INTERNATIONAL JOURNAL of RENEWABLE ENERGY RESEARCH
Ali Siavoshi Kord et al., Vol.2, No.1, 2012

Optimization and Analysis of a Vertical Ground-Coupled Heat Pump

Ali Siavoshi Kord*, Seyed Ali Jazayeri*

*K.N.Toosi University of Technology, Department of Mechanical Engineering;

‡ Corresponding Author; Ali Siavoshi Kord ,K.N.Toosi University of Technology,Tehran-Iran, +989133475613, asiavoshi@gmail.com, sajazayeri@hotmail.com

Received: 01.10.2011 Accepted:29.10.2011

Abstract- In this study a ground-coupled heat pump (GCHP) system with single U-bend ground heat exchanger has been used for heating applications. The mass, energy, entropy and exergy balance relations are derived and applied to GSHP system. The performance characteristics of this GSHP system are evaluated in terms of exergetic aspects for city of kerman and the optimum size for ground heat exchanger is calculated. The effect of different rock or soil and water-antifreeze inlet temperature on optimum length has been shown. Results show that there is an optimum length and mass for vertical ground heat exchanger

Keywords- Geothermal energy; Ground-coupled heat pump; Energy; Optimization.

1. Introduction

Ground-coupled heat pump (GCHP) systems play a key role in development of using clean and low cost energy. These systems collect and transfer heat from the earth through a series of buried pipes containing a working fluid, which is typically composed of a mixture of water and antifreeze solutions such as methanol, ethanol or glycol. GCHP are now increasingly used for space heating, cooling and to provide domestic heat water [1].

Lund and Freeston [2, 3] have reviewed the worldwide application of geothermal energy for direct utilization. They concluded that GCHP have had the largest growth since 1995, almost 59%, representing 9.7% annually. Most of this growth occurred in the United States and Europe, though interest is developing in other countries, such as Japan and Turkey. In recent years, investigations have been conducted by various researchers on design, modeling and experimental performance evaluation of GSHP systems (e.g. Kavanaugh, 1992; Kavanaugh and Rafferty, 1997; Healy and Ugursal, 1997; Hepbasli, 2002; Hepbasli et al., 2003; Sanner et al., 2003; Hepbasli and Akdemir, 2004; Bi et al., 2004; Yumrutas and Kaska, 2004; Ozgener and Hepbasli, 2004, 2005).

In Iran, in spite of very good opportunity, only a few efforts have been done for using these systems like 18000 BTU/hr air-to-air heat pump which was designed and installed in Tabriz Engineering Research Center [4].

In this study a simulation program has been used for determining the optimized dimensions of a (GCHP) with single U-bend ground heat exchanger as well as estimating exergy performance of a typical heat pump in city of Kerman in different heating season.

The first law of thermodynamics was used to balance the energy of the system and obtain the coefficient of performance (COP), and the second law was used to obtain the exergy and exegretic efficiency of heat pump also optimum length and diameter of the heat exchanger [5]. Exergy or availability analysis is a powerful tool in the design, optimization and performance evaluation of energy systems. This analysis could be used to identify the main sources of irreversibility that is exergy loss and to minimize the generation of entropy in a given process where the transfer of energy and material take place [6, 7].

2. System Description

The schematic of a vertical GCHP system is illustrated in Fig.1. The main components of GCHP systems are compressor, condenser, expansion valve, and an evaporator. The components are connected to a U-bend ground heat exchanger, as shown in Figure 1. Ground heat flows from the circulating fluid through the ground U-bend to an evaporator that uses heat to increase the temperature of a refrigerant, causing it to evaporate. The evaporated refrigerant moves into the compressor, and refrigerant is compressed to a higher pressure and temperature. The refrigerant then flows into the condenser where it condenses and gives useful heat. The expansion valve reduces the refrigerant's pressure, subsequently reducing its temperature. The refrigerant flows back to the evaporator and the process repeats again.

Following assumptions has been made in this research:

- A safety margin of 5°C superheat to prevent liquid droplets entering the compressor
- The refrigerant in evaporator is subcooled 5°C
- Condenser and evaporator have a 5% pressure drop

Table 1. Specification of the system

Heat pump	
Average compressor adiabatic power input	0.96 kW
Compressor adiabatic efficiency	0.85
Refrigerant type	R134a
Refrigerant mass flow rate	0.0192 kg/s
Condenser mass flow rate	0.149 kg/s
Ground heat exchanger	
Configuration type	Vertical
Water-antifreeze type	Propylene glycol solution
Water-antifreeze mass flow rate	0.147 kg/s
Rock type	sandstone

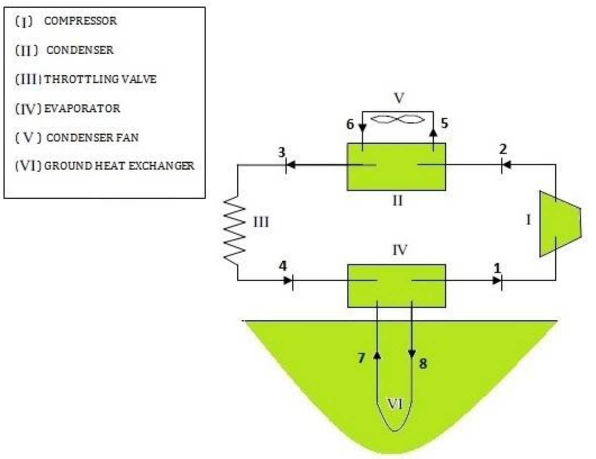

Fig.1. Schematic diagram of the system

Determining the length and diameter of a ground heat exchanger is an important aspect of ground-coupled heat pump design [8]. The equation of determining the optimum Reynolds number is given by Ali Sharifzadegan et al [9], the optimum length and diameter could be derived as:

$$D_{opt} = \frac{4\dot{m}_w}{\pi\mu \,\mathrm{Re}_{opt}} \tag{1}$$

$$L_{opt} = \frac{2n}{D_{opt}} \tag{2}$$

3. Thermodynamic Analysis

3.1. Energy and exergy balances

The governing equations of mass, energy and exergy conservation for a steady state flow rate are;

$$\sum \dot{m}_i = \sum \dot{m}_o \tag{3}$$

The first law of thermodynamic can be expressed as;

$$\dot{E}_i = \dot{E}_o \tag{4}$$

The general exergy balance can be expressed in the rate form as;

$$\dot{E}x_i - \dot{E}x_o = \dot{E}x_d \tag{5a}$$

Or

$$\dot{\mathrm{E}}\mathrm{x}_{heat} - \dot{\mathrm{E}}\mathrm{x}_{\mathrm{work}} + \dot{\mathrm{E}}\mathrm{x}_{\mathrm{mass,i}} - \dot{\mathrm{E}}_{\mathrm{xmass,o}} = \dot{\mathrm{E}}\mathrm{x}_{\mathrm{d}}. \tag{5b}$$

Using Equation (5b), the rate form of the general exergy balance can also be written as;

$$\sum (1-\frac{T_o}{T_r})\dot{Q}_k - \dot{W} + \sum \dot{m}_i\psi_i - \sum \dot{m}_o\psi_o = \dot{E}x_d, \tag{6a}$$

With

$$\psi = \varepsilon x^{PH} = (h-h_o) - T_o(s-s_o) \tag{6b}$$

3.2. Reference environment

Exergy is always evaluated with respect to a reference environment. The reference environment is in stable equilibrium, acts an infinitive system, is a sink or source for heat and materials, and experiences only internally reversible processes in which its intensive properties remains constant. In the calculations, the temperature To and pressure Po of the environment are often taken as standard-state values. If the system uses atmospheric air, To might be specified as the average air temperature. If both air and water from the natural surroundings were used, To would be specified as the lower values of the average temperatures for air and water [10].

Some important informations like average air temperature, air temperature annual amplitude, and heating design day, soil or rock temperatures and types are essential for our calculation. kerman has been chosen as typical region and relevant specifications are taken from Iranian meteorological organization [11]. The daily soil temperature is calculated using equation [12];

$$T_{\infty}(0,t) = T_a + A_o \sin\left[\frac{2\pi(t-t_0)}{365}\right] \quad (8)$$

3.3. Energy and exergy efficiencies

The energy (or first law) efficiency is simply a ratio of useful output energy to total input energy and is referred to as a coefficient of performance (COP) for refrigeration systems.

$$COP = \frac{\dot{\mathrm{Q}}\mathrm{condenser}}{\dot{\mathrm{W}}\mathrm{compressor}} \quad (9)$$

On the product/fuels (P/F) basis, exergy efficiency can be written as the ratio of total exergy output to total exergy input:

$$\varepsilon = \frac{\dot{\mathrm{E}}\mathrm{x}_o}{\dot{E}x_i} \quad (10)$$

3.4. Exergy analysis of the system studied

The mass and energy balance equations as well as the exergy destructions obtained using the entropy and exergy balance equations for each of the GCHP components illustrated in Fig. 1 are listed as follows, respectively.

For compressor:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_r \quad (11a)$$

$$\dot{W}_{comp} = \dot{m}_r(h_{2a} - h_1) \quad (11b)$$

$$\dot{E}x_{dest,comp,ent} = T_0 \dot{m}_r(s_1 - s_{2a}) \quad (11c)$$

$$\dot{E}x_{dest,comp} = \dot{m}_r(\psi_1 - \psi_{2a}) + \dot{W}_{comp} \quad (11d)$$

where the heat transfer versus the environment was neglected.

For condenser:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_r \;, \dot{m}_5 = \dot{m}_6 = \dot{m}_a \quad (12a)$$

$$\dot{\mathrm{Q}}_{\mathrm{cond}} = \dot{\mathrm{m}}_{\mathrm{r}}(\mathrm{h}_{2a}\text{-}\mathrm{h}_3); \quad \dot{\mathrm{Q}}_{\mathrm{cond}} = \dot{\mathrm{Q}}_{cfan} = \dot{\mathrm{m}}_{\mathrm{w}}\mathrm{C}_{\mathrm{p,w}}(\mathrm{T}_5\text{-}\mathrm{T}_6) \quad (12b)$$

$$\dot{E}x_{dest,cond,ent} = T_0[\dot{m}_r(s_3 - s_2) + \dot{m}_w(s_5 - s_6)] \quad (12c)$$

$$\dot{E}x_{dest,cond} = \dot{m}_r(\psi_{2a} - \psi_3) + \dot{m}_w(\psi_6 - \psi_5) \quad (12d)$$

For throttling valve;

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_r \quad (13a)$$

$$h_3 = h_4 \quad (13b)$$

$$\dot{E}x_{dest,tv,ent} = T_0 \dot{m}_r(s_4 - s_3) \quad (13c)$$

$$\dot{E}x_{dest,tv} = \dot{m}_r(\psi_3 - \psi_4) \quad (13d)$$

For evaporator;

$$\dot{m}_4 = \dot{m}_1 = \dot{m}_r \quad (14a)$$

$$\dot{Q}_{eva} = \dot{m}_r(h_1 - h_4); \; \dot{Q}_{eva} = \dot{Q}_{ghe} \quad (14b)$$

$$\dot{E}x_{dest,eva} = T_0[\dot{m}_r(s_1 - s_4) + \dot{m}_w(s_7 - s_8)] \quad (14c)$$

$$\dot{E}x_{dest,eva} = \dot{m}_r(\psi_4 - \psi_1) + \dot{m}_w(\psi_8 - \psi_7) \quad (14d)$$

For condenser fan;

$$\dot{\mathrm{m}}_{air,i} = \dot{\mathrm{m}}_{\mathrm{air,o}} = \dot{\mathrm{m}}_{air} \quad (15a)$$

$$\dot{\mathrm{Q}}_{cfan} = \dot{\mathrm{m}}_{\mathrm{a}}\mathrm{C}_{\mathrm{p,a}}(\mathrm{T}_{\mathrm{o,a}}\text{-}\mathrm{T}_{\mathrm{i,a}}) \quad (15b)$$

$$\dot{\mathrm{Q}}_{cfan} = \dot{\mathrm{Q}}_{\mathrm{cond}}; \; \dot{\mathrm{Q}}_{sph} = \dot{\mathrm{Q}}_{\mathrm{cond}} \quad (15c)$$

$$\dot{E}x_{dest,cfan,e} = T_0[\dot{m}_a(s_6 - s_5) + \frac{\dot{Q}_{cfan}}{T_{i,a}}], \quad (15d)$$

$$\dot{E}x_{dest,cfan} = \dot{m}_a(\psi_5 - \psi_6) - \dot{Q}_{cfan}\left(1 - \frac{T_o}{T_{i,a}}\right). \quad (15e)$$

For ground heat exchanger;

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_w \quad (16a)$$

$$\dot{\mathrm{Q}}_{ghe} = \dot{\mathrm{m}}_{\mathrm{w}}\mathrm{C}_{\mathrm{p,w}}(\mathrm{T}_8\text{-}\mathrm{T}_7), \quad (16b)$$

$$\dot{E}x_{dest,ghe,e} = T_0[\dot{m}_w(s_8 - s_7) + \frac{\dot{Q}_{ghe}}{T_{ground}}], \quad (16c)$$

$$\dot{E}x_{dest,ghe} = \dot{m}_w(\psi_7 - \psi_8) - \dot{Q}_{ghe}\left(1 - \frac{T_o}{T_{ground}}\right). \quad (16d)$$

4. Result and discussion

The effect of some rock or soil type on heat exchanger's length is shown in Fig.2. As can be seen for sandstone type with more thermal conductivity [13], the less length is required. The effect of ground temperature which is function of climate and rock or soil type and the water-antifreeze inlet temperature to the ground heat exchanger on the optimum length has been shown respectively on fig.3. The last plot shows that there is an optimum mass for decreasing heat exchanger's size and optimum length can't be reduced by increasing mass after a specified value. Results of exergy destruction of system components is given in in table 2. exergy analysis for selected system are shown table 3, and shows that the highest lost work occurs in condenser fan unit. Exergy efficiency of system studied is calculated to be 0.405 and COP is derived as 3.707.

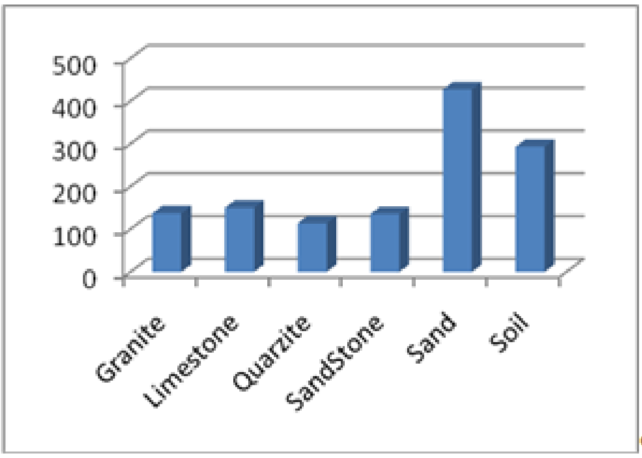

Fig.2. System exergy destruction

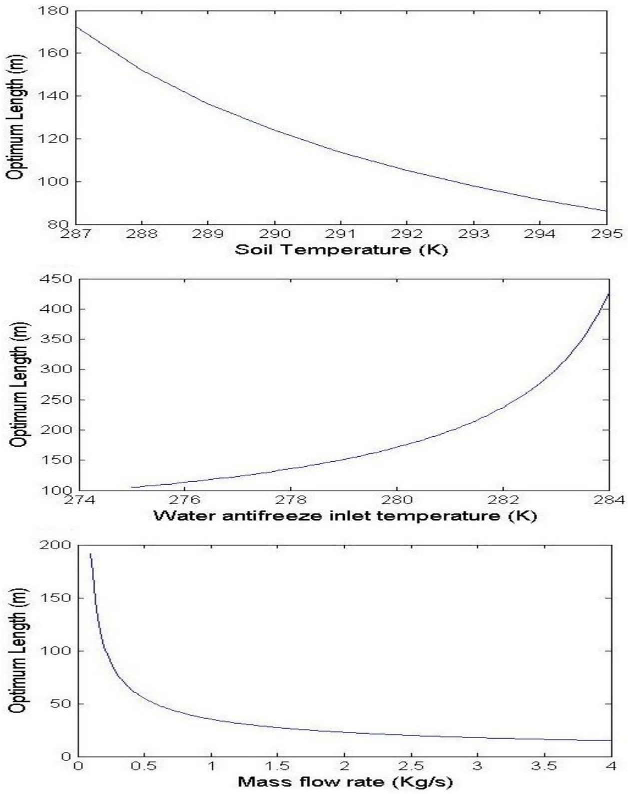

Fig.3. Parameters affecting the optimum length

Table 2. System exergy destruction

No.	Description	Exergy destruction(KW)
1	Evaporator	0.2166
2	Compressor	0.1134
3	Condenser	0.0943
4	throttling valve	0.1798
5	Condenser fan	0.8521
6	Ground Heat Exchanger	0.0629

Table 3. System exergy analysis

No.	Description	Fluid	Phase	Temp. $T(^0c)$	Pressure (KPa)	Specific enthalpy $h(KJ/Kg)$	Specific Exergy $\psi(KJ/Kg.K)$	Exergy rate $\dot{m}\psi(KW)$
0	-	R-134a	Dead state	2	101.325	403.3	0	0
0	-	Air	Dead state	2	101.325	275.4	0	0
0	-	Water-antifreeze	Dead state	2	101.325	8.503	0	0
1	Evaporator outlet	R-134a	Sup.heated vapor	-10.6	369.6	384.4	33.84	0.6494
2	Compressor outlet	R-134a	Sup.heated vapor	58.69	2032	432.3	75.8	1.455
3	Condenser outlet	R-134a	Liquid	34.4	1931	254.8	51.52	0.9887
4	throttling valve	R-134a	Mixture	-15.6	351.1	254.8	42.15	0.8089
5	Condenser fan inlet	Air	Gas	47.2	101.3	320.9	3.353	0.4997
6	Condenser fan outlet	Air	Gas	24.5	101.3	298.1	0.859	0.128
7	water-antifreeze outlet	Water-antifreeze	Liquid	4.8	300	20.38	0.1582	0.0233
8	water-antifreeze inlet	Water-antifreeze	Liquid	8.8	200	37.25	0.5454	0.0804

5. Conclusions

The optimization and analysis of a vertical ground-coupled heat pump system in kerman province was investigated and optimum length for a ground heat exchanger was calculated. The effects of different parameters such as rock and soil type, water-antifreeze inlet temperature and water-antifreeze mass flow rate on length of ground heat exchanger have been studied. Results shows that decreasing water-antifreeze inlet temperature can decrease required length of a heat exchanger. In addition the exergy losses of each component and exergy efficiency together with coefficient of performance of the system could be calculated.

Nomenclature

C	specific heat $(KJ/Kg.K)$
COP	coefficient of performance of heat pump (dimensionless)
$\dot{E}x$	exergy rate (kW)
h	specific enthalpy (KJ/Kg)
$\dot{m}$	mass flow rate (Kg/S)
p	pressure (kPa)
$\dot{Q}$	heat transfer rate (kW)
S	entropy $(KJ/Kg.K)$
$\dot{S}$	entropy rate (KW/K)
T	temperature (^{0}c or K)
$\dot{W}$	work rate or power (kW)

Greek letters

ε	exergy (second law) efficiency (dimensionless)
η	efficiency (dimensionless)
ψ	specific exergy (KJ/Kg)

Subscripts

opt	optimum
act	actual
comp	compressor
cond	condenser
dest	destruction
ent	entropy
eva	evaporator
fc	fan-coil
ghe	ground heat exchanger
cfan	condenser fan
tv	throttling valve
i	inlet
w	water
o	outlet
r	refrigerant

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