^{1/2}}\cos\left(2\pi\frac{t}{365*24*60*60}-x\left(\frac{\pi}{0.00000068*365*24*60*60}\right)^{1/2}\right) \quad (2)$$

The variation of ground surface temperature is given in Eq. (3). This equation is obtained from Eq. (2) by writing x=0.

$$T(t) = 12.95 + 11.69cos\left(2\pi\frac{t}{365*24*60*60}\right) \quad (3)$$

Where, t= 181st day of year (July, 1) then *t=181+n*30* (*n=1…12*). Temperature variations versus months are seen in Fig. 3 [26].

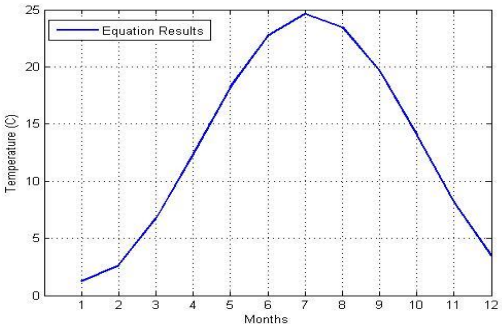


Fig. 3. Temperature distribution of soil surface for Bolu City [27].

3. Mathematical Modeling of GSHP Components

The mathematical model developed by Ayyildiz is used for heating cycle [2,19]. The mathematical models of the system equipment such as compressor, evaporator, condenser, and expansion valve are developed at first. Then, general mathematical model of GSHP is derived depending on system equipment models. System details and its T-s diagram are shown in Figures 4 and 5 respectively.

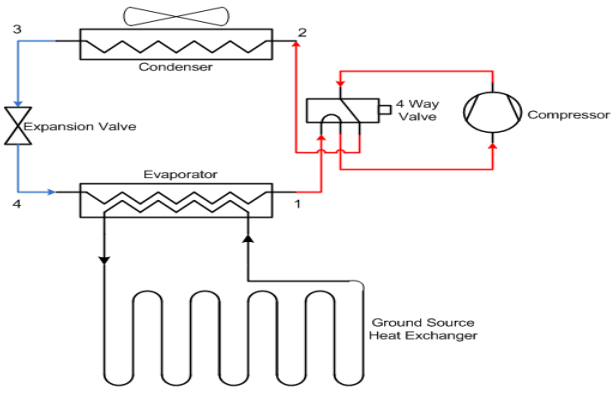


Fig. 4. Heat pump with a GSHE [19].

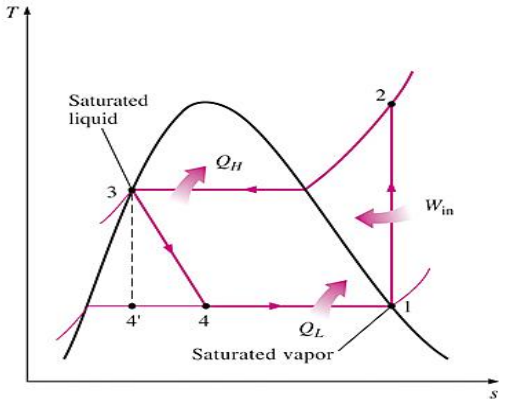


Fig. 5. T-S diagram of an ideal cycle for a heat pump.

3.1. Compressor Model

The network that must be given from compressor to refrigerant can be written as Eq. (4) [19,28].

$$\dot{W}_{net} = \dot{m}_r(h_2-h_1) \quad (4)$$

The exit refrigerant enthalpy of the compressor can be written as Eq. (5) depending on isentropic efficiency of compressor.

$$h_2 = h_1 + \frac{(h_{2s}-h_1)}{\eta_{isn}} \quad (5)$$

Consumption of energy for the compressor is determined according to Eq. (6) [19, 28].

$$\dot{W}_{comp} = \frac{W_{net}}{\eta_m\eta_e} \quad (6)$$

3.2. Condenser Model

The heat given by refrigerant to air on the condenser can be calculated using Eq. (7) and the heat taken by air can be calculated Eq. (8) [19, 28].

$$\dot{Q}_r = \dot{m}_r(h_2 - h_3) \quad (7)$$

$$\dot{Q}_c = \dot{m}_a c_{p,a}(h_{ao} - h_{ai}) \quad (8)$$

Since it is approached that there is no heat loss through the condenser, Eq. (9) can be written as follows:

$$\dot{Q}_r = \dot{Q}_c \quad (9)$$

The heat transfer from condenser to air can be written by log mean temperature difference method (LMTD) as follows:

$$\dot{Q}_c = K_c A_c F_c \Delta T_{lmc} \quad (10)$$

$$\Delta T_{lmc} = \frac{\left(T_c - T_{ao}\right) - \left(T_3 - T_{ai}\right)}{\ln \frac{\left(T_c - T_{ao}\right)}{\left(T_3 - T_{ai}\right)}} \quad (11)$$

3.3. Evaporator Model

The heat transferred by brine to refrigerant on the evaporator is calculated by Eq. (12) and the heat given by brine to refrigerant is calculated by Eq. (13).

$$\dot{Q}_e = \dot{m}_r \left(h_1 - h_4\right) \quad (12)$$

$$\dot{Q}_e = \dot{m}_b c_{p,b} \left(T_{bi} - T_{bo}\right) \quad (13)$$

The heat transferred by brine to refrigerant on the evaporator can be written by log mean temperature difference method (LMTD) as follows [19, 28]:

$$\dot{Q}_e = K_e A_e F_e \Delta T_{lme} \quad (14)$$

$$\Delta T_{lme} = \frac{\left(T_{bi} - T_1\right) - \left(T_{bo} - T_4\right)}{\ln \frac{\left(T_{bi} - T_1\right)}{\left(T_{bo} - T_4\right)}} \quad (15)$$

3.4. Expansion Valve Model

The heat transfer and work are not occurred while refrigerant is passing through the expansion valve. So, enthalpy remains constant as seen in Eq. (16) [19, 28]:

$$h_3 = h_4 \quad (16)$$

4. Application of Energy Conservation Law to Cooling Cycle

When the energy conservation law (the first law) is applied to a cooling cycle, Eq. (17) can be written as follows. COP of the system can also be written as Eq. (18) [19, 28]:

$$\dot{W}_{comp} + \dot{Q}_e = \dot{Q}_c \quad (17)$$

$$COP = \frac{\dot{Q}_e}{\dot{W}_{comp}} \quad (18)$$

5. Obtaining Parameters of the System

When relating equations are placed into Eq. (17), Eq. (19) is obtained. Also, Eq. (20) is taken from Eqs. (12) and (14). So, after writing the Eqs. (4), (6); (14), (15); (10) and (11) into Eq. (17), Eq. (21) is obtained. The temperatures of evaporator (T_e), condenser (T_c) and the exit temperature of compressor (T_2) in Eqs. (19)–(21) are defined as unknown parameters. So, the unknown temperatures can be solved iteratively by using developed codes in the computer program [19,28].

$$\frac{\dot{m}_r (h_2 - h_1)}{\eta_m \eta_e} + \dot{m}_b C_{p,b} (T_{bi} - T_{bo}) - \dot{m}_r (h_2 - h_3) = 0 \quad (19)$$

$$\frac{\dot{Q}_c}{\left(h_2 - h_3\right)} \left(h_1 - h_4\right) - K_e A_e F_e \Delta T_{lme} = 0 \quad (20)$$

$$\frac{\dot{m}_r (h_2 - h_1)}{\eta_m \eta_e} + K_e A_e F_e \frac{(T_{bi} - T_1) - (T_{bo} - T_4)}{ln \frac{(T_{bi} - T_1)}{(T_{bo} - T_4)}} - K_c A_c F_c \frac{(T_c - T_{ao}) - (T_3 - T_{ci})}{ln \frac{(T_c - T_{ao})}{(T_3 - T_{ci})}} = 0 \quad (21)$$

$$h_1 = h_1(T_1, P_1) = h_1(T_e, P_1);\ h_2 = h_2(T_2, P_2);\ h_3 = h_3(T_c) \quad (22)$$

5.1. Determination of Length for GSHE

The length of ground source heat exchanger can be determined by using data in the literatures [29,30]. So, the length of GSHE for two kinds of modes which are heating and cooling can be written as Eqs. (23) and (24) [19, 28]

$$L_{heating} = \frac{Q_{heating} \frac{COP_{heating} - 1}{COP_{heating}} \left[R_p + \left(R_s F_o\right)\right]}{T_{s,\min} - T_{b,\min}} \quad (23)$$

$$L_{cooling} = \frac{Q_{cooling} \frac{COP_{cooling} + 1}{COP_{cooling}} \left[R_p + \left(R_s F_o\right)\right]}{T_{b,\max} - T_{s,\max}} \quad (24)$$

5.2. Determination of Excavated Ground Area for GSHE

After the calculation of GSHE pipe length, the calculation of excavated ground area must be done. The parameters of GSHE geometric parameters are seen in Fig. 6. So, this area can be determined by using the formula given below [28]:

$$A_{soil} = \underbrace{\left(d + 3r + e + b\right)}_{height} \underbrace{\left(a + 2rn + c\right)}_{width} \quad (25)$$

In Eq. (25), e is an unknown parameter.

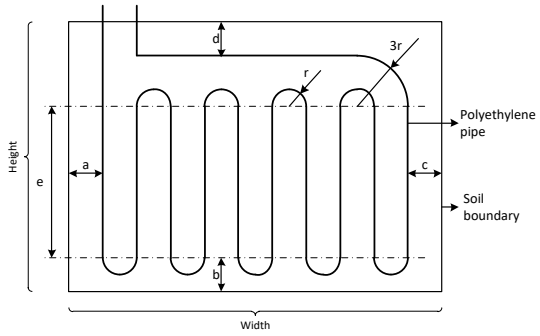


Fig. 6. The geometry of GSHE placement [28].

As it is known, one of the height or width of ground to excavate, the other dimension can be calculated by using Eq. (23) or (24). For this purpose, GSHE length can be obtained from the Equations given below [28]:

$$L=(n+1)e+3r+n\left(\frac{2\pi r}{2}\right)+\frac{2\pi 3r}{4}+(n-2)2r+r \tag{26}$$

When this formula is rearranged, Eq. (26) is obtained in another form:

$$L=(n+1)e+\left[(\pi+2)n+\frac{3\pi}{2}\right]r \tag{27}$$

The unknown parameter (e or n) can be calculated by using Eq. (27). Then, ground area is determined according to Eq. (25).

5.3. Capacity of the Circulation Pump

The losses of brine cycle are defined as uniform and local losses. The uniform losses are existed through the pipe. Local loses are existed on the elements such as T fittings, valve, etc. The pump head is calculated by the formula given below [28,31,32]:

$$h_{pump}=h_{K,uniform}+h_{K,local} \tag{28}$$

5.4. Uniform Losses of the Pipes

Uniform losses are obtained by using Eq. (29) given below [28]:

$$h_{k,uniform}=f\frac{L}{D_h}\frac{V^2}{2g} \tag{29}$$

In Eq. (29), f is the friction factor. This factor is calculated by using Eqs. (30)–(31) for laminar and turbulent flows respectively.

$$f=\frac{64}{\mathrm{Re}} \tag{30}$$

$$f=\frac{8}{\left[2.457ln\left\{\frac{1}{\left(\frac{7}{Re}\right)^{0.9}+0.27\left(\frac{\varepsilon}{D}\right)}\right\}\right]^2+\left(\frac{37530}{Re}\right)^{0.75}} \tag{31}$$

Local losses of the pipes and elbows

Local losses are calculated by using follow equation [28].

$$h_{k,local}=K_k\frac{V^2}{2g} \tag{32}$$

The coefficients of some fittings are obtained from relevant literatures. The term K_K given in Eq. (32) refers to the loss coefficient of some fittings. Loss coefficients for various fittings are presented in Table 1.

Table 1. Loss coefficients for various components [32]

Fittings	Loss Coefficients (K_K)
90° smoot bend-flanged	0.3
90° smoot bend-threaded	0.9
90° smoot bend-without vanes	1.1
90° smoot bend-with vanes	0.2
45° threaded elbow	0.4
180° return bend-flanged	0.2
180° return bend-threaded	1.5
Tee (branch flow)-flanged	1.0
Tee (branch flow)-threaded	2.0
Tee (line flow)-flanged	0.2
Tee (line flow)-threaded	0.9
Threaded union	0.08
Globe valve-fully open	10
Angle valve-fully open	5
Ball valve, fully open	0.05
Gate valve-fully open	0.2
Swing check valve	2

6. Results

The codes in the computer program are written in order to solve the accomplished models for GSHP. First heating and cooling values, compressor mechanical efficiency, isentropic efficiency, efficiency of electric engine, the heat transfer coefficients of evaporator and condenser, inlet and exit temperatures of brine and air and tolerance etc. are used as data in the computer program. Then, evaporator and condenser temperatures are estimated. Compressor, evaporator and condenser capacities up to four cooling cycles are computed according to estimated values. Until converge is achieved, iterations are proceeded with increasing of evaporator and condenser temperatures. Finally, system parameters such as the temperatures of evaporator and condenser, GSHE (ground source heat exchanger) length, operating factor, pipe and soil

resistances etc. are calculated. The evaporator and condenser temperatures are taken as input parameters for *a computer program*. After that the refrigerant properties of the cycle at some points, evaporator and condenser capacities, compressor power, pressure ratio, pressure difference, refrigerant flow rate, COP etc. values are obtained by using this trading the computer program in order to compare the values obtained from a *computer program*. System parameters such as COP, compressor power, refrigerant flow rate and GSHE lengths are calculated for some refrigerants. Some results for heating mode can also be shown in Table 2 for R134a [28]. The values obtained are depicted in Fig. (7–8) [19,28].

Table 2. Input & Output Values of the GSHE for Heating Mode [28]

INPUT PARAMETERS

Aim of the GSHE System	:	Heat. Mode	Power (kW)	:	1.4
Brine Type	:	Ethy. Glycol	Refrigerant	:	R134a
Mass Ratio (%)	:	20			

Compressor			**Condenser**		
Isometric Efficiency	:	0.70	Tot. Heat Trans. Coef.	:	0.02
Mechanical Efficiency	:	0.80	($kW/m^2°C$)		
Electric Motor Efficiency	:	0.85	Heat Transfer Surface Area (m^2)	:	10.00
Evaporator			Air Inlet Temperature (°C)	:	20.00
Total Heat Transfer Coefficient ($kW/m^2°C$)	:	0.64	Air Outlet Temperature (°C)	:	30.00
			Sub-Cooling (°C)	:	0.00
Heat Transfer Surface Area (m^2)	:	0.35			
Brine Inlet Temperature (°C)	:	5.00	**Coefficients**		
Brine Inlet Temperature (°C)	:	-5.00	Specific Heat of Air (kJ/kg K)	:	1.005
Super Heating (°C)	:	0.00			

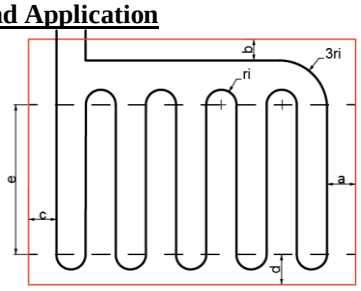
Building Properties			**Piping and Soil Resistances**		
Building inside Temperature (°C)	:	22	Placement	:	Horizontal
Minimum Outside Temperature (°C)	:	5	Number of Passes	:	4
			Soil Type	:	Light Soil–with Moisture
Limit Heating Temperature (°C)	:	23	Pipe Class	:	PE SCH-40
			Pipe Diameter (inch)	:	3/4”
			Soil Depth (m)	:	1.8

RESULTS

Cycle

	P(bar)	T(°C)	v(dm^3/kg)	h(kJ/kg)	s(kJ/kg K)
1	2.28	-6.60	90.53782	394.696	1.7317
2s	8.41	38.38	25.36812	421.736	1.7316
2	8.41	49.58	27.11838	433.333	1.7658
3	8.41	33.20	0.85056	246.676	1.1594
4	2.28	-6.60	24.52485	246.676	1.1753

GSHE Ground Application



Evaporator Temperature (°C)	:	-6.60
Condenser Temperature (°C)	:	33.20
Air Flow Rate (g/s)	:	139.30
Brine Flow Rate (g/s)	:	29.04
Evaporator Capacity (kW)	:	1.13
Condenser Capacity (kW)	:	1.41
Compressor Power (kW)	:	0.43
Evaporator Pressure (bar)	:	2.28
Condenser Pressure (bar)	:	8.41
Pressure Ratio	:	3.69
Refrigerant Flow Rate (g/s)	:	7.50
Inlet Pipe Diameter (mm)	:	7.86
Outlet Pipe Diameter (mm)	:	4.46
Pipe Diameter of Cond. (mm)	:	3.41
Pipe Diameter of Evap. (mm)	:	17.67
COP	:	3.29

a (cm) :10 b (cm):10 c (cm):10
d (cm):25 e (cm): 220 ri (cm):15

Number of Pipe Spacing	:	20
Width (m)	:	6.320
Height (m)	:	3.000
Area (m^2)	:	18.961

Pump Load

	Fitting Type	Loss Coefficient	Num.	Total Loss (mwc)
1	90° smoot bend-flanged	0.30	32	9.600
2	Globe valve-fully open	10.00	2	20.000

Piping (Building) and Soil Properties			Additional Pipe Length (m)	:	26.000
Building Coefficient	:	0.824	Elevation Difference (wmc)	:	2.3288
Pipe Resistance (m °C/W)	:	0.098	Continuous Losses (wmc)	:	0.2523
Soil Resistance (m °C/W)	:	0.861	Local Losses (wmc)	:	0.0146
Minimum Soil Temperature (°C)	:	7.782	Pump Load/Losses (wmc)	:	2.5957
Maximum Soil Temperature (°C)	:	18.118	Brine Flow Rate (m^3/hour)	:	0.1009
Total (Building) Pipe Length (m)	:	61.527			
Brine Properties					
Density (kg/m^3)	:	1036.18			
Dynamic Viscosity (kg/m s)	:	0.0034			
Flow Properties					
Average Velocity (m/s)	:	0.10			
Reynolds Number	:	574			

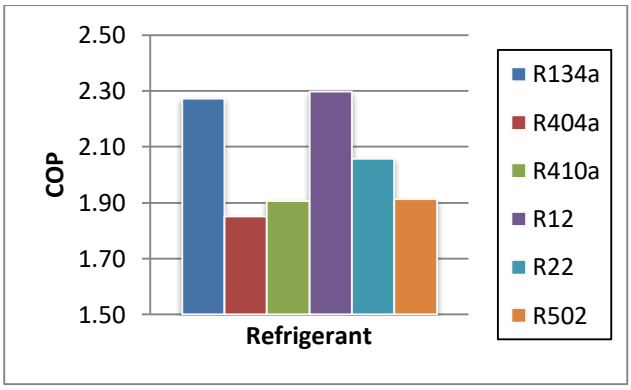


Fig. 7. COP variations depending on refrigerants for heating mode [28].

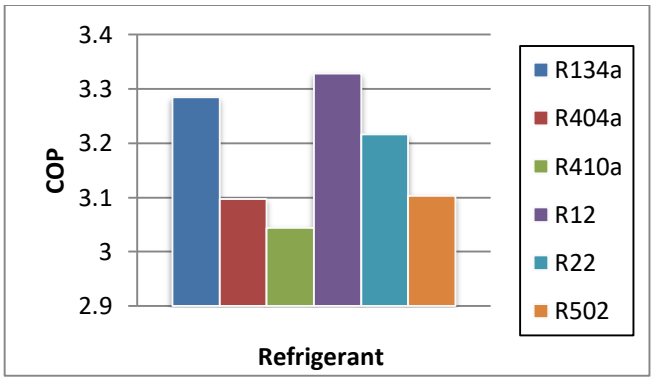


Fig. 8. COP variations depending on refrigerants for cooling mode [28].

7. Conclusions

As it is known, a significant part of the world's energy consumption is provided by renewable energy sources [33–35]. In this study, the system parameters of the ground source heat pump in Bolu province were obtained numerically. The real system parameters of GSHP in Bolu City condition are derived for this purpose, non-linear equations are determined by using available GSHP models in relevant literatures for heating and cooling capacity of 1.4 and 2.2 kW respectively for a building having dimensions as 4.4 m x 2.2 m x 3.6 m. Actual COP values of GSHP's for a desired refrigerant type can be obtained more precisely by using the derived system parameters. As a whole, exploration and activation of renewable energy potential, such as ground sources for heating and cooling purposes, is emphasized in this study.

Nomenclature

A_c, A_e : Total heat transfer area for the condenser and the evaporator [m^2]
a, b, c, d : Space between GSHE and soil boundary
$c_{p,a}$, $c_{p,b}$: Specific heat for air and brine [$kJ/kg°C$]
$COP_{cooling}$: Performance coefficient for cooling and heating [%]
$COP_{heating}$: Performance coefficient for heating [%]
e : Vertical length of pipe
F_c, F_e : Correction coefficients of the condenser (F_c=1) and the evaporator (F_e=1)
F_o : Operating factor
GSHE : Ground source heat exchanger
GSHPs : Ground source heat pumps
GWP : Global warning potential
HPs : Heat pumps
h : Pipe losses [mwc]
h_1, h_2 : Inlet and exit refrigerant enthalpy [kJ/kg]
h_{2s} : Isentropic enthalpy for refrigerant [kJ/kg]
h_3 : Exit refrigerant enthalpy of the condenser [kJ/kg]
h_4 : Inlet refrigerant enthalpy of the evaporator [kJ/kg]
K_c : Total heat transfer coefficient for the condenser [kW/m² °C]
K_e : Total heat transfer coefficient for the evaporator [$kW/m^{2}°C$]
K_K : Losses coefficients of fittings
$\dot{m}_a$,$\dot{m}_b$,$\dot{m}_r$: Mass flow rate of air, brine and refrigerant [kg/s]
mwc : Meter water column
P : Period [s: 365*24*60*60]

$\dot{Q}_c$: Condenser capacity [kW]
$Q_{cooling}$: Cooling capacity of the heat pump [W]
$Q_{heating}$: Heating capacity of the heat pump [W]
$\dot{Q}_r$: The heat transferred from refrigerant to air [kW]
R_p, R_s : Pipe and soil resistance $[mK/W]$
r : Turning radius of pipe
T_a : Average ground surface temperature [°C]
T_{ai} : Inlet air temperature of the condenser [°C]
T_{bi} : Inlet brine temperature of the evaporator [°C]
T_{amp} : Amplitude temperature of ground surface [°C]
T_{bo} : Exit brine temperature of evaporator [°C]
$T_{b,max}$: The highest brine temperature inlet evaporator [°C]
$T_{b,min}$: The minimum brine temperature inlet evaporator [°C]
$T_{s,max}$: The highest soil temperature on a year [°C]
$T_{s,min}$: The minimum soil temperature during a year [°C]
T_1 : Super-heated refrigerant exit from the evaporator [°C]
T_3 : Cooling liquid temperature exiting from the condenser [°C]
T_4 : Refrigerant temperature exiting from the expansion valve [°C]
T_{ao} : Exit air temperature of condenser [°C]
ΔT_{lmc} : Logarithmic temperature differences of the condenser [°C]
ΔT_{lme} : Logarithmic temperature differences of the evaporator [°C]
t : Duration from initial time [s]
x : Burying depth (m)

Greek symbols

α_t : Thermal diffusivity of soil (m^2/s)
ε / D : Relative roughness ratio.
$\eta_e; \eta_{isn}; \eta_m$: Efficiencies for electric engine, isentropic process and mechanical for compressor

References

[1] U. Camdali, E. Tuncel,, "System analysis of ground source heat pumps (GSHPS) for Bolu: an economical approach", Engineer and Machinery, vol. 51(604), pp. 22-35 2010 (in Turkish).

[2] I. Ayyildiz, Determination Of Performance Characteristics For The Steady-State Operation Of A Vapor-Compression Heat Pump Using Geothermal Waste Water Through A Simulation Model, Ph. D. Thesis, Gazi University, Institute of Science and Technology. 1996

[3] PF. Healy, Ugursal, VI. "Performance and economic feasibility of ground source heat pumps in cold climate", International. Journal of Energy Research vol. 21, pp. 857-870, 1997.

[4] PJ. Petit, JP. Meye, "Economic potential of vertical ground-source heat pumps compared to air-source air conditioners in South Africa", Energy, vol. 23(2), pp. 137-143 1998.

[5] O. Zogou, A. Stamatelos, "Effect of climatic conditions on the design optimization of heat pump systems for space heating and cooling", Energy Conversion and Management, vol. 39(7), pp. 609-622, 1998

[6] M. Piechowski, "Heat and mass transfer model of a ground heat exchanger: theoretical development", O 23(7), pp. 571-588, 1999.

[7] Y. Noorollahi, P. Bigdelou, F. Pourfayaz, H. Yousefi, "Numerical modeling and economic analysis of a ground source heat pump for supplying energy for a greenhouse in Alborz province, Iran", Journal of Cleaner Production; vol. 95, pp.413-421, 2016.

[8] U. Akbulut, U, Z. Utlu, O. Kincay, "Exergo-environmental and exergo-economic analyses of a vertical type ground source heat pump integrated wall cooling system", Applied Energy, vol. 86, pp.2560-2565, 2016.

[9] A. Hepbasli, "Performance evaluation of a vertical ground-source heat pump system in Izmir, Turkey", International. Journal of Energy Research, vol. 26(13), pp. 1121-1139, 2002.

[10] A. Balbay, M. Esen, "Temperature distributions in pavement and bridge slabs heated by using vertical ground-source heat pump systems". Acta Scientiarum-Technology, vol. 35(4), pp. 677-685, 2013.

[11] H. Esen, M. Inalli, M. Esen, "Numerical and experimental analysis of a horizontal ground-coupled heat pump system", Building and Environment, vol. 42(3), pp. 1126-1134, 2007.

[12] A. Hepbasli, A. Ozgener, "Development of geothermal energy utilization in Turkey: a review", Renewable and Sustainable Energy Reviews, 8(5), pp. 433-460, 2004.

[13] M. Inalli, H. Esen, "Experimental thermal performance evaluation of a horizontal ground-source heat pump system", Applied Thermal Engineering, vol. 24, pp. 2219-2232, 2004.

[14] S. Coskun, E. Pulat, K. Unlu, R. Yamankaradeniz, "Experimental performance investigation of a horizontal ground source compression refrigeration machine", International. Journal of Energy Research, vol. 2(1), pp. 44-56, 2008.

[15] A. Hepbasli, Y.A. Kalinci, "Review of heat pump water heating systems", Renewable and Sustainable Energy Reviews, vol. 13, 6-7, pp.1211-1229, 2009.

[16] E. Pulat, S. Coskun, K. Unlu, N. Yamankaradeniz "Experimental study of horizontal ground source heat

pump performance for mild climate in Turkey", Energy vol. 34(9), pp.1284-1295, 2009.

[17] H. Demir, A. Koyun, G. Temir, "Heat transfer of horizontal parallel pipe ground heat exchanger and experimental verification", Applied Thermal Engineering vol. 29(2-3), pp. 224-233, 2009.

[18] U. Camdali, E. Tuncel, "An Economic analysis of horizontal ground tsource heat pumps (GSHPs) for use in heating and cooling in Bolu, Turkey", Energy Sources Part B, vol. 8(3), pp. 290-303, 2013.

[19] U. Camdali,, M. Bulut, N. Sozbir, "Numerical modeling of a ground source heat pump: the Bolu case", Renewable Energy; vol. 83, pp.352-361, 2015.

[20] M.D. Paly, J. Hecht-Méndez, M. Beck, P. Blum, A. Zell, P. Bayer, "Optimization of energy extraction for closed shallow geothermal systems using linear programming", Geothermics, vol. 43, pp. 57-65, 2012.

[21] M. Esen, H. Esen, O. Ozsolak, "Modelling and experimental performance analysis of solar-assisted ground source heat pump system", Journal of Experimental & Theoretical Artificial Intelligence vol. 29(1), pp.1-17, 2017.

[22] Z. Dong, Q. Wu, J.Li, and X. Kong, "Effects of refrigerant charge and structural parameters on the performance of a direct-expansion solar-assisted heat pump system", Applied Thermal Engineering, vol. 73, pp.522-528, 2014.

[23] T. Sivasakthivel, K. Murugesan, H. Thomas, "Optimization of operating parameters of ground source heat pump system for space heating and cooling by Taguchi method and utility concept", Applied Energy, vol. 116, pp. 76-85, 2014.

[24] H. Esen and E. Turgut, "Optimization of operating parameters of a ground coupled heat pump system by Taguchi method", Energy and Buildings, vol. 107, pp.329-334, 2015.

[25] Z. Qiao, T. Long, W. Li, L. Zeng, Y. Li, J. Lu, Y. Cheng, L. L. Xie,, L.Yang, "Performance assessment of ground-source heat pumps (GSHPs) in the Southwestern and Northwestern China: In situ measurement" Renewable Energy, vol. 153, pp.214-227, 2020

[26] V. Arpaci, A. Selamet, S.H. Kao, Introduction to Heat Transfer, Prentice Hall, 2000.

[27] U. Camdali, M. Tunc, "Ground temperature estimations for Bolu, Turkey", Environ Monit. Assess. 2012; vol. 184(6): 3721-3729.

[28] E. Tuncel, Numerical Analysis of Ground Source Heat Pump in the Case of Bolu and Its Environment, MSc thesis at Sakarya University 2011; (in Turkish).

[29] L. Miles, L. Heat Pumps Theory and Service, Delmar Publishers Inc, 1994.

[30] I. Ersoz, Cooling A Zone by Using A Ground Source Heat Pump, Master of Science Thesis 2000, Ege University Institute of Science and Technology.

[31] M. Ozgeren, Heat Pump Modelling for Heating and Cooling, MS Thesis 1996, Cukurova University Institute of Science and Technology (in Turkish).

[32] Y.A. Cengel, and JW. Cimbala, Fluid Mechanics, Fundamentals and Applications, McGraw Hill; 2006.

[33] M. Akkaya and U. Camdali, "Thermodynamic analysis for a solar assisted absorption cooling system: energy and exergy base", International Journal of Renewable Energy Research-IJRER, vol. 12(4), 2189-2199, 2022.

[34] H.S. Mutashar and A.M. Shakir, "Robustness analysis of elm-based fault detection in PV systems", International Journal of Smart Grid, vol.7, No.4, pp. 189-199, December, 2023.

[35] R. Al-Hajj, M.M. Fouad, A. ASSI, E Mabrouk, "Short-term wind energy forecasting with independent daytime/nighttime machine learning models", 11th IEEE International Conference on Renewable Energy Research and Applications, pp. 186–191,September 18-21, 2022, Istanbul, Turkey.